

ABSTARCT

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Thermodynamic analysis of Kalina Cycle

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This very thesis report based on the analysis of Kalina power Cycle, a constructive advance in the classical Rankine Cycle. The main objective of this course is to discuss the energy and exergy analysis that the Kalina Power Cycle showcases especially when employed with a low temperature heat source. An advanced equation solving software has been used for simulation, which is known as Engineering Equation Solver (EES). The main aim of this cycle is to show the use to the renewable energy sources such as sun and earth's thermal energy to produce high grade energy. For this the temperature of heat source is taken as a constant value which can be output from a solar collector or a parabolic collector or some underground heat source. The report explores the thermodynamic properties of the working fluid mixture and discusses the impact of varying concentration ratios on cycle performance. Additionally, the study delves into parametric study and the effects of the parameters on the performance of the cycle.

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LIST OF SYMBOLS

Symbols

T	temperature (K)
P	pressure (kPa)
m	mass flow rate (kg/s)
Q	heat transfer rate (kW)
X	exergy destroyed (kW)
x	mass fraction NH ₃ (kg/kg)
I	Irreversibility (kW)
s	specific entropy (kJ/K)
h	specific enthalpy (kJ)
W	work transfer (kW)
η_I	first law efficiency
η_{II}	second law efficiency

Subscripts

o	Reference or dead state
tur	turbine
pum	pump
vg	vapor generator
reg	regenerator
abs	absorber
con	condensor
sep	separator

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1. INTRODUCTION

1.1) GENERAL INTRODUCTION

With the rise in demand of energy and depleting natural resources such as coal and petroleum, it has become necessary to find other means to produce energy through renewable resources and to increase the efficiency of current processes.

This project focuses on one such cycle i.e., Kalina cycle which can be used as a stand-alone process as well to increase the efficiency of current steam and gas turbines by replacing the Rankine cycle as bottoming cycle.

Kalina cycle was created by Dr. Alexander Kalina, a Russian scientist in 1983. Kalina cycle is a modified Rankine cycle which uses binary fluids mixture instead of a single fluid as in Rankine Cycle.

1.2) KALINA CYCLE

The Kalina cycle is one of the better cycles to convert renewable energy sources such as geo-thermal and solar energy to produce electrical energy. This cycle can effectively convert low temperature energy. The Kalina cycle can increase thermal power output efficiencies by up to 30% in suitable installations, and is ideally suited for applications such as coal, oil refineries, steel plants and cement production plants.

The Kalina cycle is a modified Rankine cycle that uses mixture of two different fluids combination as the working fluid: water and ammonia while Rankine cycle uses pure water to produce energy. The Kalina cycle exergy efficiency is 15% higher as compared to the steam power cycle.

The concentration of ammonia provides the property to increase the exergy efficiency and decrease the irreversibility. The ammonia and water have large difference in their boiling points which makes it easy to separate while expanding

or producing power and can easily be compressed as ammonia readily dissolves in water.

1.3) COMPARISON OF RANKINE AND KALINA CYCLE

Kalina cycle is a modified Rankine cycle, since it is a mixture of two fluids, the phase change does not take place at a constant temperature but at a varying temperature, as shown in the temperature(T) – entropy(S) diagram.

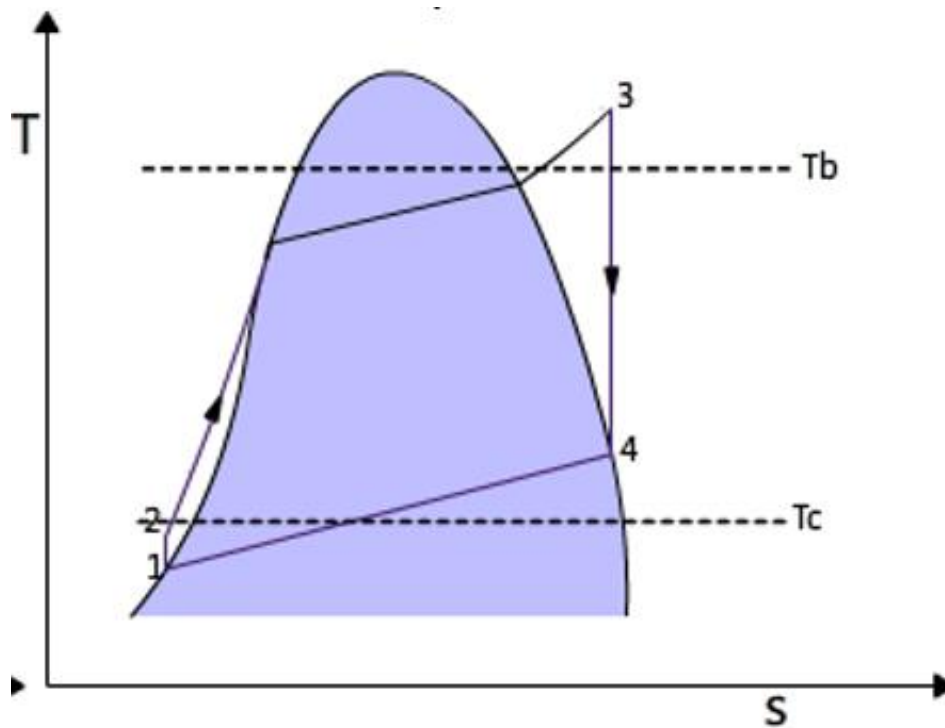


Figure 1-T-S Diagram for simple kalina cycle

1-2: Isentropic Compression (pump)

2-3: Isobaric Heat Supply

3-4: Isentropic Expansion (Turbine)

4-1: Isobaric Heat rejection

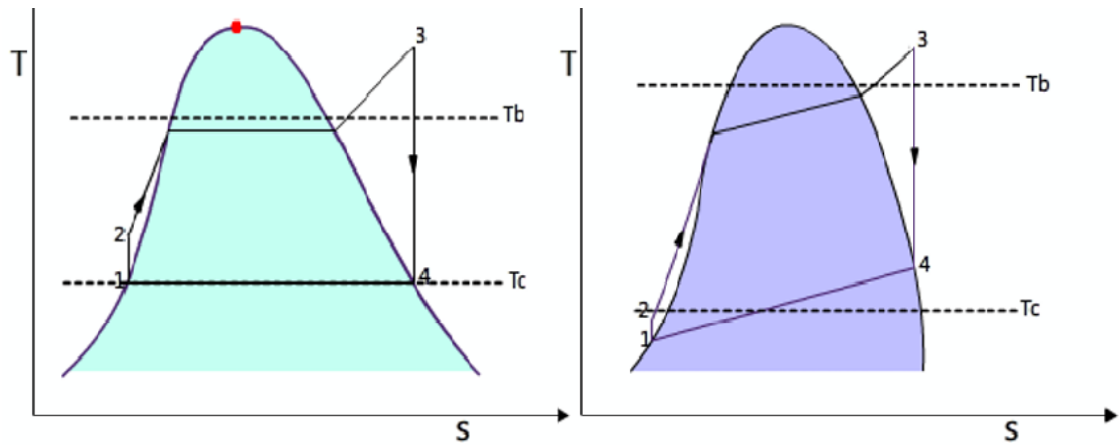


Figure 2-Comparison of simple Rankine and kalina cycle

Since the Kalina cycle operates in greater range than normal Rankine Cycle, its efficiency is also greater. As,

$$\text{Thermal Efficiency} = 1 - (T_c/T_b)$$

The working fluid that is most efficient for using in kalina cycle is aqueous Ammonia ($\text{NH}_3\text{-H}_2\text{O}$)

1.4) OBJECTIVES

- Mathematical modeling of Kalina cycle applying energy and exergy balance approach.
- To develop computer code in Engineering Equation Solver (EES) Package.
- To determine the energy and exergy efficiencies of the Kalina Power cycle.

2. LITERATURE REVIEW

The Kalina cycle, a promising thermodynamic process for power generation, has attracted significant attention in recent years due to its potential for enhanced efficiency and environmental sustainability. The first paper to be published on kalina cycle was back in 1984. Kalina [1] proposed a cycle based on binary mixture of $\text{NH}_3/\text{H}_2\text{O}$ to utilize the waste heat which required a low boiling working fluid. The results demonstrated higher efficiency in comparison to rankine cycle as bottoming cycle.

In recent years, the demand of green energy also pitched kalina cycle as it can produce energy with low temperature heat sources such as geothermal, solar energy using solar collectors [2] and parabolic collectors. Sun et al. [3] has developed a cycle using KCS-11 system and verify the correctness of the model by sampling data from the Kumejima island in Japan. They based their cycle on solar collectors as the heat source and found out the exergy efficiency can be as high as 63.5 %.

The kalina cycle can also be used as combined cooling and power cycle Ghaebi et al. [4] has proposed on such cycle to use LNG (Liquified natural gas) as heat sink and thus easily transported to large distances. Energy, exergy and exergoeconomic analysis of the proposed system was conducted using EES package and first and second law efficiencies were calculated for the combined cooling and power cycle. Further effect of some key parameters was observed on the performance of the system. Rostamzadeh et al. [5] also proposed one combined cooling and power cycle (CCP) based on ammonia water binary mixture using ejector refrigeration cycle. The proposed cycle is cooling dominant. They obtained thermal efficiency as 17.6 %. They also found out the effect of key parameters such as turbine expansion ratio (TER), pinch point temperatures on the system.

Nag and Gupta [6] analysed Kalina cycle to reduce the exergy loss of the bottoming cycle in power plants. They found out that nearly 50% exergy loss takes place in heat recovery steam generator, and the cycle efficiency to be decreasing with increase in ammonia mass fraction and efficiency to increase with increase in turbine and separator temperature.

Ogriseck [7] integrated kalina cycle in a combined heat and power plant to utilize the heat of the flue gases exhausted in the power plant. He found the efficiency of the

combined cycle in between 12.3% and 17.1% depending on the cooling water temperature. He also found out that the operation can be performed using conventional steam turbines. The cycle become is cheaper than other cycles such as organic rankine cycles as ammonia is easily available and inexpensive and its use in industrial process is already proven.

Rodríguez et al. [8] has done exergetic and economic comparison of organic rankine cycle (ORC) and kalina cycle for enhanced geothermal system in Brazil. For this they evaluated 15 different working fluids for ORC and three different compositions for ammonia-water mixture in kalina cycle. For simulation they used Aspen-HYSYS software package. They found out for 100 °C the kalina cycle offers 18% more power output than ORC and requires 37% less mass flow rate of working fluid. The best performance was obtained for R-290 fluid in case of ORC and composition of 84% ammonia mass fraction in case of kalina cycle.

Lin et al. [9] also compared kalina cycle and organic cycle thermodynamically in their research they found the maximum power output in kalina cycle for temperature range 140-200 °C. Bombarda et al. [10] in their paper have also compared kalina and ORC to recover heat energy exhausted from two diesel engines each with 8900 KW capacity. They concluded that kalina cycle requires very high maximum pressure to obtain high thermodynamic performance, the turbine is a critical component which must be either multistage or rotate at very high speed. They also found that the kalina components faced corrosion problems which must be considered for economic aspects. Modi and Haglind in their paper have done optimisation and analysis of four different kalina layouts and compared them thermodynamically. They have optimised the kalina cycle for high temperatures around 500 °C. The components are arranged according to different kalina systems for high temperatures and are labelled in that manner. They obtained maximum efficiency for kalina system KC1234 layout to be 31.47% at turbine inlet pressure 140 bar and mass fraction of ammonia as 0.8. They optimised the cycles for fixed generator power rating. The lowest efficiency they obtained was for the system KC234 which was 27.35%.

Shankar and Srinivas [12] have done their research for the cooling cogeneration aspect of kalina cycle. For this they have used ammonia-water and LiBr-water mixture as working fluid. The COP of the proposed cycle is in the range of 0.46-0.54 for the aqueous ammonia cycle and 0.74-0.82 for the LiBr-water cycle. They found that from the power generation angle LiBr-water cycle shows better performance than ammonia-

water cycle while ammonia-water shows better performance in terms of solar collector area and cooling temperature.

The enthalpy, entropy and specific heat values for the ammonia mixture are obtained from the work of Ziegler and Trepp [13]. They have obtained the co-relation of equilibrium properties of ammonia-water mixtures up to 500 K temperature range and pressure range up to 50 bar.

3. THERMODYNAMIC ANALYSIS

3.1) CYCLE DESCRIPTION AND ASSUMPTIONS

The Kalina cycle consists of several key components: a vapor generator, a separator, a turbine, a regenerator, a throttling valve, a condenser, a mixer, and a pump. The cycle operates as follows:

1. A two-phase mixture (state 1) in the vapor generator is directed to the separator. In the separator, it separates into a rich ammonia-water mixture in the form of saturated vapor (state 2) and a lean ammonia-water mixture as saturated liquid (state 3).
2. The saturated vapor (state 2) is expanded through a turbine to a lower pressure (state 6) to produce power output. Subsequently, it enters the mixer.
3. Simultaneously, the saturated liquid (state 3) flows into the regenerator, where it heats the pressurized mixture (state 4). It is then throttled back to the mixer (state 10) through a throttling valve (TV).
4. The two-phase flows combine in the mixer, and the mixed flow enters the condenser (state 7), releasing heat to the environment during states 13 and 14.
5. The saturated flow (state 8) is pumped to become compressed liquid (state 9) and then reheated by the regenerator before being returned to the generator (state 5), thus completing the Kalina cycle process.

Several assumptions were made during the thermodynamic analysis of the Kalina cycle in this project:

1. Steady-state operation of the cycle is assumed.
2. It is assumed that the working fluid at the outlet of the condenser is in the form of saturated liquid.
3. The working fluid at the inlet of the turbine is considered to be saturated vapor.
4. The throttling process is assumed to be isenthalpic.
5. The separator is assumed to completely separate the liquid and vapor phases.
6. Pressure losses and heat losses in pipes are neglected.
7. All the devices in the cycle are considered adiabatic.
8. Changes in kinetic energy and potential energy in the devices are neglected.

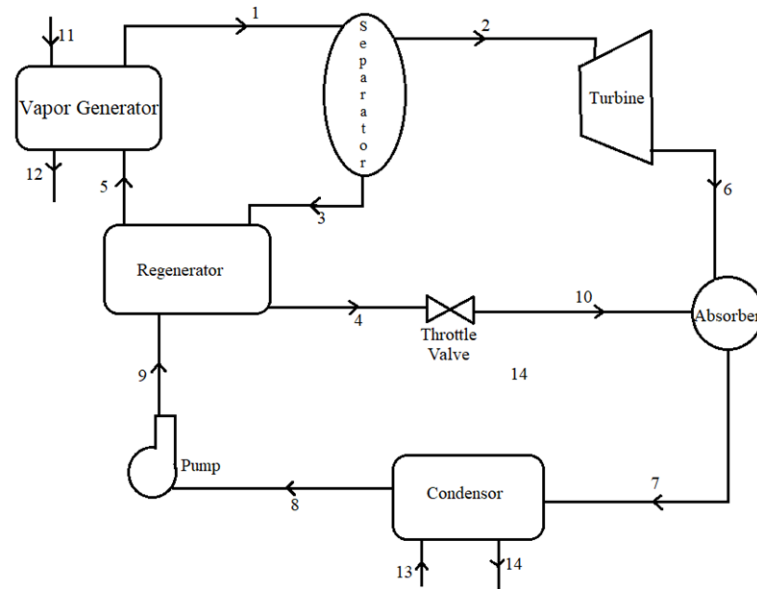


Figure 3-Schematic Diagram of Kalina Cycle

3.2) MATHEMATICAL MODEL

Mass Analysis and Energy analysis-

The mass balance equation for each component of the power cycle follows from the below written equation:

$$\sum m_{in} = \sum m_{out}$$

The energy balance equation can be written as, neglecting kinetic and potential energy:

$$\sum (mh)_{in} - \sum (mh)_{out} + Q - W = 0$$

For vapor generator –

$$m_{11} = m_{12}, m_5 = m_1$$

$$Q_{vg} = m_5(h_1 - h_5) = m_{11}(h_{11} - h_{12})$$

For separator –

$$m_1 = m_2 + m_3$$

$$m_1 h_1 = m_2 h_2 + m_3 h_3$$

For turbine-

$$m_2 = m_6$$

$$W_{tur} = m_2(h_2 - h_6)$$

For mixer-

$$m_7 = m_6 + m_{10}$$

$$m_7 h_7 = m_6 h_6 + m_{10} h_{10}$$

For throttle valve-

$$m_4 = m_5$$

$$h_4 = h_5$$

For condenser-

$$m_8 = m_7; m_{13} = m_{14}$$

$$Q_{\text{con}} = m_7(h_7 - h_8) = m_{13}(h_{14} - h_{13})$$

For pump-

$$m_9 = m_8$$

$$W_{\text{pum}} = m_8(h_9 - h_8)$$

For regenerator

$$m_4 = m_3; m_5 = m_9$$

$$Q_{\text{reg}} = m_4(h_3 - h_4) = m_9(h_5 - h_9)$$

Exergy Analysis-

Exergy destroyed for a system can be written as

$$I = X_{\text{in}} - X_{\text{out}} - W_{\text{cv}} + \left(1 - \frac{T_0}{T_1}\right) Q_1$$

where

$$X_i = m_i(h_i - h_0 - T_0(s_i - s_0)) \quad ; \text{ here } 0 \text{ subscript represents the reference state property.}$$

For vapour generator -

$$I_{\text{vg}} = (X_5 - X_1) - (X_{11} - X_{12})$$

For separator -

$$I_{\text{sep}} = X_1 - (X_2 + X_3)$$

For turbine-

$$I_{\text{tur}} = (X_2 - X_6) - W_{\text{tur}}$$

For mixer-

$$I_{\text{mix}} = X_6 + X_{10} - X_7$$

For throttle valve-

$$I_{\text{tv}} = X_4 - X_5$$

For condenser-

$$I_{\text{con}} = (X_7 - X_8) + (X_{13} - X_{14})$$

For pump-

$$I_{\text{pum}} = (X_8 - X_9) + W_{\text{pum}}$$

For regenerator

$$I_{\text{reg}} = (X_3 + X_9) - (X_5 + X_4)$$

First law efficiency = W_{net}/Q_s

where $W_{net} = W_{tur} - W_{pum}$; $Q_s = Q_{vg}$

Second Law efficiency = $(X_{in} - \sum I_o) / X_{in}$

Where $\sum I$ is the sum of exergy loss and X_{in} the exergy input at vapour generator

3.3) RESULT AND DISCUSSION

To find the performance of the proposed cycle, the code has been developed in Engineering Equation Solver (EES) based on iterative methods using the mathematical model shown previously. The input parameters for the model are given below-

Parameters	Values	Unit
Reference Temperature, T_o	303	K
Reference Pressure, P_o	101.3	kPa
Temperature of heat source, T_{hs}	393	K
Temperature of cooling water, T_{cw}	293	K
Pressure of vapor generator, P_{vg}	3500	kPa
TER	5	-
Mass flow rate of heat source	35	kg/s
TTD _{vg}	5	K
PPTD _{reg}	5	K
PPTD _{vg}	5	K
Mass fraction of ammonia, (x)	0.75	%
Isentropic efficiency of turbine, (η_{tur})	0.85	%
Isentropic efficiency of pump, (η_{pump})	0.85	%

Table 1- Mean Values of input parameters

Based on input parameters thermodynamic properties at different states are found using EES software, these are presented in the table below-

And different results such as anergy efficiency, exergy efficiency, net work output, work at turbine, work by pump, Irreversibility of components are calculated.

State	T (K)	h (kJ/kg)	m (kg/s)	s (kJ/kg-K)	x (%)	P(kPa)
1	383.0	787.968	5.705	2.651	0.75	3500
2	383.0	1440.301	2.491	4.272	0.9819	3500
3	383.0	282.294	3.214	1.394	0.5703	3500
4	319.2	-25.074	3.214	0.5235	0.5703	3500
5	360.3	196.866	5.705	1.199	0.75	3500
6	326.2	1306.950	2.491	4.344	0.9819	1167
7	329.6	556.589	5.705	2.196	0.75	1167
8	313.7	19.811	5.705	0.5266	0.75	1167
9	314.2	23.718	5.705	0.5285	0.75	3500
10	319.2	-25.074	3.214	0.5269	0.5703	1167
11	393.0	505.466	35.00	1.523	-	3500
12	370.3	409.527	35.00	1.272	-	3500
13	293.0	83.301	30.50	0.294	-	101.3
14	317.2	184.359	30.50	0.6255	-	101.3

Table 2- Thermodynamic properties for proposed cycle

From the properties table-

Energy efficiency (η_I) is calculated to be 9.19 %.

Exergy efficiency (η_{II}) is calculated to be 50.76%.

Work output at Turbine (W_{tur}) is calculated to be 309.93 kW.

Net work output (W_{net}) is calculated to be 332.22 kW.

The high energy and exergy efficiency is because of the varying boiling point of ammonia mixture. The loss of exergy in different components is shown next, which shows that the maximum exergy is destroyed in vapor generator followed by regenerator and condenser.

Component	Irreversibility	% Irreversibility
Vapor Generator	5187	89.31%
Separator	3.62E-13	0.00%
Turbine	54.31	0.95%
Absorber	4.129	0.07%
Regenerator	311.8	6.43%
Throttle Valve	3.285	0.06%
Pump	3.225	0.11%
Condenser	7.22	3.09%
	5570.969	100.00%

Table 3- Irreversibility of all components in cycle.

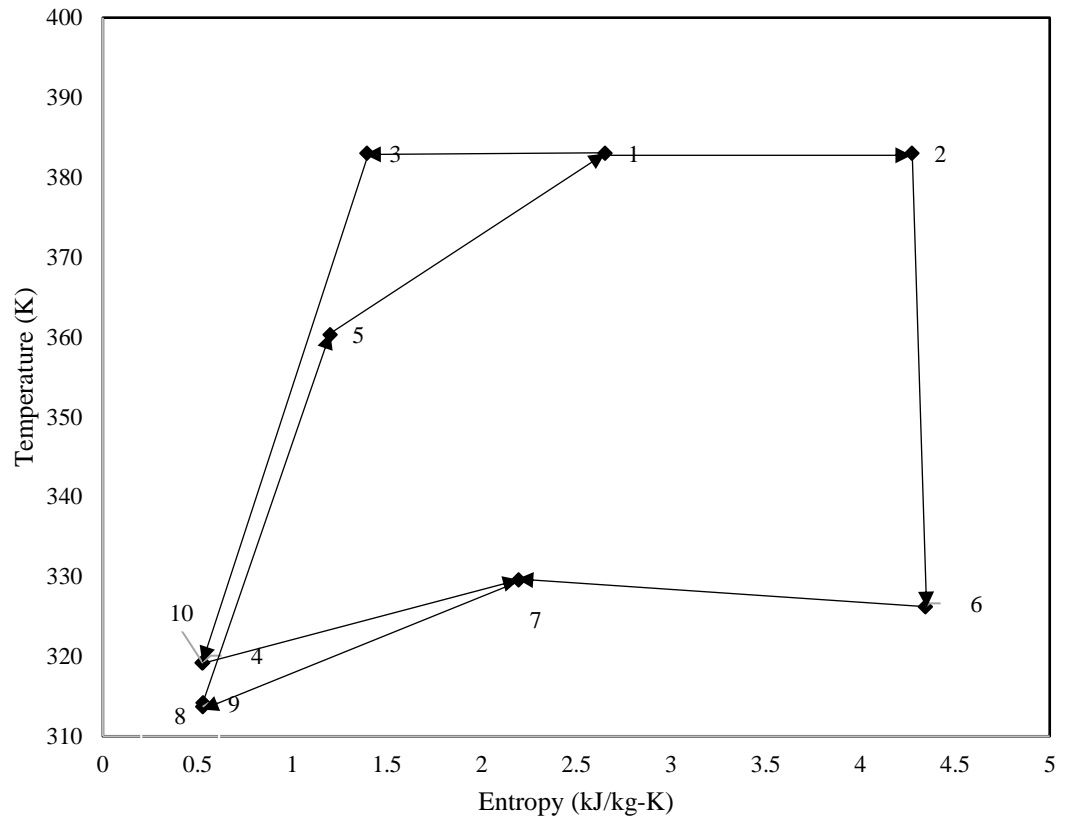


Figure 4- T-S diagram for the proposed cycle

4. PARAMETRIC STUDY

4.1) Effect of Ammonia mass fraction on Cycle

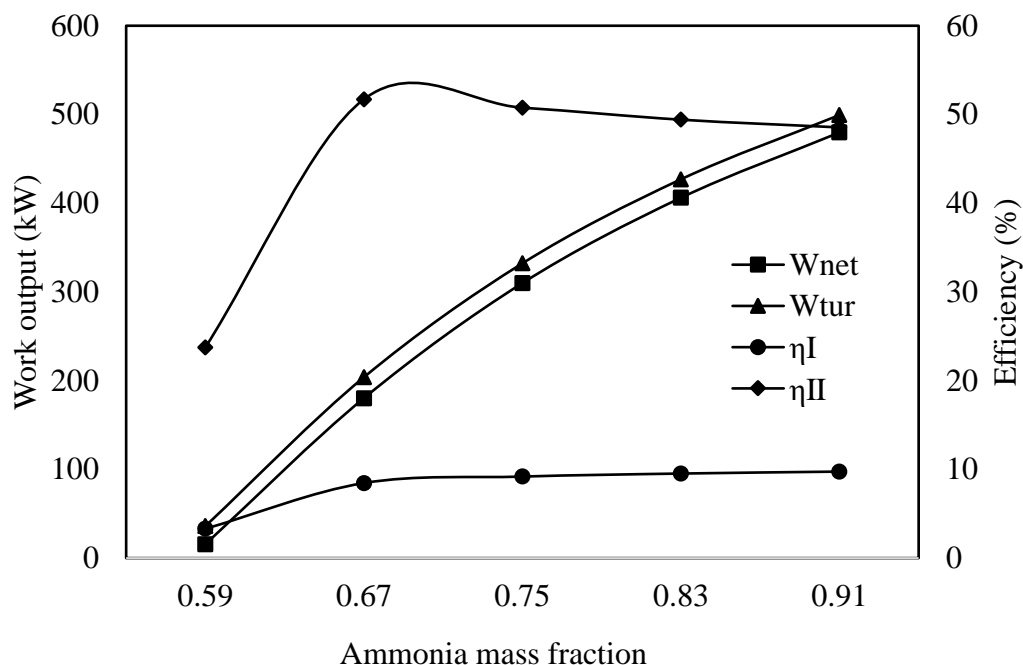


Figure 5- Effect of Ammonia mass fraction on Work output and efficiency

On increasing the ammonia concentration (x), second law efficiency first increases then decreases with increase in Ammonia mass fraction, while first law efficiency increases throughout. The reason may be that for low values of x , the turbine irreversibility is high due to more entropy generation, resulting in low second law efficiency. For high values of x , the VG irreversibility is high, and it decreases with the increase of x .

Thus, the optimum value of x (Ammonia mass fraction at condenser output) can be calculated.

4.2) Effect of Turbine Expansion ratio on cycle

As the TER increases the working fluid expands more thus the increase in energy and exergy efficiency. TER value should be optimised further so that the capital required for the turbine can be minimised.

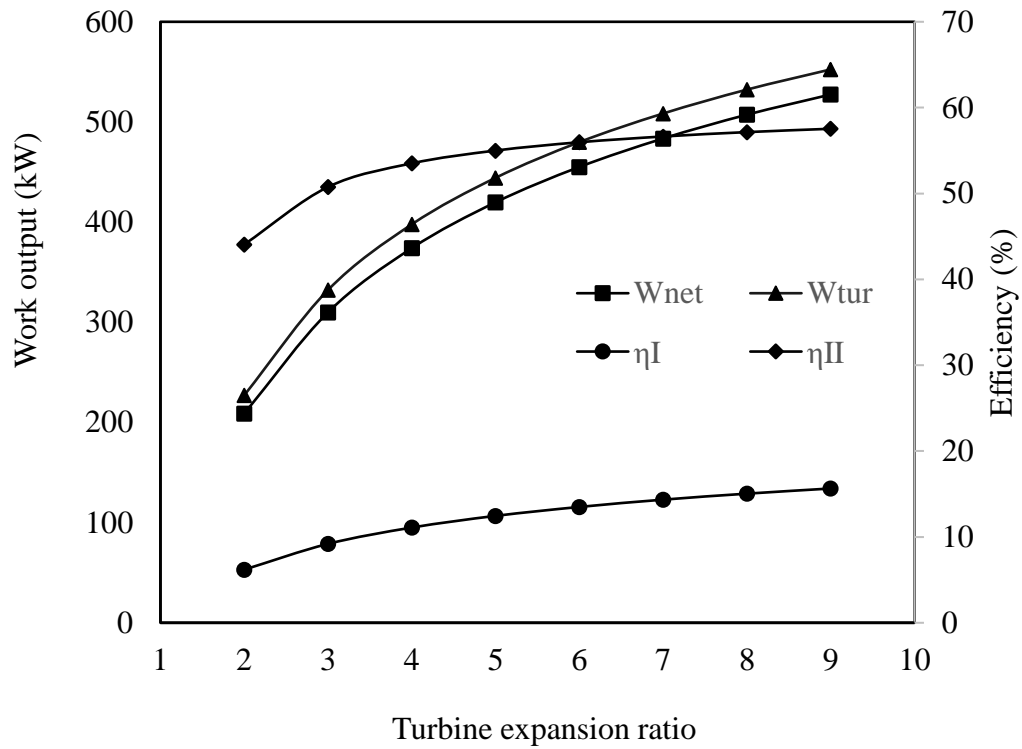


Figure 6- Effect of Turbine Expansion ratio on Work output and efficiency

4.3) Effect of Vapour Generator Pressure on cycle

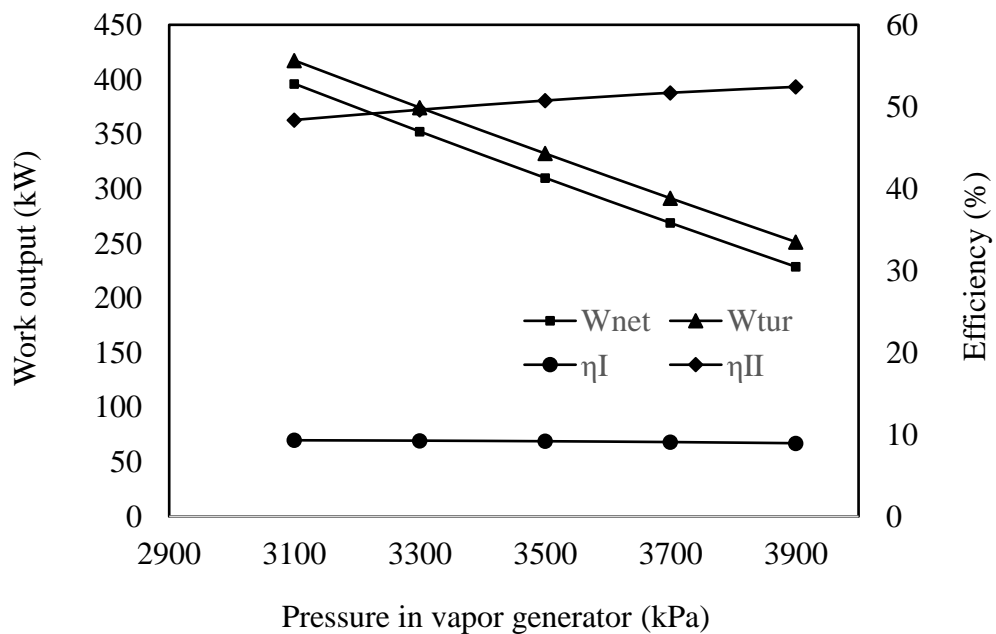


Figure 7- Effect of Pressure in Vapour Generator on Work output and Efficiency

As the pressure in vapor generator increases energy efficiency almost remains constant while exergy efficiency increases. The net output decreases as the turbine expansion ratio is kept constant while the Pvg is increased so net loss in power output.

4.4) Effect of Temperature of heat source on cycle

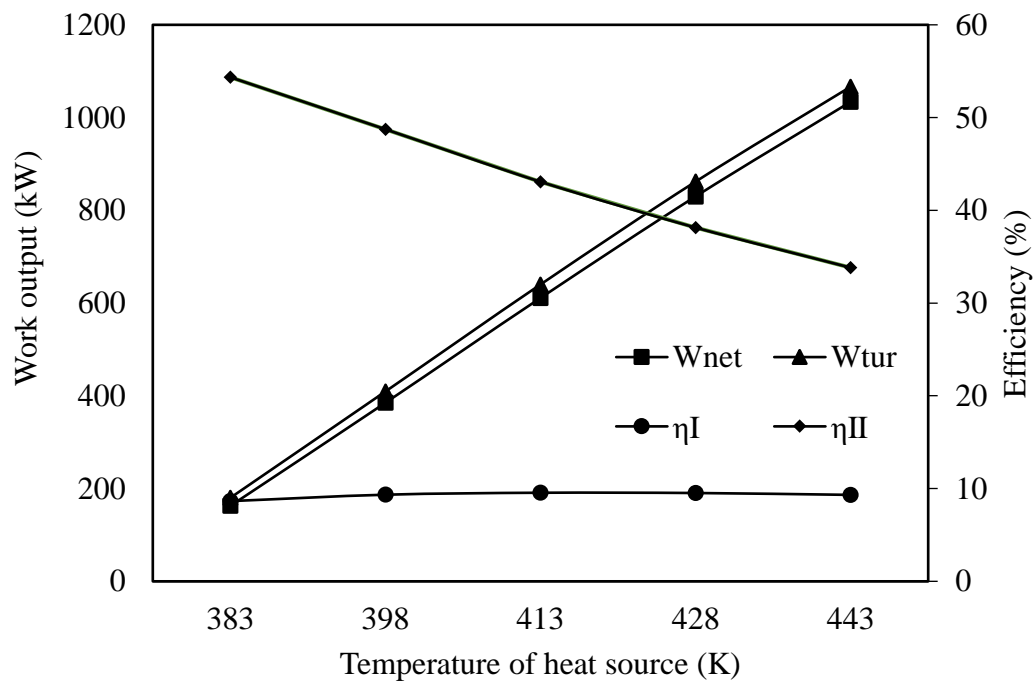


Figure 8- Effect of Temperature of heat source on Work output and Efficiency

As the temperature of heat source increases the range of temperature in which the cycle works increases which increases the first law efficiency, but the exergy destroyed in vapor generator also increases which shows the reduction in exergy efficiency. This shows how the cycle would react to different heat sources such as parabolic or solar collectors.

4.5) Effect of Pinch point temperature at regenerator

As PPTDreg increases the heating curve and the boiling curve of ammonia-water mixture becomes far away so the heating becomes lesser efficient which is shown by decrease in energy and exergy efficiency.

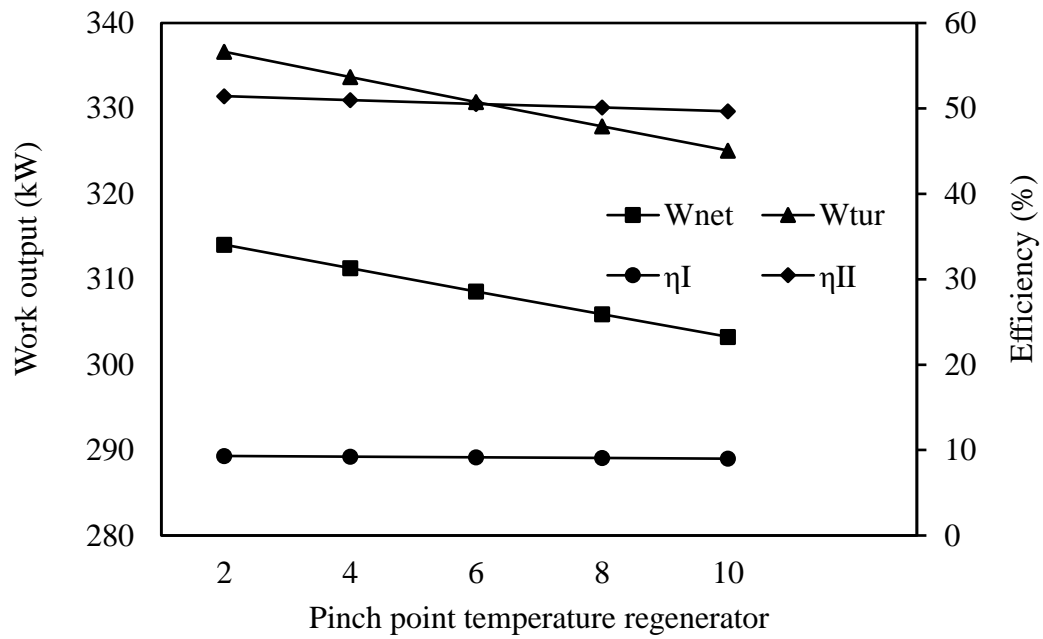


Figure 9- Effect of Pinch point temperature at regenerator on work output and efficiency.

4.6) Effect of Turbine Inlet Pressure on cycle

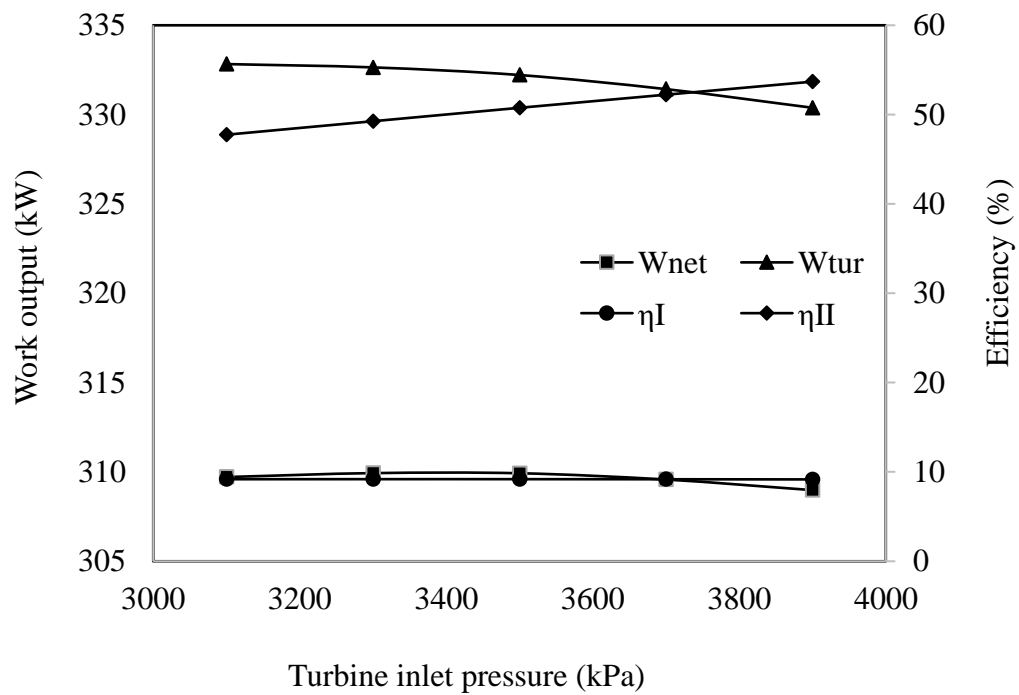


Figure 10- Effect of Turbine Inlet Pressure on Work output and efficiency

As the turbine inlet pressure increases the energy efficiency decreases as the expansion ratio is same so turbine cannot take advantage of high pressures. The work required by pump also increases as the pump must pump to a higher pressure i.e., to the turbine pressure which can be seen by decrease in net work.

4.7) Effect of Pinch point temperature difference at Vapour Generator

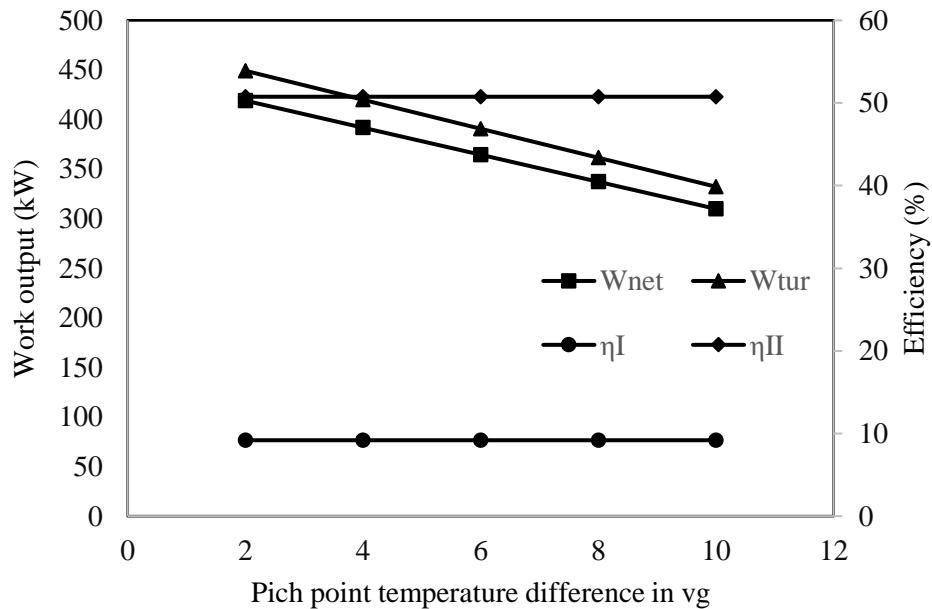


Figure 11- Effect of Pinch point temperature difference at Vapour Generator on Work output and efficiency

As the pinch point temperature at vapor generator increases the heating efficiency decreases thus decreasing the work outputs. The effect on efficiencies is almost negligible.

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