Water, Energy, Food and Environment Journal
An International Journal

http://dx.doi.org/10.18576/wefej/010102

Organic Rankine Cycle and Steam Rankine Cycle for Waste Heat Recovery in a Cement Plant in Egypt: A Comparative Case Study

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Received: 28 Jun. 2019, Revised: 22 Sep. 2019, Accepted: 30 Oct. 2019.
Published online: 1 Jan. 2020..

Abstract: Investigations of the Rankine cycle for power generation using waste heat from a cement plant were performed, while considering three waste streams of three different temperature ranges. Six organic fluids, in addition to water, were selected to be employed as working fluids in five suggested Rankine cycle schemes built and simulated with Aspen HYSYS v8.8. The suggested schemes were made to suit the types of the working fluids used in the investigations. Working fluids were compared based on efficiency to select the best organic fluid to be compared with water. The basic comparisons resulted in the selection of seven methanol points in scheme 5 (based on the turbine inlet pressures) to be compared with nine points of water in schemes 1 and 4. The comparisons showed that methanol resulted in the highest net power output and cycle efficiency with the lowest irreversibility. Furthermore, the selected methanol points showed the highest net power output values, in addition to the lowest payback period of 4 years with high net present values. Consequently, methanol is suggested as a better choice for working fluid in Rankine cycle for waste heat recovery in the cement industry.
Keywords: Waste heat recovery; Cement Industry; Organic Rankine Cycle; Power generation.

1 Introduction

The cement industry consumes extensive amounts of energy which accounts for about 12-15% of the total industrial energy [1]. In addition, the cement industry is one of the most energy-intensive industries with energy represents about 50-60% of the production costs [2]. Each produced tonne of clinker consumes about 110-120 kWh of electric energy and 4-5 GJ of thermal energy [2,3]. Among which, 40% of the total input energy is being lost through hot exhaust gases, 60% of this energy is lost in the form of the hot flue gases from the pre-heater and the hot air from the clinker [4].

The cement industry in Egypt is considered the most energy consuming sector with more than 9% of the total primary energy [5]. Thus, the recovery of waste heat from the cement industry presents an appealing solution that may result in clean ways for power generation, reduction of the consumption of fossil fuel, and mitigation of many environmental problems [6].

Most of waste heat recovery (WHR) projects in the cement industry have been for power generation as the industry has not any significant low temperature heating requirements [3]. The main power generation technologies are

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steam Rankine cycle (whether single flash or dual pressure), organic Rankine cycle (ORC) and Kalina cycle [2]. The steam Rankine cycle is the most common applied technology and it accounts for 99% of the existing WHR installations, while there are only nine ORCs and two Kalina cycles in the cement industry worldwide [3].

In recent years, there is an improvement in the overall efficiency of cement manufacture; thus the newer cement plants are of higher efficiencies. This improvement results in lowering the temperatures of exhaust gases. Consequently, the opportunities of both ORC and Kalina cycle will increase due to the advantage of these cycles for low-grade temperature applications [7,8]. Compared to the ORC, the Kalina cycle gives more power output for the same heat input, but it is much more complex and needs more maintenance [9,10]. On the other hand, the produced energy by the ORC can fulfill about 10-20% of the total electrical energy demand in the cement plant [11].

The use of ORC for power generation has attracted more and more attention in recent years due to the suitability of the cycle to recover heat from a wide range of heat sources such as waste heat, solar radiation, biomass and geothermal energy [12,13]. The operation efficiency of the cycle is very dependent on the operation conditions and the thermodynamic properties of the working fluid [14,15]. The selection of the working fluid is very dependent on the target application, the working conditions and the considered criteria [16]. General criteria considered in the selection of the most suitable organic fluid for any application include: thermodynamic and physical properties of the fluid, fluids stability and compatibility with materials in contact, safety (flammability, toxicity and corrosivity), environmental aspects (Ozone Depletion Potential (ODP) and Global Worming Potential (GWP)), and the availability of the fluid and costs [9,17].

In the past few years, many researchers focused on the selection of working fluids for the ORC from different aspects. Drescher and Brüggemann [18] considered 700 substances for the high grade temperatures from biomass plants and found that the alkylbenzenes family shows highest thermal efficiencies. Siddiqi and Atakan [19] made a systematic study to compare the use of hydrocarbons with water as working fluids for the heat recovery of sources with different temperature ranges and found that hydrocarbons such as n-hexane and n-pentane are promising in the low temperature range, n-dodecane and toluene are promising in the higher temperature ranges, while octane, heptanes and water are well suited working fluids for the medium temperature ranges. Khatita et al. [20] utilized the ORC for the heat recovery of a high temperature waste heat in an existing gas treatment plant in Egypt. Eleven organic fluids and water were considered to select the best working fluid and benzene and cyclohexane were proposed as the most promising choices. Finally, Bao et al. [21] reviewed the working fluids investigated for the ORCs and summarized the recommended candidates for different heat sources in different applications.

Only few researches in the literature considered ORC for the WHR in the cement industry. Citrin [22] presented the application of ORMAT energy converter based on the ORC to recover the waste heat in the Heidelberger Cement plant in Lengfurt, which was the first application of the ORC in the cement industry with 1.5 MWe electricity generation. Wang et al. [2] compared single flash steam cycle, dual-pressure steam cycle, ORC and the Kalina cycle for the recovery of waste heat in the cement industry based on energy analyses under the same conditions. R123 was the only employed working fluid for the ORC, which resulted in bad efficiency results for the ORC. This is attributed to that the usage of a single candidate working fluid in their investigations without considering that there is no universal optimum organic fluid that can be used for all ORCs since the suitable working fluid differs according to the operation conditions of the cycle and other criteria. Furthermore, Amiri and Vaseghi [23] reviewed the steam Rankine cycle, the ORC, the Kalina cycle, and the supercritical CO_2 cycle without recommending the best cycle from them. Karellas et al. [8] tried to identify the best practice system for the recovery of waste heat in the cement industry using the steam Rankine cycle and the ORC employing an intermediate heat transfer fluid. They examined R245fa, neopentane, pentane and isopentane as working fluids for the ORC. Isopentane was selected as the working fluid because it gave the maximum system efficiency. Mortada et al. [24] reported that it was possible to generate up to 2.5 MW electric from flue gases in the cement plant they studied. Moreover, Wang et al. [25] investigated ORCs for the recovery of waste heat from a typical cement production line in China and chose hexane, isohexane, R601, R123 and R245fa as working fluids to select the most suitable one for the ORC. They found that R601 resulted in the best economic performance with significant reduction of gas emissions. Sánchez et al. [26] performed mass and energy balances analyses of a typical rotary kiln cement plant to determine the process overall energy efficiency and to enable WHR using an ORC cogeneration system. The results showed that about 5.5 GWh/year of electricity and 23.7 GWh/year of thermal energy can be produced in the cement plant by the ORC cogeneration system with expected payback period of 4.2 years. Fergani et al. [27] employed three working fluids for a parametric study of an ORC used in the cement industry and evaluated the effects of some key parameters on the system performance using the exergy, exergo-economic and exergo-environmental approaches. They found that best thermodynamic performances were obtained with higher turbine inlet and that the best fluid from the thermodynamic and exergo-economic viewpoints was cyclohexane, while the best from exergo-environmental point of view was benzene. Finally, Aboelwafa et al. [28] employed methanol and water as working fluid for the recovery of waste heat in a cement plant in Egypt and reported that water resulted in lower payback period while the higher power output resulted with methanol.

The main objective of this paper is to compare between ORC and steam Rankine cycle in WHR for power generation in a cement plant in Egypt, while considering three waste streams of three different temperature ranges. Based on the previous literature review, there is no work reported to deal with recovering the heat from multiple streams with different temperatures in such heavy industry. Because of such, five different schemes of the Rankine cycle have been suggested based on the type of working fluid and the operating conditions. Comparisons, based on thermodynamic efficiency and economics, have been applied to select the best organic fluid to be studied economically in more details. A detailed economic study has been performed to compare the organic fluid resulted in the highest power output and efficiency with water to select the best candidate for the cement plant.

2 Methodoloy

2.1Working Fluid Selection

The Rankine cycle, whether it is the conventional steam Rankine cycle employing water as a working fluid or the organic Rankine cycle employing organic fluid, consists of four components: pump, evaporator, turbine and condenser. In these components, four processes occur: pressurizing, evaporation, expansion, and cooling.

Three types of working fluids are employed for the Rankine cycle. They are classified according to the slope of the vapor line curve in the temperature-entropy diagram. The three types are: (1) wet fluids with negative slopes, (2) dry fluids with positive slopes, and (3) isentropic fluids with slopes of infinite value as they have nearly vertical saturated vapor curves. Figure 1 shows these three types.

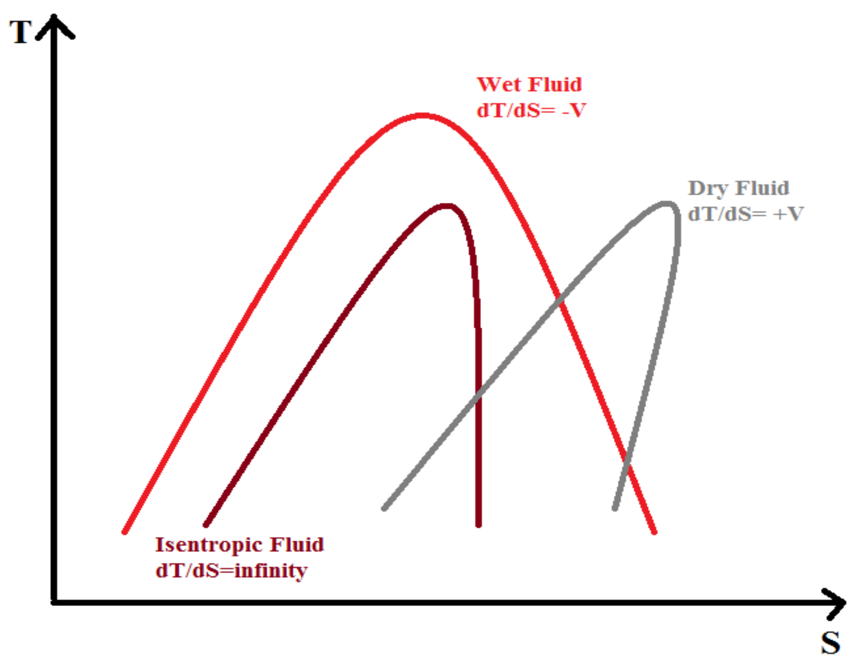


Fig. 1: Temperature-entropy diagram of dry, wet, and isentropic working fluids classified according to the value of the slope (dT/dS).

Starting from previous work in the same ranges of waste heat temperature [20,29], and the fact that water is the most used working fluid in waste heat recovery applications [3], water, as a wet fluid, and six organic working fluids have been selected for the current work to compare the ORC with the steam Rankine cycles. The six organic fluids are four dry fluids namely: benzene, cyclo-hexane, n-hexane and n-heptane and two wet fluids: methanol and ethanol. The selection criteria were: the physical properties, thermal and chemical stability, environmental aspects and the fluids availability. Table 1 shows the physical properties of the selected fluids.

Table 1: Properties of the selected working fluids based on component list properties of Aspen Hysys V8.8.

Working Fluid	Tc (°C)	Pc (bar)	Normal Boiling Point (°C)
Benzene	288.9	49.24	88.09
Cyclo-hexane	280.1	40.53	80.73
n-heptane	267	27.37	98.43
n-hexane	234.7	30.32	68.73
Ethanol*	247.465	69.46	78.05
Methanol	239.4	73.76	64.48
Water	374.1	221.2	100

* Ethanol used as a mixture of ethanol/water (95/5 wt%) to simulate commercial ethanol.

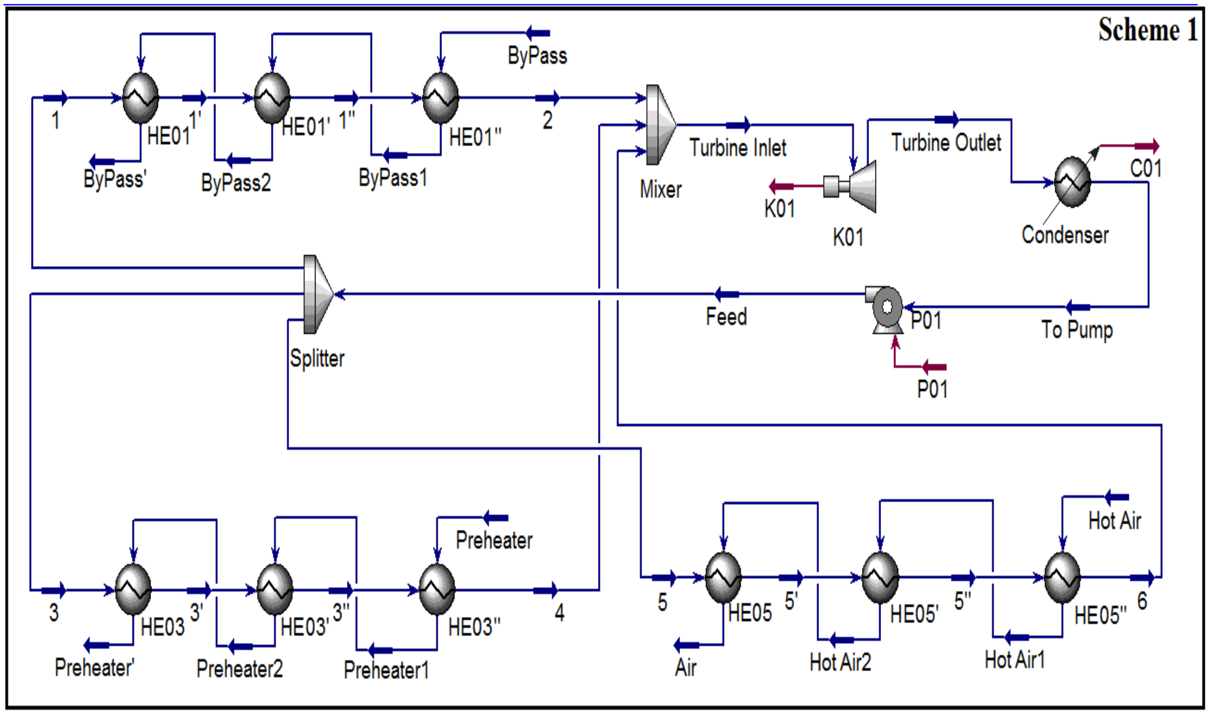
2.2 Rankine Cycle Schemes

Aspen HYSYS, as a proven industry/standard, v8.8 was used to simulate the five Rankine cycle schemes suggested for the recovery of heat from three waste streams in a cement plant, of 4500 tonnes per day capacity, located in Beni Suef in middle Egypt. Waste heat streams from three waste streams from the cement plant were considered in the suggested simulation models. These streams are: 1) hot air from the clinker cooler at 360°C, 2) hot flue gases from the preheater at 270°C. and 3) hot flue gases from the bypass exhaust on the preheater at 650°C. Each working fluid type is investigated in the suitable scheme.

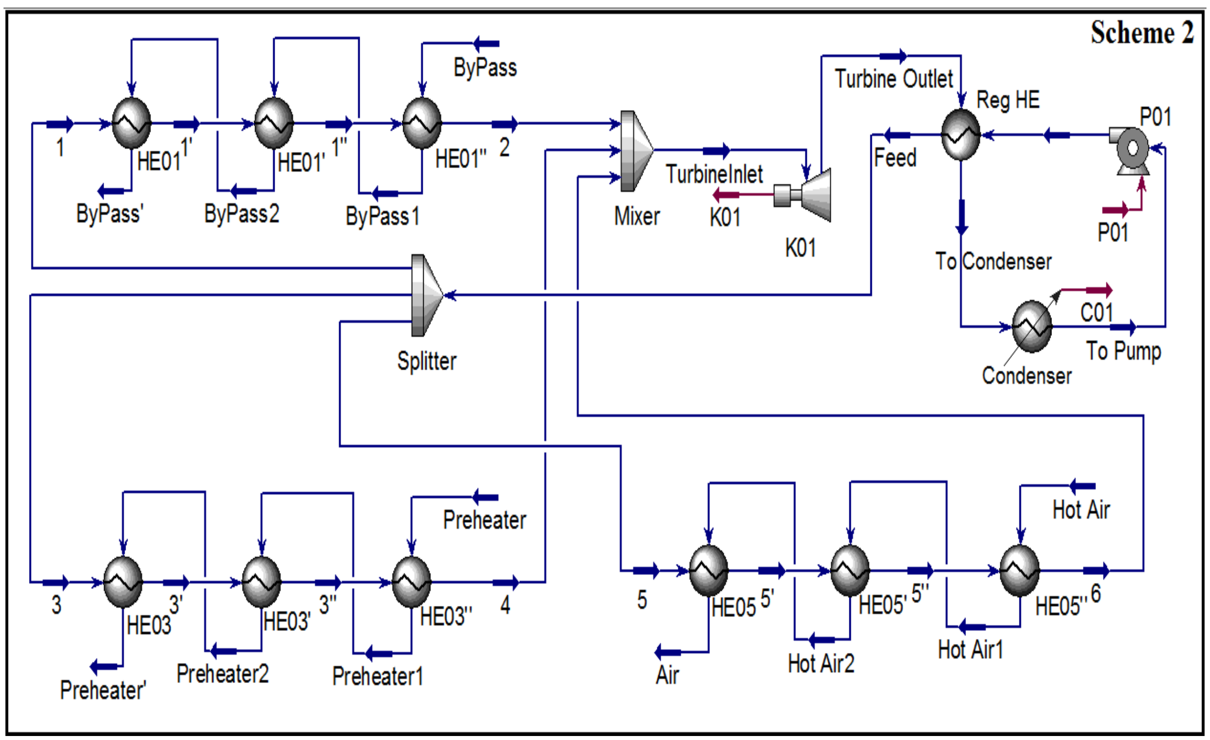
Figures 2 (A-E) shows the five suggested Rankine cycle schemes. Scheme 1 (Figure 2A) is the basic scheme, which consists of: (1) nine heat exchangers, three of them for each waste stream, acting as economizers, evaporators and superheaters, if required, (2) a turbine for expansion and power generation, (3) a condenser for condensation. and (4) a pump in which the working fluid is pressurized. Other components are the mixer before the turbine to mix the hot working fluids before the turbine and the splitter to split the feed stream into three flows to be heated by the hot waste streams. In this scheme, the wet fluids: ethanol, methanol, and water were investigated. These working fluids were superheated to a sufficient degree to ensure that the stream leaving the turbine is a saturated vapor. Scheme 2 (Figure 2B) is similar to scheme 1, but it has a regeneration heat exchanger (Reg HE). The dry fluids were investigated in such scheme. The Reg HE is used in the recovery of heat from the superheated stream leaving the turbine.

Based on the positive effect of superheating the wet fluids on the cycle efficiency [9], scheme 3 (Figure 2C) was proposed to maximize this advantage. Thus, methanol and ethanol were employed in scheme 3. Water was not employed in scheme 3 since it requires heating stream (2) higher than 550˚C to obtain suitable higher temperatures before the turbine to capture the advantage of superheating on the efficiency. These higher temperatures are not suitable from the common operation temperatures of water in superheaters since it results in high thermal stresses in the superheater [30].

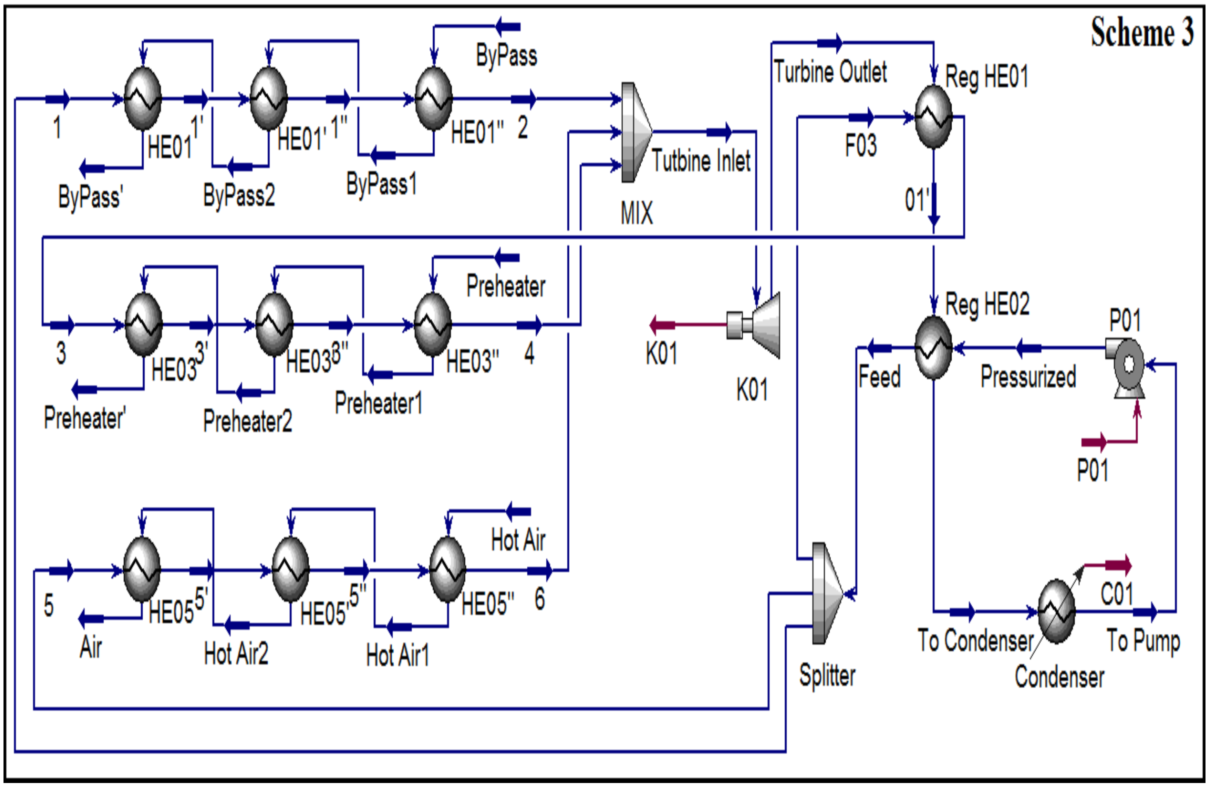
Water is characterized with high critical pressure of water compared to other chosen working fluids (Table 1). Accordingly, scheme 4 (Figure 2D) was proposed to use three-stage turbine to take advantage of this. Taking into consideration the maximum temperature of steam at the turbine inlet of 550˚C and that the steam should leaves the



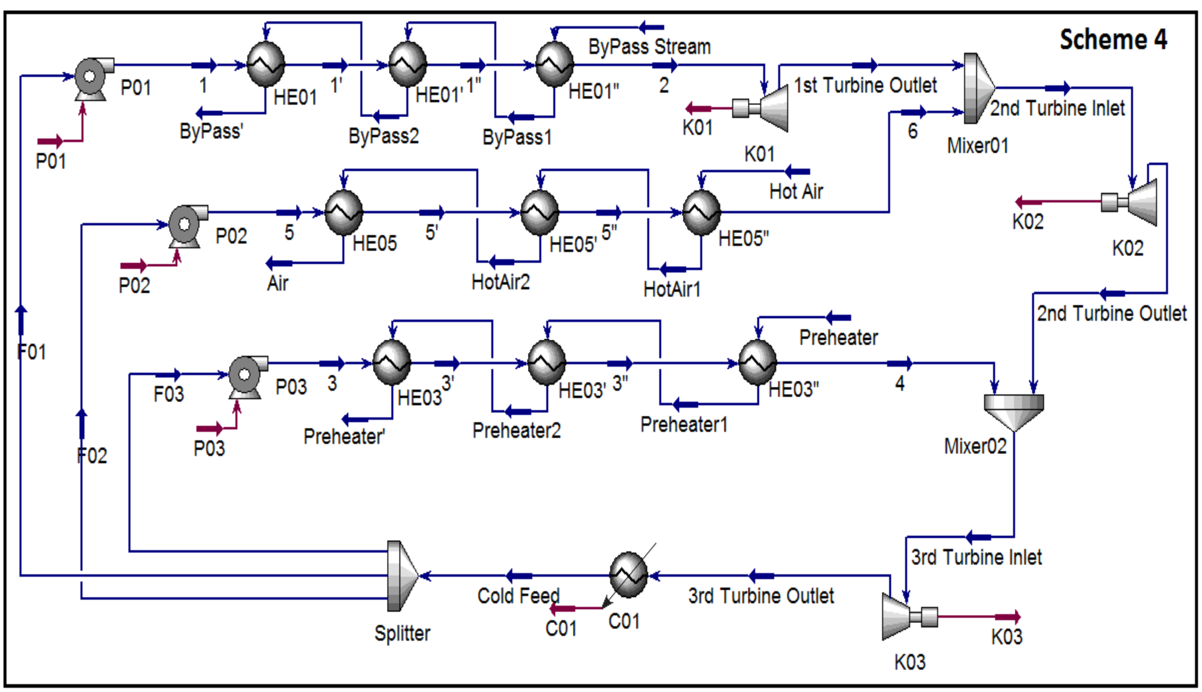
A



B



C



D

third stage without condensation, the inlet pressure of the first stage was adjusted accordingly to keep the stream leaving the turbine as saturated vapor. The inlet pressures of the second and third stages were varied accordingly to cover wide range of inlet pressure values.

Methanol and ethanol have relatively high critical pressures compared to the dry hydrocarbons investigated in the current work. Hence, scheme 5 (Figure 2E) is presented to take the advantage of the high critical pressures using multi stage turbines as those used in scheme 4, in addition to the superheating used in scheme 3. However, the configuration of scheme 5 differs a lot from schemes 3 and 4 in is the way of heat recovery from the stream leaving the third turbine using the regeneration heat exchangers.

In the five suggested schemes, the turbine outlet pressure is set to the minimum suitable values according to the type of the working fluid. For the organic working fluids, the turbine outlet pressure is set to be 1.2 bar with the consideration of ±0.1 bar in the control system and the consideration of sustaining a positive pressure difference for safety reasons. In case of water, the turbine inlet is set to 1 bar.

For each scheme, the organic fluids and water are investigated and the net power output and then cycle efficiency are calculated. The organic fluid resulted in the highest cycle efficiency, will then be chosen to be compared with water. The comparison will be based on the cycle irreversibility and economic study, in which the rate of return and net present value shall be calculated. Finally, the parameters considered in simulation are shown in Table 2.

Table 2: Parameters used in the simulation model.

Parameter	Value
Minimum Approach for Heat Exchangers	20°C
Heat Exchanger Pressure Drop (Working Fluid)	50 kPa
Heat Exchanger Pressure Drop (Waste Streams)	15 kPa
Cooler Pressure Drop	20 kPa
Turbo Expander Adiabatic Efficiency	75%
Temperature of Preheater Stream	230°C
Temperature of ByPass Stream	150°C
Temperature of Air Stream	150°C
Turbine Outlet Pressure	1.2 bar
Turbine Outlet Pressure (in case of water)	1 bar

where: I: irreversibility, S_{gen}: the entropy generation rate [kW/K], T_o: surrounding temperature [K], m^o: the total mass flow rate of the working fluid [kg/s], Ex_{in}: incoming exergy flows, Ex_{out}: outgoing exergy flows, q_i: the specific heat of the jth component of the cycle [kJ/kg], and T_j: The temperature of the j_{th} component of the cycle [K].

After the application of the first law of thermodynamics to calculate the rate of heat transfer and power in each component of the cycle, the following equations were generated:

- The heat exchangers:

$$q_{HEn} = h_{inlet\ stream} - h_{outlet\ stream} \tag{3}$$

- Turbo Expander (K01, K02, K03)

$$p_{k01,02,03} = h_{Inlet} - h_{Outlet} \tag{4}$$

- Condenser (C01)

$$q_C = h_{condenser\ inlet} - h_{condenser\ outlet} \tag{5}$$

- Pump (P01,P02,P03)

$$P_{P01,02,03} = h_{inlet} - h_{outlet} \tag{6}$$

q_{HEn}: absolute values of specific heat in the heat exchanger (n) whether evaporators or the after turbine heat exchangers.

$P_{K01,02,03}$: absolute values of turbo expander specific work (K01,K02,K03).

q_{C01}: absolute values of specific heat in the condenser (C01).

$P_{P01,02,03}$: absolute values of pump specific work (P01,P02,P03).

$h_{inlet\ stream}$: specific enthalpy for the working fluid at the inlet of the equipment (HE_n).

$h_{outlet\ stream}$: specific enthalpy for the working fluid at the inlet of the equipment (HE_n).

$h_{condenser\ inlet}$, $h_{condenser\ outlet}$: specific enthalpy for the condenser inlet and outlet streams.

$h_{pump\ inlet}$, $h_{pump\ outnlet}$: specific enthalpy for the pump inlet and outlet streams.

4 Parameters for Comparison

4.1 Power Output and Efficiency

The comparison between the investigated working fluids in the suggested schemes is based firstly on the power output and cycle efficiency to select the best organic fluid for further comparison with water.

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

$$\text{Net Power Output (kW)} = \text{Turbine Power Output (kW)} - \text{Pump Power (kW)} \tag{7}$$

b) Efficiency (η_{th}):

The efficiency of the cycle is a very important factor as it represents the relation between the net power output and the thermal efficiency. The higher the thermal efficiency the greater the net power output. The efficiency is calculated as follow [31]:

$$\eta_{th} = P_{net}/Q^o_{in} \tag{8}$$

$$Q^o_{in} = \sum m^o \times \sum(h_{Eij} - h_{Eoj}) \tag{9}$$

Where: Q^o_{in}: the heat gained from the exhaust gases, [kW], $\sum m^o$: the sum of the mass flow rate of the exhaust gases [kg/s], h_{Eij}: specific enthalpy for the exhaust gas (j) in [kJ/kg], and h_{Eoj}: specific enthalpy for the exhaust gas (j) out [kJ/kg].

4.2 Irreversibility

The irreversibility is a representation of the lost opportunity to do work. In a steady state flow condition, the irreversibility rate for a cycle can be expressed as:

$$I_{tot} = I_{HE} + I_{Condenser} \tag{10}$$

Where: I_{tot}: irreversibility [kW], I_{HE}: irreversibility in the heat exchangers [kW] and $I_{Condenser}$: irreversibility in the condenser [kW].

4.3 Capital Cost and Profitability

Capital cost was estimated using Aspen Process Economic Analyzer software v8.8 integrated with Aspen Hysys v8.8. The economic analyzer obtained the required data from the simulator to perform the capital cost estimation of the equipment.

After calculating the capital cost, profitability study have been carried out on the organic working fluid compared to water. The rate of return and the net present value have been used for the economic comparison:

a) Rate of Return

Rate of return on investment is expressed on an annual percentage basis. The yearly profit is divided by the total initial investment necessary times 100 represents the standard percent return on investment. To determine the profit, estimates are made for direct production costs, fixed charges including depreciation, plant overhead costs, and general expenses [32].

$$\text{Rate of Return} = (\text{Profit} / \text{Capital Costs}) \times 100\% \tag{11}$$

b) Net Present Value (NPV)

Net cash flows and net benefits are important for the project developers. Net Present Value (NPV) method is a powerful indicator of the viability of the projects and can be determined from the following equation [20,33]:

$$NPV = \sum (B - C)_j \, a_j \tag{12}$$

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:

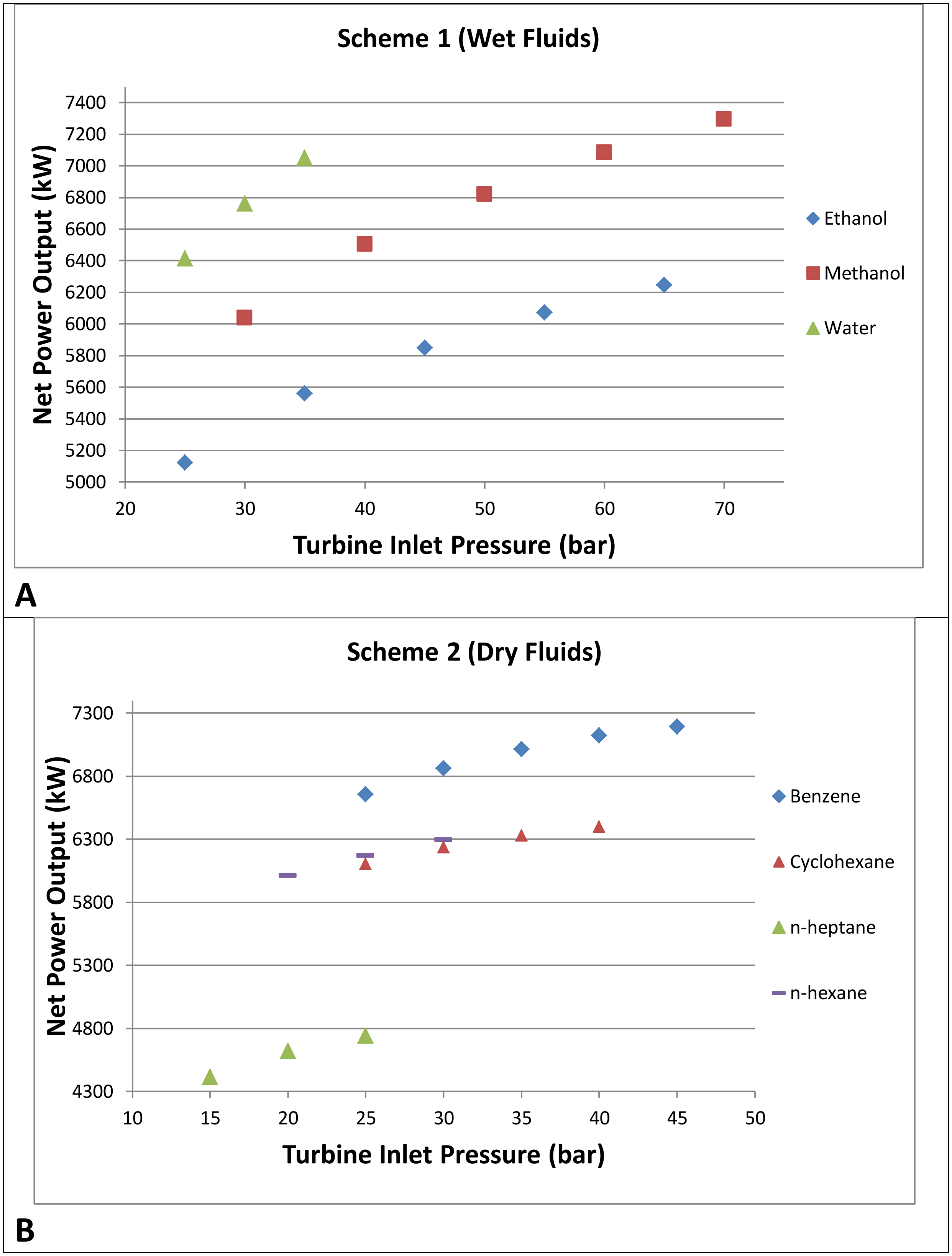
$$a = 1 / (1+i)^p \tag{13}$$

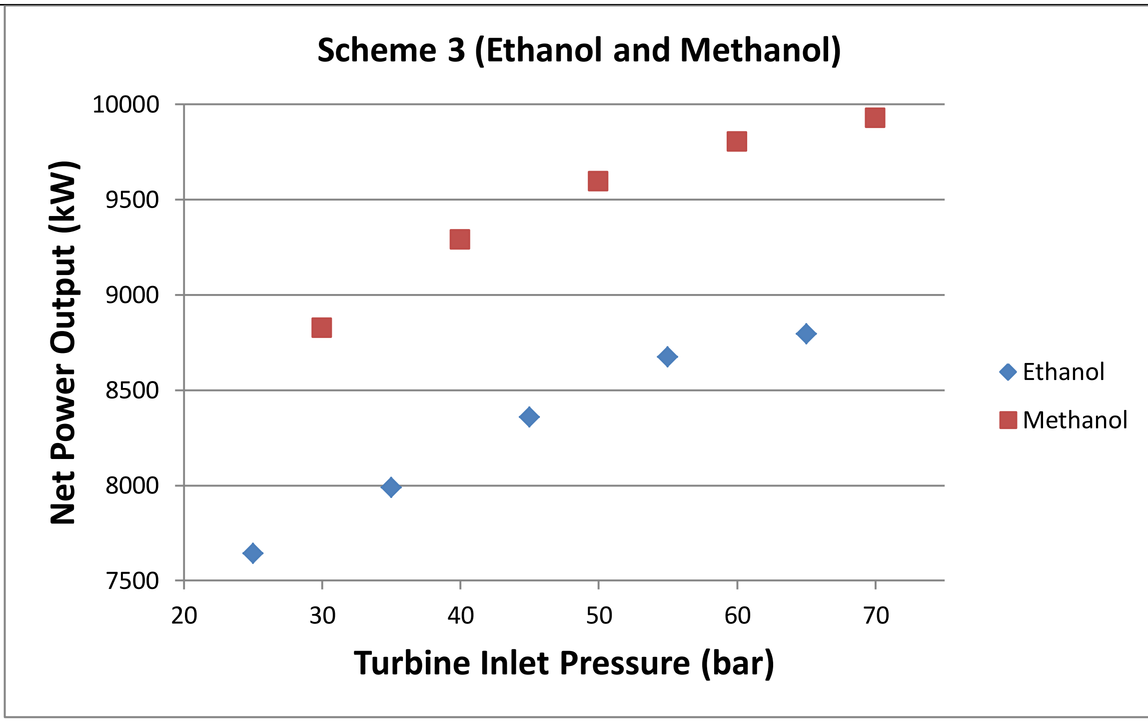
where i is the interest rate and p is the period.

5 Results and Discussion

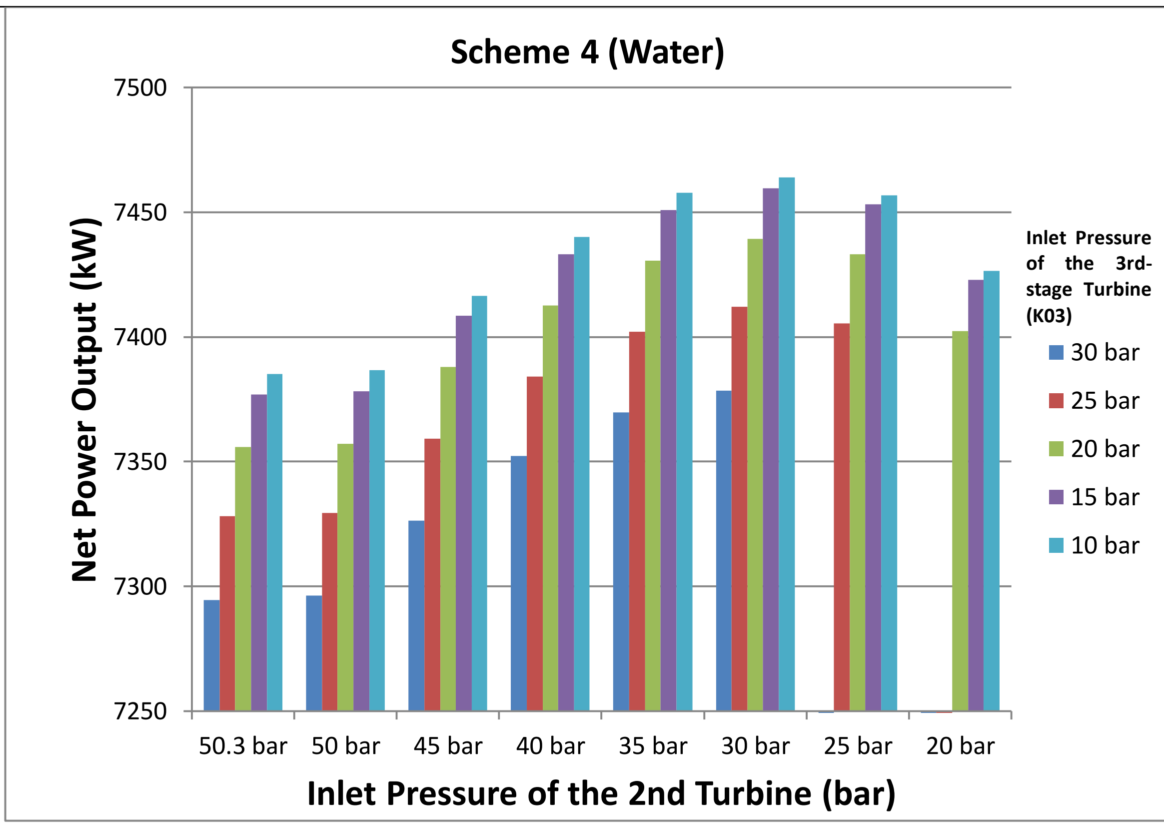
5.1 Net Power Output and Efficiency

Figures 3(A-F) show the net power output results of the working fluids investigated in schemes (1-5). In schemes 1-3, as the turbine inlet pressure increases, the net power output increases. This is attributed simply to the increase of the pressure ratio (the turbine inlet pressure / the turbine outlet pressure). In scheme 1, water results in high net power output values compared to ethanol and methanol. However, water cannot be investigated at turbine inlet pressures greater than 35 bar due to the liquid formation at these pressures resulted from Stream 4 of the working fluid. Thus, methanol results in higher net power output values as it is investigated at higher pressures than that of water.


Scheme 1 (Wet Fluids)
Net Power Output (kW)
7400
7200
7000
6800
6600
6400
6200
6000
5800
5600
5400
5200
5000
20
30
40
50
60
70
Turbine Inlet Pressure (bar)
Ethanol
Methanol
Water
A
Scheme 2 (Dry Fluids)
Net Power Output (kW)
7300
6800
6300
5800
5300
4800
4300
10
15
20
25
30
35
40
45
50
Turbine Inlet Pressure (bar)
Benzene
Cyclohexane
n-heptane
n-hexane
B


Scheme 3 (Ethanol and Methanol)
Net Power Output (kW)
10000
9500
9000
8500
8000
7500
20
30
40
50
60
70
Turbine Inlet Pressure (bar)
Ethanol
Methanol

C


Scheme 4 (Water)
Net Power Output (kW)
7500
7450
7400
7350
7300
7250
50.3 bar
50 bar
45 bar
40 bar
35 bar
30 bar
25 bar
20 bar
Inlet Pressure of the 2nd Turbine (bar)
Inlet Pressure of the 3rd-stage Turbine (K03)
30 bar
25 bar
20 bar
15 bar
10 bar

D

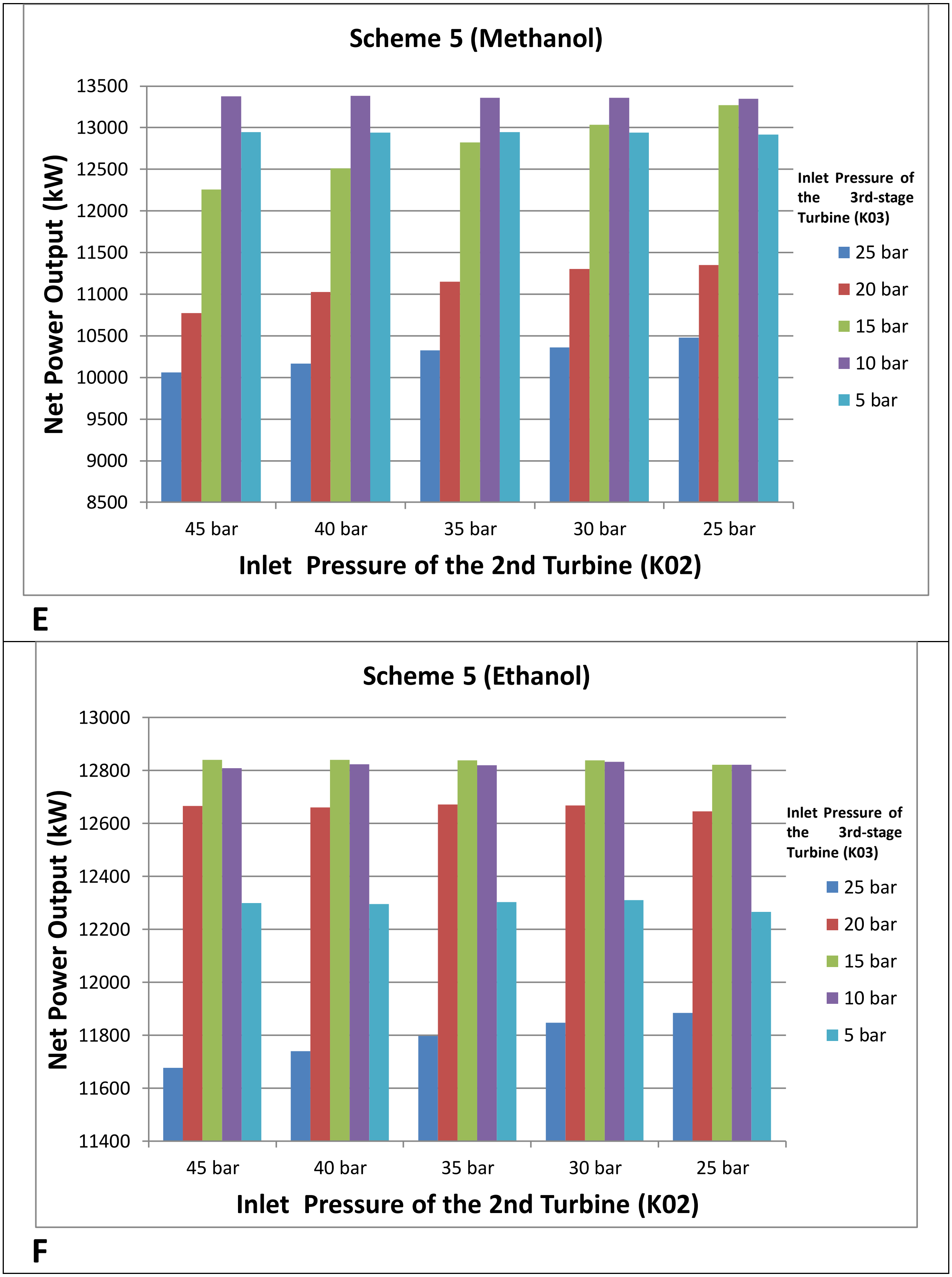


Fig.3: Net power output of the Rankine cycles of schemes (1-5).

In scheme 2, benzene results in the highest net power output values followed by n-hexane, then cyclo-hexane and finally n-heptane, which results in the lowest net power output values. The comparison between the results of scheme 2 with that of scheme 1 shows that benzene results in the highest net power output followed by water and methanol while ethanol and n-heptane result in the lowest net power output values. Thus, dry fluids can be better than wet fluids if the investigations are done in the basic Rankine cycle configuration suitable for each type. However, results of scheme 3, in which wet fluids are superheated and involves double recovery of heat from the turbine outlet stream using regeneration heat exchangers, show that wet fluids can result in higher net power output values. In case of ethanol, this ranges from 40-49% increase, while in case of methanol, the increase in the net power output ranges from 36-46%. This matches with that superheating has positive effect on the system power output.

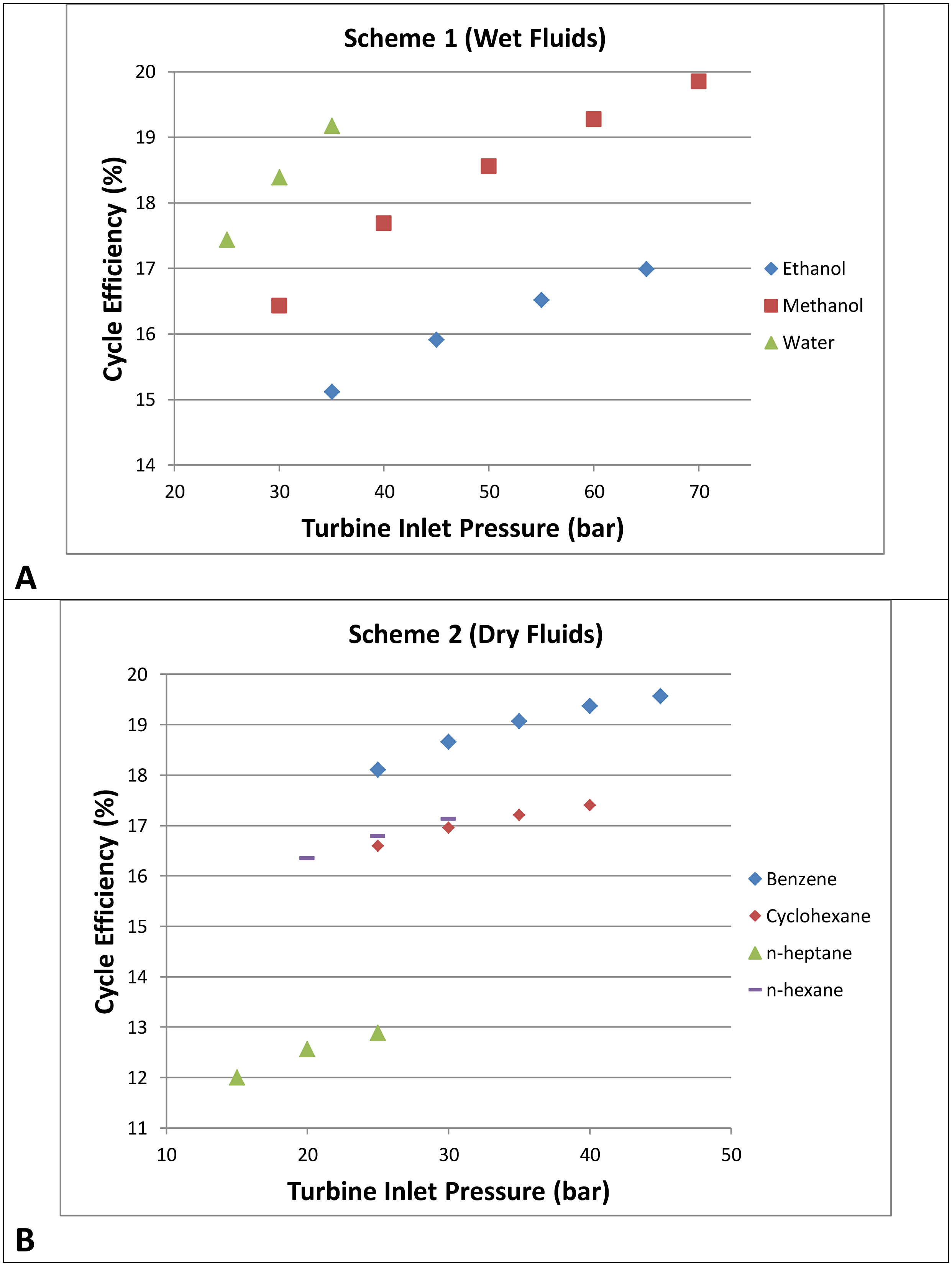
In Scheme 4, including the use of multistage turbines, water is investigated at different values of the inlet pressure of the second-stage turbine (K02). For each value of the inlet pressure of the second-stage turbine, the value of the inlet pressure of the third-stage turbine (K03) is varied to cover wide range of pressures. The inlet pressure of the first-stage turbine (K01) is adjusted to the value sufficient for the stream leaving the third-stage turbine to be saturated vapor. Results show that there is an optimum value of the inlet pressure of the second-stage at 30 bars, at which the net power output values is the highest. However, there is not any optimum value of the inlet pressure of the third-stage turbine (K03) within the covered range. However, it may be at about 10 bars, as the difference between the net power output values decrease with decreasing the value of the inlet pressure of the third-stage turbine. For example, the difference between the power output at 20 and 15 bars (20kW) is greater than that between 15 and 10 bars (4kW) at inlet pressure of the second-stage turbine of 30 bars. The results of scheme 4 also show that the use of multistage turbines does not affect greatly the net power out values when compared to the results of water in scheme 1. The increase in net power output is only about 5-15%.

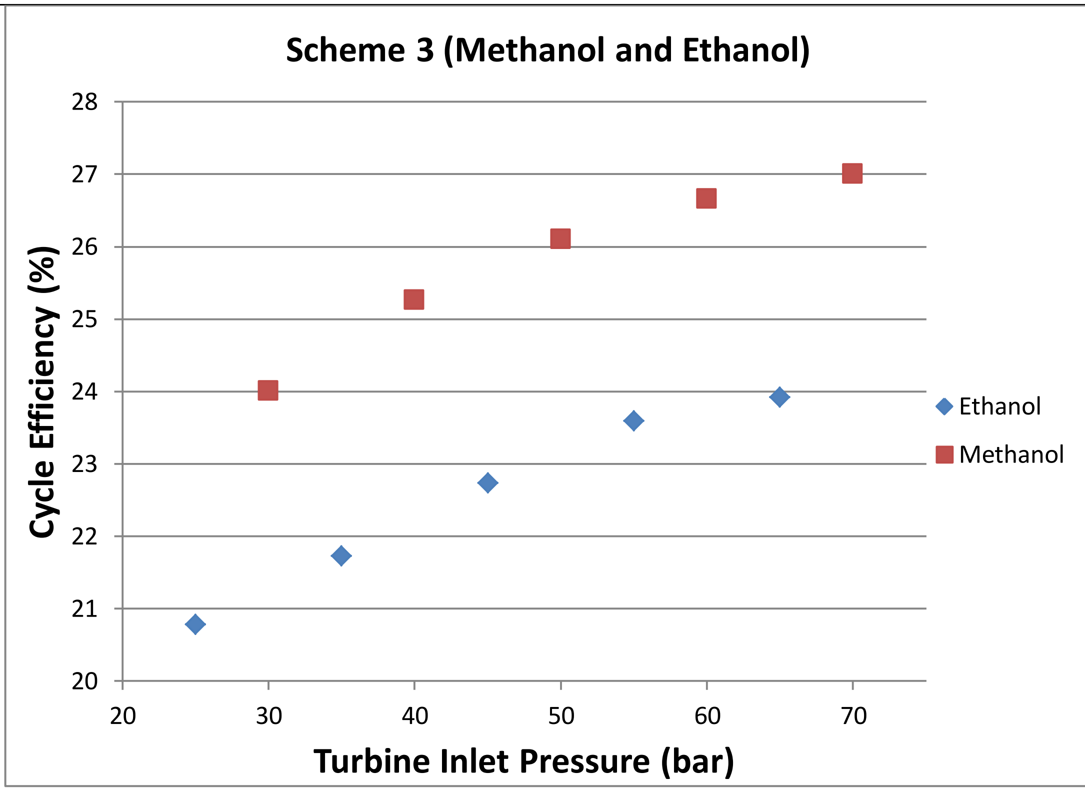
On the other hand, the same concept of multistage turbines has applied for methanol and ethanol in scheme 5. However, with the consideration of the effect of superheating on the power output of the system, the net power output has shown a dramatic increase. The net power output for methanol and ethanol in scheme 5 compared to scheme 1 is roughly more than the double. Moreover, if the results of scheme 5 are compared to those of scheme 3, the increase in the net power output for ethanol and methanol is about 40-55% and 18.7-30%, respectively. Finally, there is an optimum net power output at inlet pressure of the 3rd turbine (K03) equals 10 bars and 15 bars for methanol and ethanol, respectively. The high net power output values might be attributed to: (1) the use of multi-stage turbine, which enables the optimization based on expansion ratios in the turbines, (2) the use of superheating and regeneration together, and (3) the recovery of a part of sensible heat in the regeneration heat exchanger; thus, the stream enters the condenser partially condensed.

For the cycle efficiency, Figures 4(A-E) show the results of cycle efficiency of schemes (1-5). The same trend of results in case of net power output is found in the results of cycle efficiency as the cycle efficiency is a function of net power output. The cycle efficiency of methanol and ethanol in scheme 5 is the highest followed by that of the same fluids in scheme 4. Water in scheme 3 also results in efficiencies higher than that of water in scheme 1. For dry fluids investigated in scheme 2, benzene results in the highest cycle efficiency compared to the other hydrocarbons or to methanol, ethanol and water investigated in scheme 1.

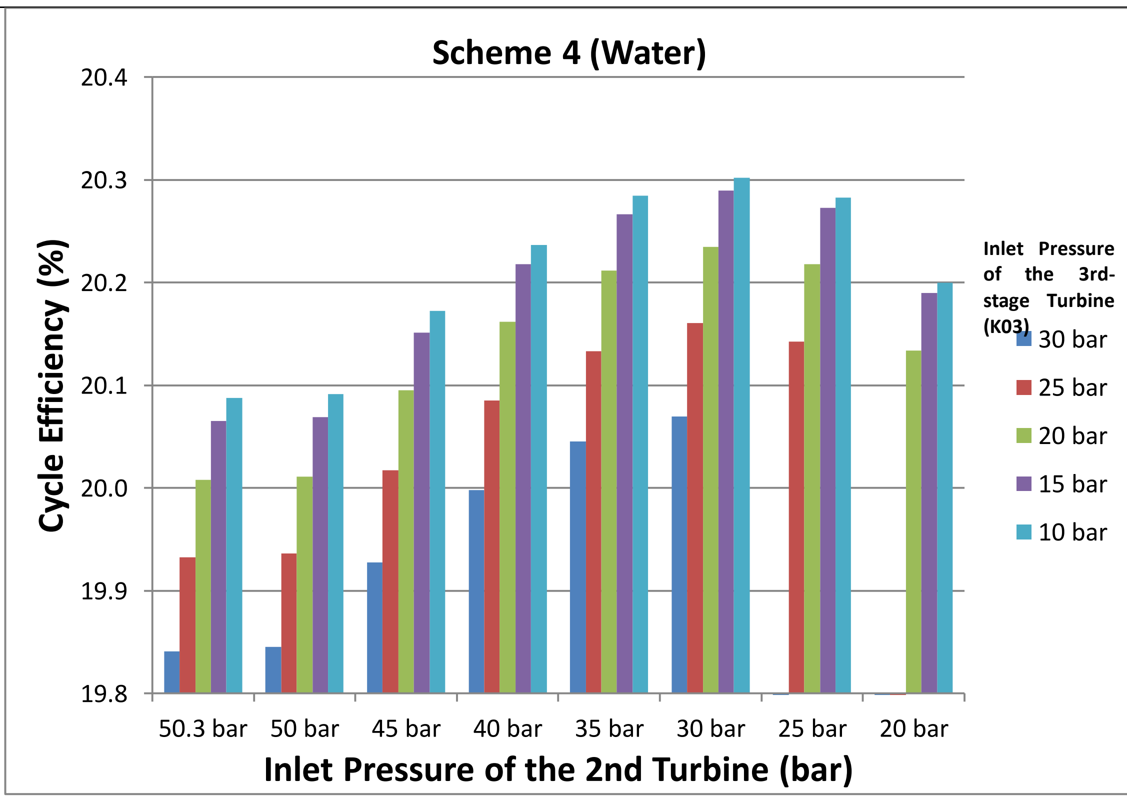
Based on these results, further comparison is performed between the best organic fluid and water. Water is selected as the comparison reference due to the fact that the most waste heat recovery systems for power generation are based on steam Rankine cycle. The aforementioned results show that methanol in scheme 5 has the best results. On the other hand, water in scheme 4 results in better results than that of scheme 1. However, since the difference between efficiency results of water in scheme 1 and scheme 4 are comparable (5 - 15% as referred in the discussion), both the schemes of 1 and 4 are to be compared with methanol in scheme 5.

For methanol in scheme 5, the points at which the cycle efficiency is greater than 35.5% are to be considered in the comparison. For water, the three turbine inlet pressure values in scheme 1 and those in scheme 4 are to be included in the comparison since the efficiencies are very close. The selected points are those with cycle efficiency 20.25% or greater. Table 3 shows the selected points for comparison.

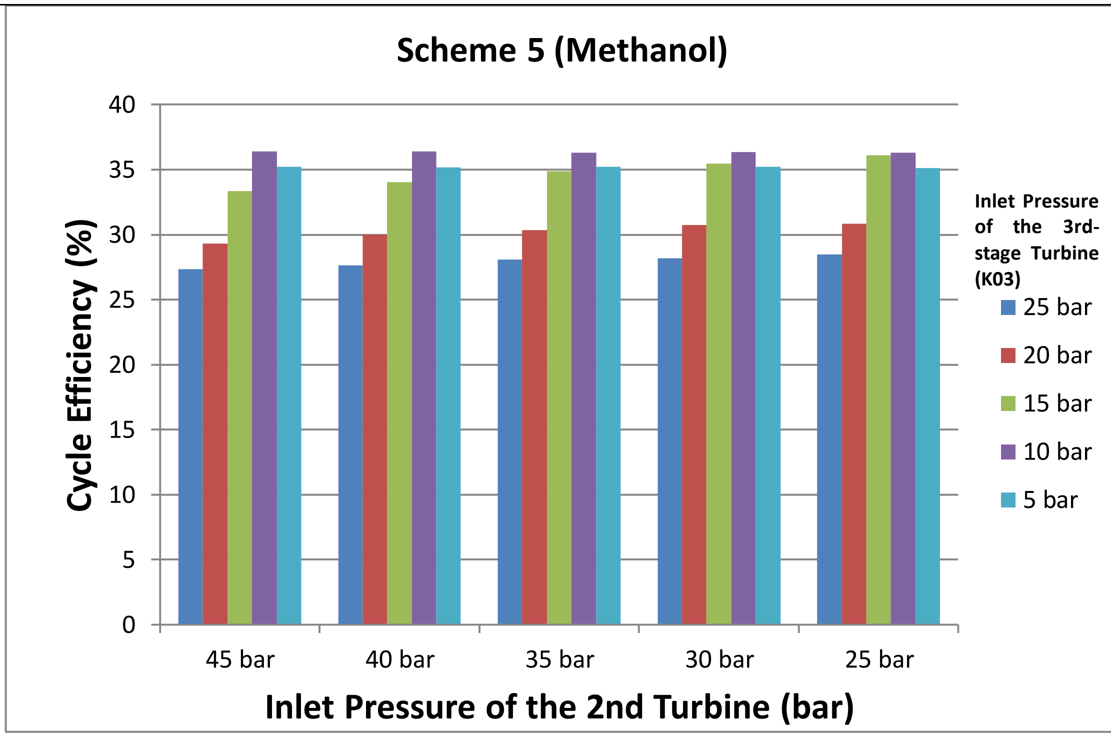

Scheme 1 (Wet Fluids)
Cycle Efficiency (%)
20
19
18
17
16
15
14
20
30
40
50
60
70
Turbine Inlet Pressure (bar)
Ethanol
Methanol
Water
A
Scheme 2 (Dry Fluids)
Cycle Efficiency (%)
20
19
18
17
16
15
14
13
12
11
10
20
30
40
50
Turbine Inlet Pressure (bar)
Benzene
Cyclohexane
n-heptane
n-hexane
B


Scheme 3 (Methanol and Ethanol)
Cycle Efficiency (%)
Turbine Inlet Pressure (bar)
Ethanol
Methanol
28
27
26
25
24
23
22
21
20
20
30
40
50
60
70

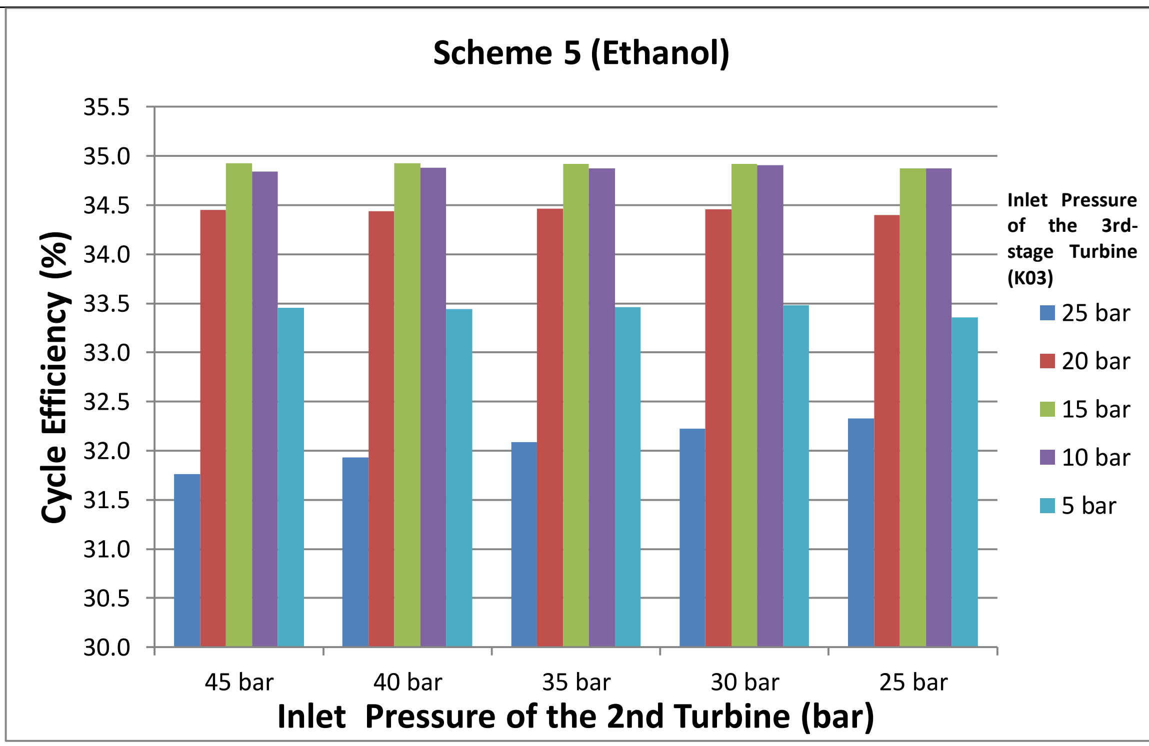
C


Scheme 4 (Water)
Cycle Efficiency (%)
Inlet Pressure of the 2nd Turbine (bar)
Inlet Pressure of the 3rd-stage Turbine (K03)
30 bar
25 bar
20 bar
15 bar
10 bar
20.4
20.3
20.2
20.1
20.0
19.9
19.8
50.3 bar
50 bar
45 bar
40 bar
35 bar
30 bar
25 bar
20 bar

D



E



F

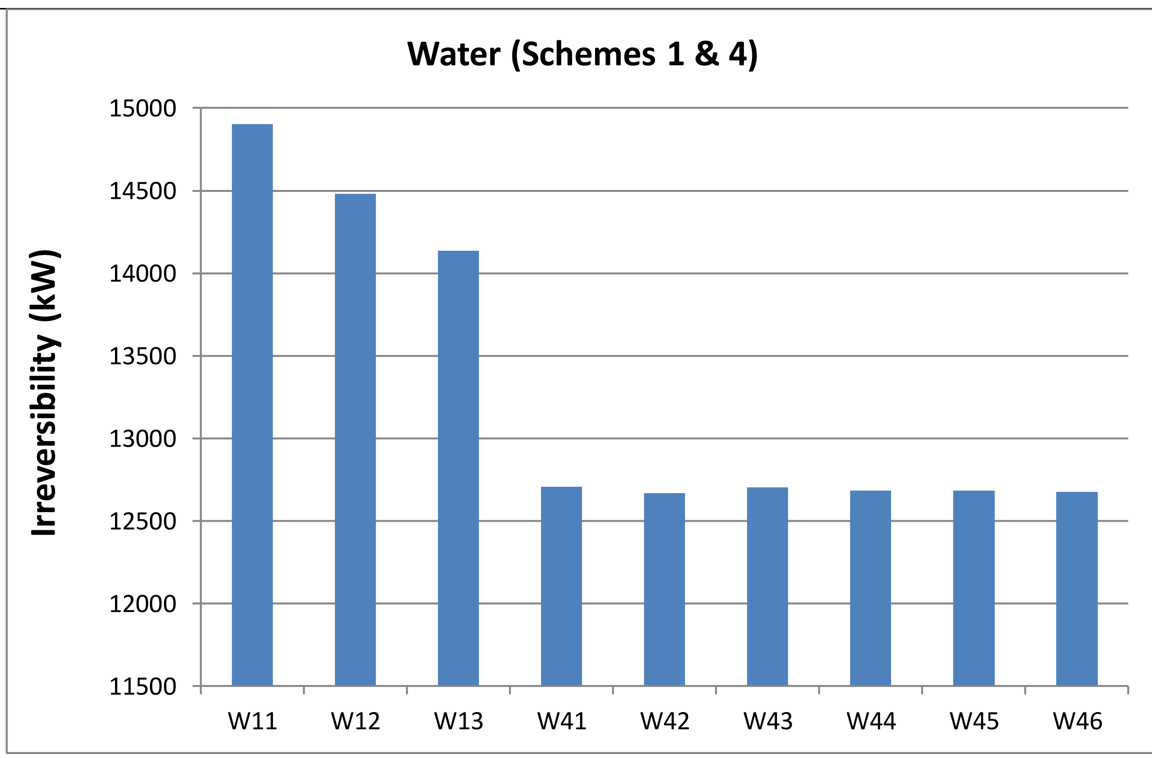
Fig.4: Cycle efficiency of the Rankine cycles of schemes (1-5).

Table 3: Selected turbine inlet pressure points for the comparison between methanol and water.

Water							Methanol			
Scheme 1			Scheme 4				Scheme 5			
Symbol	**Inlet Pressure of the Turbine**	**Efficiency**	**Symbol**	**Inlet Pressure of the 2nd-stage Turbine (k02)**	**Inlet Pressure of the 3rd-stage Turbine (k03)**	**Efficiency**	**Symbol**	**Inlet Pressure of the 2nd-stage Turbine (k02)**	**Inlet Pressure of the 3rd-stage Turbine (k03)**	**Efficiency**
W11	25 bar	17.44%	**W41**	35 bar	15 bar	20.27%	**M51**	30 bar	15 bar	35.68%
W12	30 bar	18.39%	**W42**	35 bar	10 bar	20.28%	**M52**	25 bar	15 bar	36.16%
W13	35 bar	19.18%	**W43**	30 bar	15 bar	20.29%	**M53**	45 bar	10 bar	36.41%
			W44	30 bar	10 bar	20.3%	**M54**	40 bar	10 bar	36.42%
			W45	25 bar	15 bar	20.27%	**M55**	35 bar	10 bar	36.41%
			W46	25 bar	10 bar	20.28%	**M56**	30 bar	10 bar	36.38%
							M57	25 bar	10 bar	36.31%

5.2 Irreversibility

Figures 5(A&B) show the cycle irreversibility of the selected points of water and methanol. The irreversibility of methanol is nearly half that of water as the power output resulted from methanol in nearly the double of that resulted from water. The values of irreversibility of points M53-M56 are very close as the net power output values of these points are very close. The same words can be said for water for points W41-W46. The main cause of irreversibility is the condenser and as in scheme 5, the methanol stream enters the condenser partially condensed, which makes the irreversibility low. Thus, from the net power output, efficiency and irreversibility points of view (thermodynamic point of view) methanol is better than water.



A

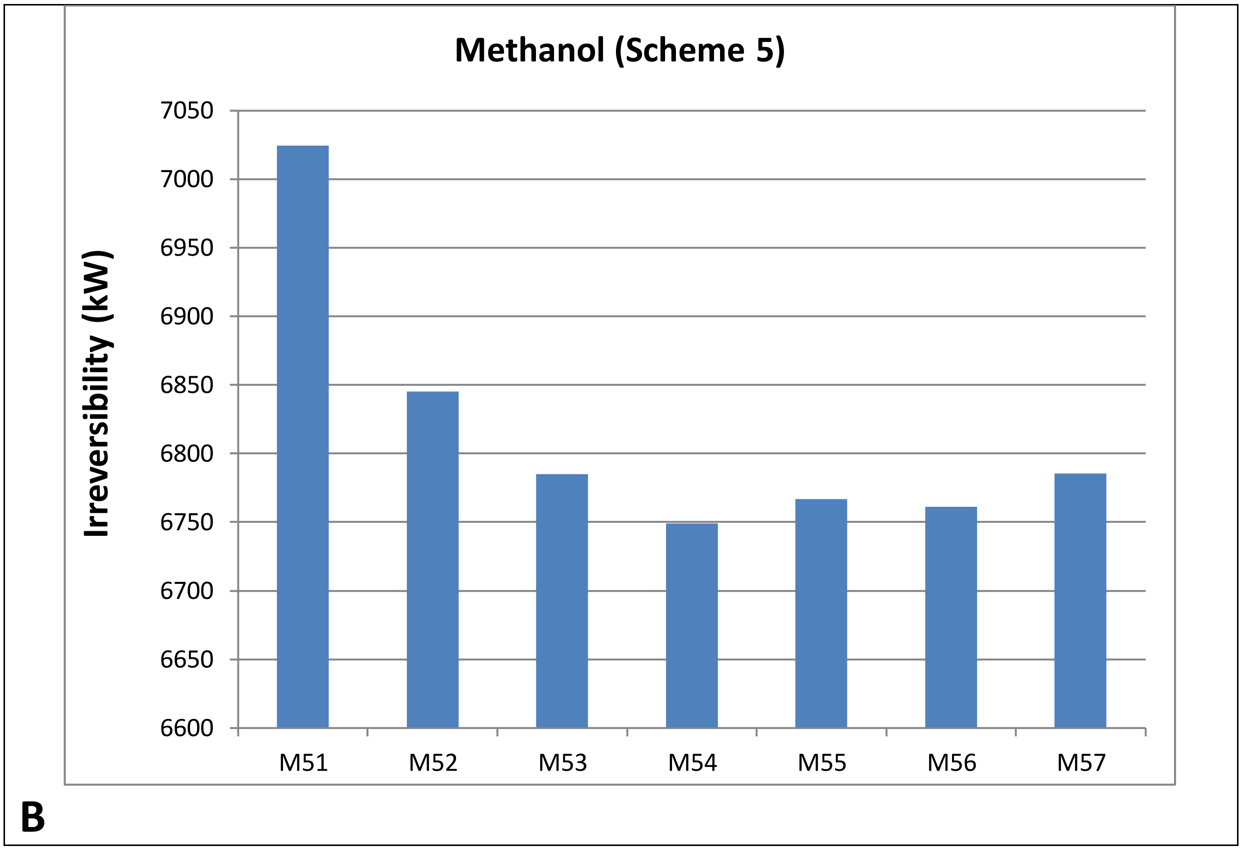


Fig. 5: Cycle irreversibility of the investigated points of water and methanol.

5.3 Capital Cost and Profitability Analysis

Total project capital cost has been estimated using Aspen Process Economic Analyzer V8.8. Figures 6(A&B) show the estimated values of capital cost of the investigated water and methanol points. Generally, the capital cost of methanol is greater than that of water due to the excess equipments used, such as the regeneration heat exchangers and the pumps, in addition to the high flow rates of methanol, which enlarges the size of equipment and in turn the cost. It is also shown that the highest capital cost values are those of the same points at which the highest net power output values are obtained and vice versa.

Figures 7(A&B) show the rate of return (ROR) of the investigated water and methanol points that are calculated at electricity price of 0.1US$/kWh. It worth noting that the breakeven for the electricity price for investigated methanol points is about 0.021US$/kWh, while for water it is in the range of 0.083-0.467 US$/kWh. As shown, the rate of return values of methanol is always greater than that of water by about 50%. This is attributed to the great profit of methanol due to the high net power output values. Also, it is noticed that the difference between the ROR results in not great, especially for methanol, due to the effect of the profit values (based on the calculated net power output values), which are nearly close. The highest ROR value for methanol is that of M52 and M57 and for water is that of W12 and W13. Thus, the net present value is calculated at these points so as to complete the comparison between methanol and water.

Figures 8(A&B) show the net present value (NPV) of water and methanol, respectively, calculated at points W12, W13, M52, and M57. It is found that at both points M52 and M57, the payback period is about4 years only, but with different present valued of 1.94 Million US $ for M52 and 2.13 Million US $ for M57. For water, the payback period is 7 years with present value 0.976 Million US $ for W12 and 1.36 Million US $ for W13. It is clear that from the economic point of view methanol is better than water.

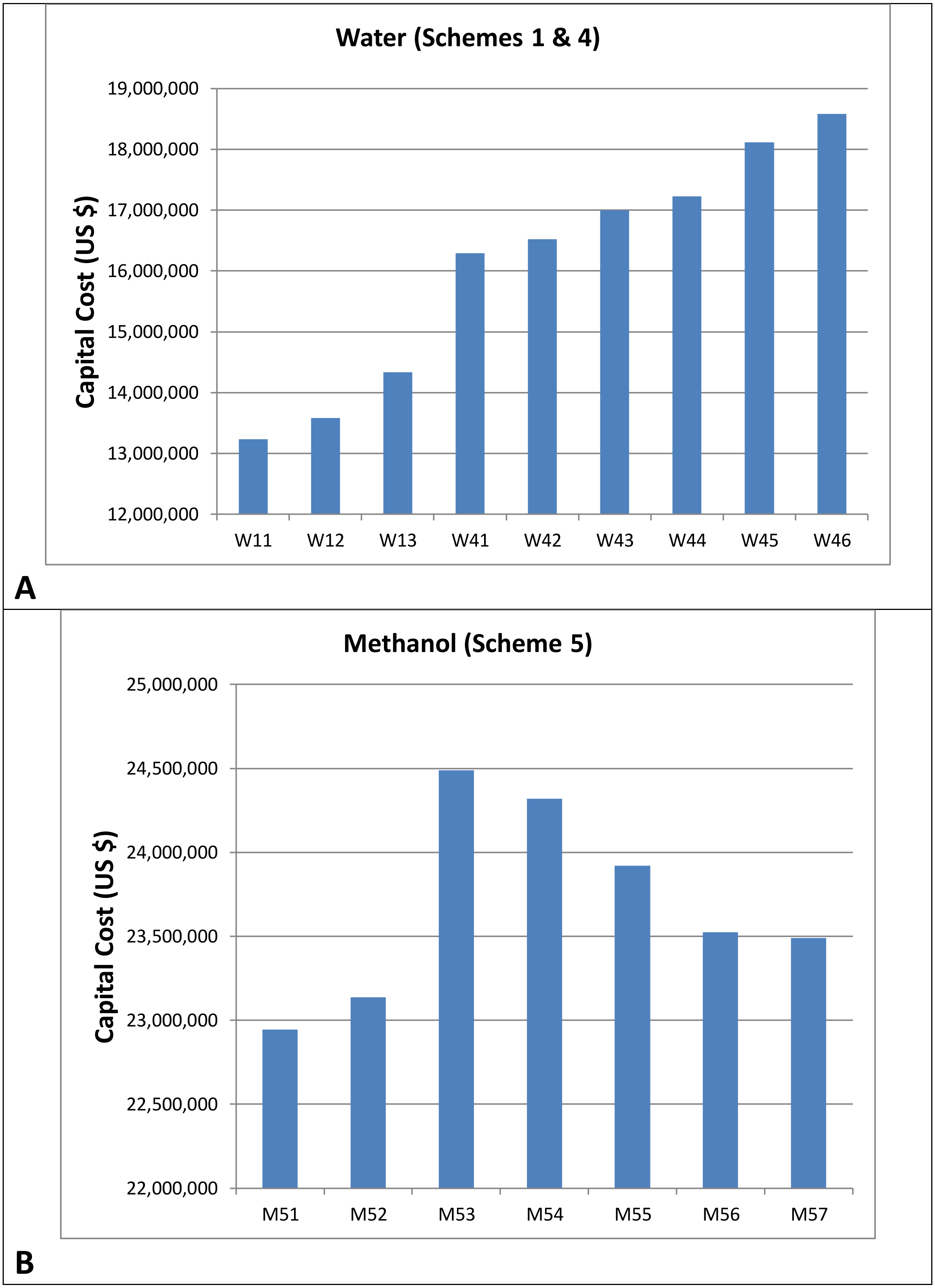


Fig. 6: Total project capital cost of the investigated points of water and methanol.

5 Conclusions

Investigations of the Rankine cycle for power generation using waste heat from a cement plant in Egypt were performed. Six organic fluids and water were selected to be employed as working fluids in five suggested Rankine cycle schemes built and simulated using Aspen HYSYS v8.8. The suggested schemes were made to suit the types of the working fluids used in the investigations: dry or wet fluids.

Working fluids were compared based on the efficiency to select the best organic fluid to be compared with water. The basic results showed that efficiencies of wet fluids in scheme 1 were 13.93-19.85%, for dry fluids in scheme 2 were 12-19.6%, for ethanol and methanol in scheme 3 were 20.78-27%, for water in scheme 4 were 19.84-20.29%, and for ethanol and methanol in scheme 5 were 27.48-36.42%. Thus, seven methanol points in scheme 5 were selected with nine points of water in schemes 1 and 4 to perform comparison between them based on irreversibility and profitability analysis. The comparisons showed that generally, methanol resulted in net power output and cycle efficiency nearly double that of water. Also, the irreversibility from methanol was nearly half of that of water.

Furthermore, the profitability analysis showed that rate of return using methanol was 32.8-34.5%, while that of water was 18.7-22.5%. Finally, methanol in scheme 5 at values of inlet pressure of the 2nd- and 3rd-stage turbine of 25 and 10 bars, respectively (point M52) resulted in the lowest payback period of 4 years with high net present value, while the lowest payback period resulted while using water at turbine inlet pressure 35 bars in scheme 1 (Point W13) and was 7years. Hence, the organic fluid, methanol, presented a better choice as a working fluid to be employed in the Rankine cycle for waste heat recovery in the cement industry.

Notations

a	Discount Rate
B	Benefit
C	Cost
Ex	Exergy [kW]
h	Mass specific enthalpy [kJ/kg]
i	The interest rate
I	Irreversibility [kW]
mo	Mass flow rate [kg/s]
p	The period
P	Power [kW]
Q	Specific Heat [kJ/kg]
Qo	Heat transfer rate [kW]
Sgen	Entropy generation rate [kW/K]
T	Temperature [K]
T0	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
C01	Condenser duty
Cr	Critical
E_i	The exhaust gas in
E_o	The exhaust gas out
H	Heat source
in	Incoming
J	j^{th} component of the cycle
out	Outgoing
P01	Pump (P01)
P02	Pump (P02)
P03	Pump (P03)
K01	Turbo expander (K01)
K02	Turbo expander (K02)
K03	Turbo expander (K03)
th	Thermal
tot	Total
a	Discount Rate
B	Benefit
C	Cost
Ex	Exergy [kW]
h	Mass specific enthalpy [kJ/kg]
i	The interest rate
I	Irreversibility [kW]
m^o	Mass flow rate [kg/s]
p	The period
P	Power [kW]
Q	Specific Heat [kJ/kg]
Q^o	Heat transfer rate [kW]
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
C01	Condenser duty
Cr	Critical
E_i	The exhaust gas in
E_o	The exhaust gas out
H	Heat source
in	Incoming
J	j^{th} component of the cycle
out	Outgoing
P01	Pump (P01)
P02	Pump (P02)
P03	Pump (P03)
K01	Turbo expander (K01)
K02	Turbo expander (K02)
K03	Turbo expander (K03)
th	Thermal
tot	Total

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