Optimization of Rankine Cycle

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ABSTRACT

Optimization of steam power plants has always been a major concern for industries. Even one percent increase in thermal efficiency of a power plant could save lots of money. A power plant which consumes sea water as a working fluid has been studied which simulated by VisSim software. The ideal steam power plant is called Rankine cycle. In this study, cycle efficiency and optimal placement of FWHs in three regenerative Rankine cycle from one feedwater heater (FWH) up to three feedwater heaters (FWHs) have been investigated in ideal and actual case.

Different parameters that affects cycle efficiency include boiler pressure, condenser pressure, boiler temperature, pump efficiency, turbine efficiency, cooling water temperature and the number of feedwater heaters utilized in a power plant. Based on the acquired data from VisSim software, in actual base one, two and three FWHs, the efficiency was 38.87%, 39.68% and 43.55% respectively and the change from ideal cycle was 10.16% decrease, 10.17% decrease and 2.73% decrease respectively. As the number of FWHs increased, the efficiency also increased and it got closer to ideal cycle.

The parameter which had the highest impact on increasing efficiency was turbine efficiency. By raising turbine efficiency by 5%, the efficiency moved up around 5% for one and two FWHs and 1.11% for three FWHs. After that boiler temperature had the highest impact on efficiency change which was about 3.5% increase in efficiency by rising it up to 20%. By raising boiler pressure by 20% the efficiency raised by around 1.80% and finally lowering condenser pressure by 20%, led to around 1.40% efficiency increase for actual one, two and three FWHs. Another purpose of this study was to find the optimal placement of feedwater heaters in each cycle. In 1 FWH and 2 FWHs regenerative Rankine cycle, the optimal pressure of the FWHs in both ideal and actual cycles are almost the same, however in 3 FWHs regenerative Rankine cycle, optimal pressure of the feedwater heaters in actual cycle is much less than those in ideal cycle for corresponding feedwater heaters.

Keywords: Rankine Cycle, Feedwater Heater, Vissim, Optimization, Power Plant

ÖZ

Buhar santrallerinin optimizasyonu her zaman sanayi için önemli ilgi noktalarından birisi olmuştur. Bir santralin termik verimliliğinde yüzde birlik bir artış bile çok büyük maddi tasarruf sağlayabilmektedir. İdeal buhar santraline Rankine çevrimi denir. Bu çalışmada çevrim verimi, ve besleme suyu ısıtıcılarının optimum yerleştirilmesi için 1, 2 ve 3 ara ısıtıcılı ideal ve gerçek rejeneratif (Rankine) buharlı güç çevrimleri incelendi.

Çevrim verimini etkileyen farklı parametreler kazan basıncı, kazan sıcaklığı, pompa verimi, türbin verimi, soğutma suyu sıcaklığı ve santralda kullanılan besleme suyu ısıtıcılarının (ara ısıtıcılar) sayısını içerir. Çevrim simülasyonu için VisSim yazılımı kullanıldı. Gerçek, rejenerasyonlu bir, iki ve üç ara ısıtıcılı buhar santralleri için simülasyon sonucu elde edilen optimum termal verimlilik sırasıyla % 38.87, % 39.68 ve % 43.55 olarak hesaplandı. Gerçek rejeneratif Rankine çevriminin aynı türbin basıncı ve sıcaklığı ile kondenser basınca sahip olan ideal rejeneratif Rankine çevriminden daha düşük ısıl verimliliği vardır. Bir, iki ve üç ara ısıtıcılı gerçek bir Rankine çevriminin verimliliği aynı çalışma parametreleri olan ideal çevime göre sırası ile % 10.16, % 10.17 ve % 2.73 daha düşüktür. Ara ısıtıcıların sayısı arttıkça, gerçek çevrimin verimliliği artar ve ideal çevrimin verimliliğine yaklaşır.

Gerçek çevrimde, termal verimliliğe en yüksek etkisi olan parametre türbinin verimliliğidir. Türbin verimi referans değerine göre % 5 artırılırsa, bir ve iki ısıtıcılı gerçek çevrimin termal verimliliğinde yaklaşık % 5 iyileşme olur. Türbin verimliliğinde yapılan ayni iyileştirmede ise üç araısıtıcılı çevrimin verimliliğindeki artış % 1.11 dir. İkinci en

yüksek etkiye sahip parametre kazan sıcaklığıdır, kazan sıcaklığında % 20 lik artış çevrimin termal verimliliğini % 3.5 civarında artıdı. Kazan basıncındaki % 20 lik artışın bir, iki ve üç ara ısıtıcılıdaki gerçek rejeneratif Rankine çevrimindeki verimliliğe etkisi yaklaşık % 1.80 dir. Son olarak kondenser basıncındaki % 20 lik azalmanın her üç çevrimin termal verimliliğine olan etkisi % 1.40 iyileşme olarak bulundu. Sunulan verimlilikler optimal verimlilikler olup çevrimin termodinamik optimizasyonu sonucu ara ısıtıcıların en uygun yerleştirilmesi yolu ile elde edildi.

Anahtar Kelimeler: Rankine Çevrimi, Besleme Suyu Isıtıcı, Vissim, Optimizasyon, Güç Santralı

To my family

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I should thank to my parents and my wife’s parents for all their financial and emotional support.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

q ……………………………...……………………………………heat transfer (kJ/kg)

w…………………………………………………………………………...work (kJ/kg)

h…………………………………………………………………….enthalpy (kJ/kg)

$W_{pump,in}$ ……………...…………………………………..…work input to pump (kJ/kg)

ν………..…………………………………………………...specific volume (m^3/kg)

P………………………………………………………………..……...pressure (kPa)

$W_{turbine,out}$ ……………………………………...……..……work output of turbine (kJ/kg)

q_{in} ………………………………………………...……..heat input to the boiler (kJ/kg)

q_{out} ……………………………………..………heat output from the condenser (kJ/kg)

η_{th} ………..…………………..………………………………thermal efficiency of cycle (%)

w_{net} ………………………………………………………………...the net work (kJ/kg)

η_p ……………………………………………..……. isentropic efficiency of pump (%)

η_t …………………………………………………….isentropic efficiency of turbine (%)

h_s ……………………………....…………………….enthalpy in isentropic state (kJ/kg)

h_a ……………………………………………………….enthalpy in actual state (kJ/kg)

w_s …………………………………...…isentropic exit states in pump and turbine (kJ/kg)

w_a ………………………………………….actual exit states in pump and turbine (kJ/kg)

y……………..….fraction of extracted steam from turbine (dimensionless, between 0 to 1)

$\dot{m}$…………..……………………………………….....mass flow rate of steam (kg/s)

T…………………………………………......…………………….temperature (° C)

S…………………………………………………………..…………entropy (kJ/kg.K)

Chapter 1

INTRODUCTION

From 1880s, fossil fuels are used in power plants to generate electricity for industries. Thomas Edison was the first one who opened the first generating station in 1882. Then, different power plants have been constructed all around the world. New technologies are applied to enhance power plant operations such as automation of power plants and the improvements made to have higher efficiencies in power plants and reduce the hazardous emissions (Flynn, 2003).

Rankine cycle is the ideal cycle for vapor power plants (i.e., steam power plants). The Rankine cycle was accepted as a standard for steam power plants. The simple ideal Rankine cycle basically has 4 components; steam generator, turbine, condenser and pump. The actual Rankine cycle used in power generation is more complex compared with simple ideal Rankine cycle. The Rankine cycle has been the most widely used cycle in electricity generation. Therefore, any modification made to improve the cycle thermal efficiency means large savings from energy input (i.e., fuel saving). The aim is to decrease the irreversibilities in order to improve the thermal efficiency.

The aim of this study is to optimize the thermal efficiency of regenerative Rankine cycle and investigate the impact of various parameters on the cycle thermal efficiency. VisSim simulation software is used in the optimization of the Rankine cycle. Both ideal and actual cycles are extensively investigated.

The thesis is organized as follows. Chapter 2 provides the literature review and theory of the Rankine cycle. In chapter 3, the methodology of cycle optimization is explained. The results and discussion are presented in chapter 4. The conclusions are given in chapter 5.

Chapter 2

LITERATURE REVIEW

Optimizing sources of energy regarding to condition of environment and general supply of energy is needed. So, units which provide power become more complex. The owners of power plants are asking for guaranteed and high performance power plants. Increasing fuel prices and environmental influence draw attention to energy issues considerably (Dincer, I., & Al-Muslim, H., 2001).

Overall thermal power plant optimization is a very complex process. Power plant optimization may mean; maximum thermal efficiency, minimum power generation cost, minimum downtime or lowest possible emissions. The owners of the power plants try to be more competitive and seek constantly ways to decrease designing time and costs, planning time and employ computer aided approaches to avoid delays and errors (Perz, 1991).

At present electric power is mainly produced by using natural resources. Electricity is consumed rapidly as an energy resource in the world. System simulation is one of the parts of optimizing power plants process. Most designed systems are operating at loads less than the value which had been designed. Thus operating at lower thermal efficiencies than it was predicted. (Egelioğlu, F., 2002).

Power plant manufacturers provide training manuals to their customers but most of these training manuals are not open to the public. Web-sites of power plant suppliers such as ABB [http://www.abb.com/powergeneration] provide information on power plant equipment, optimization, technology, environmental issues, project development and financing.

The Carnot cycle is the most efficient cycle operating between two given temperatures (i.e., high temperature and low temperature). However, the practical limitations of the Carnot cycle makes it an unsuitable model for a power generation cycle. By consider all theoretical and practical limitations and redesigning the cycle to eliminate the impracticalities such as superheating the steam before entering into turbine and condensing it completely after exiting the turbine, gives the idealized Rankine cycle.

The Rankine cycle (i.e., vapor power cycle) is employed in different steam power plants, (Ahlgren, 1994). The simple Rankine cycle comprises of four main parts; boiler, turbine, condenser, and pump. The efficiency of cycle performance can be improved by adding some more components to the cycle (i.e., by modifying the cycle) (Fischer DW. 1996).

The Carnot vapor cycle and the Rankine cycle are explained in brief in the following sections.

2.1 The Carnot Vapor Cycle

Carnot vapor cycle is the most productive ideal cycle, which operates between two certain temperature levels, but the Carnot cycle is not considered as a perfect model for power cycles because of several drawbacks. As it can be seen in Fig. 2.1, the thermal efficiency will be reduced in the cycle as the heat transfer process is restricted to two phase systems. The temperature levels (processes 1-2 and 3-4 which are heat addition and rejection at constant temperature) are limited. On the other hand in process 2-3 (i.e., isentropic expansion in a turbine) if the steam quality becomes less than 90% water droplets formed erodes the turbine blades. The process 4-1 (i.e., isentropic compression process in a pump) requires a compressor to manage two phases which is impractical. Therefore, applying Carnot Vapor Cycles for actual machines and real vapor power cycles is not recommended (Onkar, S., 2009).

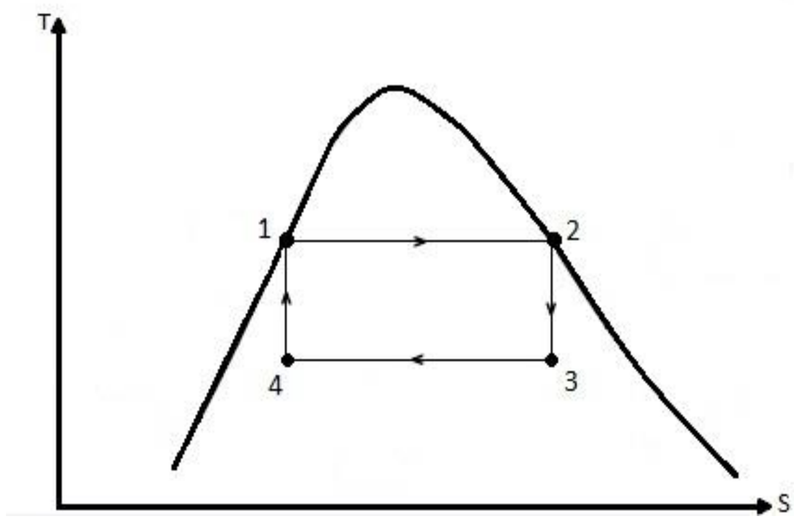

Figure 2.1. Carnot Vapor Cycle T-S Diagram (Rankine Cycle, n.d.)

2.2 The Ideal Rankine Cycle

As mentioned earlier, the Rankine cycle (i.e., vapor power cycle) is employed in different steam power plants. The ideal Rankine cycle has no internal irreversibilities. The Rankine is a vapor-liquid cycle so it is convenient to sketch the P-V (pressure-volume) and T-S (temperature-entropy) diagrams with respect to saturated-liquid and vapor lines. The simplified flow diagram of the Rankine cycle is presented in Fig. 2.2. The T-S diagram of a superheated ideal Rankine cycle are shown in Figs. 2.3. The cycle has the following processes. Process 1-2, adiabatic reversible (isentropic) compression by the pump of saturated liquid. Process 2-3, heat addition in the steam generator at constant pressure. Process 3-4, isentropic expansion through the turbine. Process 4-1 heat rejection at constant pressure in the condenser.

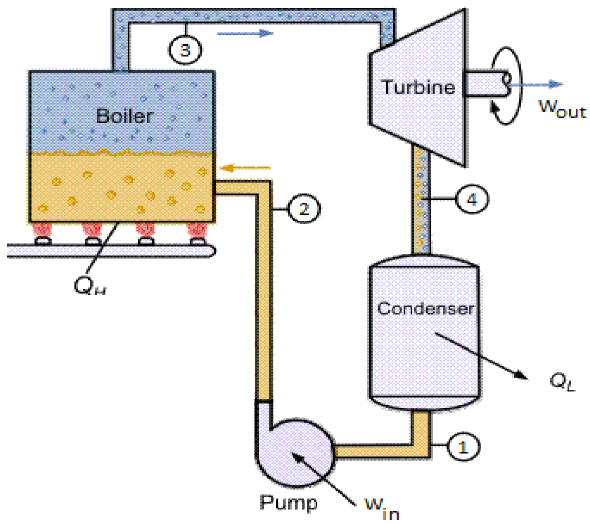

Figure 2.2. Simple Ideal Rankine Cycle (Rankine Cycle, n.d.)

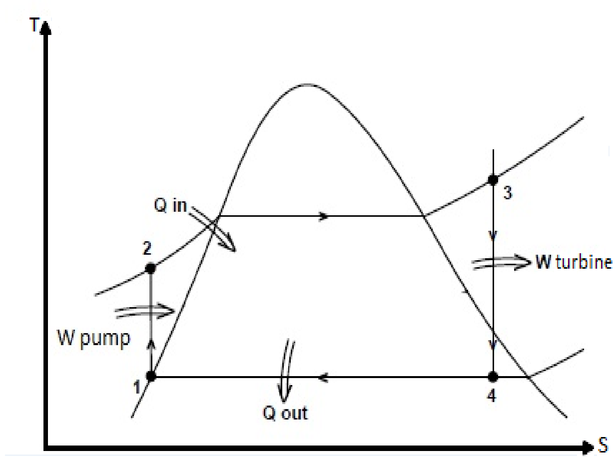

Figure 2.3. T-S Diagram of Simple Ideal Rankine Cycle (Rankine Cycle, n.d.)

In Fig. 2.3, the area under stage 2 to 3 shows the transfer of heat to water in the boiler and the area under process 4 to 1 shows rejection of heat in condenser. The net work developed within the cycle is shown by the differences between the areas which are contained by the cycle curves (Kapooria et al, 2008). Figure 2.4 shows steam turbines, condenser and generator of a real power plant.

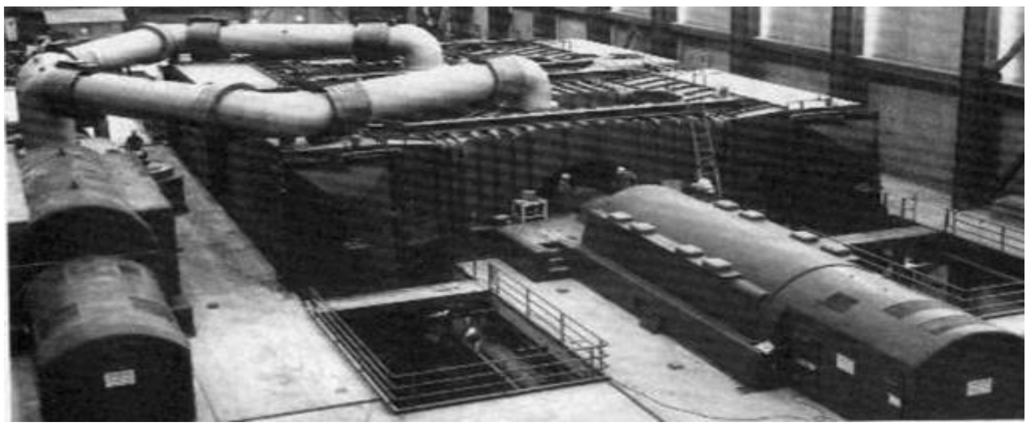

Figure 2.4. Steam Turbines, Condenser and Generator at TVA Bull Run Plant (Kapooria et al, 2008)

2.3 Ideal Rankine Cycle Energy Analysis

Pump, boiler, turbine and condenser are steady flow devices. The steady-flow energy equation per unit mass of steam will be as follows. (Ignoring change in kinetic and potential energy)

$$q - w = \Delta h \tag{2.1}$$

where, q is the heat transfer (kJ/kg), w is the work (kJ/kg) and Δh is the change in enthalpy (kJ/kg).

Ideal Rankine cycle is internally reversible so that the pump and turbine are considered to be isentropic. The condenser and boiler are not intended to do any work, so the energy analysis for each device is as follows.

Pump work $w_{pump,in} = h_2 - h_1 = v_1(P_2 - P_1)$ kJ/kg (2.2)

Where $w_{pump,in}$ is the pump work input where h_2-h_1 is the enthalpy change of feed water between the output and input of the pump, v_1 is the specific volume of feed-water at the pump inlet and P_2 and P_1 are the feed-water pressures at the outlet and inlet of the pump respectively.

Heat added $q_{in} = h_3 - h_2$ kJ/kg (2.3)

Where q_{in} is the heat added in the boiler, h_3 and h_2 are the enthalpy of the working fluid at the exit and inlet of the boiler respectively.

Heat rejected $q_{out} = h_4 - h_1$ kJ/kg (2.4)

Where q_{out} is the heat rejected at the condenser, h_4 and h_1 are the enthalpies of the working fluid at the inlet and exit of the condenser.

Turbine work $w_{turb,out} = h_3 - h_4$ kJ/kg (2.5)

Net work $w_{net} = (h_3 - h_4) - (h_2 - h_1)$ kJ/kg (2.6)

Thermal efficiency $\eta_{th} = \frac{w_{net}}{q_{in}}$ (2.7)

Work ratio is another parameter which is employed in the Rankine cycle analysis. The work ratio is defined as the ratio of net work to gross work.

The actual power cycles are not internally reversible. The deviation of actual power cycle is briefly explained below.

2.4 Deviation of Actual Vapor Cycle from Ideal Vapor Cycle

The Actual vapor cycle is different from the ideal one, because of irreversibilities in the devices and losses within pipes. Greater work input is needed for the pump and less work output is produced by the turbine because of irreversibilities. Regarding to ideal condition, isentropic flow is considered for the pump and turbine. The deviation of the actual cycle from the ideal cycle is presented in Fig. 2.5. The deviation between actual turbines and pumps from isentropic pump and turbine can be explained by applying isentropic efficiencies given below.

$$\eta_p = \frac{w_s}{w_a} = \frac{h_{2s}-h_1}{h_{2a}-h_1} \tag{2.8}$$

$$\eta_t = \frac{w_a}{w_s} = \frac{h_3-h_{4a}}{h_3-h_{4s}} \tag{2.9}$$

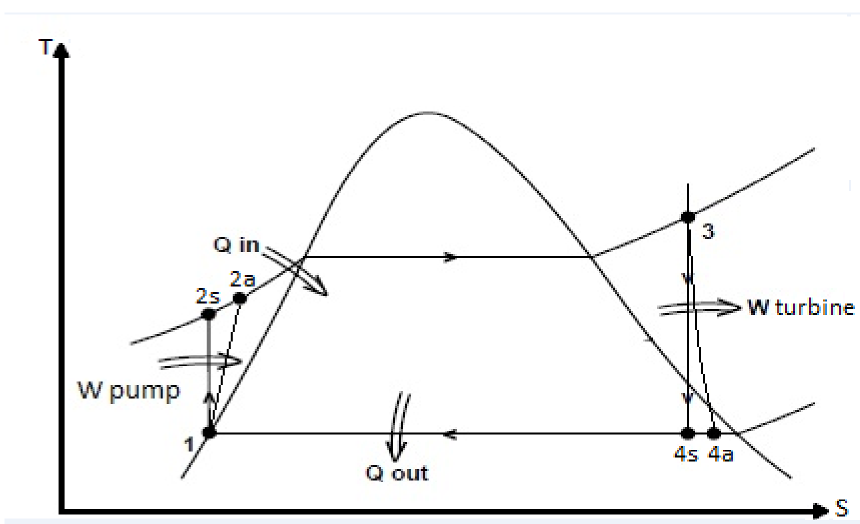

Figure 2.5. Deviation of Actual Vapor Cycle from Ideal Vapor Cycle (Rankine Cycle, n.d.)

Where η_p and η_t are the pump and turbine isentropic efficiencies respectively and h_{2a} and h_{4a} are the actual specific enthalpies at the exit of pump and turbine respectively whereas, h_{2s} and h_{4s} are corresponding isentropic specific enthalpies.

2.5 How to Increase Rankine Cycle Efficiency

Most of the electricity production all around the world is generated in steam power plants and when thermal efficiency increases, it will lead to large amounts of saving from fuel consumption. The simple rule is first to increase the average degree of temperature at which heat transfers to the fluid in boiler, second, decrease the average degree of temperature at which heat rejected from the fluid in condenser. Three ways that are recommended to increase the thermal efficiency of the Rankine cycle are as follows:

1. Reducing the condenser pressure (lowers condensing temperature)

2. Heating the steam extremely to reach to high temperature (i.e., superheating)

3. Raising the boiler pressure, increases the boiler temperature (Cengel, Y. A., & Boles, M. A., 2007).

Reducing the condenser pressure depends on the cooling water temperature in the condenser. Increasing boiler pressure increases the boiler temperature thus, efficiency can be improved. However, this procedure will increase moisture content in the exiting vapor to unwanted level. This problem can be solved by reheating.

2.6 Regeneration

A great deal of irreversibility can be avoided if regeneration is utilized to heat up feedwater before entering to the boiler. Rising the average temperature of water (as a working fluid) before sending it to the boiler, increases thermal efficiency because the working fluid has higher temperature within the boiler. This process can be carried out by dragging the temperature from the higher temperature in turbine to the lower temperature in the feedwater rather than utilizing another external source. This procedure which saves a lot of energy is called Regeneration and the steam that comes from a turbine to heat the feedwater is named extraction steam (Srinivas, T. et al, 2010).

By inclusion of feedwater heaters (FWHs), the steam power cycle efficiency can be increased (Srinivas, T. et al, 2010, Haywood, RW., 1949, Weir CD, 1960)

Regarding to thermodynamic outlook, the impact of steam regeneration with direct contact heaters on performance of combined plants was analyzed by Cerri (Cerri, 1985).

In the literature, there is an investigation regarding to relation between formulations of number of FWHs in the Rankine cycle. Many ways are mentioned in the literature about increasing steam power cycle efficiency. Batt et. al. (Batt and Rajkumar,1999) introduced the methods for increasing the cycle efficiency of coal-fired thermal power plants.

The regenerative FWHs effect on steam power plant operational cost for boosting energy efficiency was investigated by Szargut (2005).

2.7 Feedwater Heating

Feedwater heating reduces economizer irreversibility. Feedwater heating was started to be used in early 1920s. In large steam power plants 5-8 feedwater heaters are employed. There are three types of feedwater heaters, these are:

Open FWH

Closed FWH with drains pumped forward

Closed FWH with drains cascaded backward

2.8 Open Feedwater Heater

Steam extracted from the turbine, mixes directly with feedwater in an open FWH to increase its temperature.

FWH function is to use extraction steam energy to increase the temperature of the feedwater before reaching steam generator. FWH will be insulated in order to prevent loss of heat to environment so they are considered as adiabatic devices (Weston, K., 1992).

The benefits of using open feedwater heater are their low cost and capacity of transferring high heat. The drawback is the use of pump for each heater to manage stream of large feetwater (Dincer, I., & Al-Muslim, H., 2001).

2.9 Closed Feedwater Heater

The difference between open FWH and closed FWH is that the mixing does not occur in closed feedwater heater. As a result, the two streams that enter the closed feed water heater can have different pressures. The condensed steam either goes to another FWH or to the condenser through a trap. A trap is installed which can throttle the liquid to a lower amount of pressure and traps vapor.

The complexity of designing the inner tubing network of closed FWH is one of the disadvantages. Another drawback of closed FWH is its high expense and less effective heat transfer because two streams do not have direct contact. One of the advantages of closed FWHs is that they do not need a separate pump for each single heater if it drains cascaded backward (Weston, K., 1992).

The placement of FWHs affects the thermal performance of the Rankine cycle. In the following chapter, the methodology for power plant thermal efficiency optimization is explained in detail.

Chapter 3

METHODOLOGY

Steam thermal power plants have mature technology. Steam power plants are widely used for power generation, so researches in the field is continuous. In this study, VisSim simulation software (version 3.0E) is used for the Rankine cycle efficiency analysis. VisSim is a block diagram visual simulation program which can be employed for simulating complex dynamic systems. With this software, flow paradigm of graphical data is implied to run dynamical system by using variety of equations. This version is free of charge for academic purposes and add-ons can easily be downloaded for further purposes (Darnell,1996).

In this study the VisSim software was employed to simulate ideal and actual Rankine cycles having one, two and three feedwater heaters. All the components of the Rankine cycle are modeled separately under a compound block, (i.e., modular simulation) in VisSim. The power plant system modeling was performed by connecting these sub-components. The work can be extended for the cycles having more than three feedwater heaters. The working fluid (i.e., water) property tables obtained from the engineering thermodynamics book [Cengel, A. and Boles,A.,(2001)] were inserted to VisSim as property blocks. The VisSim software is capable to use tables (i.e., property blocks) and make interpolations.

3.1 Single-Staged Regenerative Rankine Cycle Simulation

In the Rankine cycle several components interact with each other. Each component is modeled separately for simulation. Both ideal and actual cases of each regenerative cycle were studied. In the ideal case the devices are considered to be internally reversible, the processes in pump and turbine are isentropic (i.e., reversible adiabatic). In actual cycles the internal irreversibilities of the devices are considered so the isentropic efficiencies of pumps and turbine are less than 100%. These irreversibilities cause the pump to require more work input and the turbine to generate less work output. The pressure drop in the boiler and the condenser was assumed to be negligible in this study. The simple Rankine cycle efficiency can be improved substantially by the addition of feedwater heaters. Figure 3.1 shows a schematic flow diagram of single-staged regenerative Rankine cycle. The processes that are taking place in the turbine and the steam generator are very important because the steam properties at the inlet of the turbine play an important role in the cycle efficiency. The thermal efficiency, η_{th}, can be defined as the net power output over boiler heat input (see Eq. 2.7).

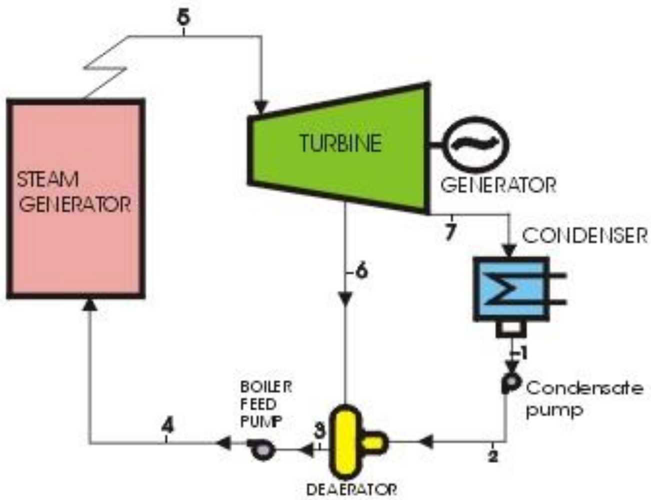

Figure 3.1. Schematic Flow Diagram of Single-Staged Regenerative Rankine Cycle

3.1.1 Mass Balance

In order to find the thermal efficiency of the Rankine cycle both mass balance and energy balance must be considered. For example, the mass balance based on a unit mass flow rate at the turbine inlet for the cycle given in Fig. 3.1 can be expressed as:

Mass flow between 5 and 6 $\dot{m} = 1$

Mass flow between 6 and 3 y

Mass flow between 6 and 7 $\dot{m} = 1\text{-}y$

Where y is the steam extraction (kg/s) from the turbine at 6.

3.1.2 Energy Balance

The processes that make up the Rankine cycle can be analyzed as steady flow processes. The potential and kinetic energy changes of the steam can be neglected as they are relatively small compared to the work and heat transfer terms. The heat and work interactions of a regenerative Rankine cycle with one feedwater heater (see Fig. 3.1) can be expressed per unit mass of the steam flowing through the boiler (i.e., $\dot{m}$ =1 kg/s) as follows:

Heat added (q_{in}) to the working fluid in the steam generator is

$$q_{in} = h_5 - h_4 \qquad \text{kJ/kg} \tag{3.1}$$

Where, h_5 and h_4 are the specific enthalpies at the exit and inlet of the steam generator respectively. Similarly, heat rejection at the condenser is

$$q_{out} = (1-y)\,(h_7 - h_1) \qquad \text{kJ/kg} \tag{3.2}$$

where y is the fraction of the steam extracted at 6 (i.e., $y = \dot{m}_1/\dot{m}$)

Turbine work (w_t, kJ/kg) can be expressed as:

$$w_{turb\ out} = (h_5 - h_6) + (1-y)\,(h_6 - h_7) \tag{3.3}$$

As seen in Fig. 3.1 two pumps are employed. Pump I pumps the condensate water from the condenser to the open FWH. Boiler feed pump (pump II), pumps the saturated liquid water from the open FWH to the steam generator. Pump work input for the two pumps can be calculated as

$$w_{pumpI,\,in} = (1-y)\,(h_2-h_1) \tag{3.4}$$

$$w_{pumpII,\,in} = h_4-h_3 \tag{3.5}$$

The energy analysis of the FWHs can be expressed as

$$\Sigma\, m_i h_i = \Sigma\, m_e h_e$$

Where the subscripts *i* and *e* stands for inlet and exit respectively.

There is one open FWH, the energy equation for the OFWH is

$$y\, h_6 + (1-y)\, h_2 = h_3 \tag{3.6}$$

Similar equations are developed for two and three FWHs regenerative Rankine cycles shown in Figs. 3.2 and 3.3. The developed equations were used in VisSim for the cycle optimization.

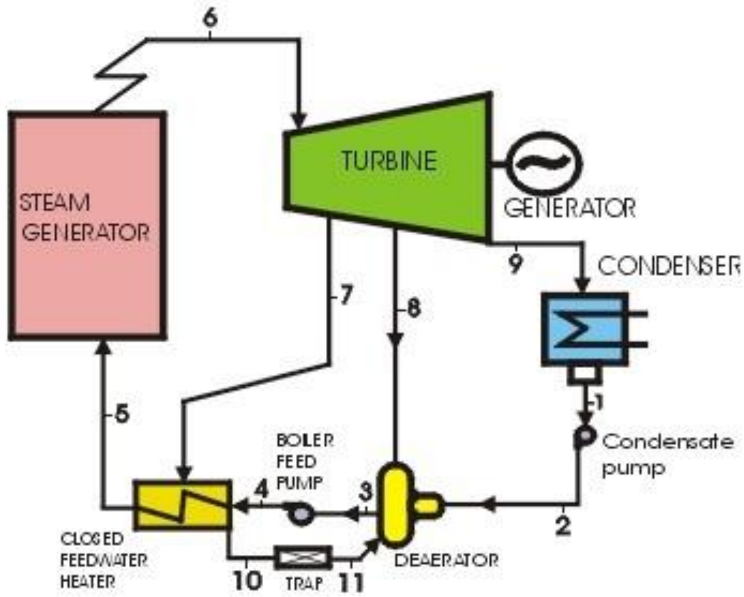

Figure 3.2. Schematic Flow Diagram of Double-Staged Regenerative Rankine Cycle

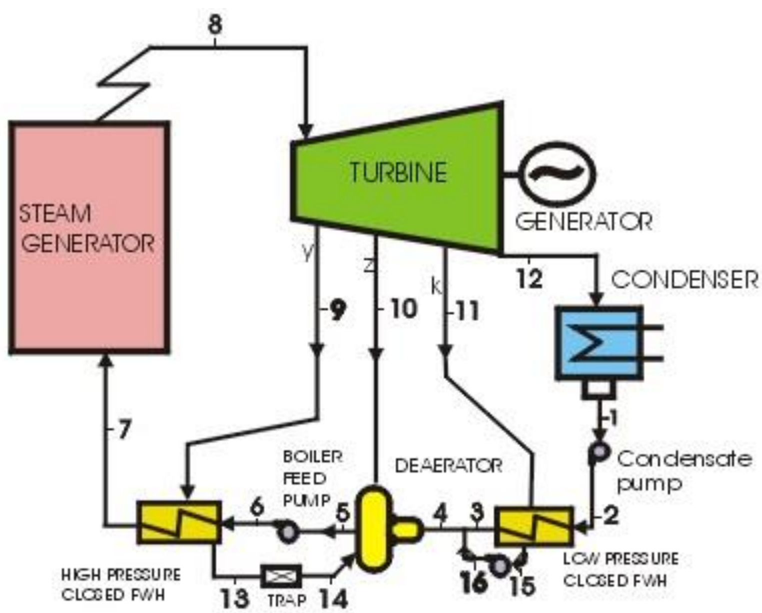

Figure 3.3. Schematic Flow Diagram of Triple-Staged Regenerative Rankine Cycle

The base case operating parameters which are used for the ideal and actual single-staged, double-staged and triple-staged regenerative Rankine cycles are presented in the following tables.

Table 3.1. Operating Parameters (Base Case) of the Single-Staged Regenerative Rankine Cycle

Condenser pressure	P_1 = P_7 =10 kPa
Open FWH (Deaerator pressure, varied)	10 kPa < P_2 = P_3 = P_6 < 4010 kPa
Boiler pressure	P_4 = P_5 = 10000 kPa
Boiler temperature	T_5 = 510 ^{o}C
Turbine efficiency	η_t = 0.9 (η_t = 1.0 for the ideal cycle)
Pump efficiency	η_p = 0.9 (η_p = 1.0 for the ideal cycle)

Table 3.2. Operating Parameters of the Double-Staged Regenerative Rankine Cycle

Condenser pressure	P_1 = P_9 = 10 kPa
Closed FWH pressure	1500 kPa < P_7 < 3500 kPa
Open FWH pressure	250 kPa < P_8 < 1500 kPa
Boiler pressure	P_6 = 10000 kPa = P_4 = P_5
Boiler temperature	T_6 = 510^{o}C
Turbine efficiency	η_t = 0.9 (η_t = 1.0 for the ideal cycle)
Pump(s) efficiency	η_p = 0.9 (η_p = 1.0 for the ideal cycle)

Table 3.3. Operating Parameters of the Triple-Staged Regenerative Rankine Cycle

Condenser pressure	P_1 = P_{12} = 10 kPa
High pressure closed FWH pressure	1000 kPa < P_9 < 6000 kPa
Open FWH pressure	300 kPa < P_{10} < 1800 kPa
Low pressure closed FWH pressure	20 kPa < P_{11} < 220 kPa
Boiler pressure	P_8 = 10000 kPa = P_6 = P_7
Boiler temperature	T_8 = 510^{o}C
Turbine efficiency	η_t = 0.9 (η_t = 1.0 for the ideal cycle)
Pump(s) efficiency	η_p = 0.9 (η_p = 1.0 for the ideal cycle)

The thermal efficiency calculations for each regenerative cycle are based on the data presented in Tables 3.1-3.3. The results are presented in the following chapter. The main purpose of this study is to seek the optimum placement of the FWHs (i.e., optimum extraction pressures for feedwater heating) which gives the maximum thermal efficiency. Thermodynamic optimization methodology for the cycles is briefly explained in the following section.

3.2 Thermodynamic Optimization Methodology

Heat balance calculations are necessary to obtain power plant efficiency or heat rate. Heat balance calculations procedures are explained below:

- The turbine expansion line is estimated by using turbine operating pressure and temperature, turbine internal efficiency, extraction and condenser pressures.

- Steam properties at various locations are determined by using the known operating parameters.

- Extraction steam flow rates are calculated, starting with the high-pressure heater closest to the steam generator.

- The turbine work output is calculated by adding the outputs of the turbine cylinders.

- The power consumed by the feedwater pumps and heat inputs supplied to the system are calculated, and finally plant thermal efficiency is estimated.

The aim is to optimize the Rankine cycle (i.e., to find the optimum design parameters that give the maximum thermal efficiency). The placement of feedwater heaters is important in cycle optimization. Optimum extraction pressures can be obtained most accurately by a complete optimization of the cycle [Wakil, 1984].

Although there are several rough methods for extraction pressure calculations, those methods are not suitable for real power plants. They can be employed for ideal systems. Rough methods include, equal temperature increase method, equipartition of enthalpy increasing method and geometry distributing method, and etc.

In the optimization of the Rankine cycles, first of all, the turbine inlet pressure, the turbine inlet temperature, and the turbine exit pressure are fixed (see Table 3.1). The pressures of the all extractions (for 2 and 3 FWHs Rankine cycles) except for the first one are fixed at estimated values as well. The pressure of the first extraction is variable. The turbine work for a specific pressure range of the first extraction is computed as the summation of turbine work at all stages by using these fixed parameters. Then, the boiler inlet enthalpy, boiler heat input, and the cycle thermal efficiency (η_{th}) are obtained as explained in the previous sections. After obtaining the optimum value of the pressure (i.e., the pressure that gives the maximum cycle thermal efficiency) for the first extraction, the pressure for the first extraction is fixed at this pressure. Then, the pressure of the second extraction is taken as a variable parameter, and the procedure described above is repeated for the second extraction pressure values to obtain the optimum value of the pressure for the second extraction pressure. The same procedure is repeated for the other extraction pressures. Now, this is the end of the first iteration. Then, the pressure of the first extraction is taken as a variable parameter. After obtaining the new first extraction pressure that gives the maximum cycle thermal efficiency, the pressure of the second extraction is taken as variable and the new optimum pressure value is obtained for the second extraction pressure. This procedure is repeated for all the other extraction pressures. Those iterations are repeated until the pressures of the extractions that give the maximum cycle thermal efficiency does not change, namely the convergence is obtained.

3.3 The Effect of the Operating Parameters on Cycle Efficiency

The effect of the operating parameters, boiler pressure and temperature and the condenser pressure on the optimum thermal efficiency are investigated in detail. First boiler pressure has been changed for the simulation. The other operating parameters are kept constant (i.e., boiler temperature and condenser pressure). The boiler pressure was increased by 5, 10, 15 and 20% from the base value (i.e., 10000 kPa) and the placement of the feedwaters and the optimum efficiency were evaluated for each case as explained in the previous section. The effect of boiler pressure was also investigated by decreasing the base value by 5, 10, 15 and 20%.

Then the effect of the boiler temperature and condenser pressure on the cycle efficiency and the placement of feedwater heaters for each case as explained in the previous paragraph were investigated. The impact of the boiler temperature and pressure, condenser pressure on the ideal cycles (i.e., 1, 2 and 3 FWHs ideal Rankine cycles) were investigated, the pump and turbine efficiencies are assumed to be 100% in ideal Rankine cycles. In actual cycles the impact of the boiler temperature and pressure, condenser pressure and the turbine and pumps isentropic efficiencies on the maximum cycle efficiency and the optimal placement of the FWHs were investigated.

The effects of pump efficiency and turbine efficiency on the actual cycles were investigated. The base values for the pump(s) and turbine efficiencies were used as 0.9. The impact on the optimum cycle efficiency of $\mp 5\%$ change in the pump and turbine isentropic efficiencies were separately investigated for all actual cases.

The simulation results are presented in the following chapter.

Chapter 4

RESULTS ANALYSIS

4.1 Regenerative Rankine Cycle with One Feedwater Heater

Single-staged regenerative Rankine cycle has one open FWH. The effect of boiler pressure, condenser pressure, boiler temperature, pump and turbine isentropic efficiencies on the cycle thermal efficiency and the optimum placement of the open FWH are presented in the following sub-sections.

4.1.1 Boiler Pressure

Figures 4.1 and 4.2 illustrate cycle thermal efficiency versus DEA (deaerator) pressure for nine different boiler pressures in the single-staged regenerative Rankine cycle. It is clear from Figs. 4.1 and 4.2 that the thermal efficiency and the DEA optimum pressure increase as the boiler pressure increases. As expected the thermal efficiency of the ideal cycle is greater than the actual cycle as there are no internal irreversibilities in the ideal cycle.

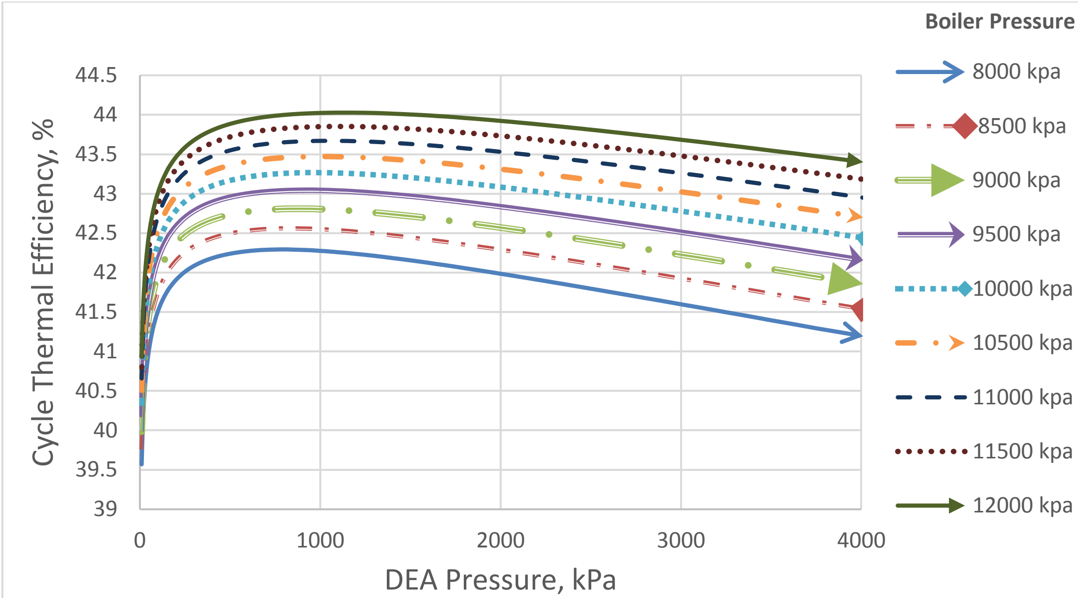

Figure 4.1. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Ideal Single-Staged Regenerative Rankine Cycle

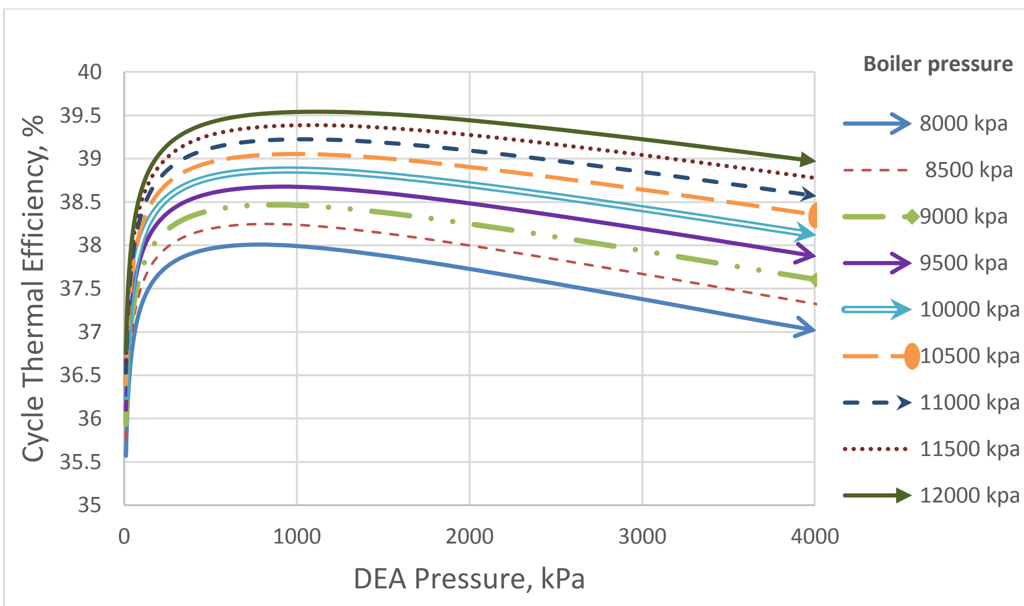

Figure 4.2. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Actual Single-Staged Regenerative Rankine Cycle

Figures 4.3 and 4.4 plot the DEA optimum pressure versus boiler pressure for the ideal and actual single-staged regenerative Rankine cycles respectively. Optimum DEA pressure increases with respect to boiler pressure in ideal and actual cycles.

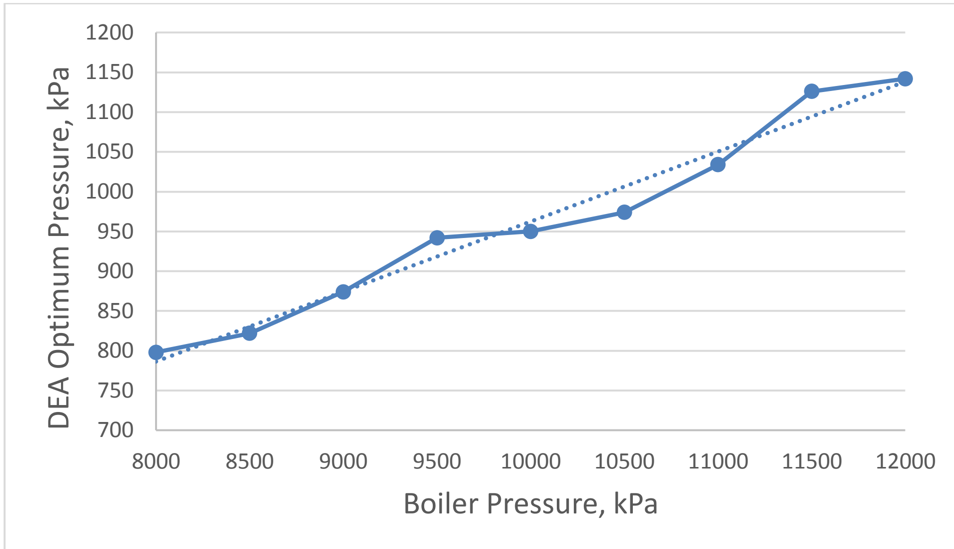

Figure 4.3. Optimum DEA Pressure vs. Boiler Pressure in the Ideal Single-Staged Regenerative Rankine Cycle

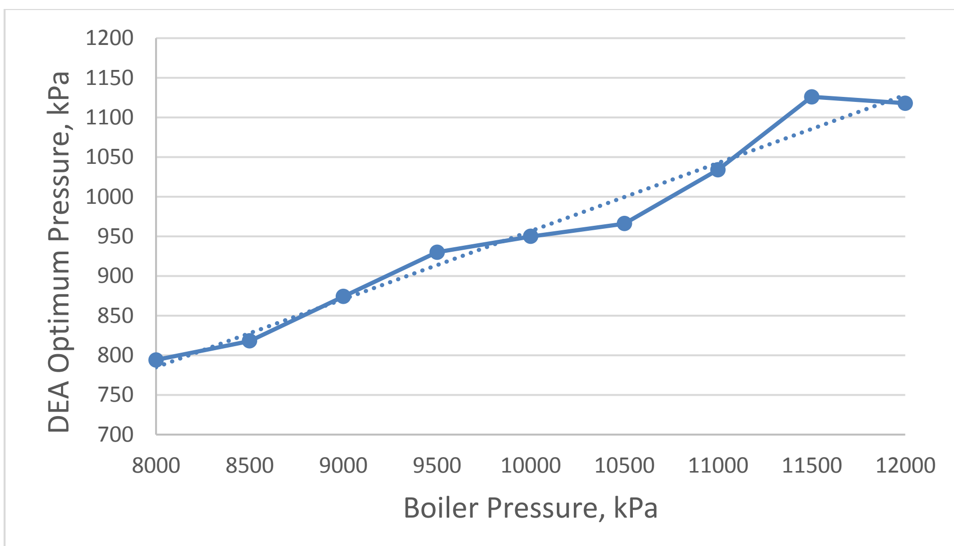

Figure 4.4. Optimum DEA Pressure vs. Boiler Pressure in the Actual Single-Staged Regenerative Rankine Cycle

Figures 4.5 and 4.6 show that the thermal efficiency varies linearly with respect to the boiler pressure in the ideal and actual single-staged regenerative Rankine cycles respectively.

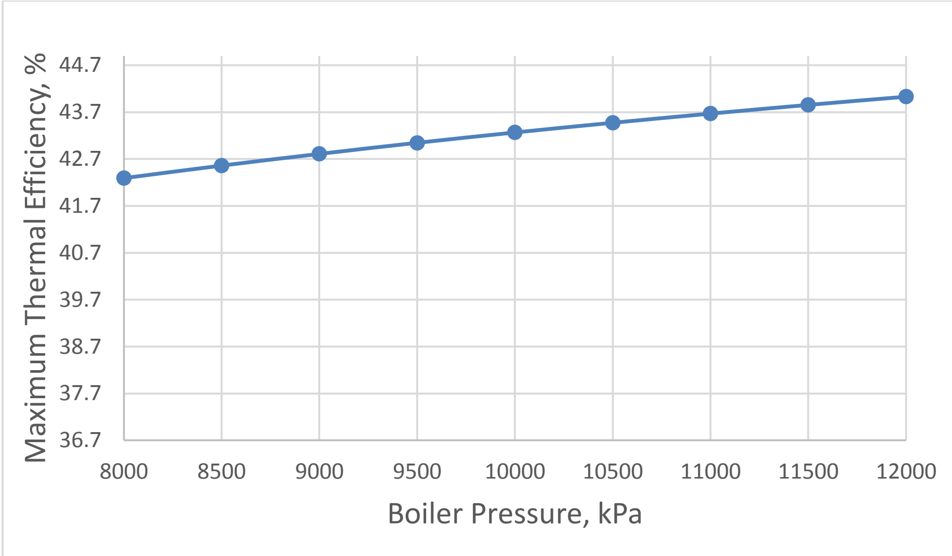

Figure 4.5. Maximum Thermal Efficiency vs. Boiler Pressure in the Ideal Single-Staged Regenerative Rankine Cycle

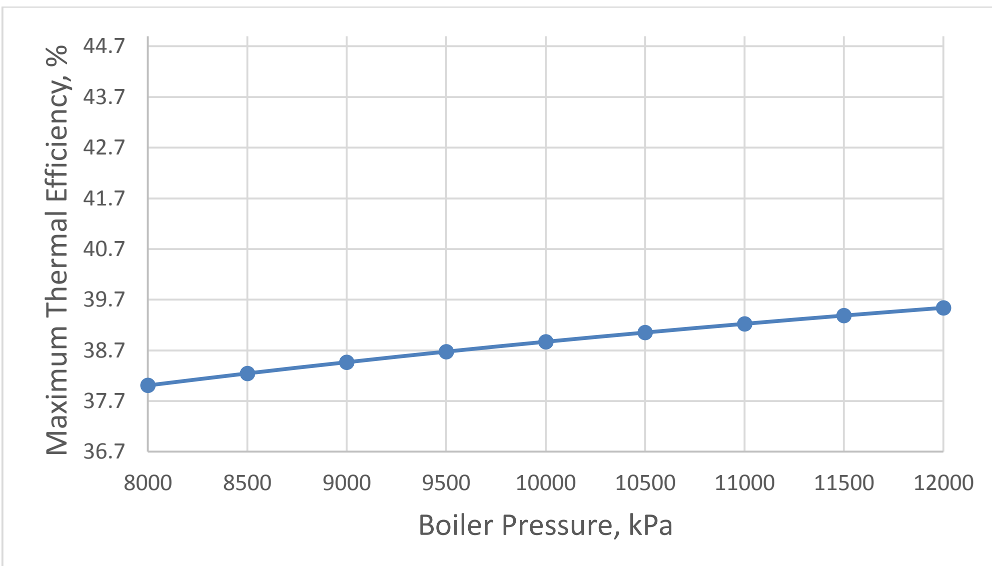

Figure 4.6. Maximum Thermal Efficiency vs. Boiler Pressure in the Actual Single-Staged Regenerative Rankine Cycle

4.1.2 Condenser Pressure

Figures 4.7 and 4.8 plot the cycle thermal efficiency versus DEA pressure at various condenser pressures for the ideal and actual single-staged regenerative Rankine cycles respectively. As the condenser pressure increases thermal efficiency decreases.

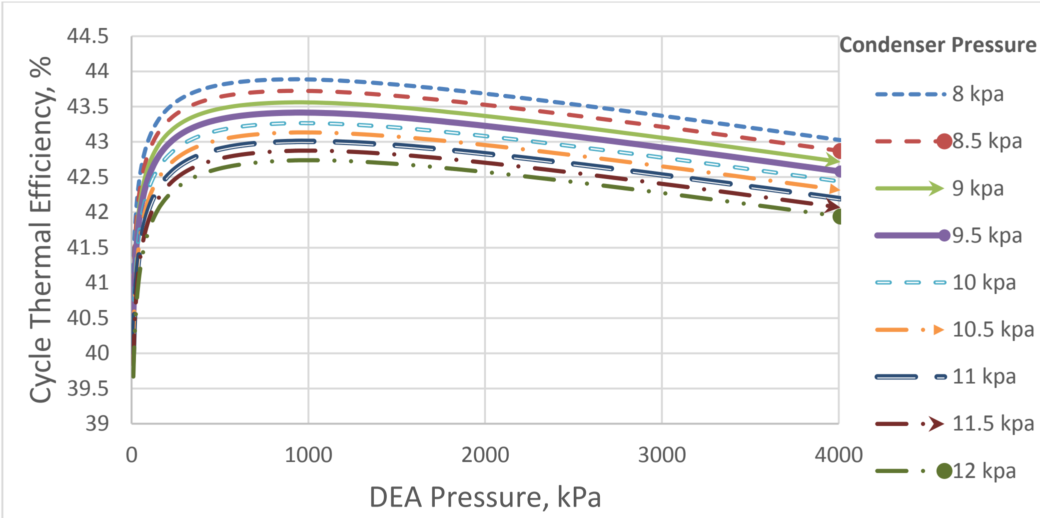

Figure 4.7. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Ideal Single-Staged Regenerative Rankine Cycle

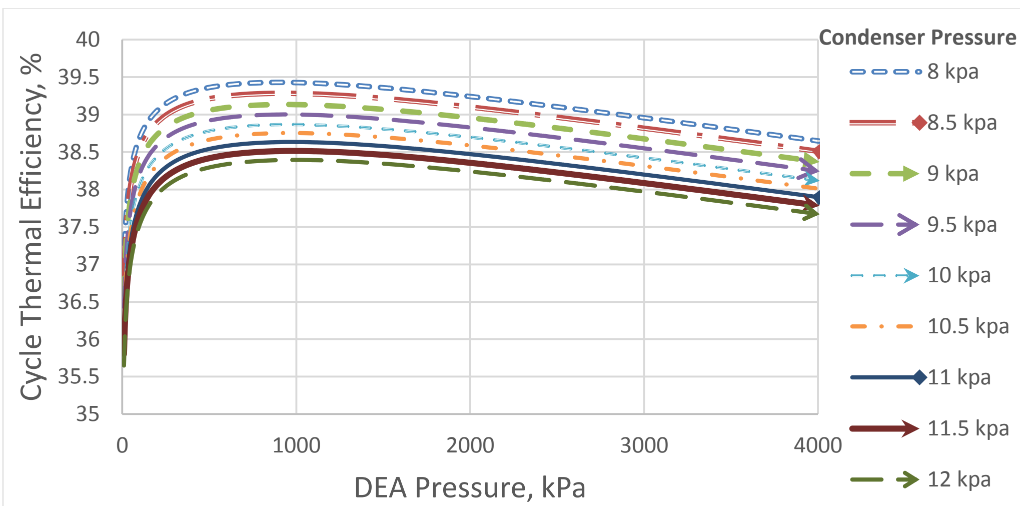

Figure 4.8. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Actual Single-Staged Regenerative Rankine Cycle

Figures 4.9 and 4.10 plot optimum DEA pressure vs. condenser pressure for the ideal and actual single-staged regenerative Rankine cycles respectively. The optimum DEA pressure increases slightly with respect to the condenser pressure in both cases.

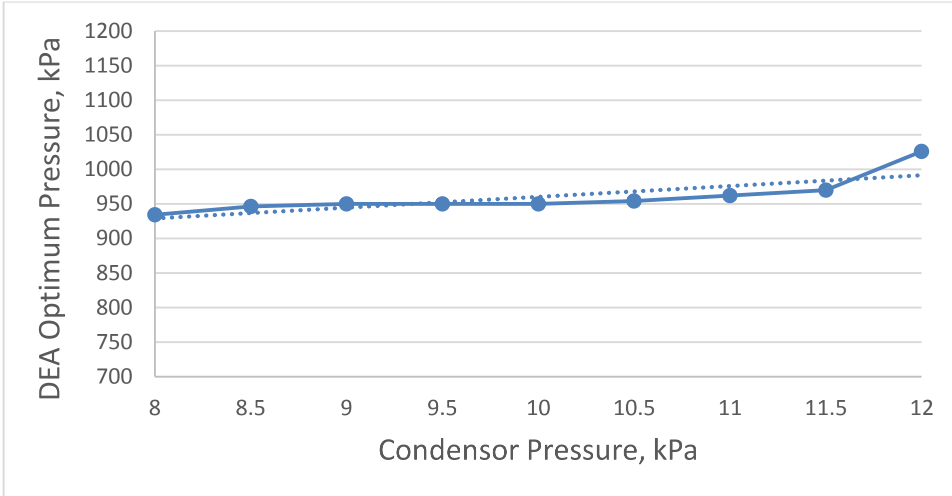

Figure 4.9. Optimum DEA Pressure vs. Condenser Pressure in the Ideal Single-Staged Regenerative Rankine Cycle

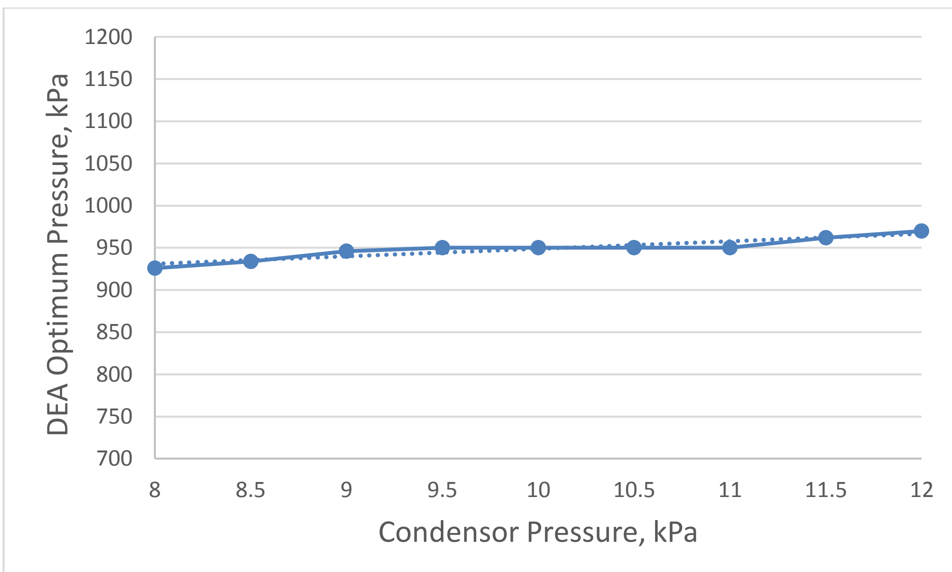

Figure 4.10. Optimum DEA Pressure vs. Condenser Pressure in the Ideal Single-Staged Regenerative Rankine Cycle

Figures 4.11 and 4.12 present the maximum thermal efficiency vs. condenser pressure for the ideal and actual single-staged regenerative Rankine cycles respectively. The cycle efficiency decreases as the condenser pressure increases.

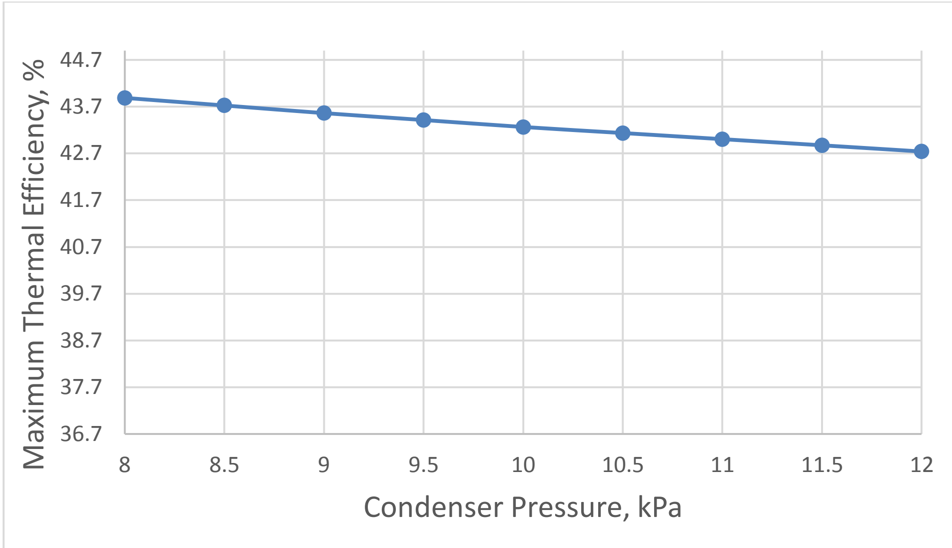

Figure 4.11. Maximum Thermal Efficiency vs. Condenser Pressure in the Ideal Single-Staged Regenerative Rankine Cycle

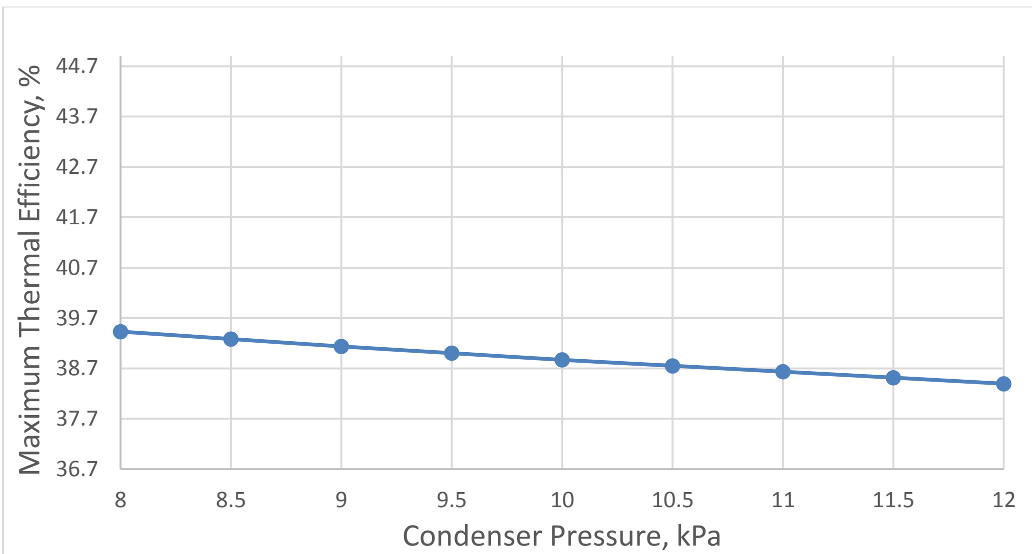

Figure 4.12. Maximum Thermal Efficiency vs. Condenser Pressure in the Actual Single-Staged Regenerative Rankine Cycle

4.1.3 Boiler Temperature

Boiler temperature is an important parameter in power generation. Figures 4.13 and 4.14 represent thermal efficiency vs. DEA pressure at 9 different boiler temperatures for the ideal and actual single-staged regenerative Rankine cycle respectively. It is clear from the plots that thermal efficiency increases as boiler temperature increases (superheating increases efficiency).

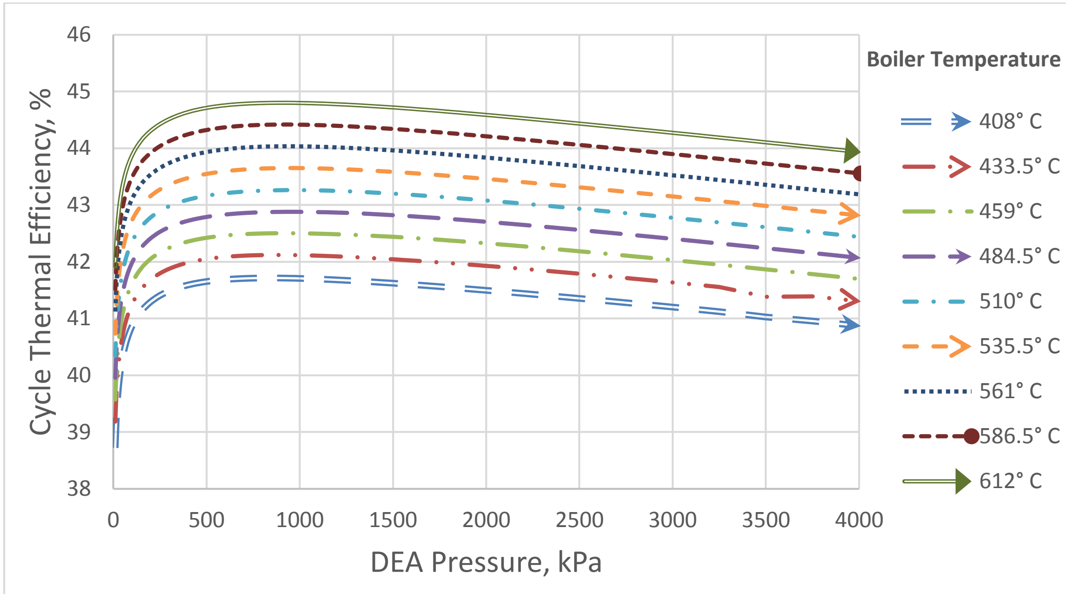

Figure 4.13. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Ideal Single-Staged Regenerative Rankine Cycle

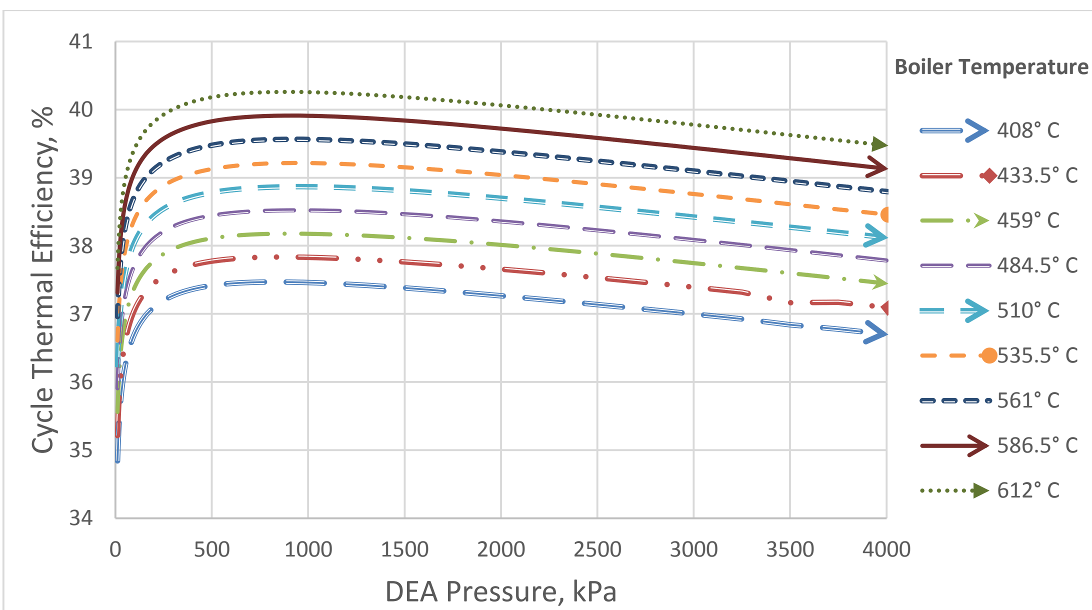

Figure 4.14. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Actual Single-Staged Regenerative Rankine Cycle

Figures 4.15 and 4.16 show the optimum DEA pressure vs. boiler temperature for the ideal and actual single-staged regenerative Rankine cycles respectively. In both cases, as the boiler temperature increases, the optimum DEA pressure first rises but then decreases.

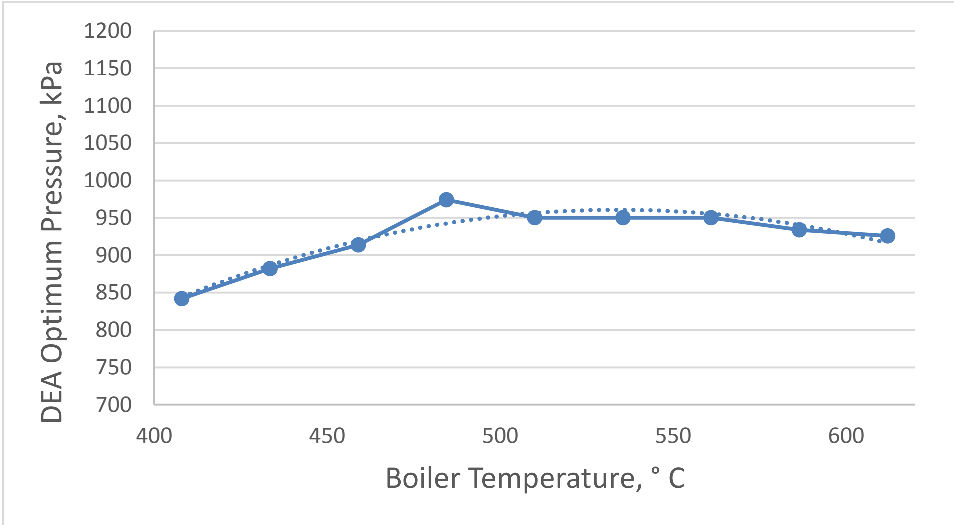

Figure 4.15. Optimum DEA Pressure vs. Boiler Temperature in Ideal Single-Staged Regenerative Rankine Cycle

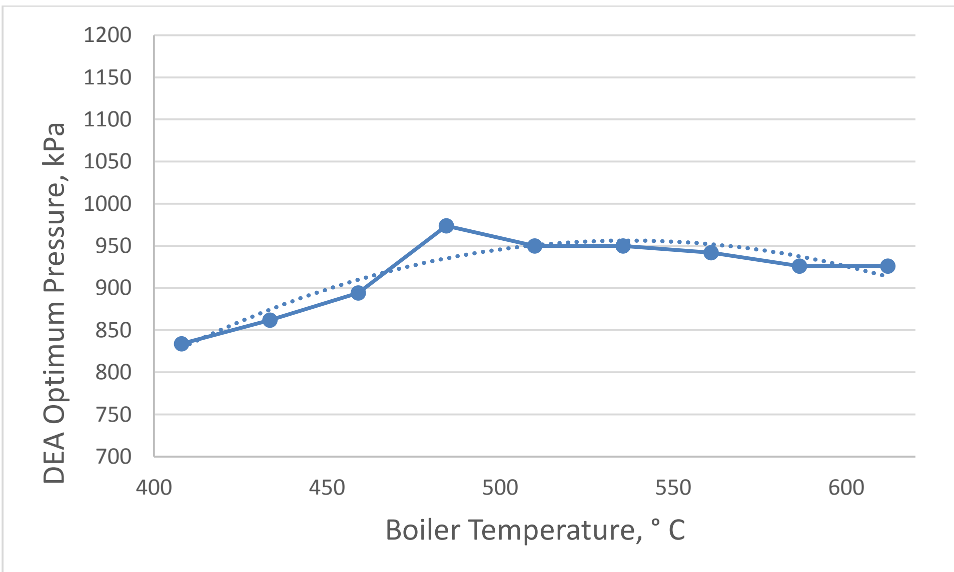

Figure 4.16. Optimum DEA Pressure vs. Boiler Temperature in Actual Single-Staged Regenerative Rankine Cycle

The following Figs. 4.17 and 4.18 demonstrate that thermal efficiency increases as boiler temperature increases.

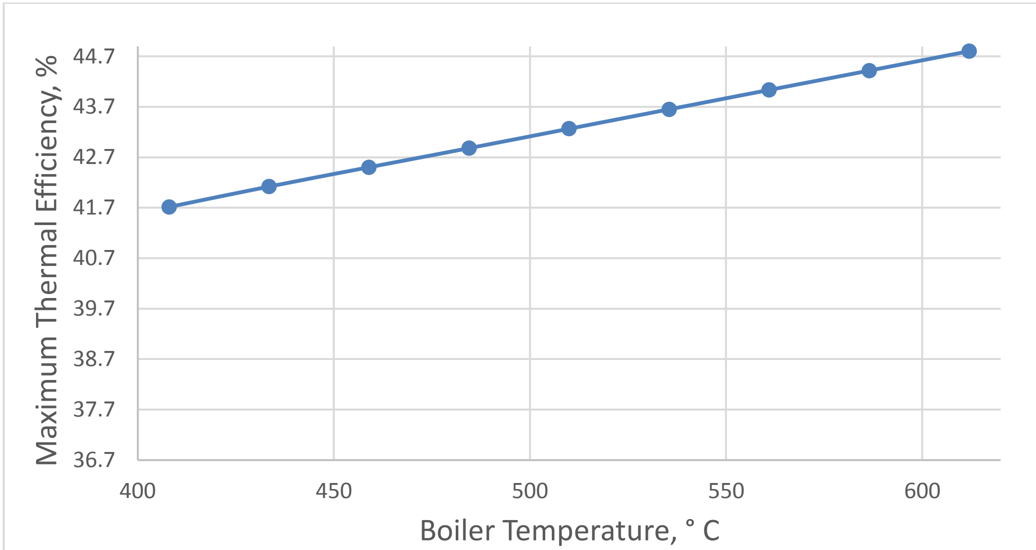

Figure 4.17. Maximum Thermal Efficiency vs. Boiler Temperature in the Ideal Single-Staged Regenerative Rankine Cycle

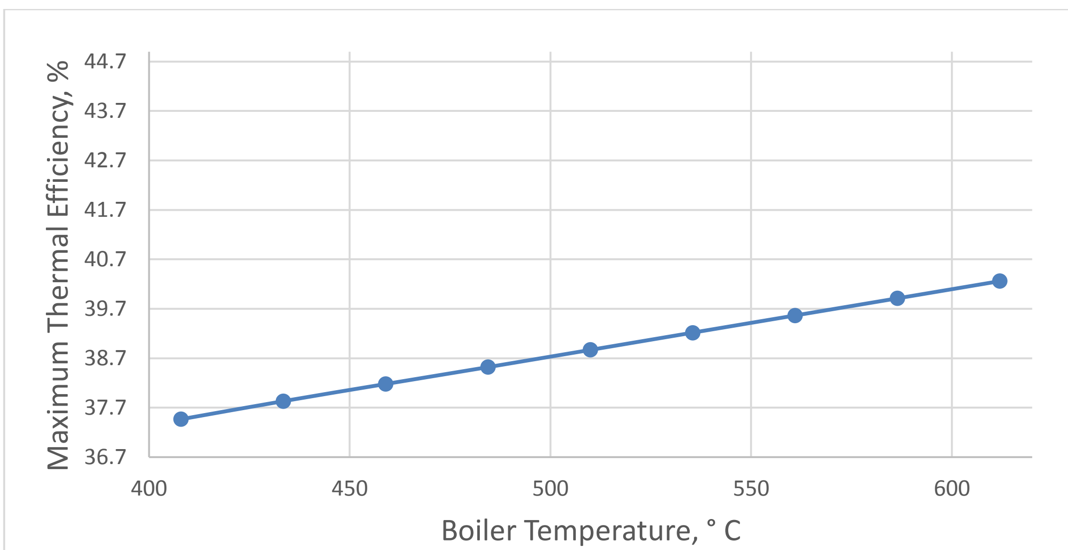

Figure 4.18. Maximum Thermal Efficiency vs. Boiler Temperature in the Actual Single-Staged Regenerative Rankine Cycle

4.1.4 Pump Efficiency

Figure 4.19 plots the cycle thermal efficiency vs. DEA pressure for 3 different isentropic efficiencies of pumps. As expected the change in the thermal efficiency is low as the pump energy consumption because the increase or decrease in the pump efficiency is not very high. The efficiency curves almost overlap each other. Figure 4.20 plots the optimum DEA pressure vs. pump efficiency which shows there is no change in optimum DEA pressure. Similarly Fig 4.21 plots the thermal efficiency vs. pump efficiency which shows that the thermal efficiency change is almost negligible when the pump efficiency changes.

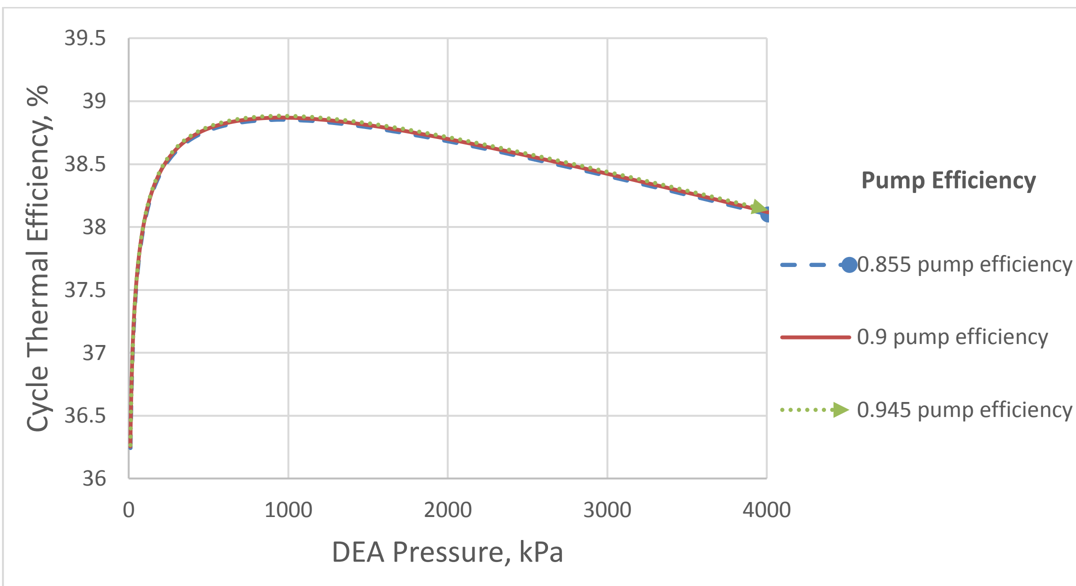

Figure 4.19. Thermal Efficiency vs. DEA Pressure at Different Pump Efficiencies for the Actual Single-Staged Regenerative Rankine Cycle

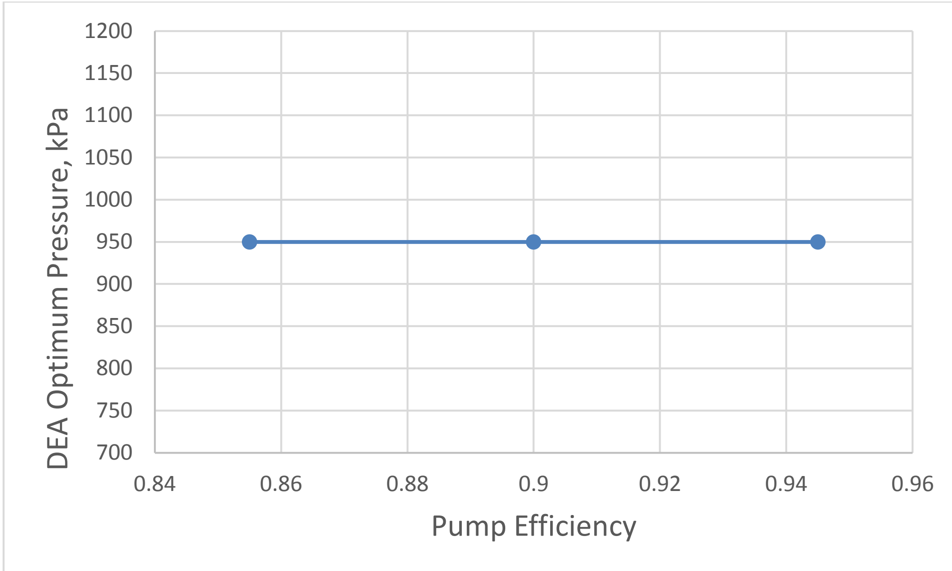

Figure 4.20. Optimum DEA Pressure vs. Pump Efficiency in the Actual Single-Staged Regenerative Rankine Cycle

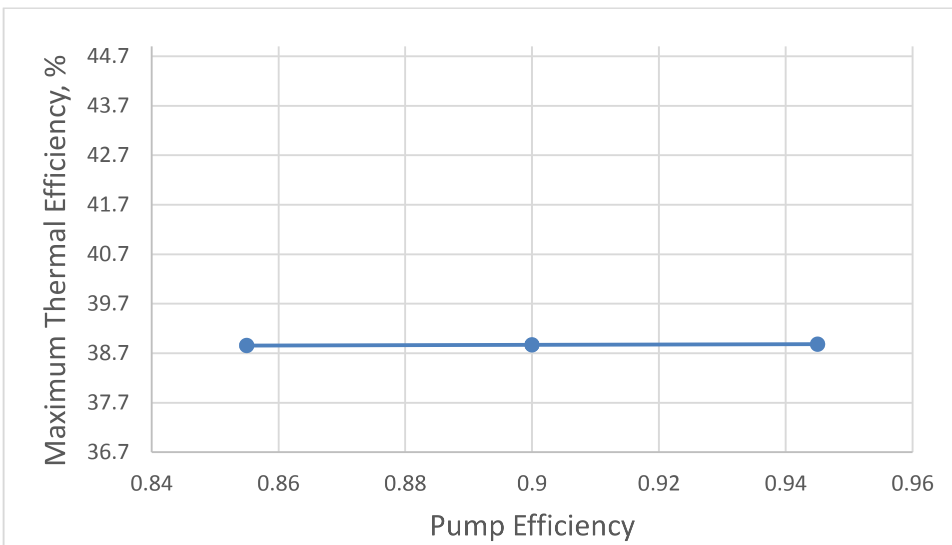

Figure 4.21. Maximum Thermal Efficiency vs. Pump Efficiency in the Actual Single-Staged Regenerative Rankine Cycle

4.1.5 Turbine Efficiency

In figure 4.22 thermal efficiency versus DEA pressure in 3 different turbine efficiencies are shown. In that figure contrary to the figure 4.19 for pump efficiency, the three curves do not overlap each other which means turbine efficiency has a huge effect on thermal efficiency (refer to figure 4.24) and no effect on DEA optimum pressure for one feedwater heater (refer to figure 4.23)

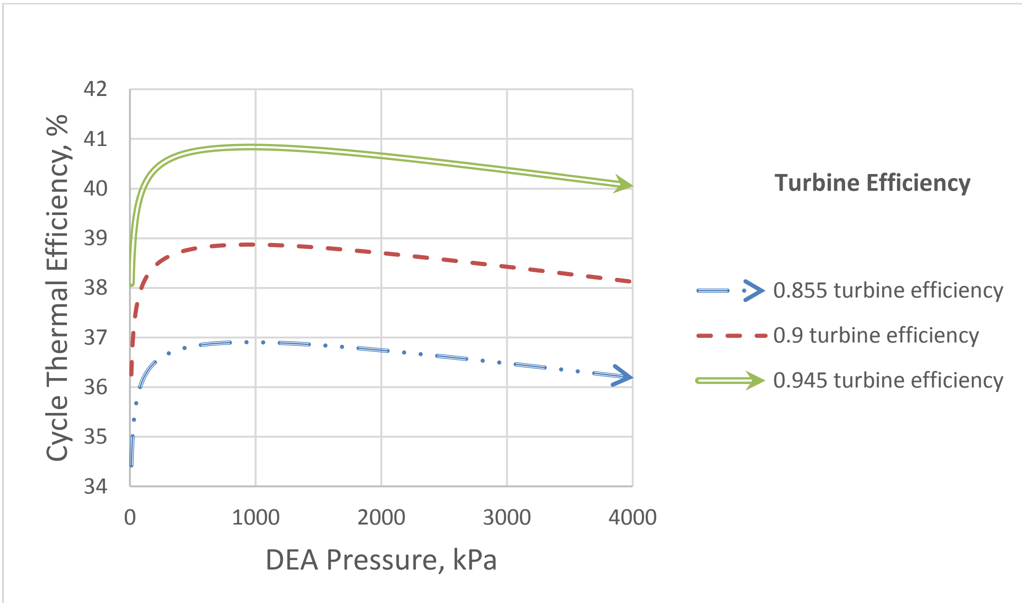

Figure 4.22. Thermal Efficiency vs. DEA Pressure at Different Turbine Efficiencies for the Actual Single-Staged Regenerative Rankine Cycle

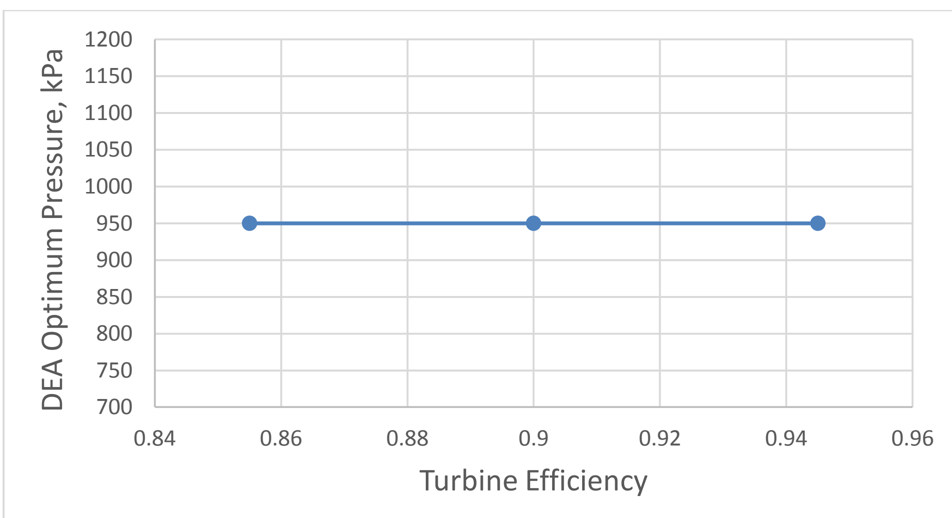

Figure 4.23. Optimum DEA Pressure vs. Turbine Efficiency in Actual Single-Staged Regenerative Rankine Cycle

In figure 4.24 as it was expected, the efficiency is raised from 36.39 to 40.27 percent by increasing turbine efficiency from 0.855 to 0.945.

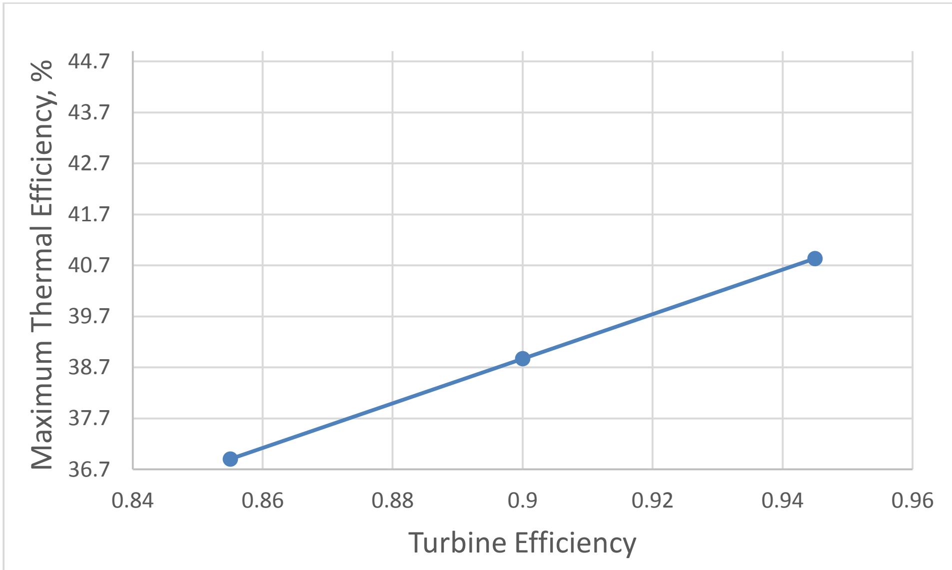

Figure 4.24. Maximum Thermal Efficiency vs. Turbine Efficiency in Actual Single-Staged Regenerative Rankine Cycle

The obtained values for one FWH has been shown in Table A.1 and Table A.2 for ideal and actual cycles respectively.

4.2 Regenerative Rankine Cycle with Two Feedwater Heaters

A regenerative Rankine cycle with 2 feedwater heaters has one closed FWH and one open FWH (Deaerator). As mentioned in chapter 3, the iteration method has been used, it means that for getting closed FWH data, optimized values for open FWH are employed and vice-versa for obtaining open FWH data, optimized values for closed FWH are employed. The effect of boiler pressure, condenser pressure, boiler temperature, pump and turbine isentropic efficiencies on the cycle thermal efficiency and the optimum placement of the feedwater heaters are presented in the following sub-sections.

4.2.1 Boiler Pressure

Figures 4.25 and 4.26 illustrate the relation between efficiency versus closed feedwater heater pressure which applied in 9 different boiler pressures for ideal and actual cycles. The lowest curve shows the lowest boiler pressure (8000 kPa) which has the lowest efficiency and the lowest closed feedwater heater (CFWH) optimum pressure. On the other hand, the highest curve illustrate the highest boiler pressure (12000 kPa) which has the highest efficiency and the highest closed feedwater heater (CFWH) optimum pressure. (As it was mentioned before in chapter 3, optimum feedwater heater pressure is where the efficiency is maximum on the curve. This is also known for optimal placement of (opened or closed) feedwater heater).

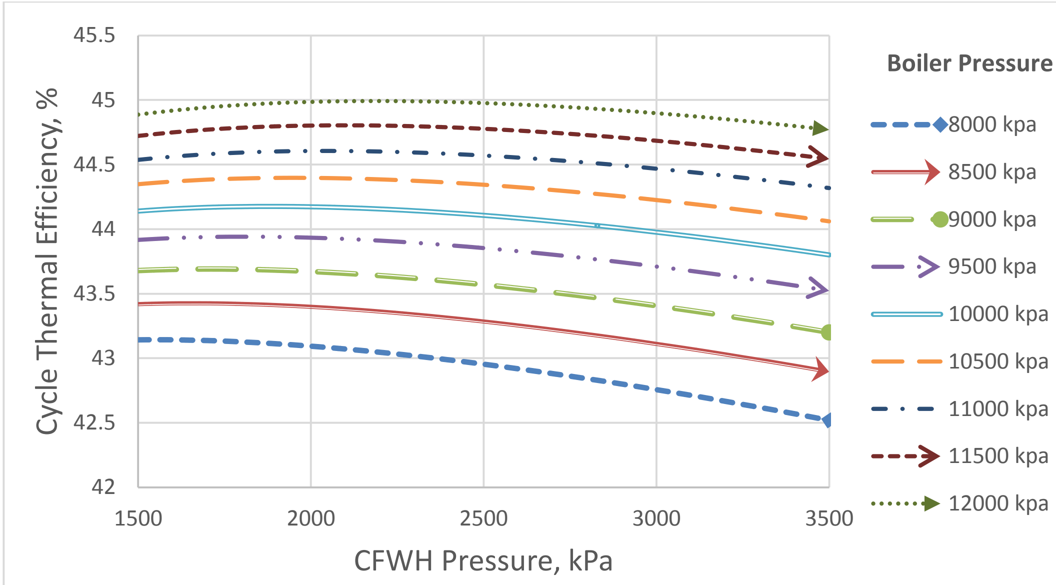

Figure 4.25. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Boiler Pressures for the Ideal Regenerative Rankine Cycle with 2 FWHs

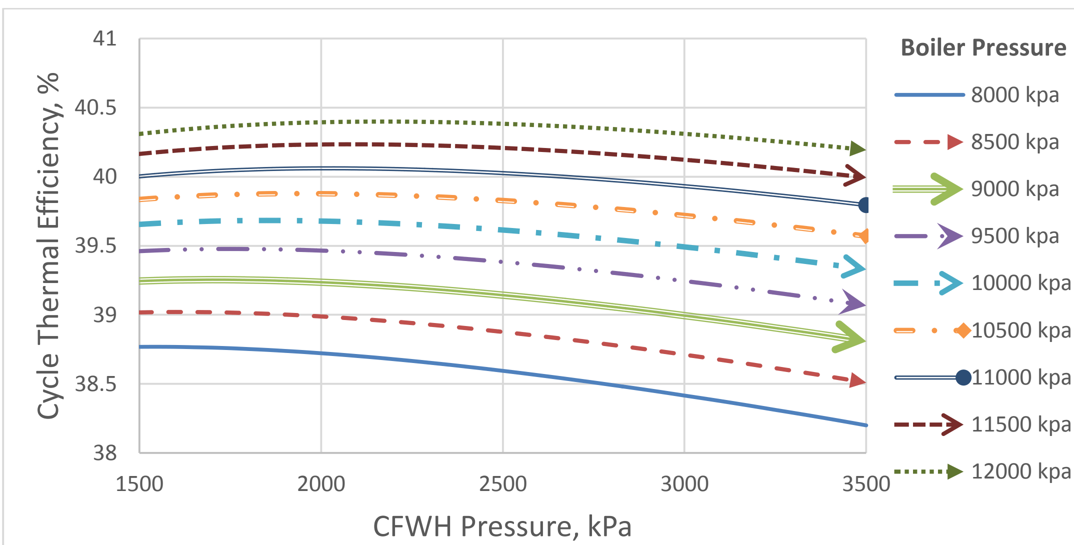

Figure 4.26. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Boiler Pressures for the Actual Regenerative Rankine Cycle with 2 FWHs

In figures 4.27 and 4.28 as mentioned above, by increasing boiler pressure, optimum closed feedwater heater pressure rises for both ideal and actual cycles. The points on the curve correspond to the optimal placement of the closed feedwater heater pressure for 9 different boiler pressures from 8000 to 12000 kPa.

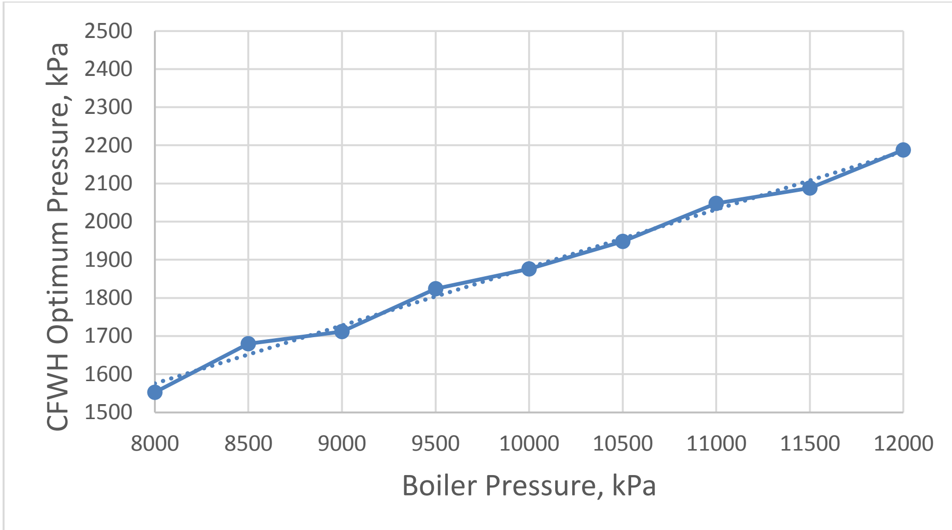

Figure 4.27. Optimum Closed Feedwater Heater Pressure vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

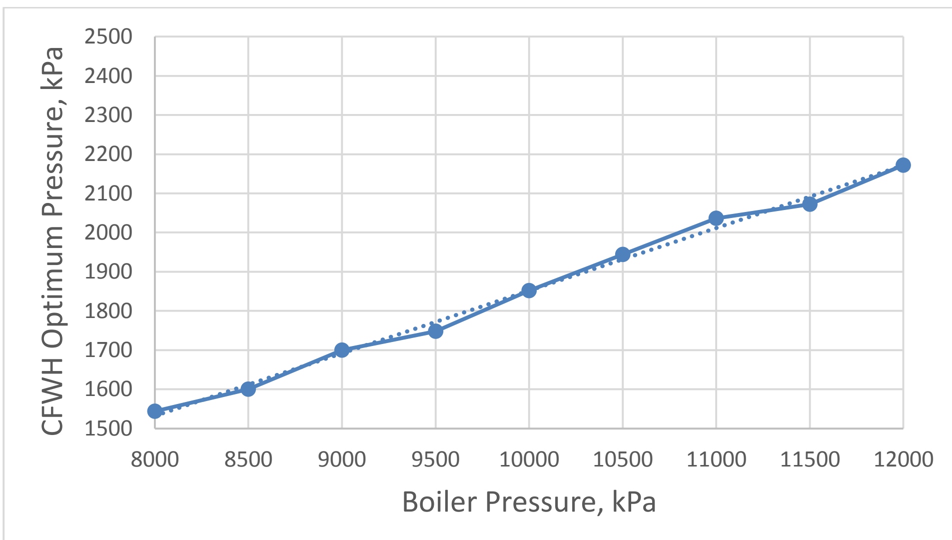

Figure 4.28. Optimum Closed Feedwater Heater Pressure vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

In figures 4.29 and 4.30 the cycle thermal efficiency versus DEA pressure has been shown in various boiler pressures. The highest curve shows the highest boiler pressure (12000 kPa). It has also the highest efficiency and optimum DEA pressure amongst these 9 different boiler pressures.

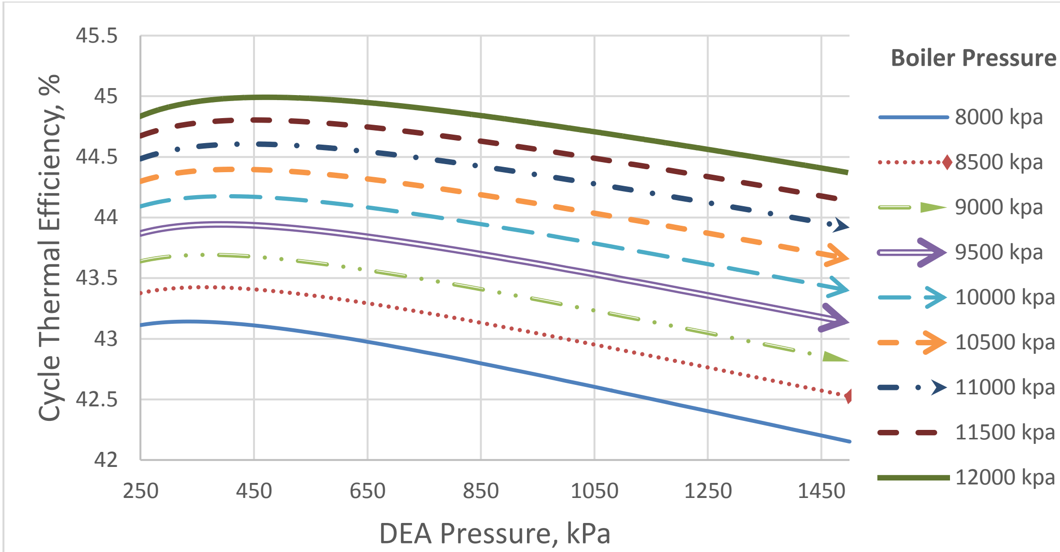

Figure 4.29. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Ideal Regenerative Rankine Cycle with 2 FWHs

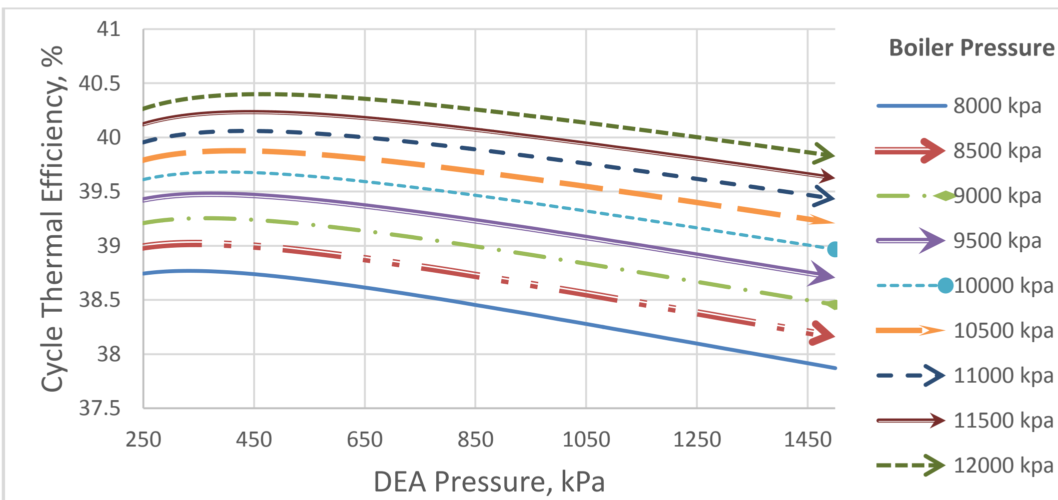

Figure 4.30. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Actual Regenerative Rankine Cycle with 2 FWHs

In figures 4.31 and 4.32 increasing boiler pressure causes rising in the DEA optimum pressure and efficiency as well (refer to figures 4.33 and 4.34). The points on the curve illustrate the optimal placement of the deaerator pressure (open feedwater heater pressure) calculated for 9 different values of the boiler pressure.

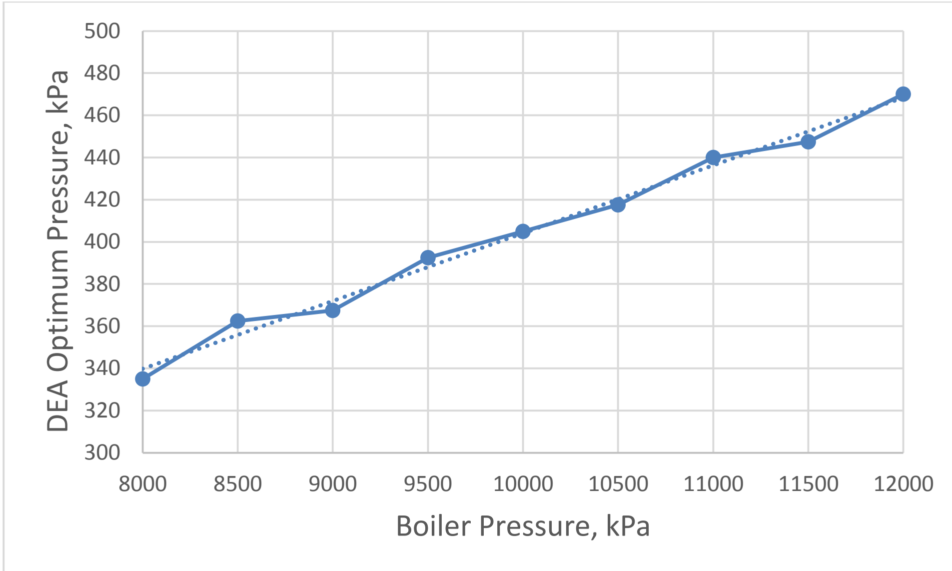

Figure 4.31. Optimum DEA Pressure vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

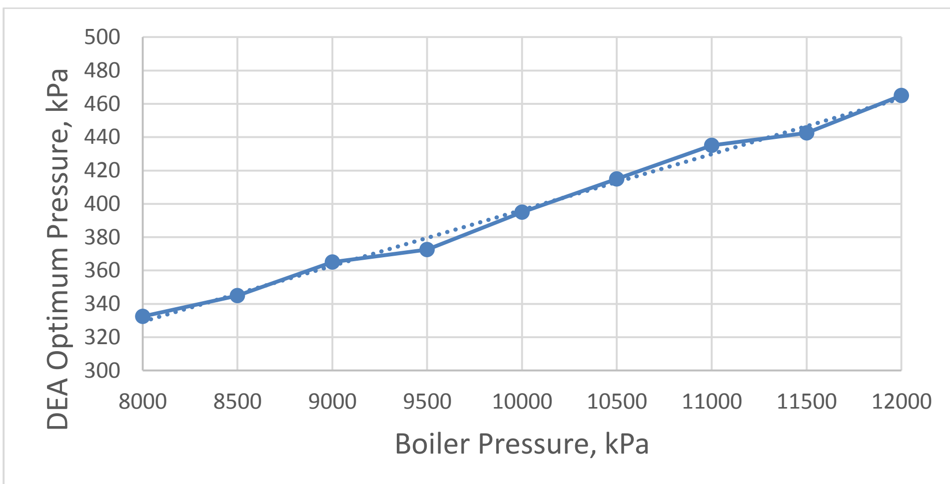

Figure 4.32. Optimum DEA Pressure vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

Figures 4.33 and 4.34 exactly shows how much thermal efficiency increases by increasing the boiler pressure. As can be seen, the efficiency rises from 43.14 percent for 8000 kPa boiler pressure to 44.99 percent for 12000 kPa boiler pressure.

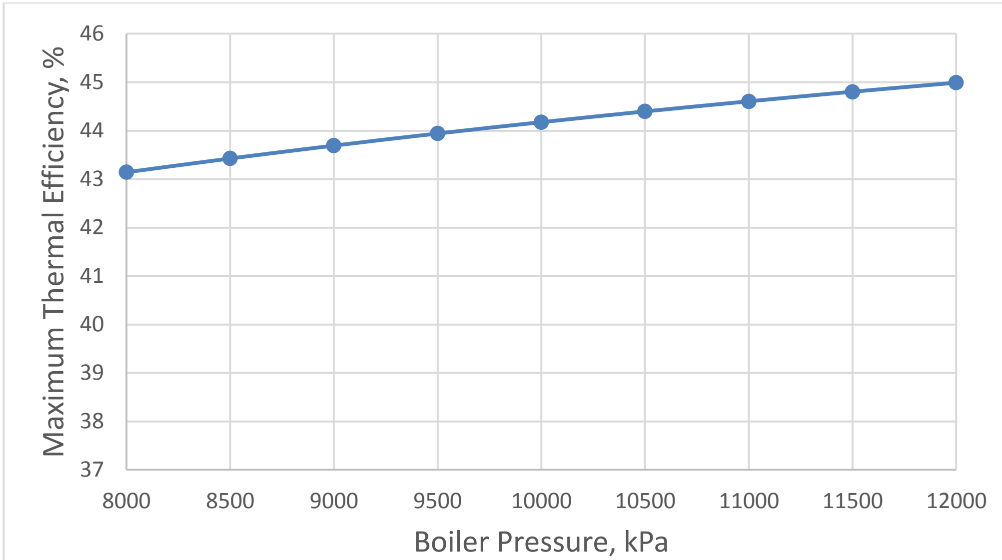

Figure 4.33. Maximum Thermal Efficiency vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

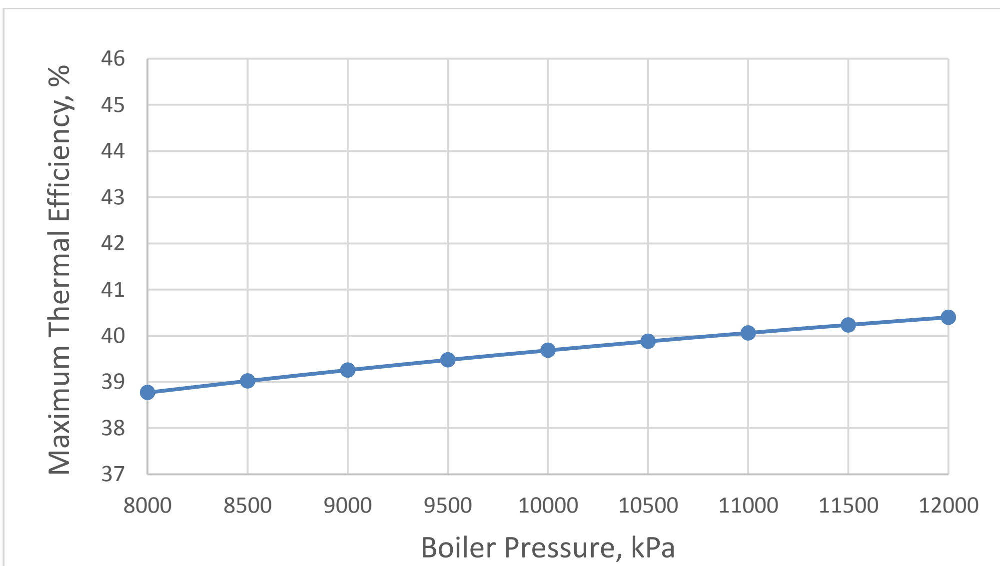

Figure 4.34. Maximum Thermal Efficiency vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

4.2.2 Condenser Pressure

In figures 4.35 and 4.36 cycle thermal efficiency versus closed feedwater heater in various condenser pressures are illustrated. It is obvious that the highest curve (8 kPa) has the highest efficiency but the lowest closed feedwater heater optimal pressure among 9 curves (from 8 to 12 kPa). On the other hand, 12 kPa curve is classified as the lowest efficiency and highest closed feedwater heater optimal pressure between the curves.

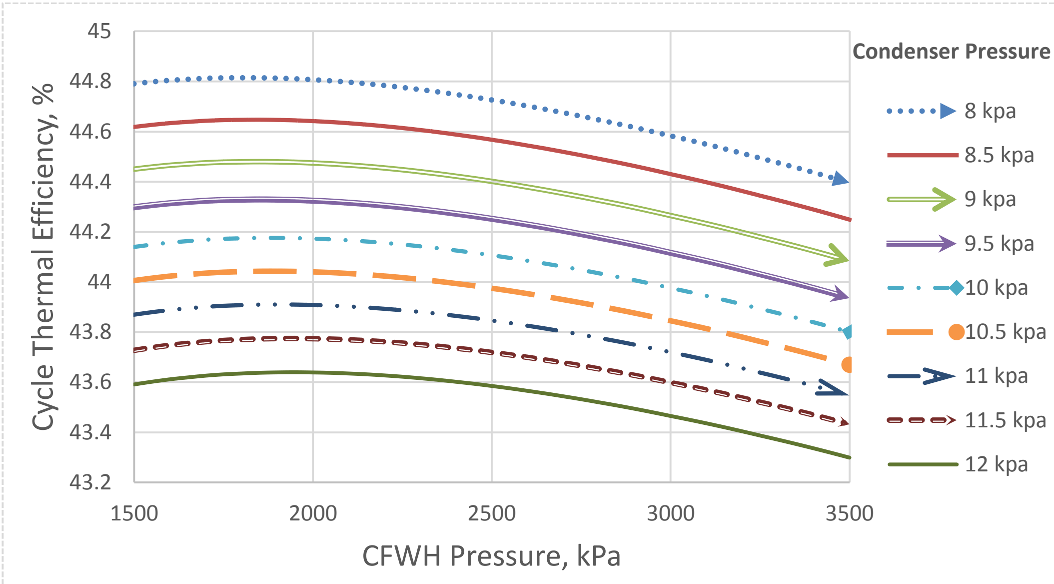

Figure 4.35. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Condenser Pressures for the Ideal Regenerative Rankine Cycle with 2 FWHs

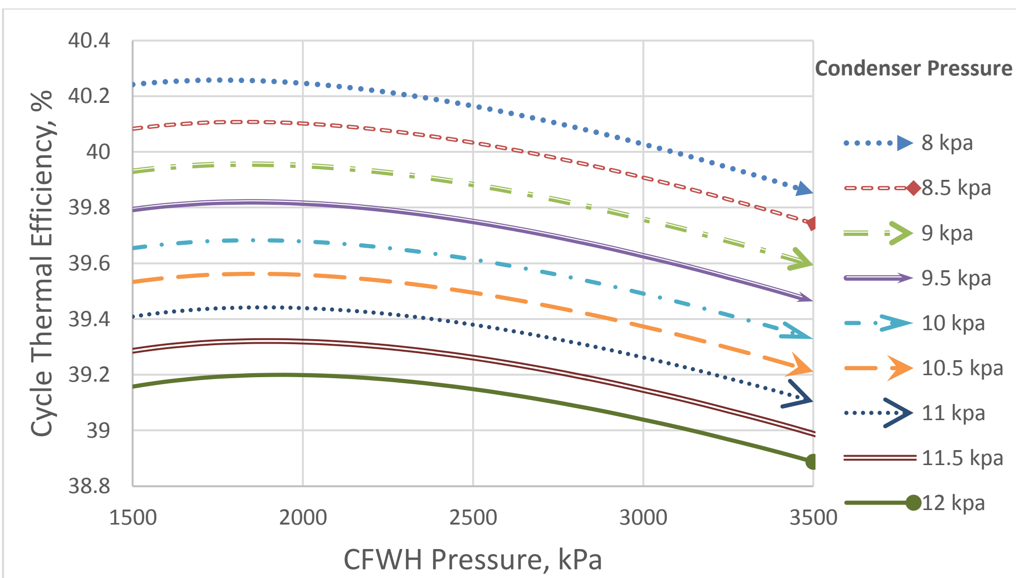

Figure 4.36. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Condenser Pressures for the Actual Regenerative Rankine Cycle with 2 FWHs

Figures 4.37 and 4.38 shows that closed feedwater heater optimum pressure increases with a slight slope by rising condenser pressure.

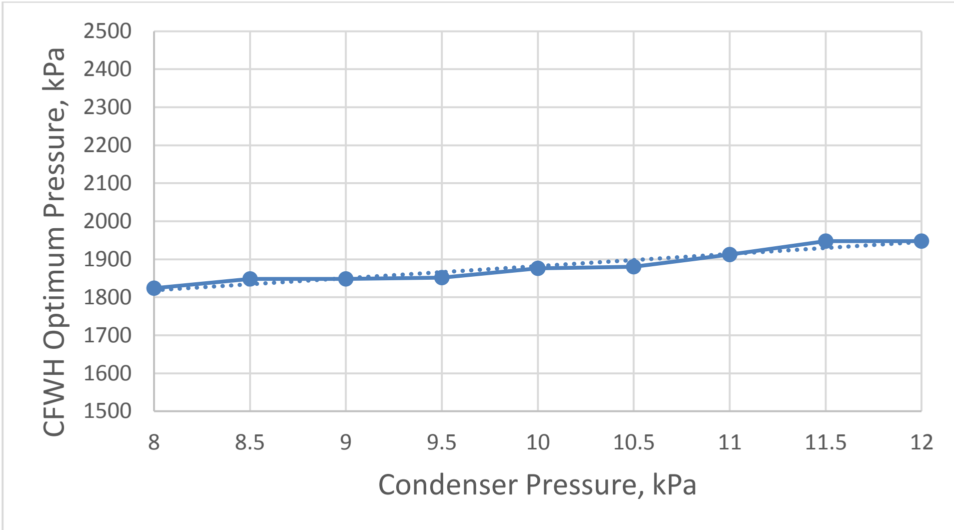

Figure 4.37. Optimum Closed Feedwater Heater Pressure vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

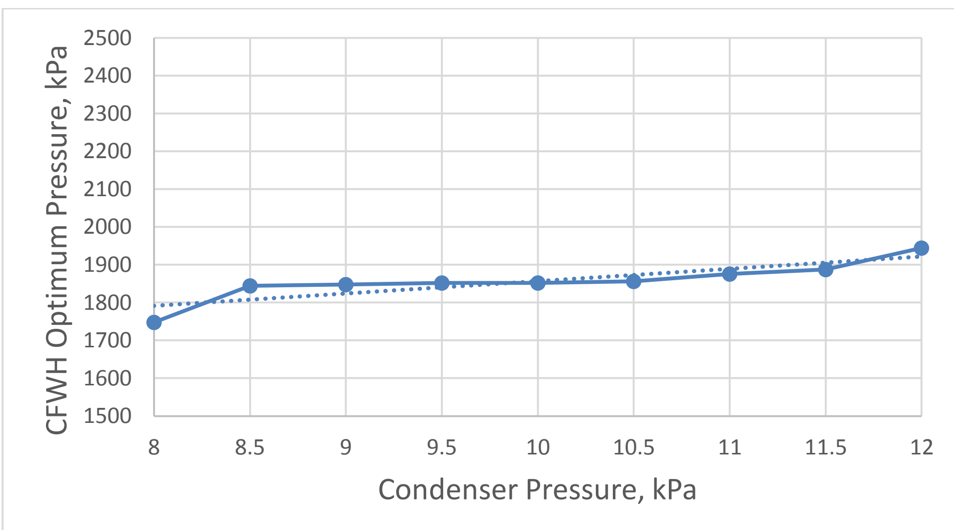

Figure 4.38. Optimum Closed Feedwater Heater Pressure vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

In figures 4.39 and 4.40 the efficiency versus DEA (deaerator) pressure is illustrated from 8 until 12 kPa for both ideal and actual cycles. Similar to the efficiency versus CFWH pressure graph the highest efficiency belongs to the 8 kPa curve and 12 kPa curve denotes for the lowest efficiency. In the figures 4.41 and 4.42, DEA Optimum Pressure increases almost gradually when the condenser pressure goes from 8 to 12 kPa for both actual and ideal cycles.

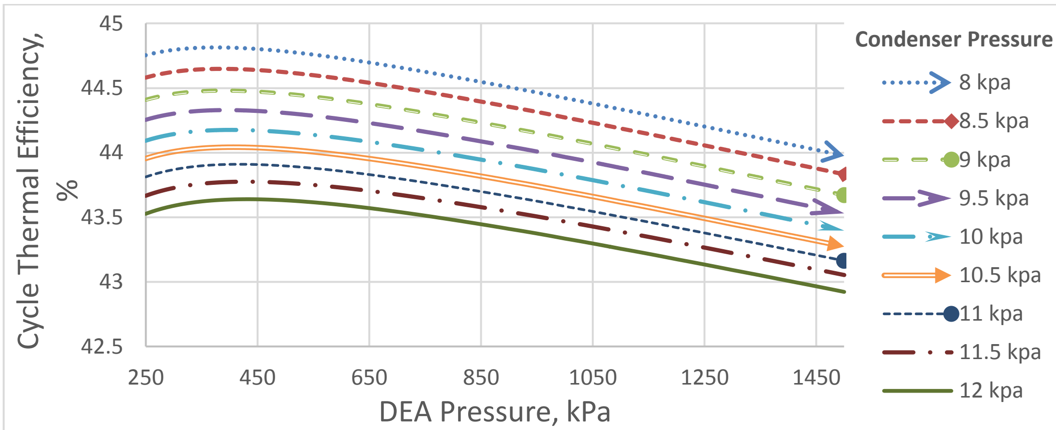

Figure 4.39. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Ideal Regenerative Rankine Cycle with 2 FWHs

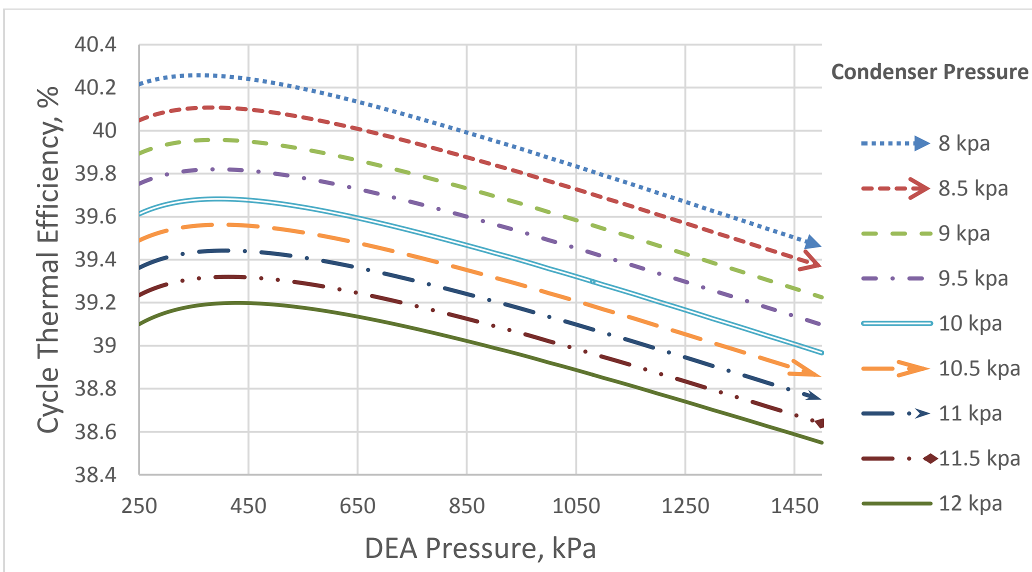

Figure 4.40. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Actual Regenerative Rankine Cycle with 2 FWHs

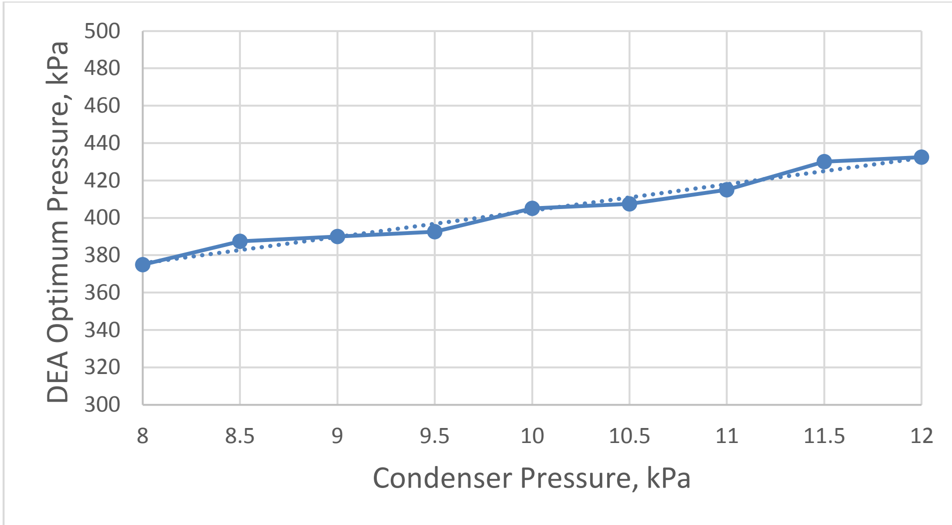

Figure 4.41. Optimum DEA Pressure vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

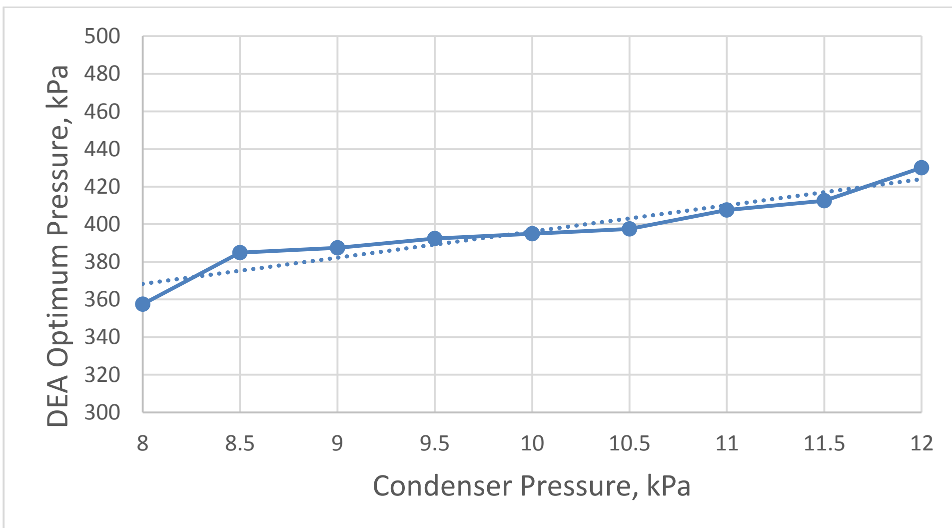

Figure 4.42. Optimum DEA Pressure vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

In figures 4.43 and 4.44 the relation between efficiency and condenser pressure is more distinguishable than above charts. As mentioned before, there is a reverse relation between efficiency and condenser pressure. Thermal efficiency decreases by increasing condenser pressure.

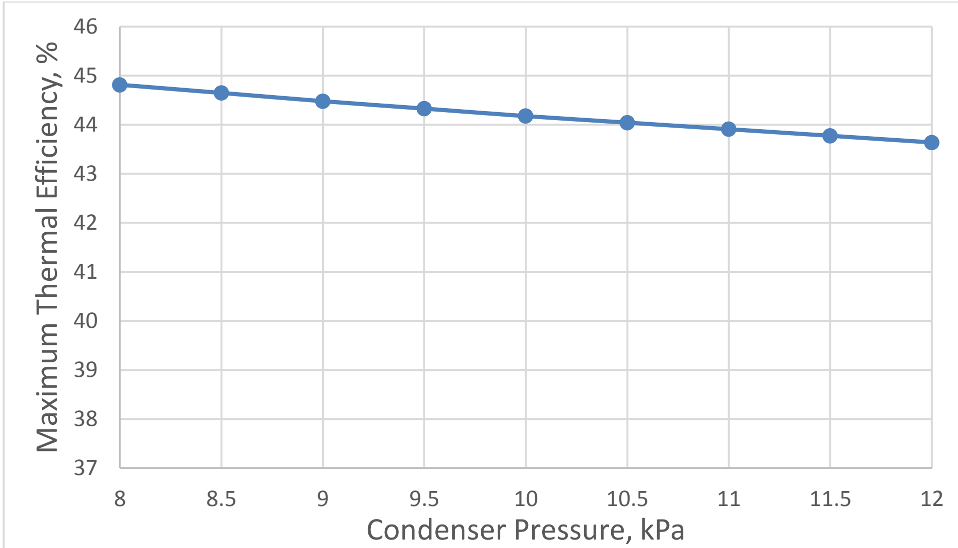

Figure 4.43. Maximum Thermal Efficiency vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 2 FWHs

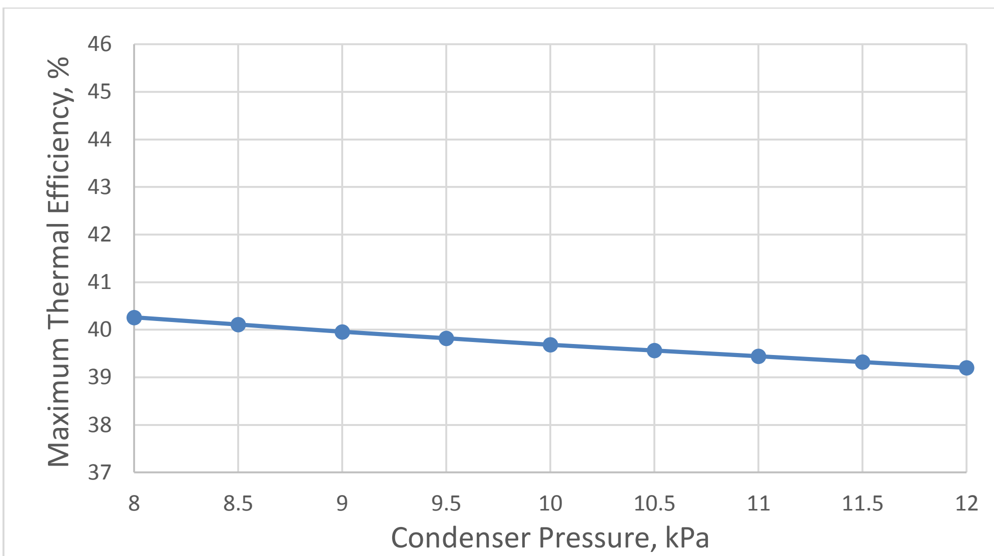

Figure 4.44. Maximum Thermal Efficiency vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 2 FWHs

4.2.3 Boiler Temperature

Figures 4.45 and 4.46 show the cycle thermal efficiency versus closed feedwater heater pressure in different boiler temperatures from 408° C to 612° C. As it is clear the highest efficiency belongs to 612° C. Obviously by increasing boiler temperature, thermal efficiency increases and closed feedwater heater optimal pressure also increases which are illustrated in figures 4.47 and 4.48.

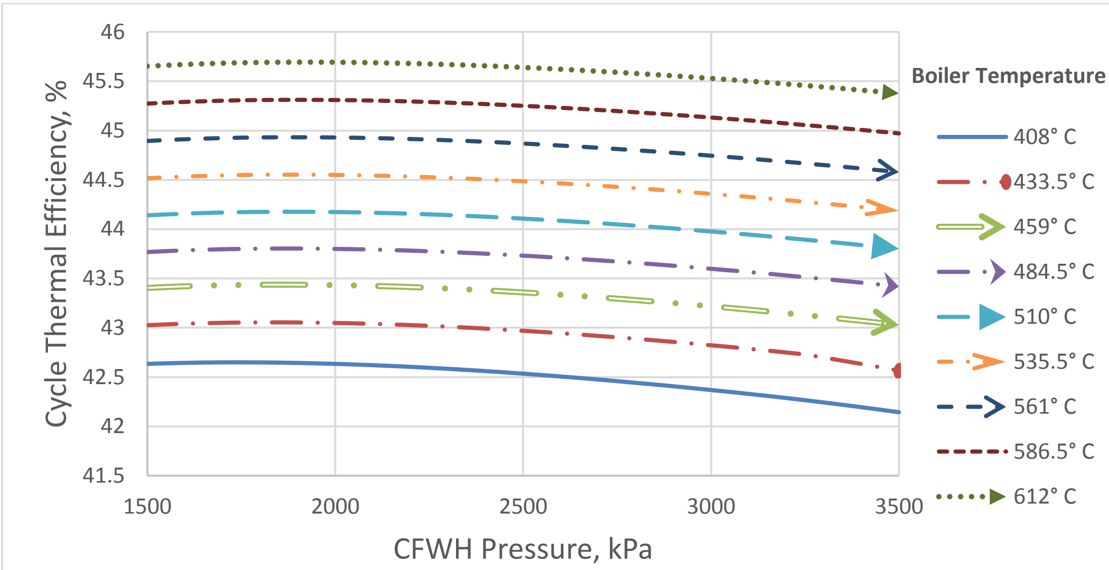

Figure 4.45. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Boiler Temperatures for the Ideal Regenerative Rankine Cycle with 2 FWHs

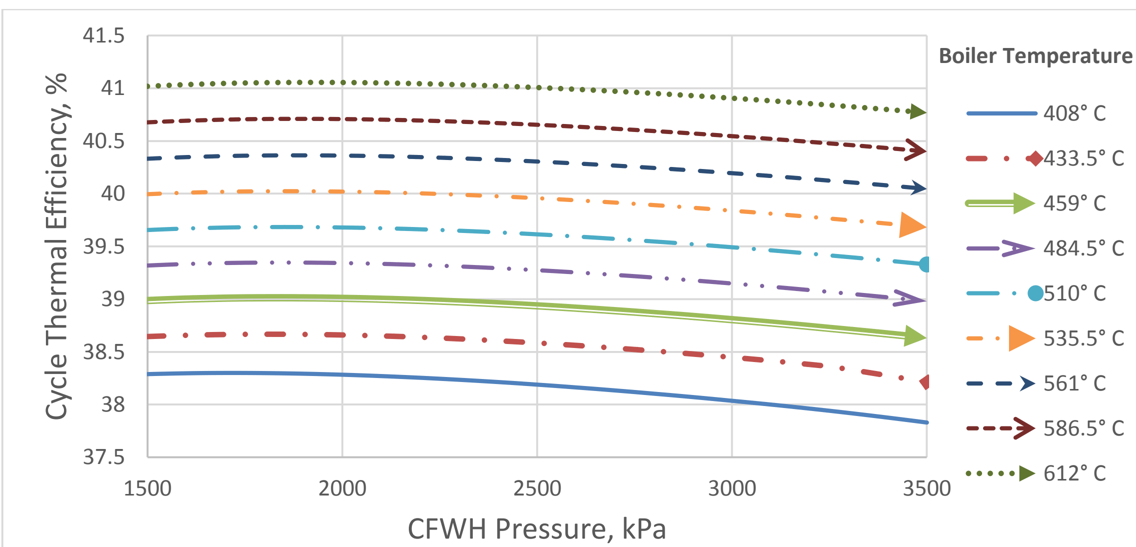

Figure 4.46. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Boiler Temperatures for the Actual Regenerative Rankine Cycle with 2 FWHs

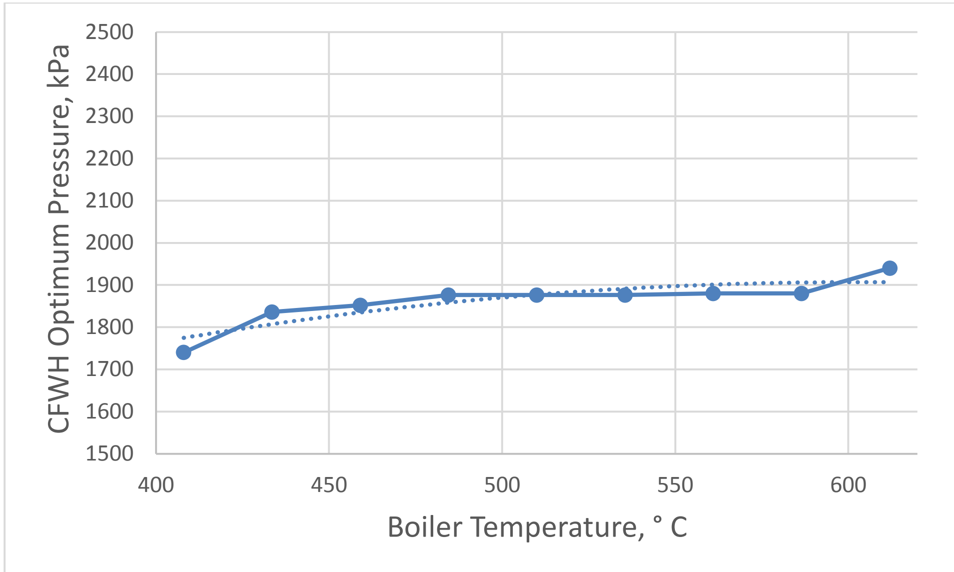

Figure 4.47. Optimum Closed Feedwater Heater Pressure vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 2 FWHs

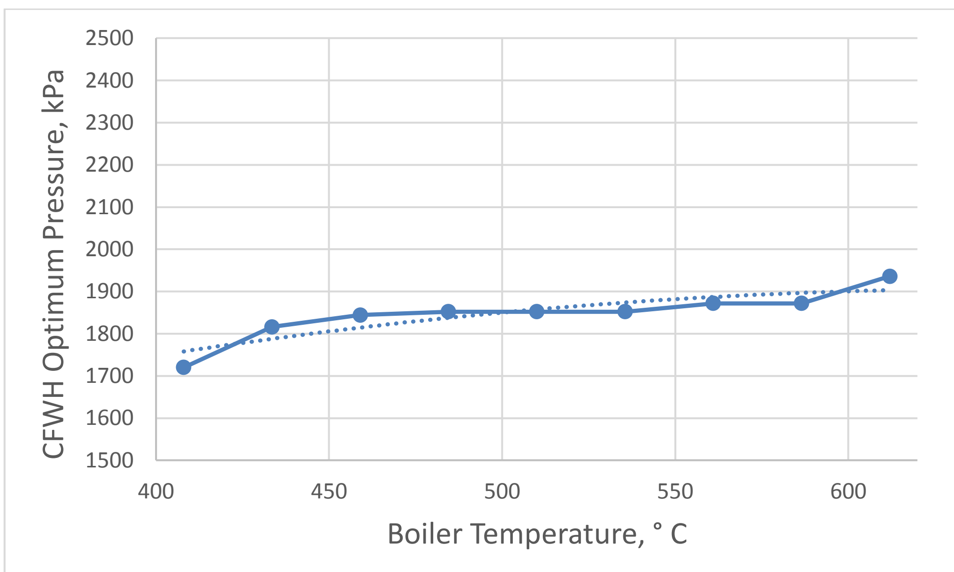

Figure 4.48. Optimum Closed Feedwater Heater Pressure vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 2 FWHs

Figures 4.49 and 4.50 illustrate the cycle thermal efficiency versus DEA (deaerator) pressure for 9 different boiler temperatures (408° C to 612° C). It is completely noticeable that by rising the boiler temperature, thermal efficiency increases, however deaerator optimal pressure increases very slightly for both ideal and actual cycles. This matter is clearer in figures 4.51 and 4.52.

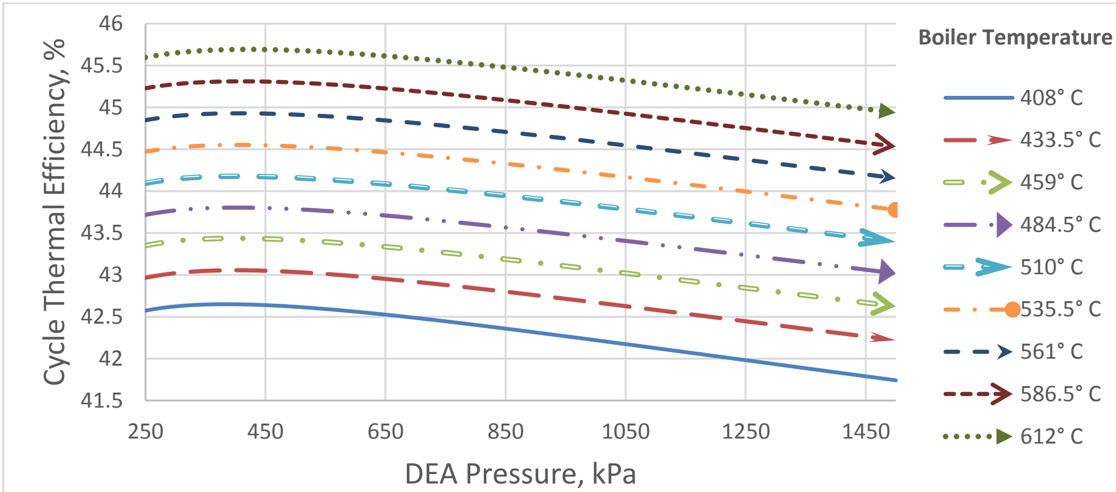

Figure 4.49. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Ideal Regenerative Rankine Cycle with 2 FWHs

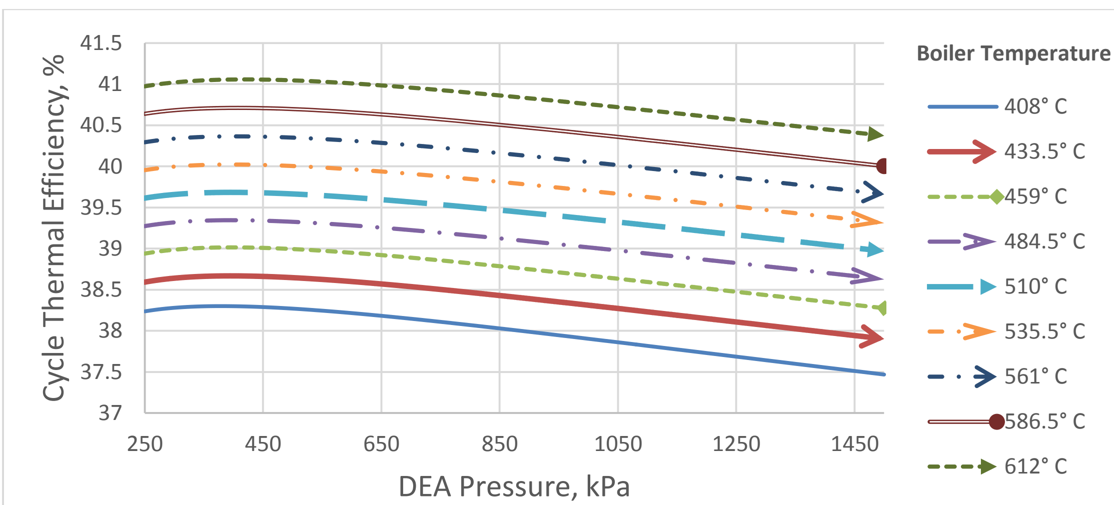

Figure 4.50. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Actual Regenerative Rankine Cycle with 2 FWHs

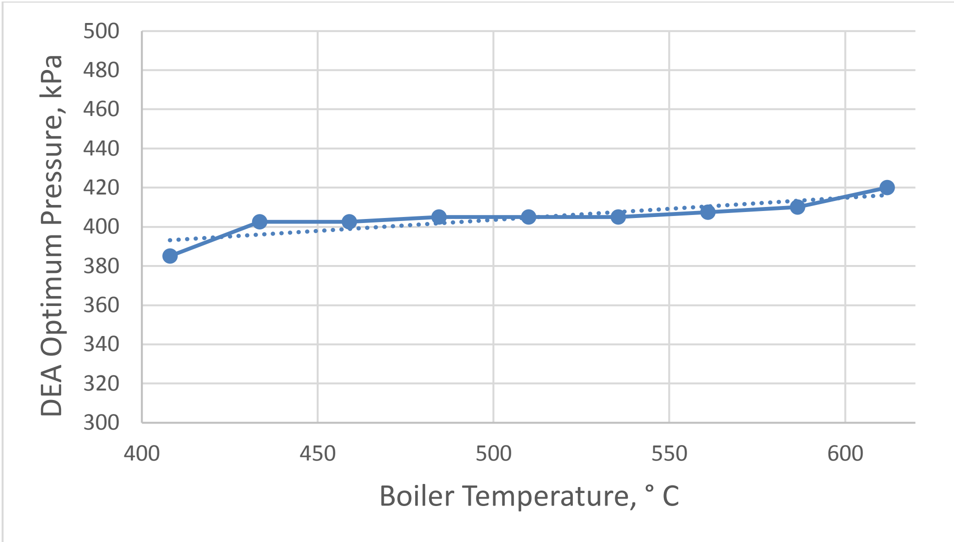

Figure 4.51. Optimum DEA Pressure vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 2 FWHs

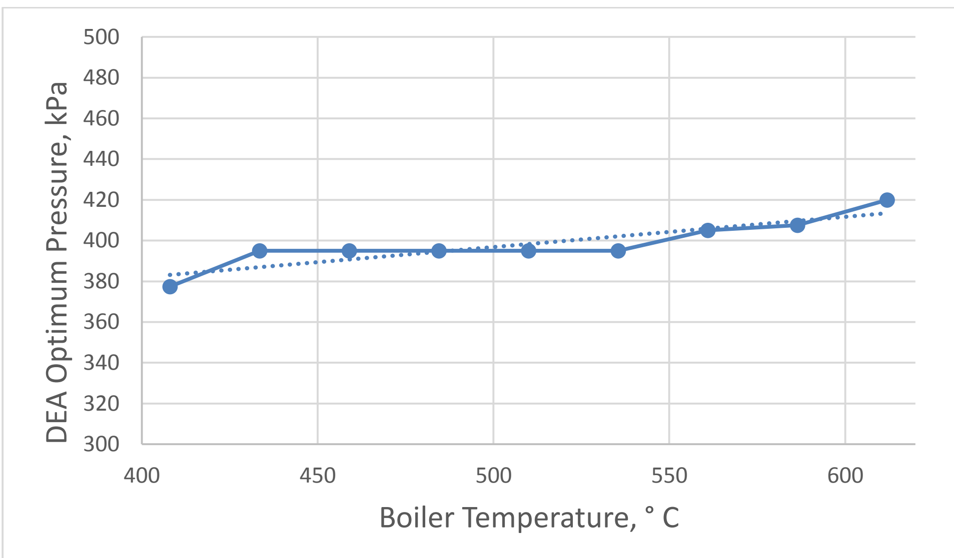

Figure 4.52. Optimum DEA Pressure vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 2 FWHs

The exact values for thermal efficiency by changing the boiler temperature are drawn in figures 4.53 and 4.54. As described above, the efficiency increases by raising the boiler temperature from about 42.65 to 45.69 percent for ideal cycle and from around 38.30 to 41.05 percent for actual cycle.

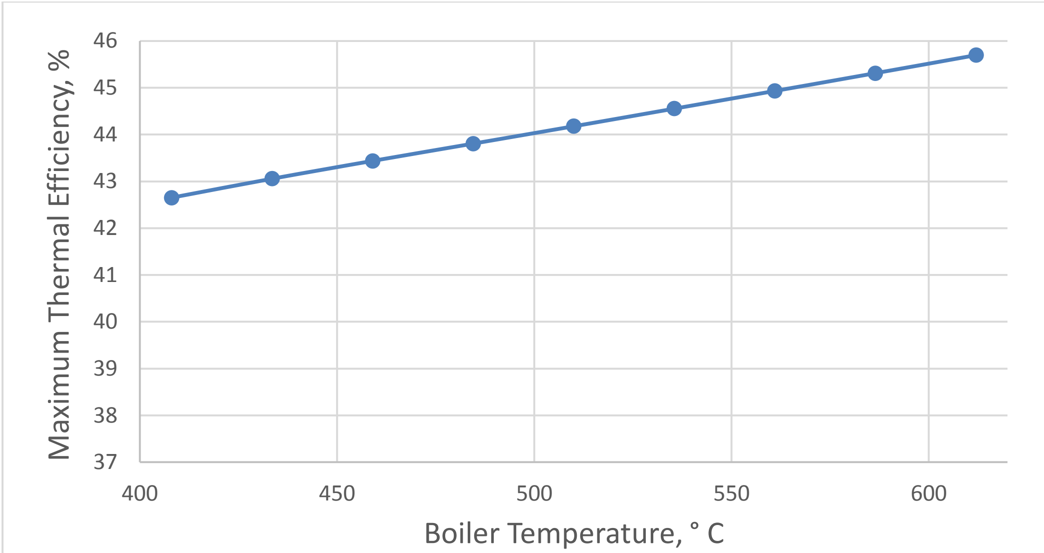

Figure 4.53. Maximum Thermal Efficiency vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 2 FWHs

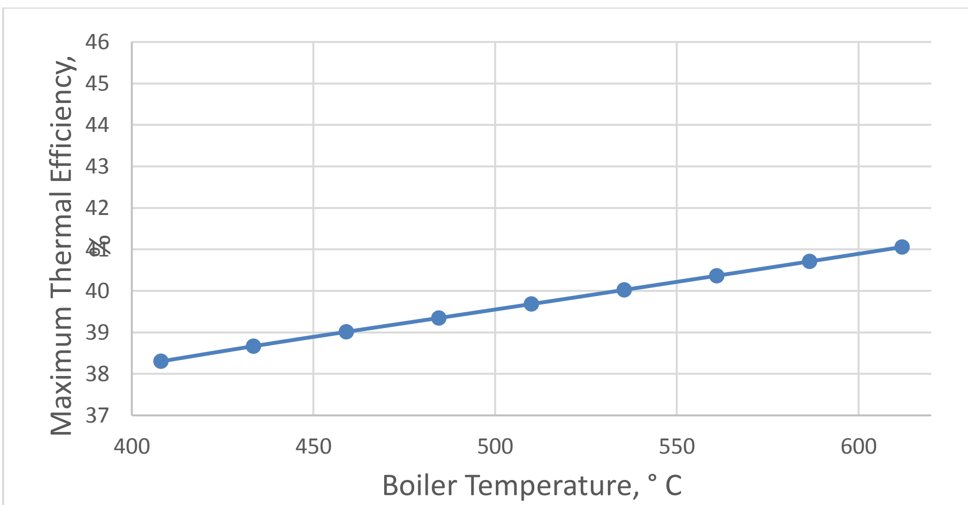

Figure 4.54. Maximum Thermal Efficiency vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 2 FWHs

4.2.4 Pump Efficiency

In figure 4.55 three different pump efficiencies has been considered. It shows the relation between cycle thermal efficiency versus closed feedwater heater. Although three curves do not completely overlap each other but the change is very slight in thermal efficiency, however no change in the closed feedwater heater optimal pressure.

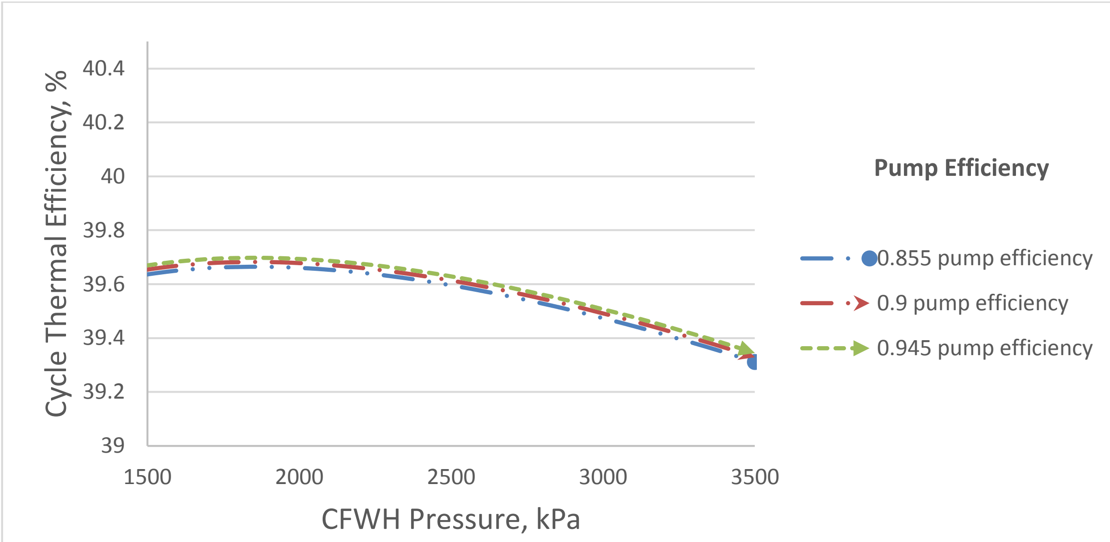

Figure 4.55. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Pump Efficiencies for the Actual Regenerative Rankine Cycle with 2 FWHs

Closed feedwater heater optimal pressure in three different pump efficiencies are equivalent to 1852 kPa (see figure 4.56).

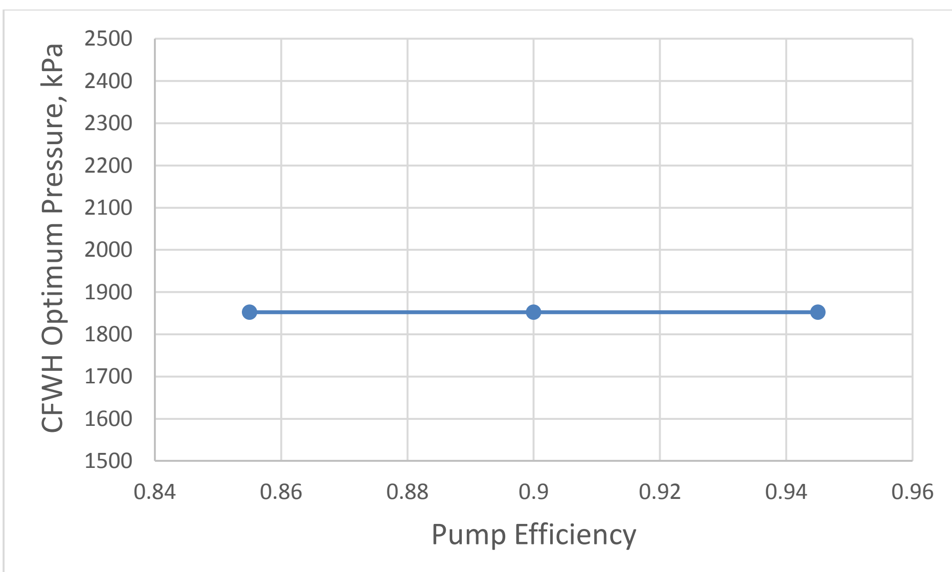

Figure 4.56. Optimum Closed Feedwater Heater Pressure vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

Figure 4.57 illustrates thermal efficiency versus Dea pressure. As mentioned before, the efficiency increases as pump efficiency rises from 0.855 to 0.945, however the dea optimum pressures are considered as constant.

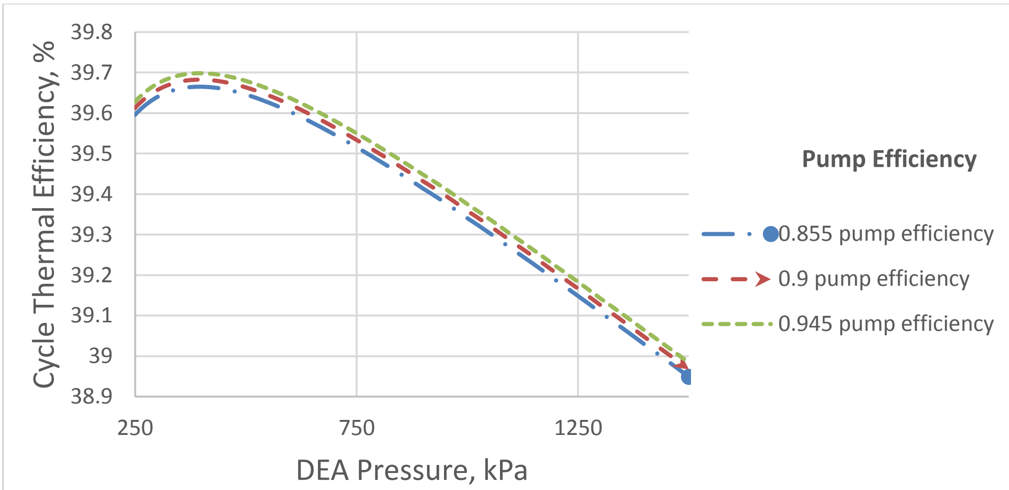

Figure 4.57. Thermal Efficiency vs. DEA Pressure at Different Pump Efficiencies for the Actual Regenerative Rankine Cycle with 2 FWHs

Deaerator optimum pressure can be considered constant (395 kPa) when pump efficiency varies from 0.855 to 0.945 (refer to 4.58).

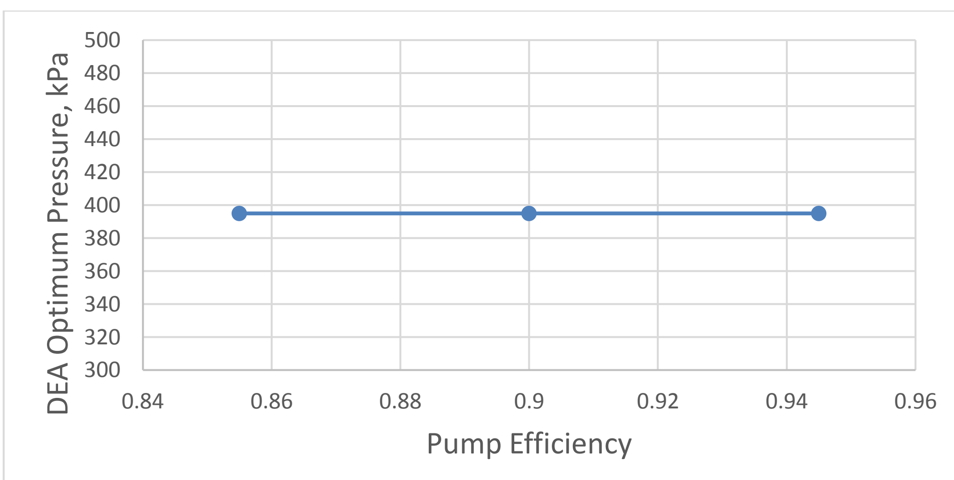

Figure 4.58. Optimum DEA Pressure vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

When the pump efficiency changes from 0.855 to 0.9 or changes from 0.9 to 0.945, the overall thermal efficiency varies only 0.04%. Therefore pump efficiency does not have much effect on the overall cycle efficiency. This matter is illustrated by figure 4.59.

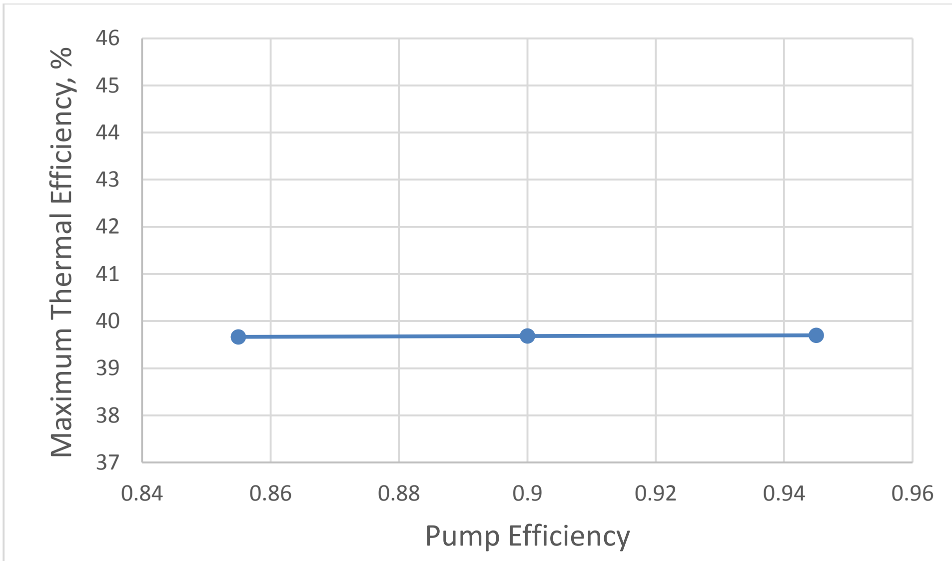

Figure 4.59. Maximum Thermal Efficiency vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

4.2.5 Turbine Efficiency

Contrary to the pump efficiency, which has a slight effect on cycle thermal efficiency, turbine efficiency has a huge effect on cycle thermal efficiency. Figure 4.60 shows that for turbine efficiencies from 0.855 to 0.945, the thermal efficiency increases from around 37.67 to 41.69 %, however closed feedwater heater optimum pressures remain almost constant which is drawn in figure 4.61.

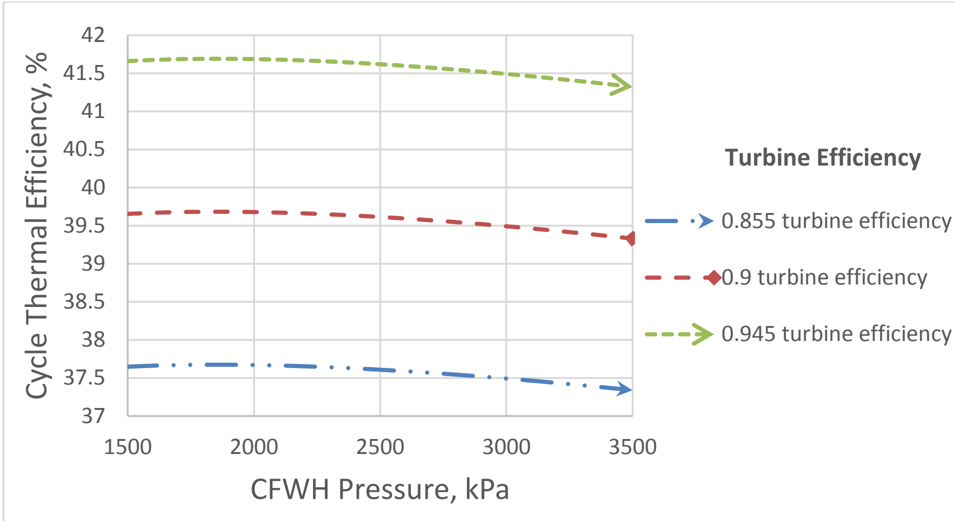

Figure 4.60. Thermal Efficiency vs. Closed Feedwater Heater Pressure at Different Turbine Efficiencies for the Actual Regenerative Rankine Cycle with 2 FWHs

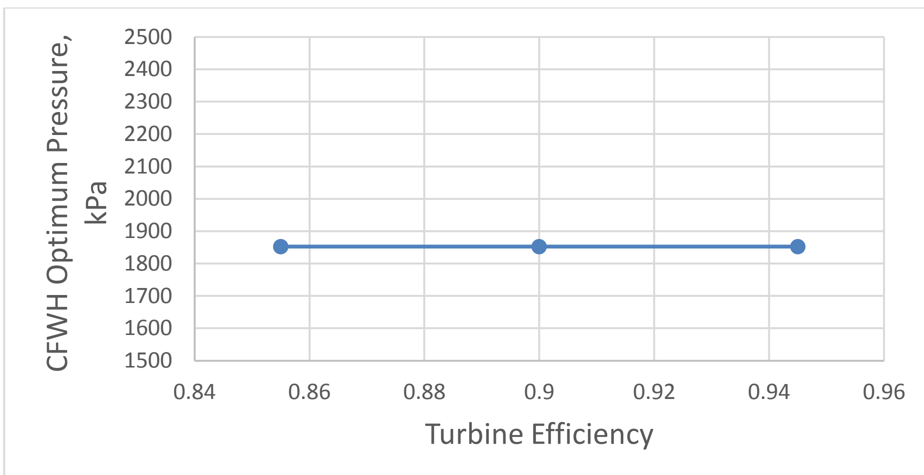

Figure 4.61. Optimum Closed Feedwater Heater Pressure vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

In figure 4.62 for three different turbine efficiencies, deaerator optimum pressure almost remains constant in spite of the fact that thermal efficiency increases. (Deaerator optimum pressure as mentioned before, is where the efficiency has the maximum amount on each curve.) This is completely observable in figure 4.63.

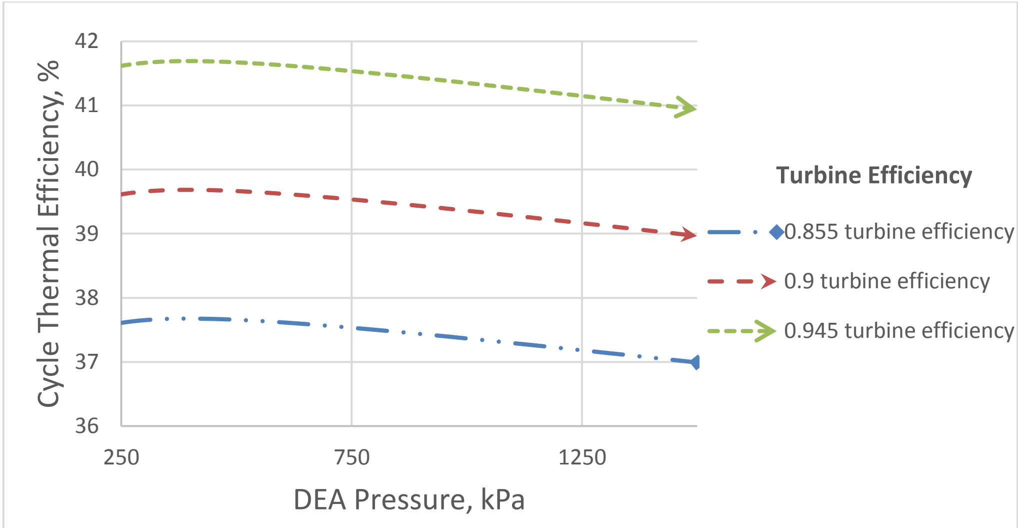

Figure 4.62. Thermal Efficiency vs. DEA Pressure at Different Turbine Efficiencies for the Actual Regenerative Rankine Cycle with 2 FWHs

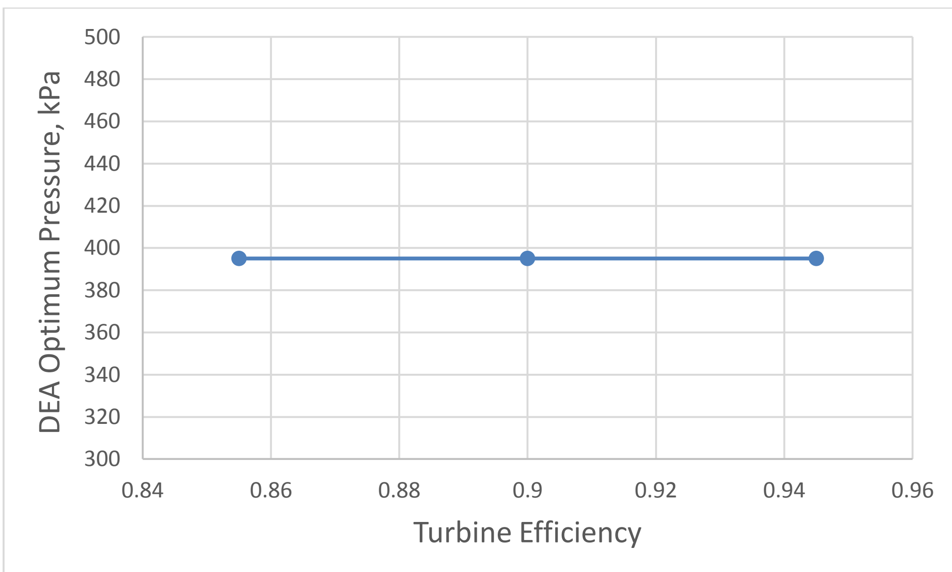

Figure 4.63. Optimum DEA Pressure vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

As can be seen in figure 4.64 turbine efficiency has a significant effect on thermal efficiency. It raised from 37.67 up to 41.69 percent.

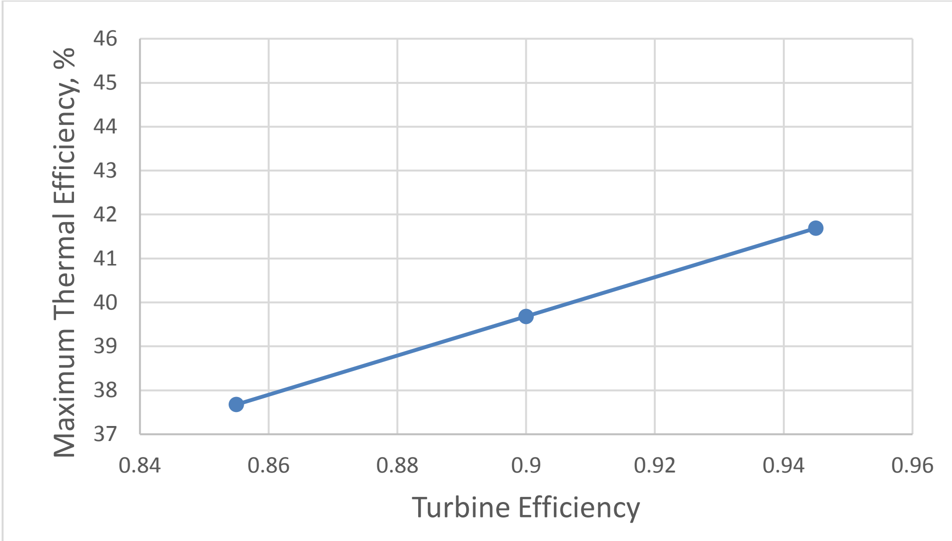

Figure 4.64. Maximum Thermal Efficiency vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 2 FWHs

The obtained values for two FWH has been shown in Table A.3 and A.4 for ideal and actual.

4.3 Regenerative Rankine Cycle with Three Feedwater Heaters

A regenerative Rankine cycle with 3 feedwater heaters has one high pressure closed FWH, one low pressure closed FWH and one open FWH (Deaerator). As mentioned in chapter 3, the iteration method has been used, it means that for getting high pressure closed FWH data, optimized values for low pressure FWH and open FWH are employed and like that for obtaining one of the FWHs data, optimized values of all the other feedwater heaters must be employed. The effect of boiler pressure, condenser pressure, boiler temperature, pump and turbine isentropic efficiencies on the cycle thermal efficiency and the optimum placement of the feedwater heaters are presented in the following sub-sections.

4.3.1 Boiler Pressure

Figures 4.65 and 4.66 show cycle thermal efficiency versus high pressure closed feedwater heater (HPCFWH) pressure which applied in 9 different boiler pressures from 8000 to 12000 kPa in ideal and actual cycles. The lowest curve shows the lowest boiler pressure (8000 kPa) which has the lowest efficiency and the lowest amount of high pressure closed feedwater heater (HPCFWH) optimal pressure. On the other hand, the highest curve illustrates the highest boiler pressure (12000 kPa) which has the highest efficiency and highest amount of high pressure closed feedwater heater (HPCFWH) optimum pressure. As mentioned before, optimum feedwater heater pressure is where the efficiency is maximum on the curve. This is also known for optimal placement of (opened or closed) feedwater heater.

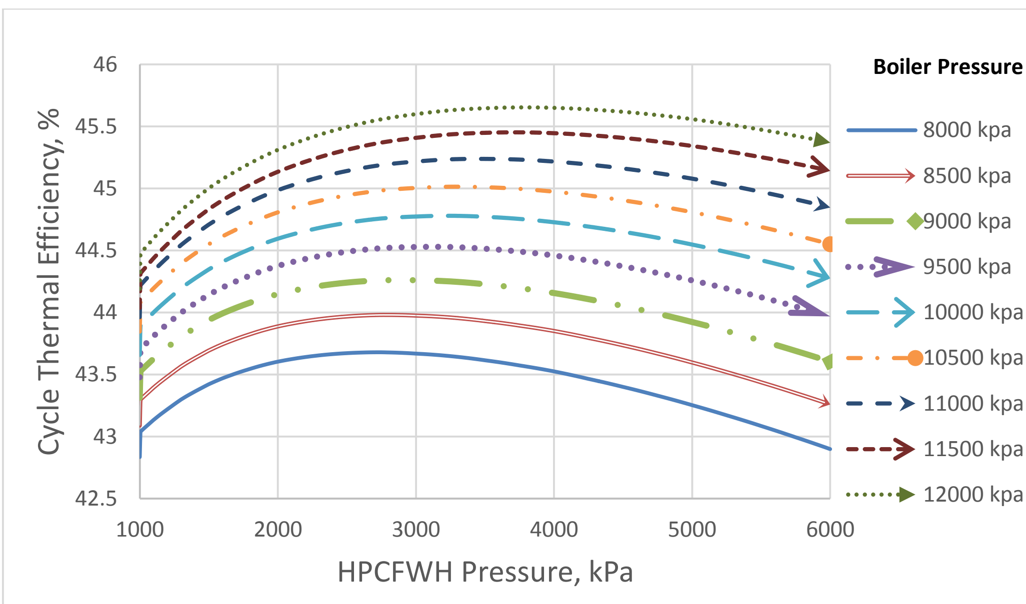

Figure 4.65. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Boiler Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

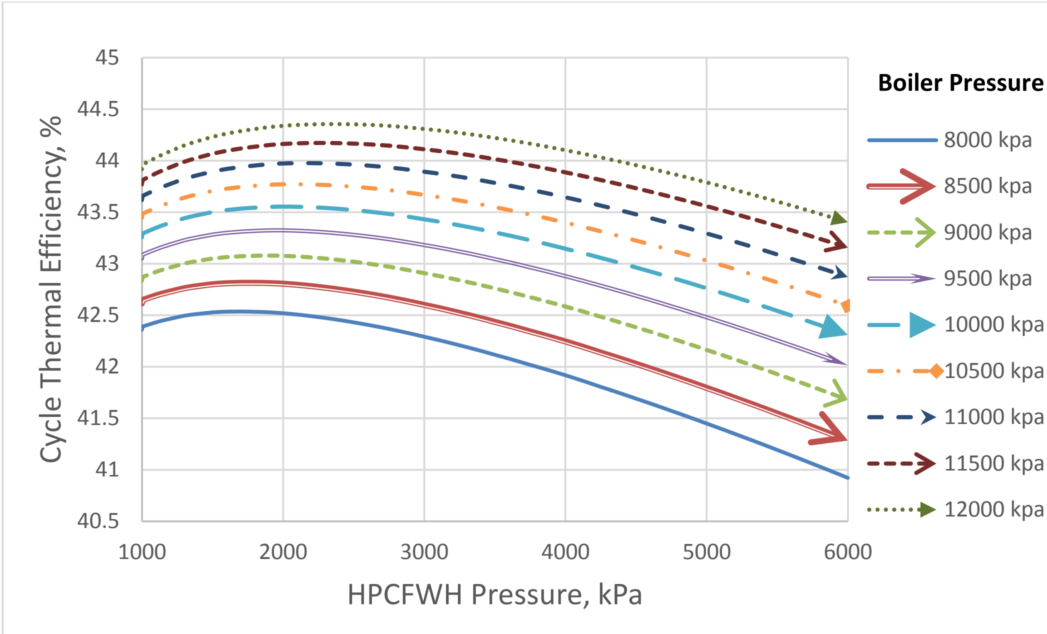

Figure 4.66. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Boiler Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.67 and 4.68 show that by increasing the boiler pressure, high pressure closed feedwater heater optimum pressure rises gradually from 2735 until 3770 kPa in ideal and from 1700 to 2350 kPa for actual cycles.

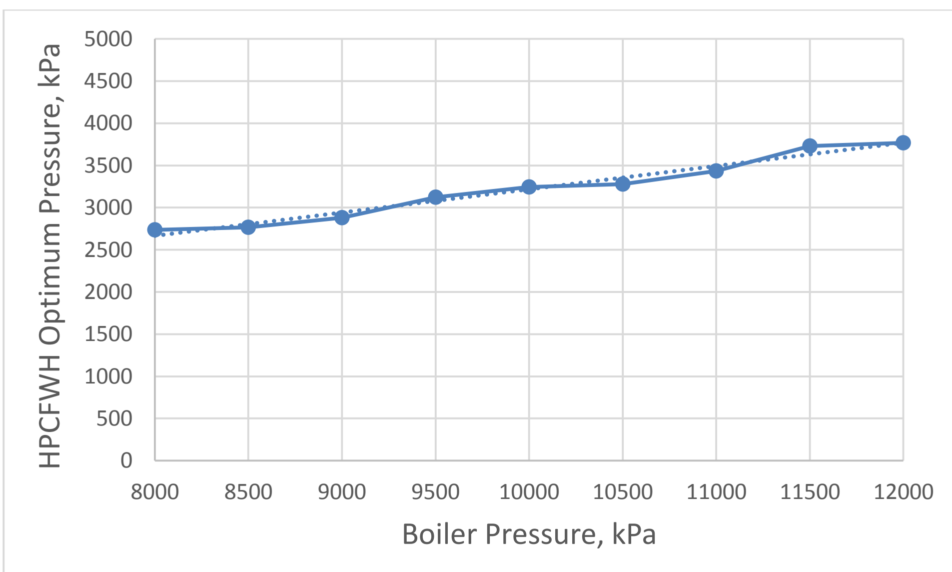

Figure 4.67. Optimum High Pressure Closed FWH Pressure vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

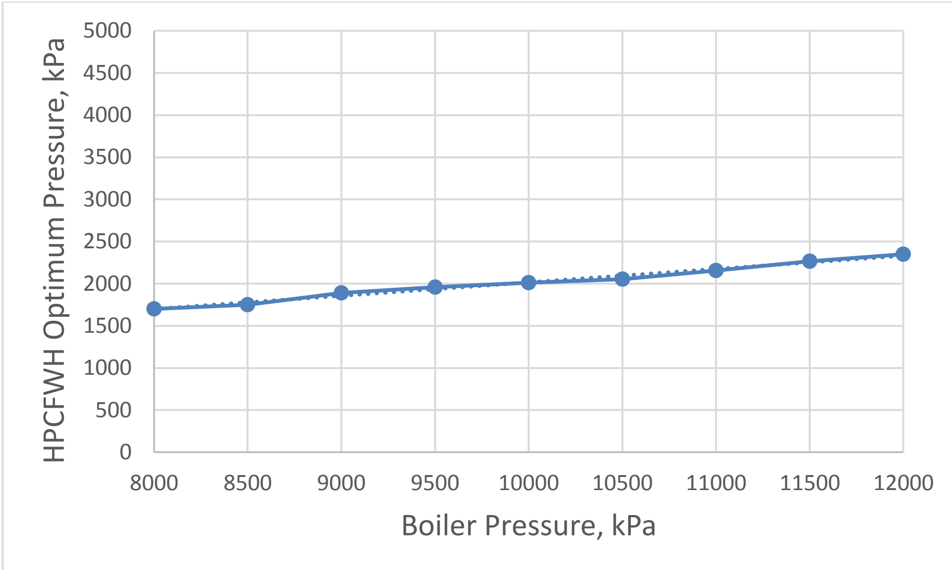

Figure 4.68. Optimum High Pressure Closed FWH Pressure vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.69 and 4.70 illustrates cycle thermal efficiency versus deaerator pressure from 8000 kPa until 12000 kPa. As it is obvious the efficiency goes up by moving from lowest curve (8000 kPa) to the highest curve (12000 kPa) boiler pressure. In both ideal and actual cycles deaerator optimum pressure (where the efficiency reaches its peak) continues to increase but with lower values in the actual cycle in equivalent boiler pressures. This is thoroughly observable in the figures 4.71 and 4.72.

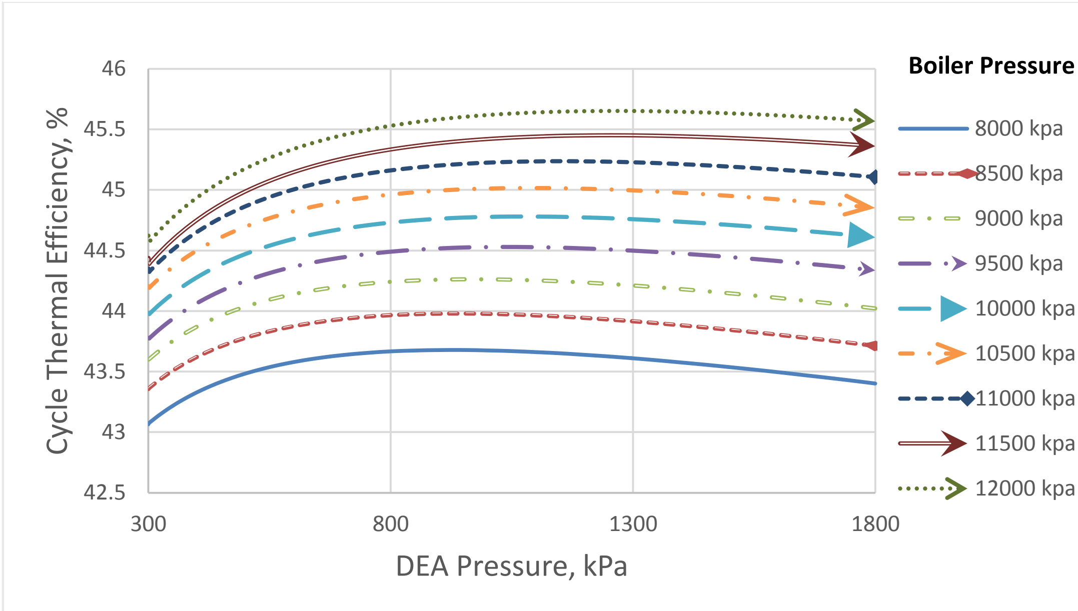

Figure 4.69. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

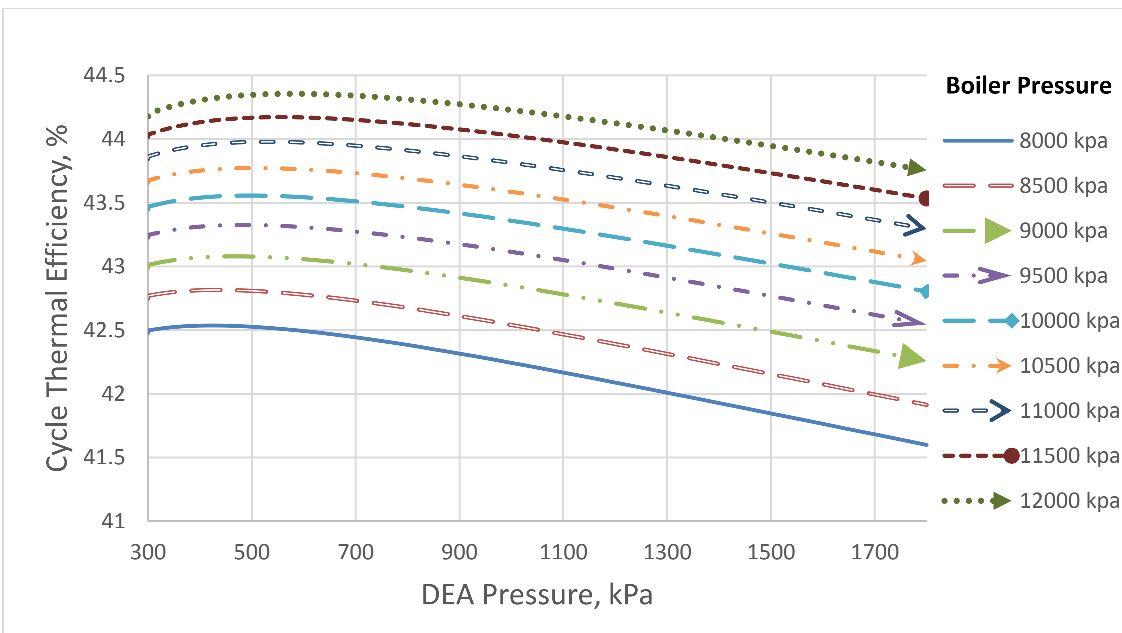

Figure 4.70. Thermal Efficiency vs. DEA Pressure at Different Boiler Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

In the figures 4.71 and 4.72 as mentioned in the preceding sentences, for the equivalent boiler pressure, the DEA optimum pressure is almost twice in ideal cycle than that in actual one. Although DEA optimum pressure in both ideal and actual cycle increases gradually.

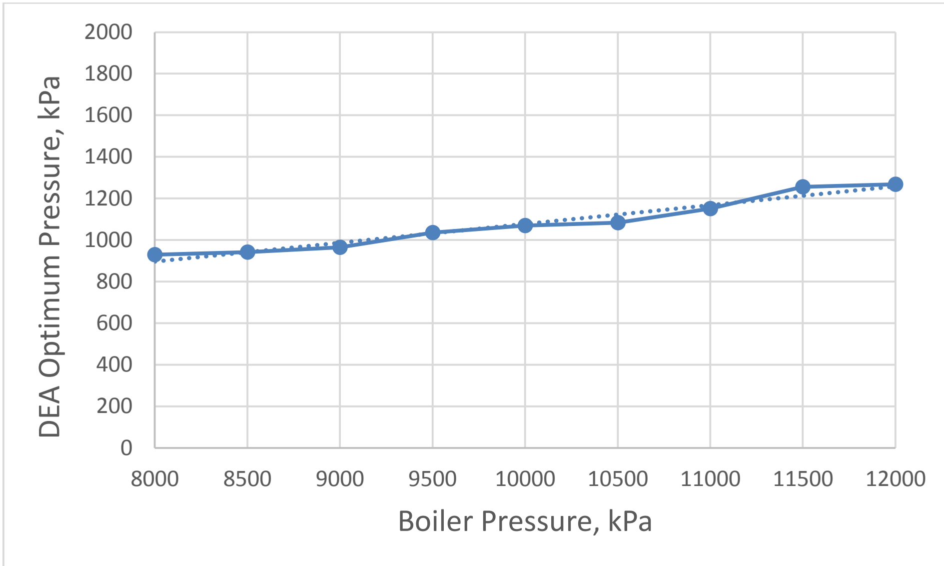

Figure 4.71. Optimum DEA Pressure vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

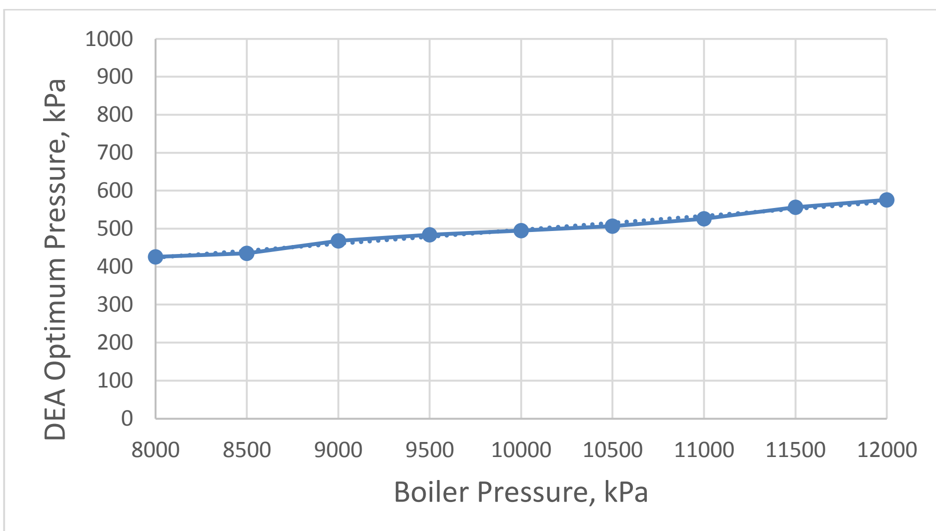

Figure 4.72. Optimum DEA Pressure vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.73 and 4.74 illustrate cycle thermal efficiency versus low pressure closed feedwater heater (LPCFWH) pressure from 8000 kPa until 12000 kPa. It is obvious that thermal efficiency rises by increasing boiler pressure. In both ideal and actual cycles low pressure closed feedwater heater (LPCFWH) optimum pressure (where the efficiency reaches its peak) continues to increase but with lower values in the actual cycle in equivalent boiler pressures. This is thoroughly observable in figures 4.75 and 4.76.

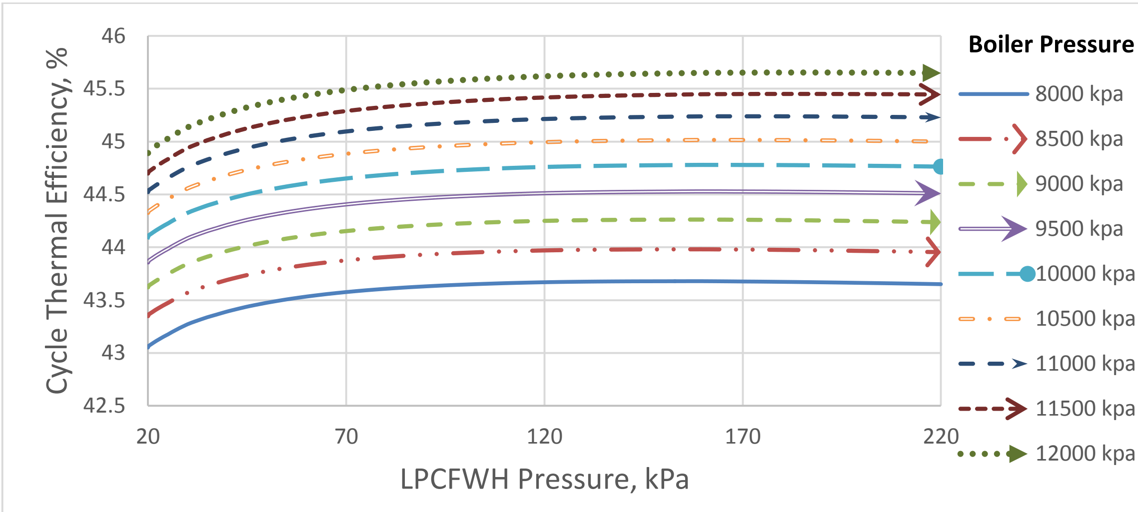

Figure 4.73. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Boiler Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

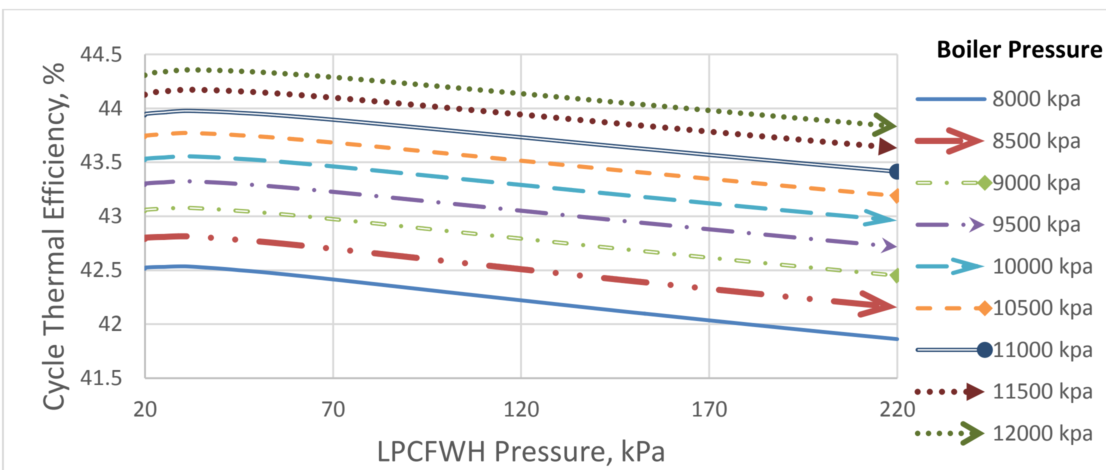

Figure 4.74. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Boiler Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

In the figures 4.75 and 4.76 the points on the curve show the exact values of low pressure closed feedwater heater (LPCFWH) optimal pressure in ideal and actual. As can be observed, there is a very slight increase in low pressure closed feedwater heater optimal pressure in ideal cycle but steady values in actual cycle. In actual cycle, low pressure closed feedwater heater optimum pressures can be considered constant and the change is negligible.

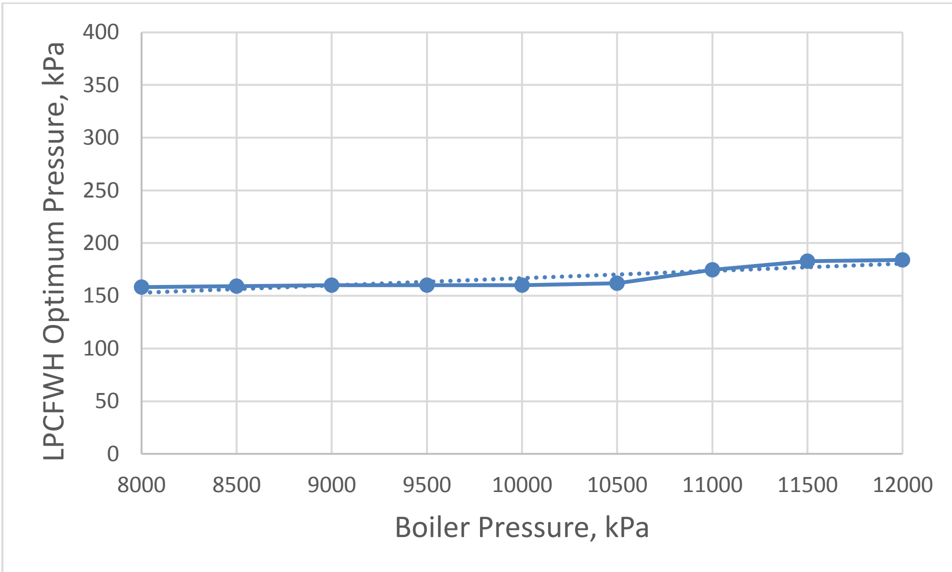

Figure 4.75. Optimum Low Pressure Closed FWH Pressure vs. Boiler Pressure Change in Ideal Regenerative Rankine Cycle with 3 FWHs

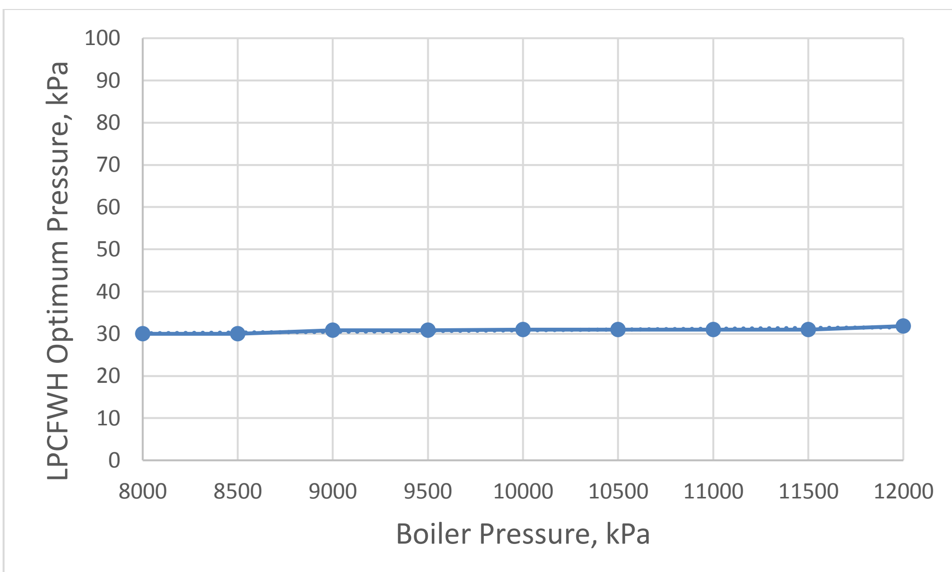

Figure 4.76. Optimum Low Pressure Closed FWH Pressure vs. Boiler Pressure Change in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.77 and 4.78 illustrate that by rising the boiler pressure from 8000 to 12000 kPa, an increase of about 2% can be noticed in both ideal and actual.

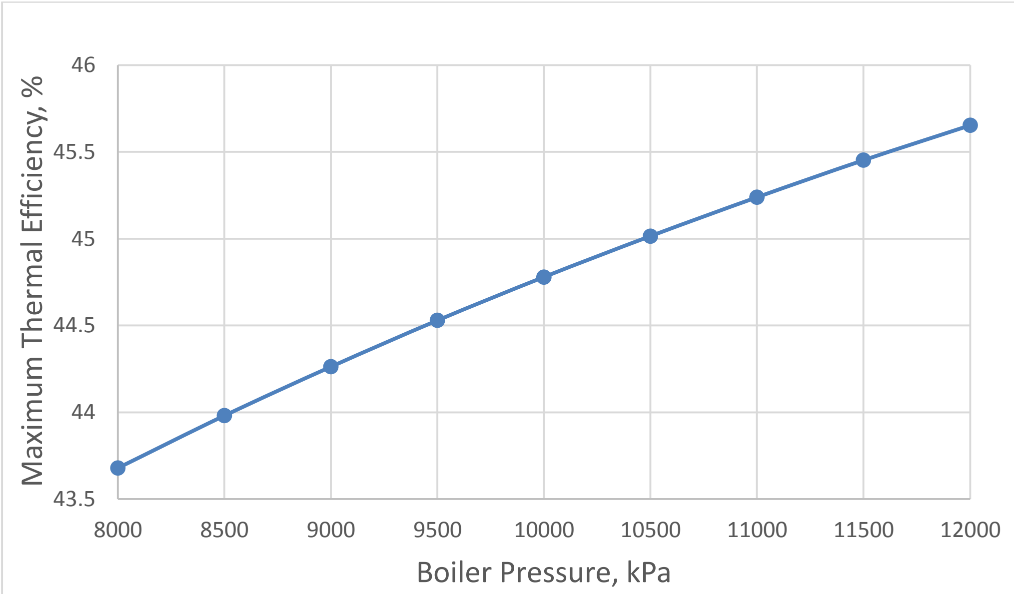

Figure 4.77. Maximum Thermal Efficiency vs. Boiler Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

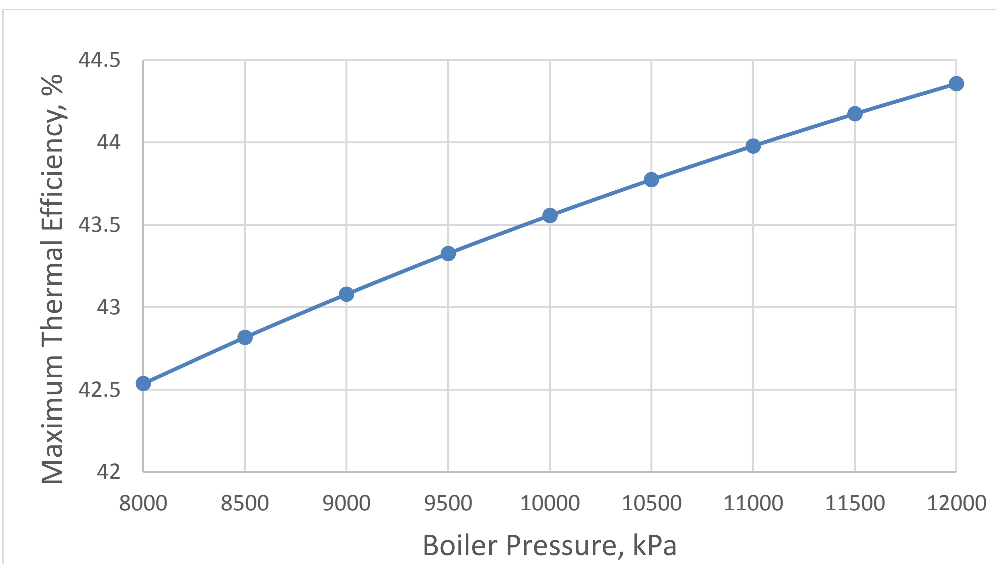

Figure 4.78. Maximum Thermal Efficiency vs. Boiler Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

4.3.2 Condenser Pressure

In figures 4.79 and 4.80 cycle thermal efficiency versus high pressure closed feedwater heater pressure in various condenser pressures are illustrated. It is obvious that the highest curve (8 kPa) has the highest thermal efficiency. On the other hand, 12 kPa curve has the lowest thermal efficiency. High pressure closed feedwater heater optimum pressure for both ideal and actual is clearer in figures 4.81 and 4.82.

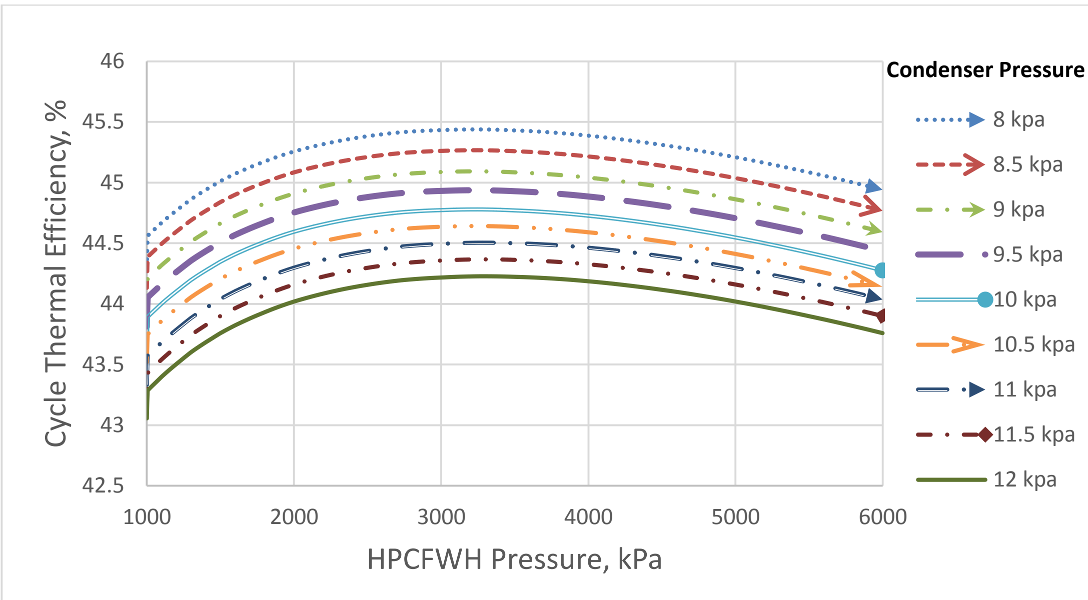

Figure 4.79. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Condenser Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

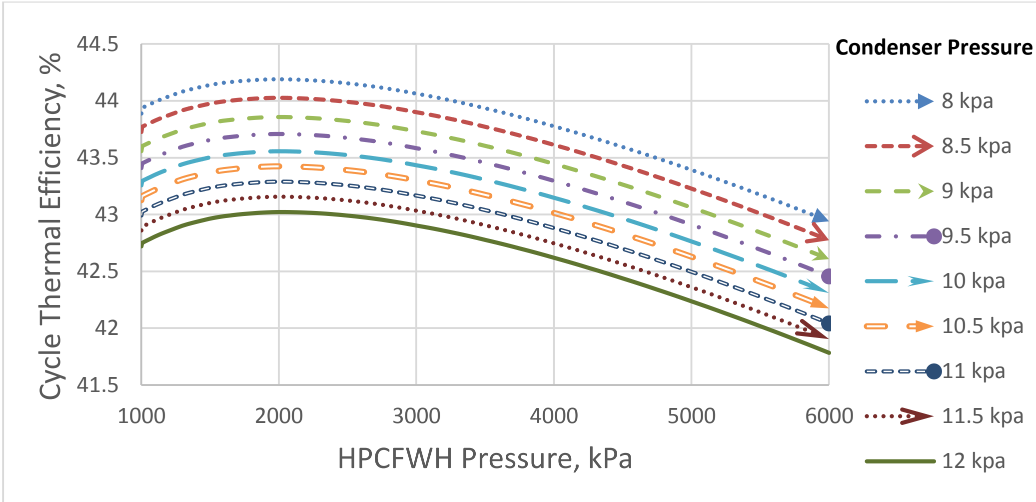

Figure 4.80. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Condenser Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

In figures 4.81 and 4.82 it is clear that there is a very slight increase in high pressure closed feedwater heater optimum pressure in ideal and actual cycle when condenser pressure goes from 8 to 12 kPa.

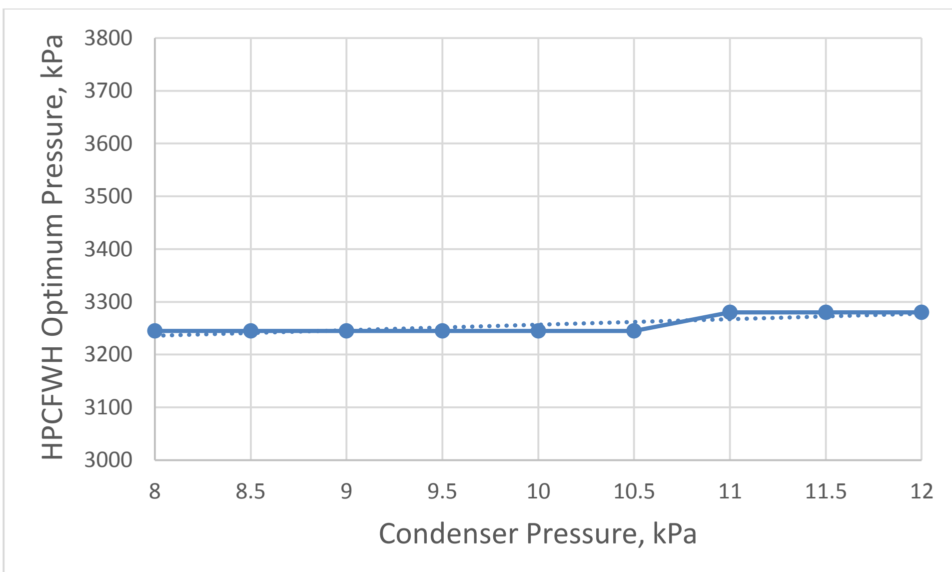

Figure 4.81. Optimum High Pressure Closed FWH Pressure vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

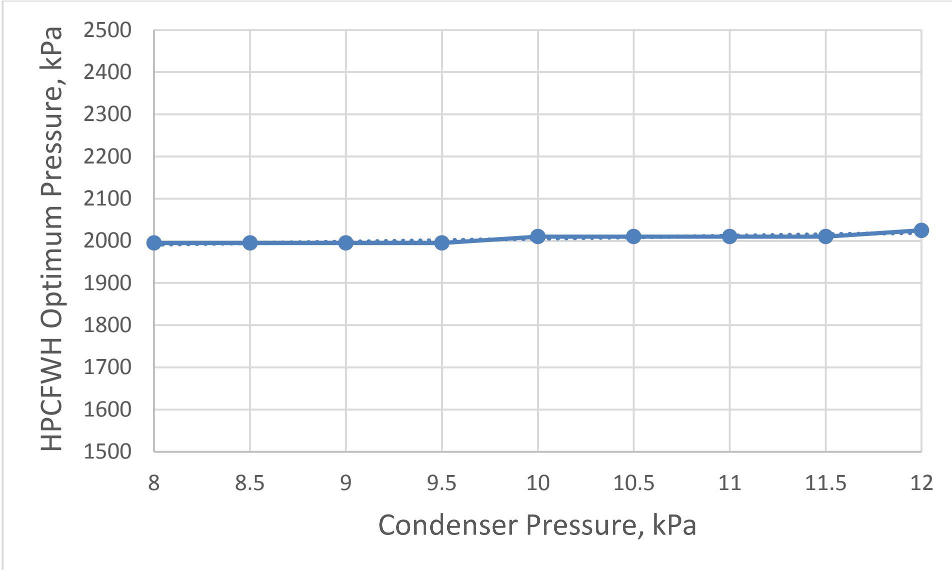

Figure 4.82. Optimum High Pressure Closed FWH Pressure vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

In figures 4.83 and 4.84 the cycle thermal efficiency versus DEA pressure (Deaerator) is illustrated from 8 to 12 kPa for both ideal and actual cycles. The highest efficiency belongs to the 8 kPa curve and 12 kPa curve denotes for the lowest efficiency. For the DEA optimum pressure please refer to the figures 4.85 and 4.86.

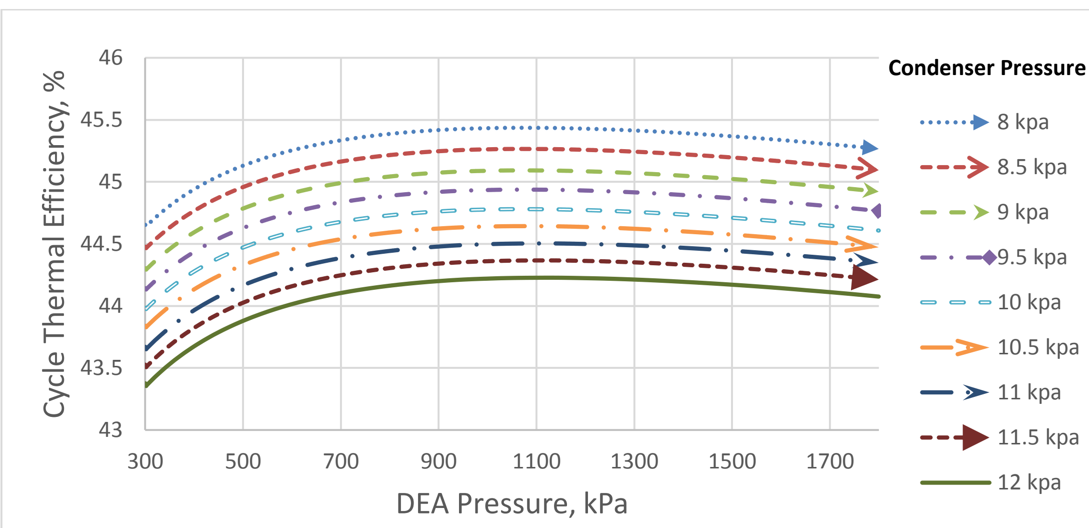

Figure 4.83. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

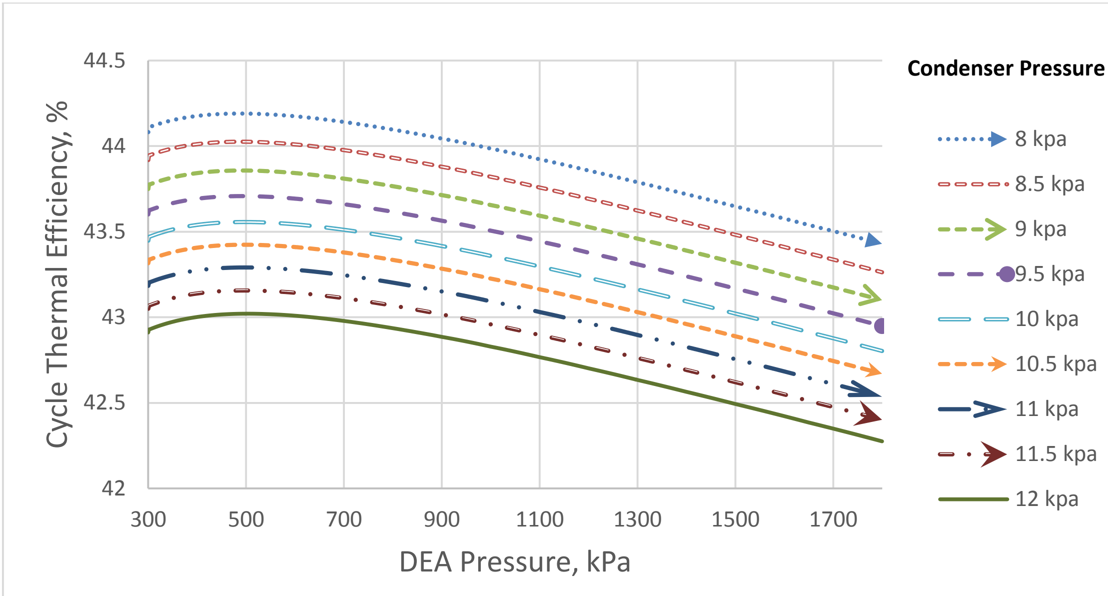

Figure 4.84. Thermal Efficiency vs. DEA Pressure at Different Condenser Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

As can be seen in figure 4.85 in the ideal cycle, the deaerator optimum pressure increases very slightly when condenser pressure goes from 8 to 12 kPa.

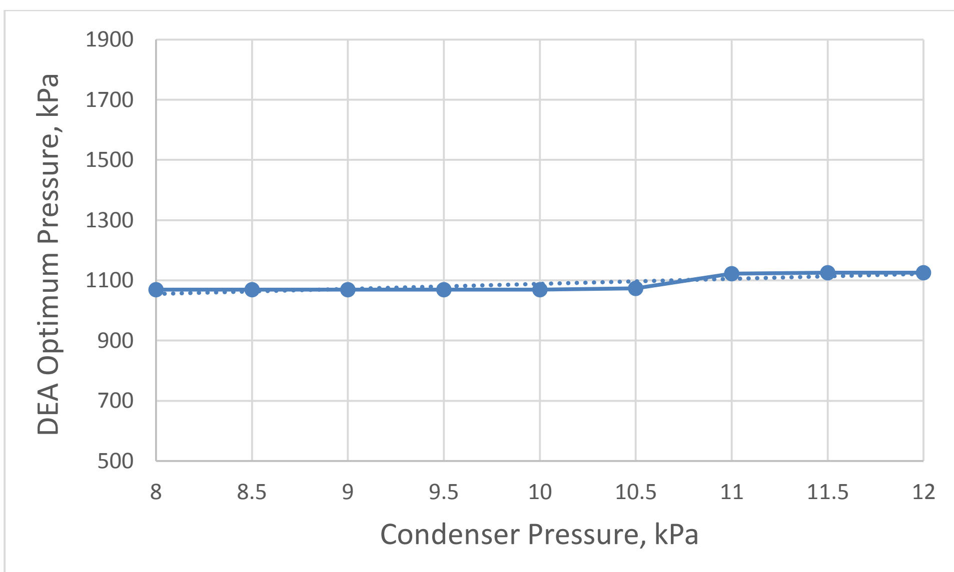

Figure 4.85. Optimum DEA Pressure vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

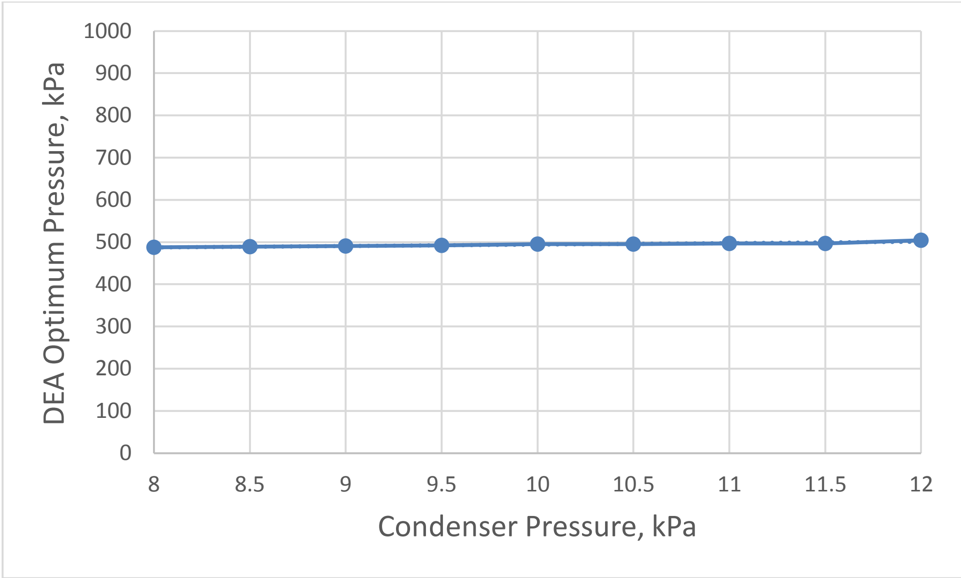

Figure 4.86. Optimum DEA Pressure vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

In figures 4.87 and 4.88 cycle thermal efficiency versus low pressure closed feedwater heater pressure in various condenser pressures are illustrated. It is obvious that the highest curve (8 kPa) has the highest thermal efficiency. On the other hand, 12 kPa curve has the lowest thermal efficiency. Low pressure closed feedwater heater optimum pressures for both ideal and actual are more obvious in figures 4.89 and 4.90.

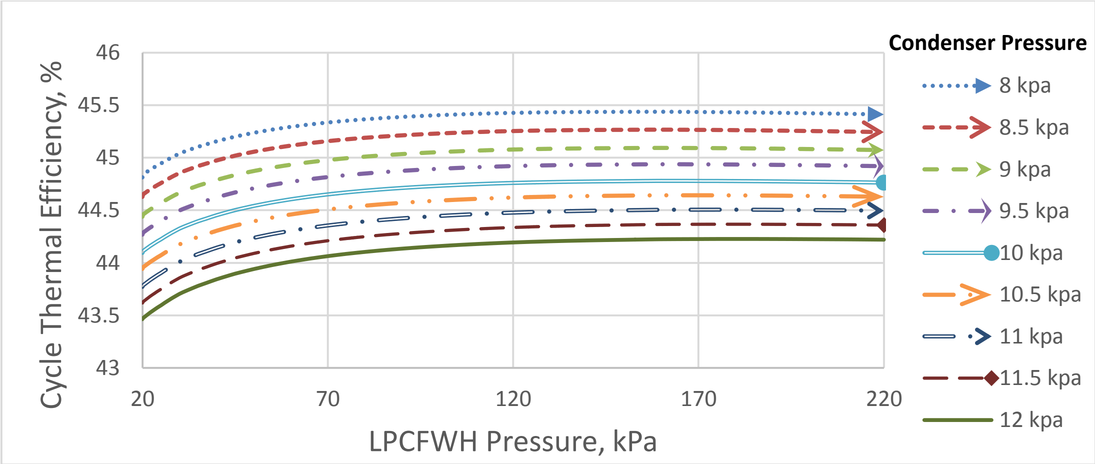

Figure 4.87. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Condenser Pressures for the Ideal Regenerative Rankine Cycle with 3 FWHs

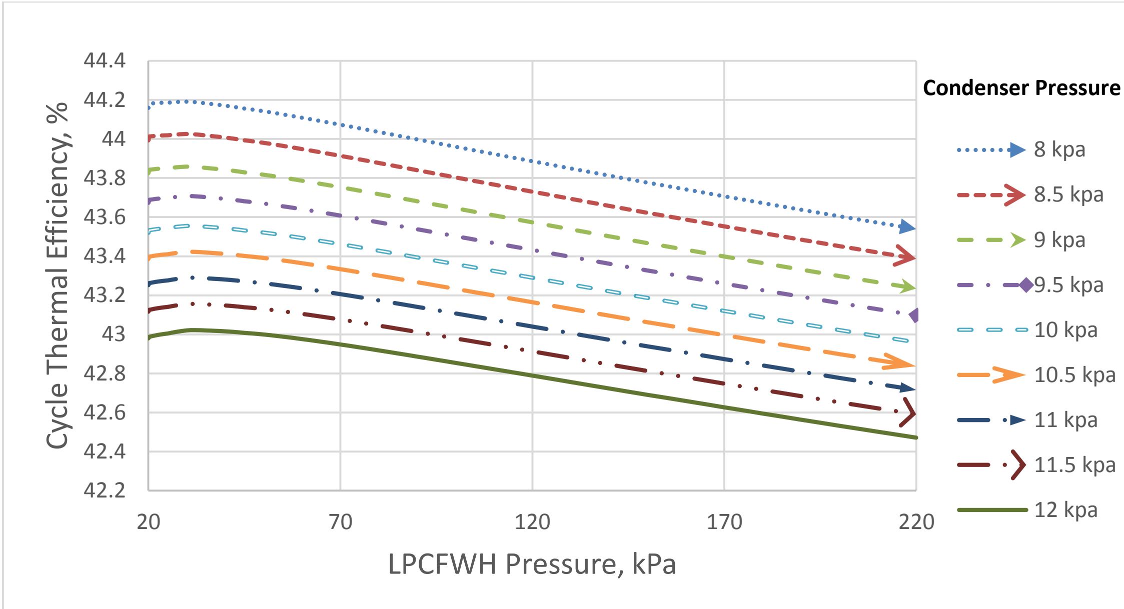

Figure 4.88. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Condenser Pressures for the Actual Regenerative Rankine Cycle with 3 FWHs

As can be seen in figure 4.89 and 4.90 for ideal cycle, low pressure closed feedwater heater optimum pressures has a very slight increase when condenser pressure increases. In actual cycle, low pressure closed feedwater heater optimum pressures can be considered constant and the change is negligible.

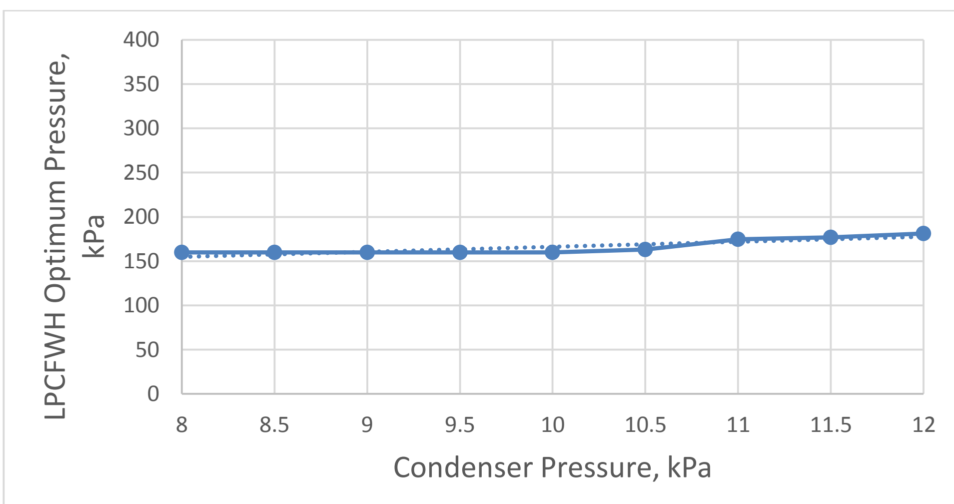

Figure 4.89. Optimum Low Pressure Closed FWH Pressure vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

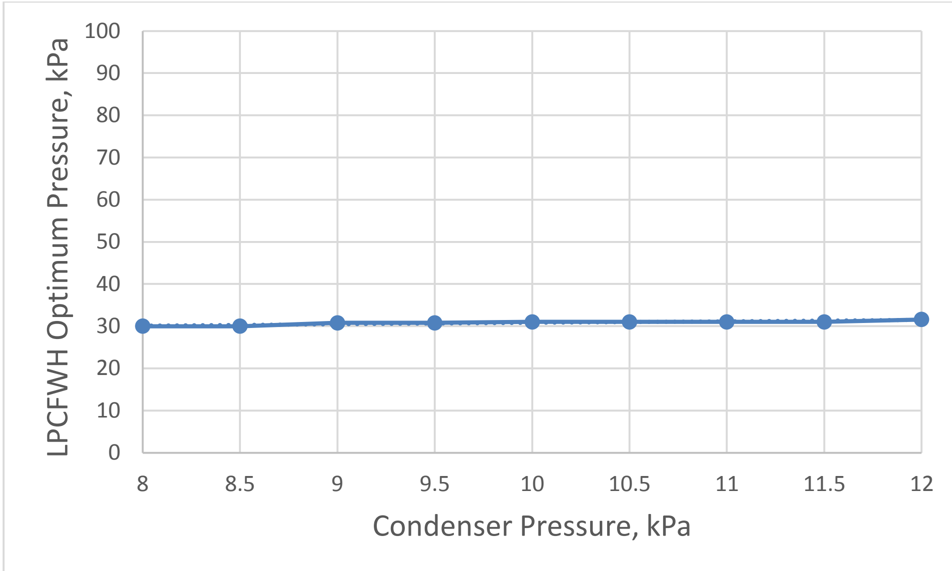

Figure 4.90. Optimum Low Pressure Closed FWH Pressure vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.91 and 4.92 show the exact amount of efficiency in those 9 condenser pressures. As can be seen, the thermal efficiency decreases by almost 1% when the condenser pressure varies from 8 to 12 kPa gradually.

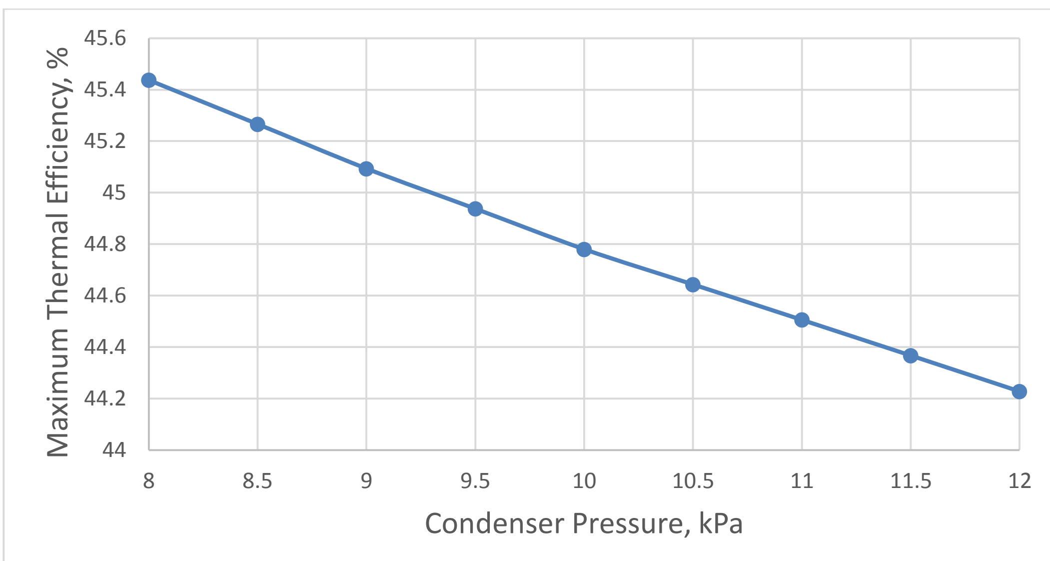

Figure 4.91. Maximum Thermal Efficiency vs. Condenser Pressure in Ideal Regenerative Rankine Cycle with 3 FWHs

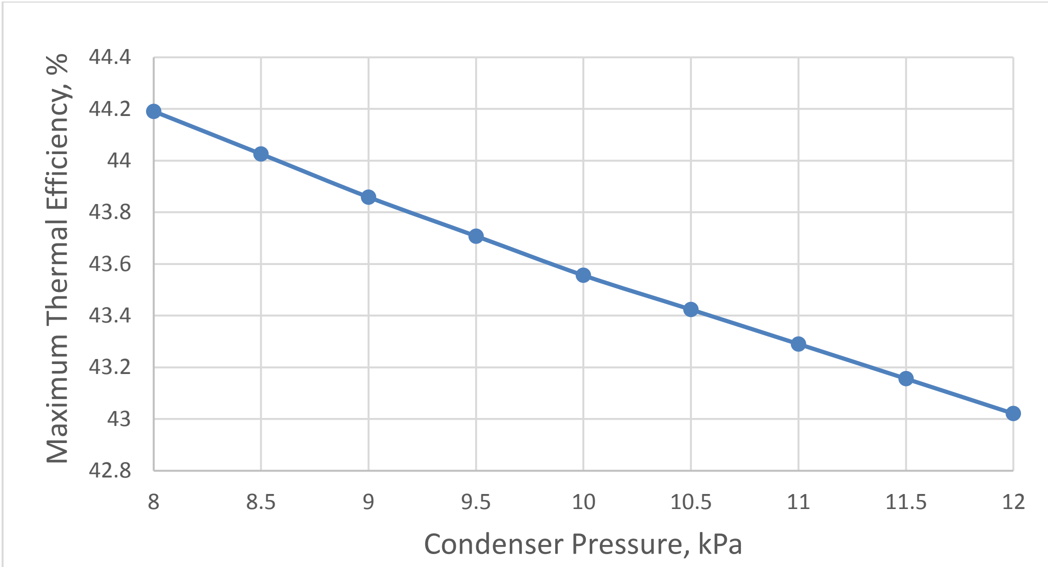

Figure 4.92. Maximum Thermal Efficiency vs. Condenser Pressure in Actual Regenerative Rankine Cycle with 3 FWHs

4.3.3 Boiler Temperature

Figures 4.93 and 4.94 illustrate cycle thermal efficiency versus high pressure closed feedwater heater (HPCFWH) pressure which applied in 9 different boiler temperatures from 408° C to 612° C in ideal and actual cycles. The lowest curve shows the lowest boiler temperature (408° C) which has the lowest efficiency and the lowest amount of high pressure closed feedwater heater (HPCFWH) optimal pressure. On the other hand, the highest curve shows the highest boiler temperature (612° C) which has the highest efficiency. As mentioned before, optimum feedwater heater pressure is where the efficiency is maximum on the curve. This is also known as optimal placement of (opened or closed) feedwater heater.

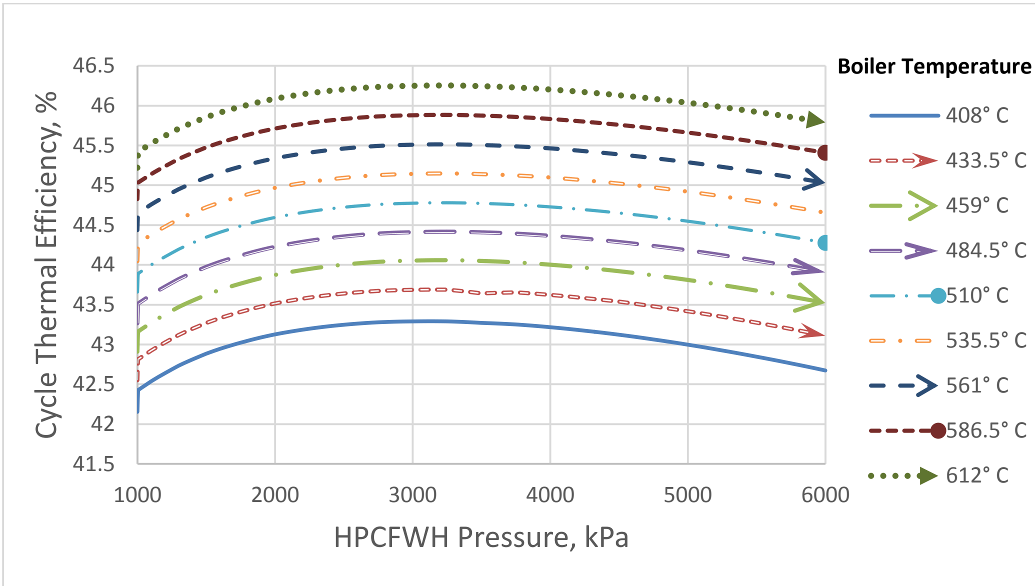

Figure 4.93. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Boiler Temperatures for the Ideal Regenerative Rankine Cycle with 3 FWHs

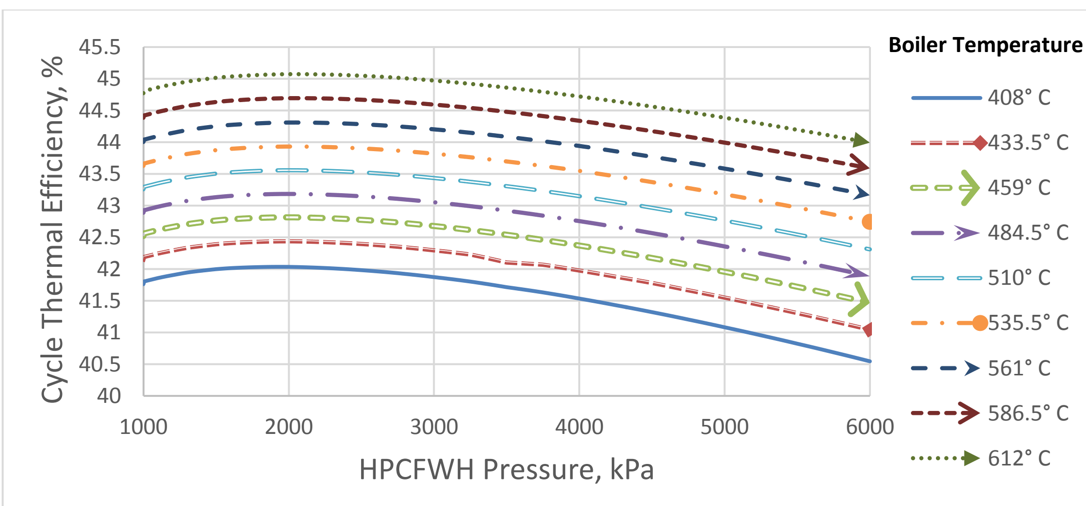

Figure 4.94. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Boiler Temperatures for the Actual Regenerative Rankine Cycle with 3 FWHs

As in figures 4.95 and 4.96 is clear, for ideal cycle between 408° C and 459° C, high pressure closed feedwater heater (HPCFWH) optimum pressure is raised from 3135 to 3235 kPa and after that it remains constant. In actual cycle, in general, high pressure closed feedwater heater (HPCFWH) optimum pressure increases gradually however it is steady between 433.5° C and 484.5° C and between 561° C and 612° C.

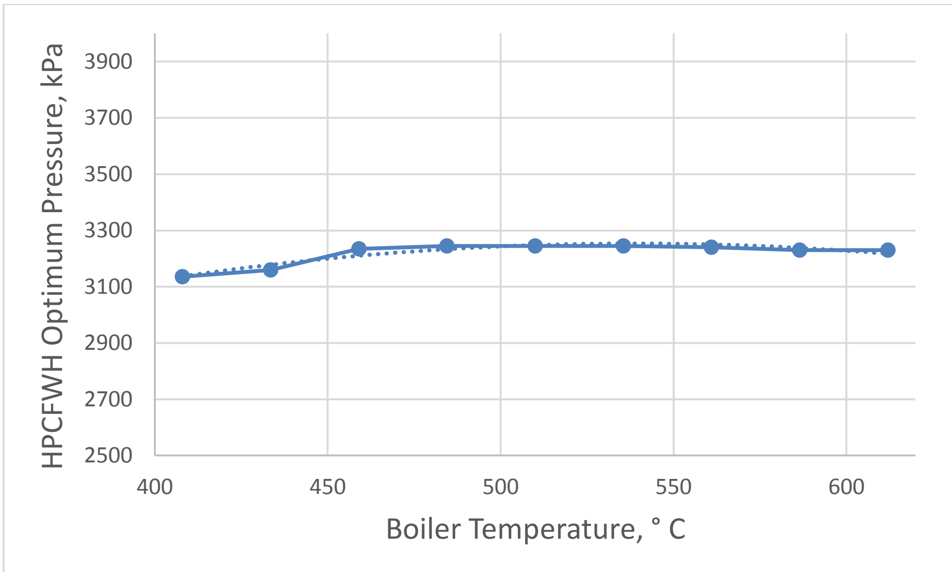

Figure 4.95. Optimum High Pressure Closed FWH Pressure vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 3 FWHs

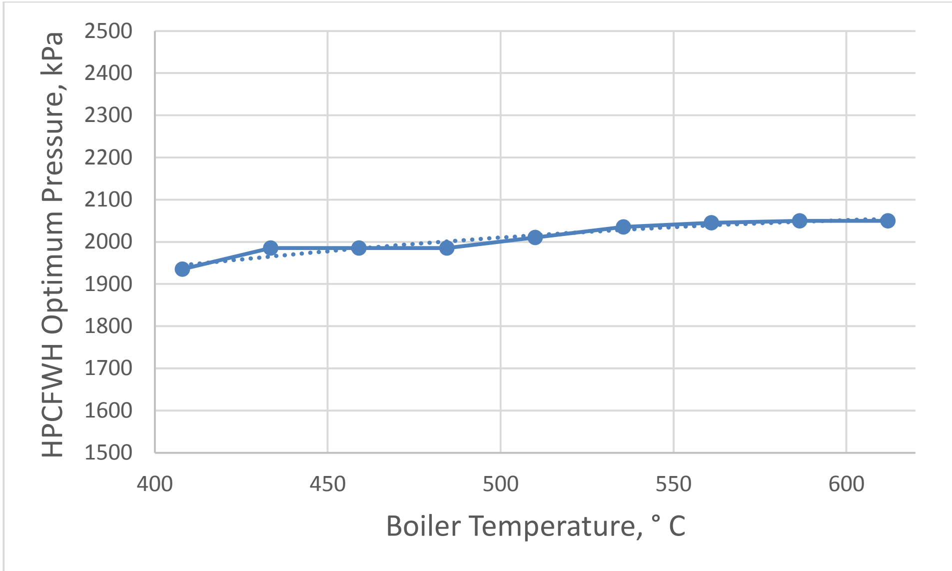

Figure 4.96. Optimum High Pressure Closed FWH Pressure vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.97 and 4.98 illustrate cycle thermal efficiency versus deaerator pressure from 408° C until 612° C. As it is obvious, the efficiency rises by moving from the lowest (408° C) to the highest (612° C) boiler temperature. In both ideal and actual cycles deaerator optimum pressures (where the efficiency reaches its peak) remain almost constant although there are slight ups and downs. Deaerator optimum pressures in actual cycle for corresponding boiler temperatures are much lower than that in ideal cycle. This is thoroughly observable in figures 4.99 and 4.100.

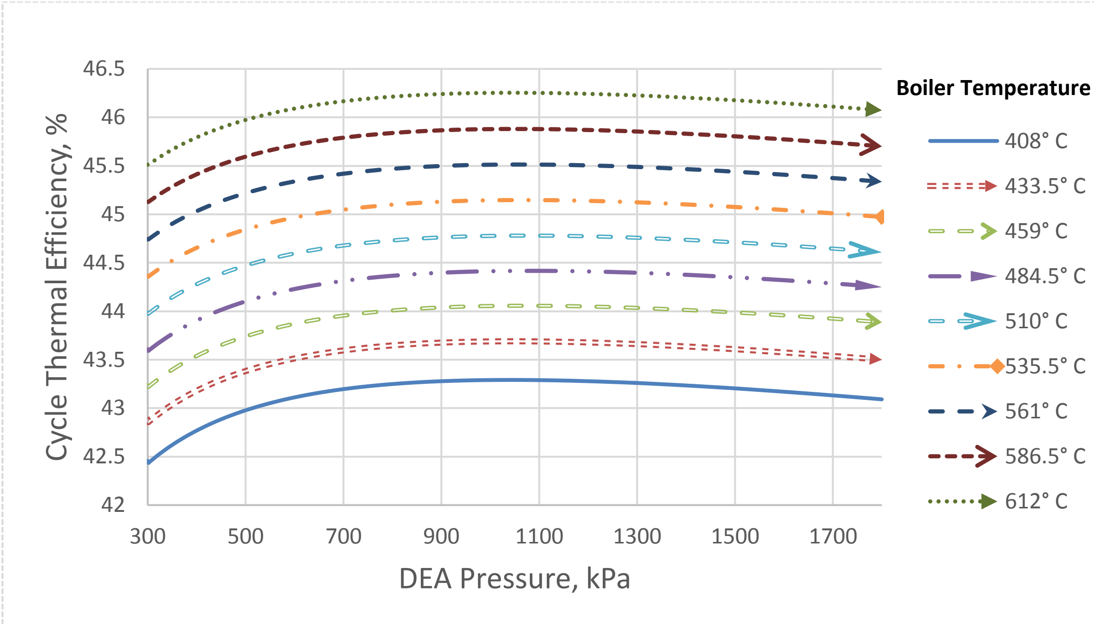

Figure 4.97. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Ideal Regenerative Rankine Cycle with 3 FWHs

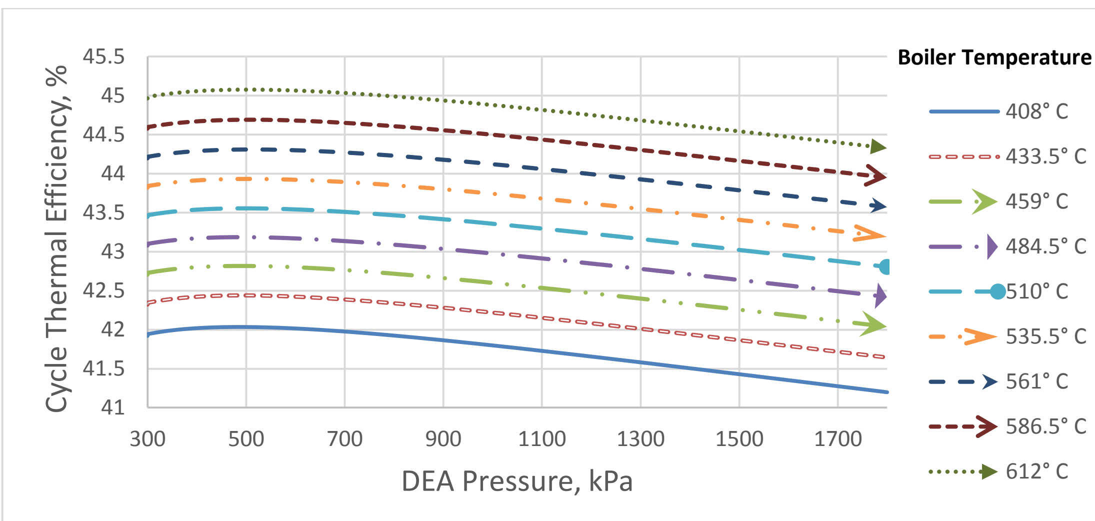

Figure 4.98. Thermal Efficiency vs. DEA Pressure at Different Boiler Temperatures for the Actual Regenerative Rankine Cycle with 3 FWHs

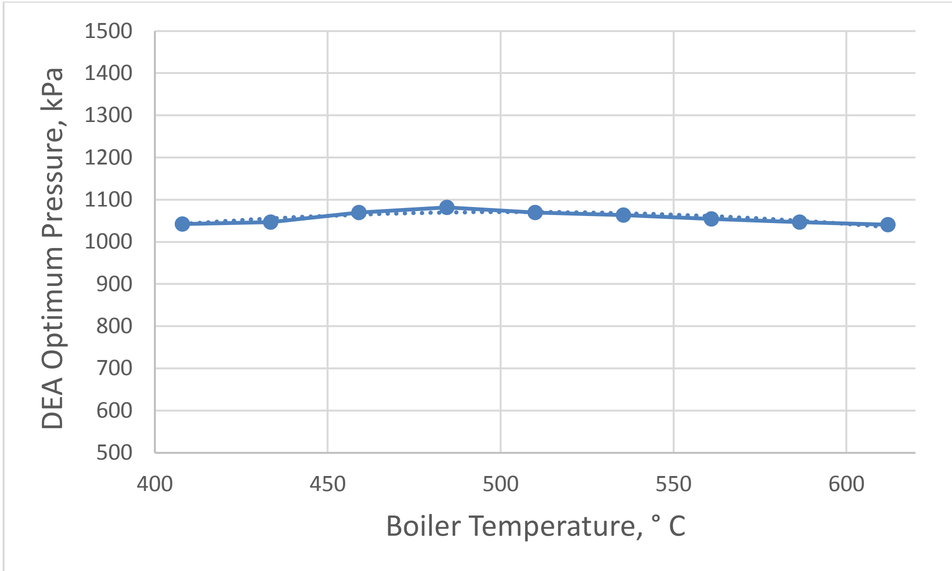

Figure 4.99. Optimum DEA Pressure vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 3 FWHs

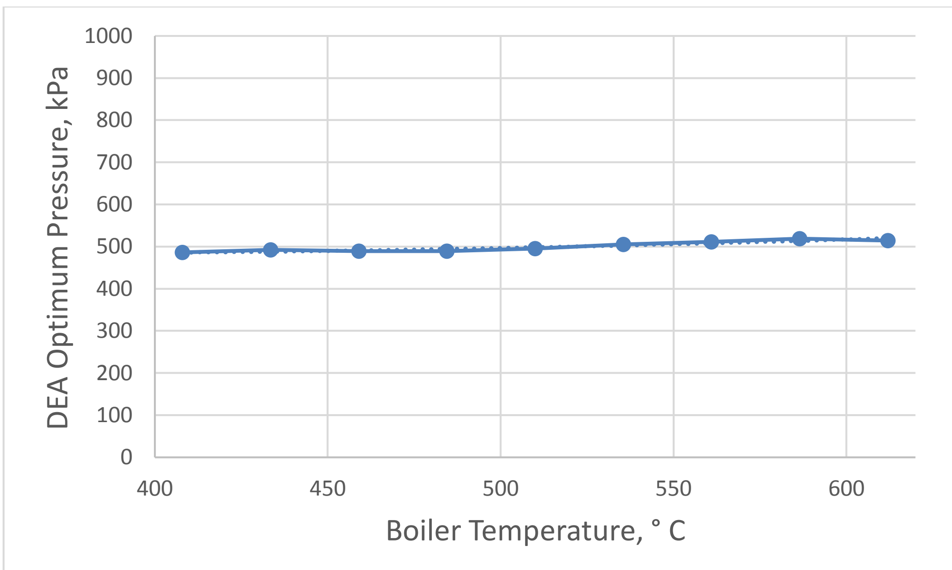

Figure 4.100. Optimum DEA Pressure vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.101 and 4.102 illustrate cycle thermal efficiency versus low pressure closed feedwater heater (LPCFWH) pressure from 408° C to 612° C boiler temperature. It is obvious that thermal efficiency increases by moving from the lowest (408° C) to the highest (612° C) boiler temperature. In both ideal and actual cycles low pressure closed feedwater heater (LPCFWH) optimum pressures (where the efficiency reaches its peak) remain at the same level but with lower values in the actual cycle in equivalent boiler temperatures. This is thoroughly observable in figures 4.103 and 4.104.

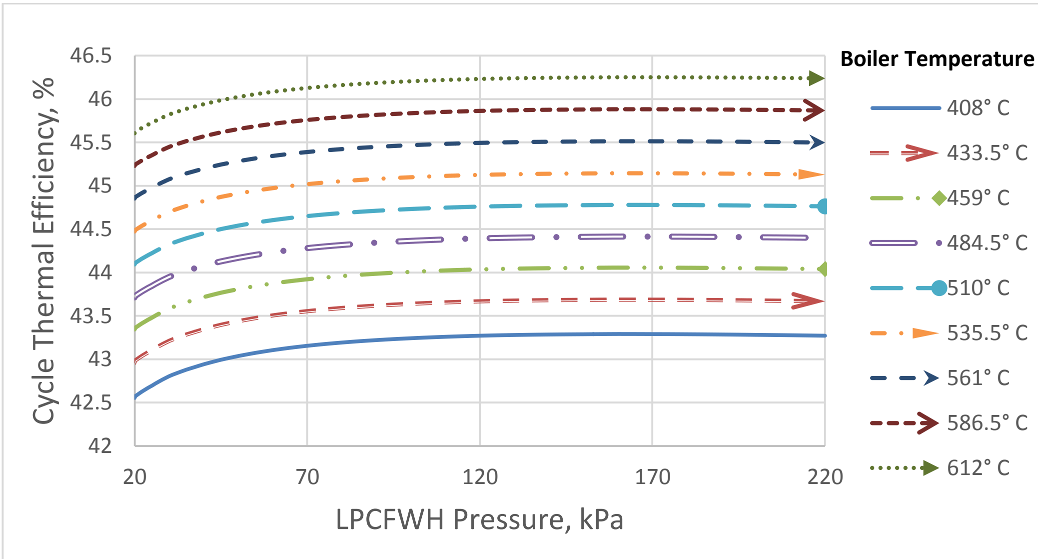

Figure 4.101. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Boiler Temperatures for the Ideal Regenerative Rankine Cycle with 3 FWHs

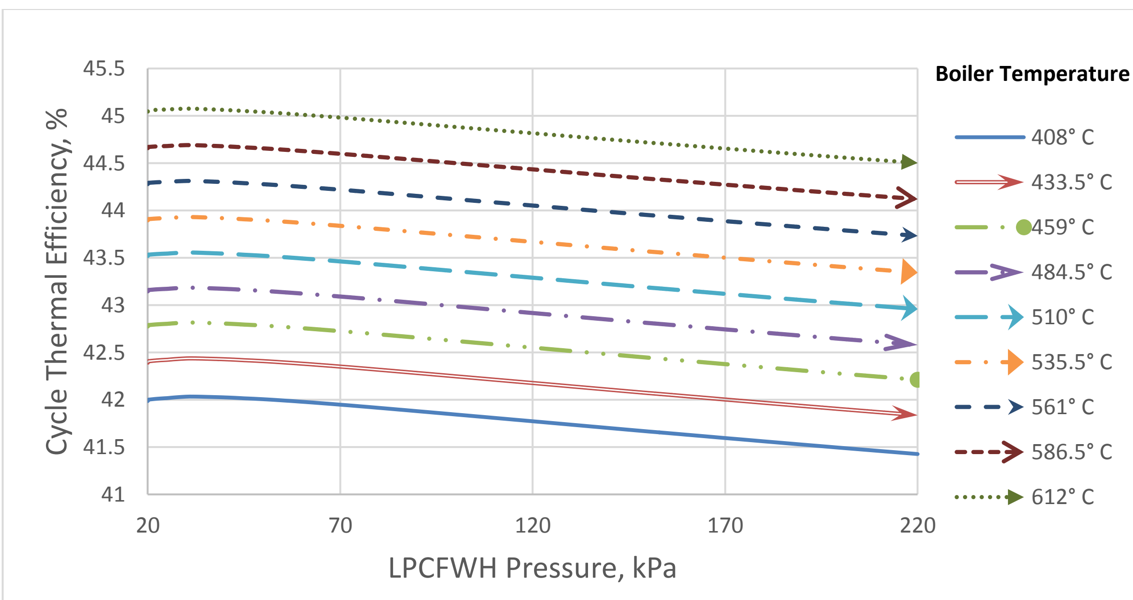

Figure 4.102. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Boiler Temperatures for the Actual Regenerative Rankine Cycle with 3 FWHs

By increasing the boiler temperature, the change in low pressure closed feedwater heater optimum pressure is absolutely negligible in both actual and ideal cycles, therefore it can be considered as constant (refer to figures 4.103 and 4.104), despite the fact that their values are much lower in actual cycle than ideal cycle in corresponding boiler temperatures.

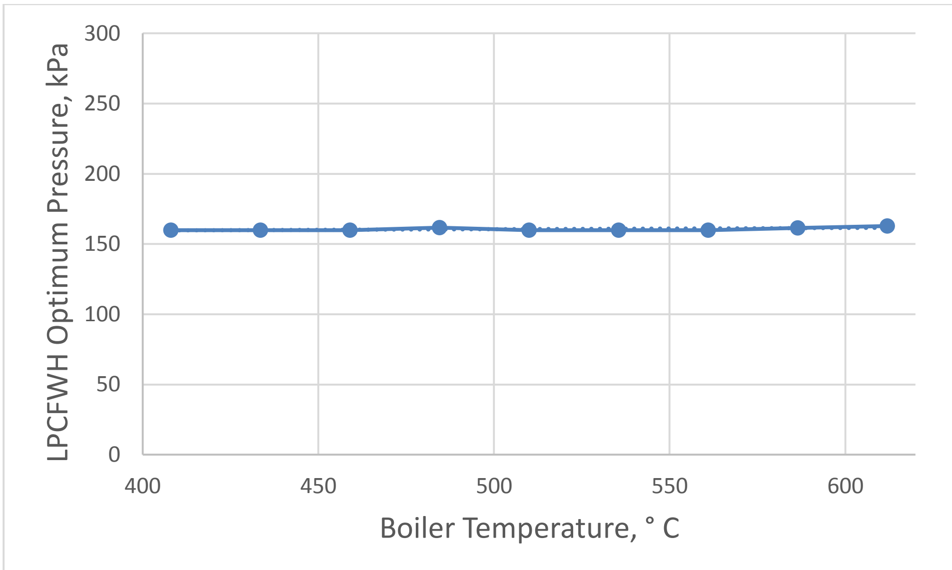

Figure 4.103. Optimum Low Pressure Closed FWH Pressure vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 3 FWHs

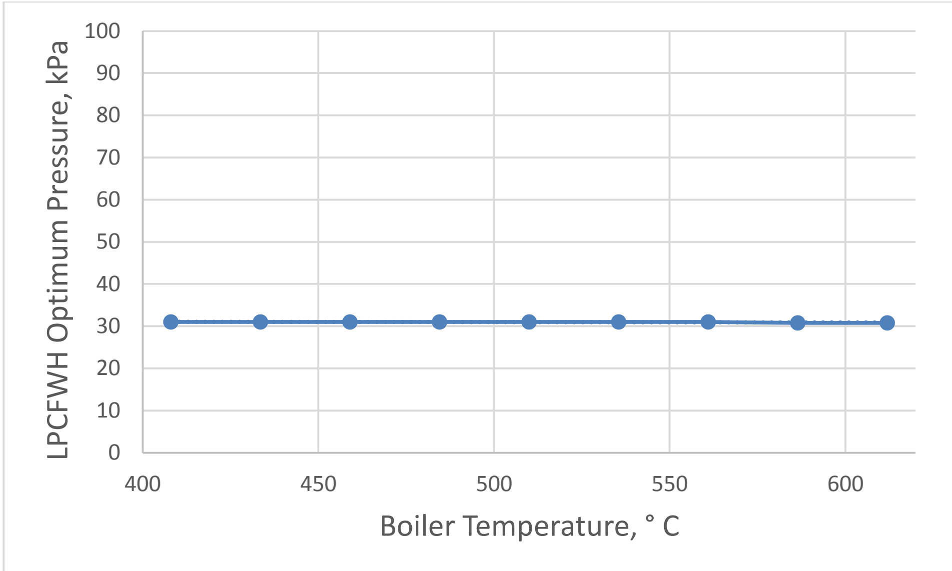

Figure 4.104. Optimum Low Pressure Closed FWH Pressure vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 3 FWHs

Figures 4.105 and 4.106 show that when boiler temperature increases from 408° C to 612° C, thermal efficiency increases from 43.29% to 46.25% in ideal cycle and from 42.03% to 45.07% in actual cycle.

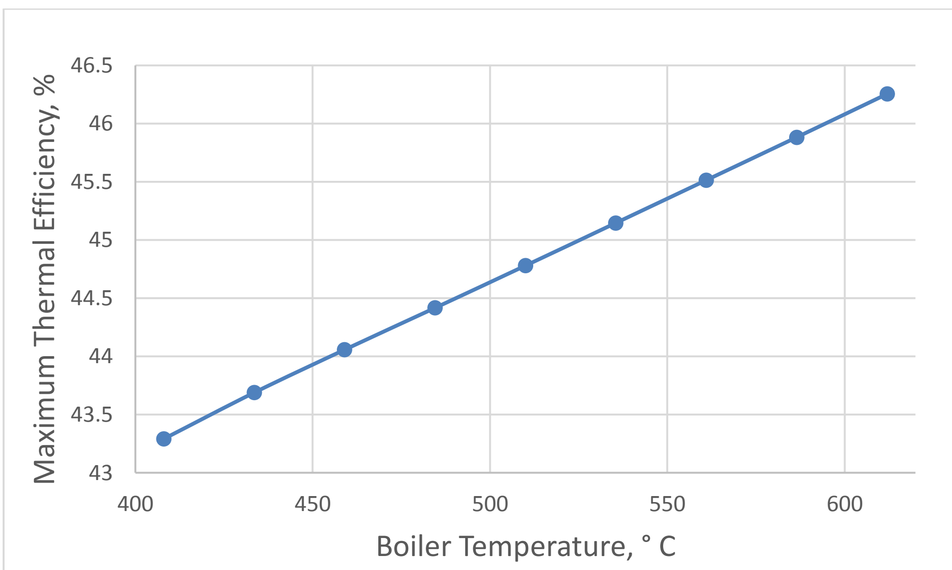

Figure 4.105. Maximum Thermal Efficiency vs. Boiler Temperature in Ideal Regenerative Rankine Cycle with 3 FWHs

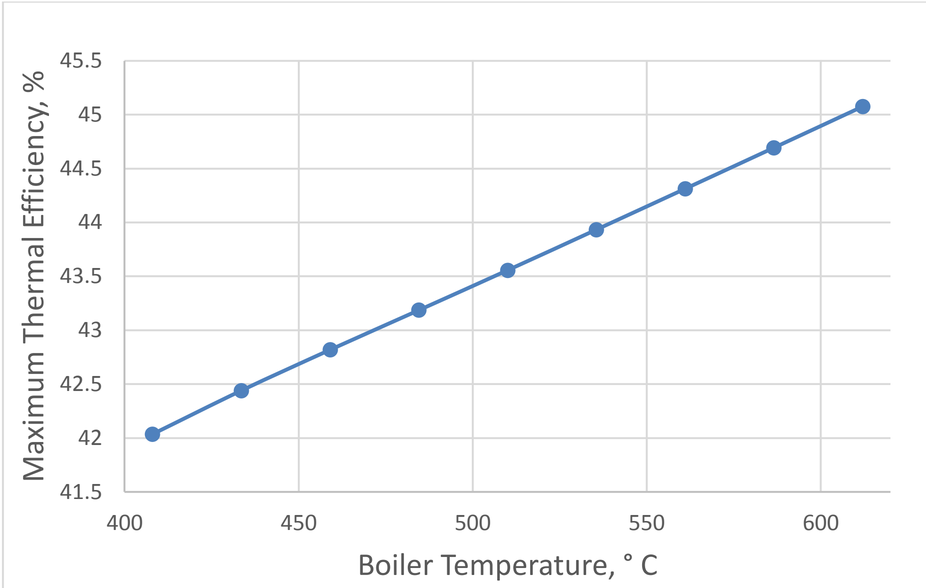

Figure 4.106. Maximum Thermal Efficiency vs. Boiler Temperature in Actual Regenerative Rankine Cycle with 3 FWHs

4.3.4 Pump Efficiency

Figure 4.107 shows cycle thermal efficiency versus high pressure closed feedwater heater pressure for three different pump efficiencies. As it is observable, all three curves almost overlaps each other which means by changing pump efficiencies the change in thermal efficiency will be insignificant additionally, high pressure closed feedwater heater optimum pressure could be regarded as constant because its change is negligible (figure 4.108).

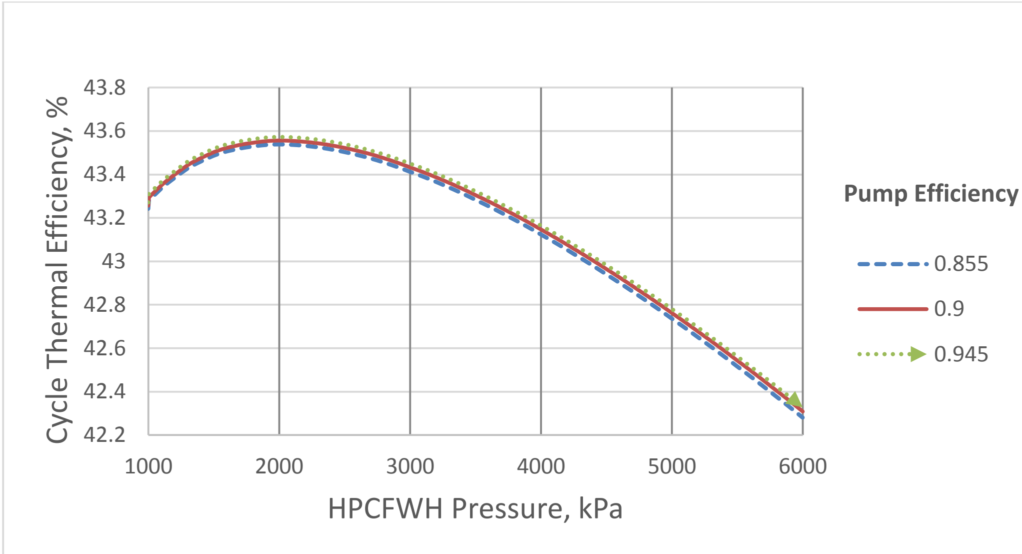

Figure 4.107. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Pump Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

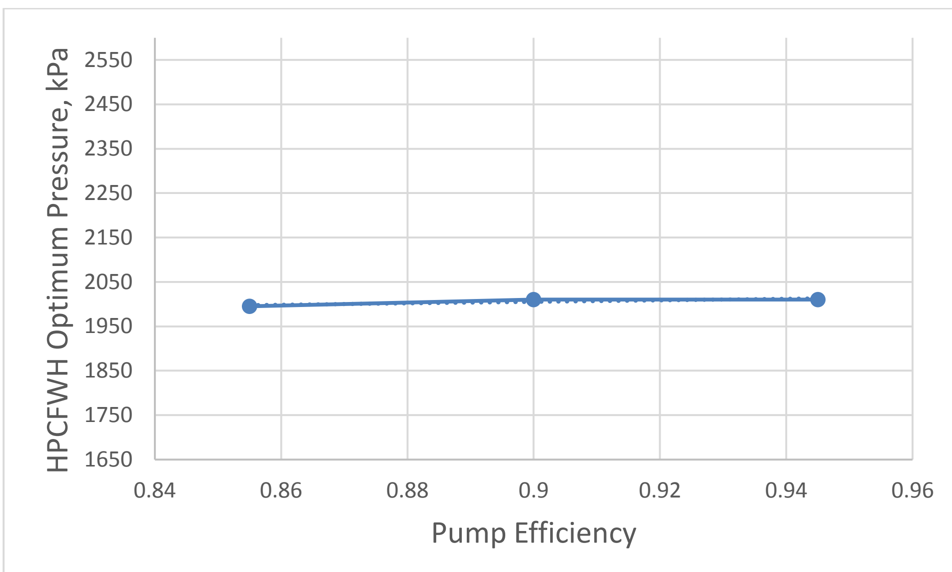

Figure 4.108. Optimum High Pressure Closed FWH Pressure vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

Figure 4.109 illustrates cycle thermal efficiency versus Deaerator pressure. As mentioned before, the efficiency increases very slightly as pump efficiency rises from 0.855 to 0.945, however the Deaerator optimum pressures are regarded as steady (figure 4.110).

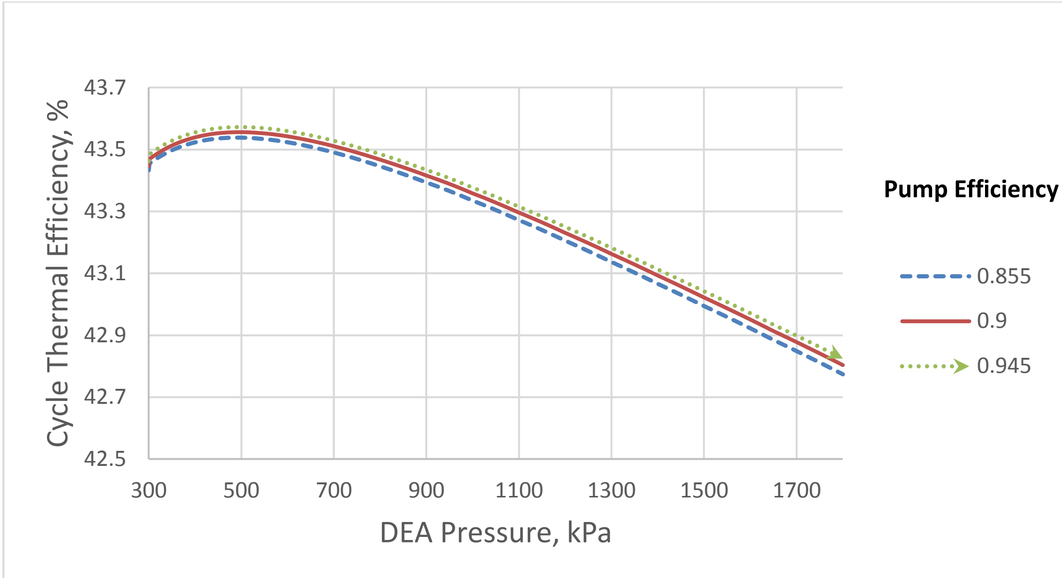

Figure 4.109. Thermal Efficiency vs. DEA Pressure at Different Pump Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

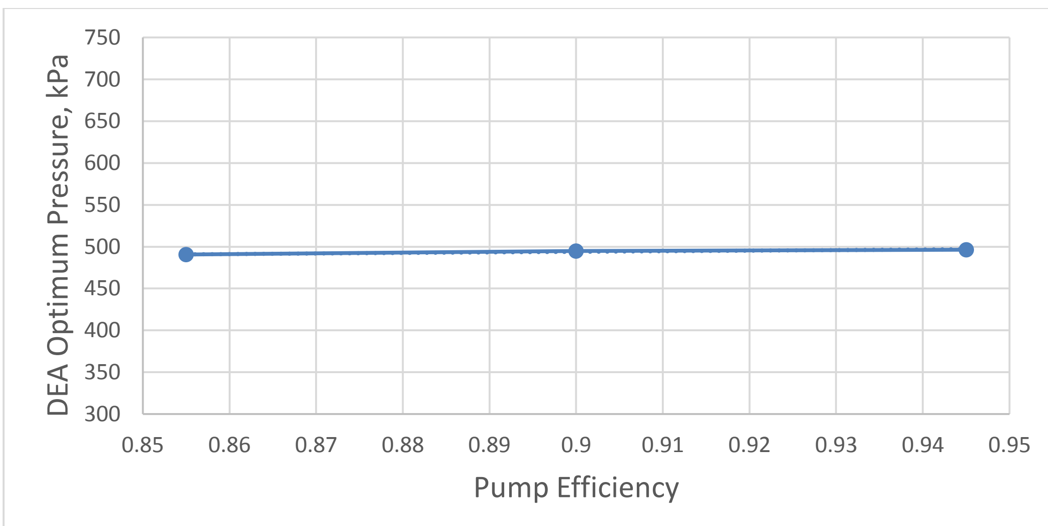

Figure 4.110. Optimum DEA Pressure vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

Figure 4.111 shows the thermal efficiency versus low pressure closed feedwater heater (LPCFWH) pressure for three different pump efficiencies. As it is clear, by changing pump efficiencies the change in thermal efficiency will be insignificant, additionally low pressure closed feedwater heater optimum pressures are absolutely constant (figure 4.112).

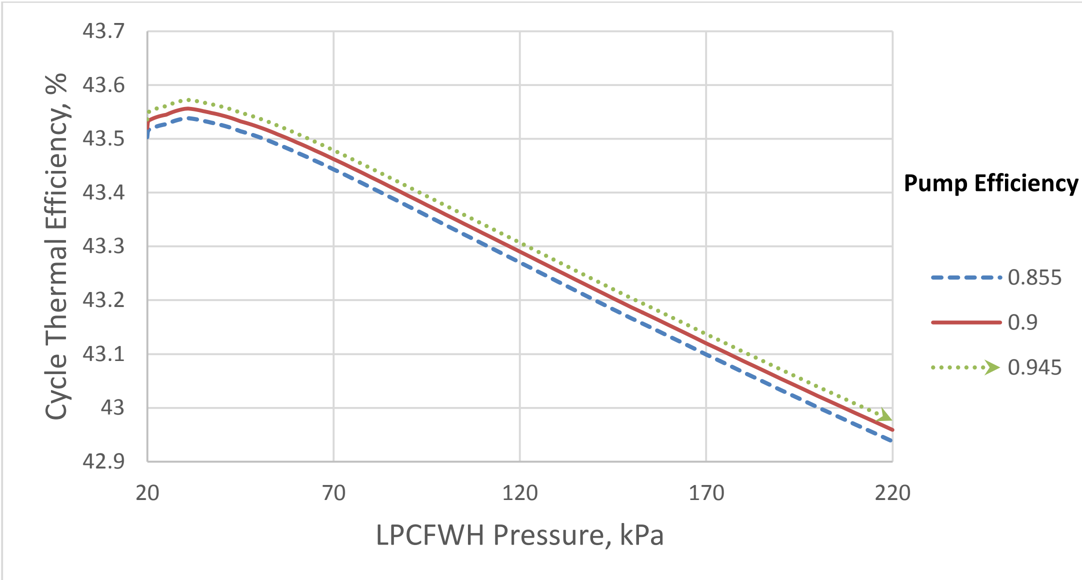

Figure 4.111. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Pump Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

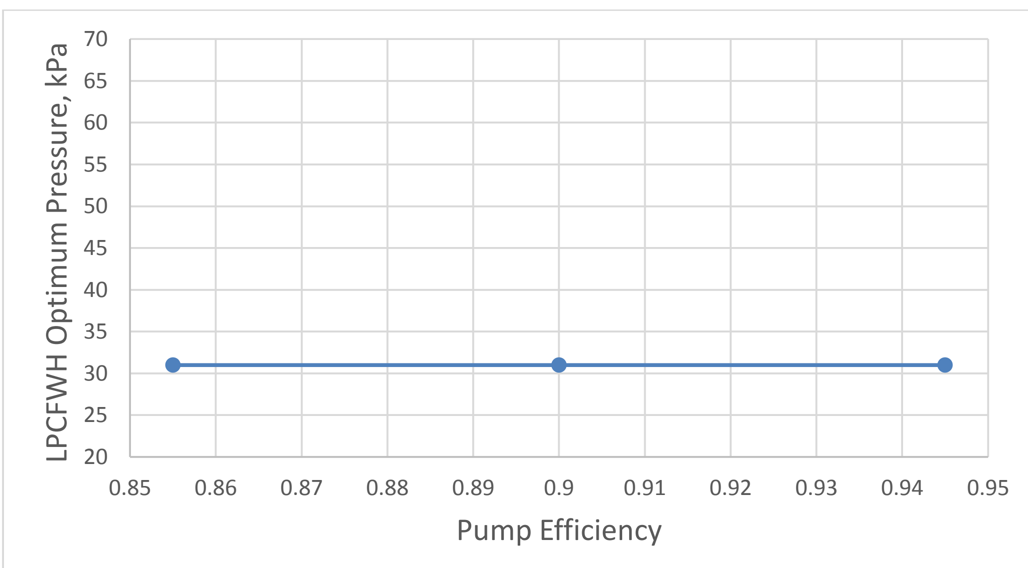

Figure 4.112. Optimum Low Pressure Closed FWH Pressure vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

When the pump efficiency changes from 0.855 to 0.945, the overall thermal efficiency varies only from 43.53% up to 43.57%. Therefore pump efficiency does not have much effect on the overall cycle efficiency. This matter is illustrated by figure 4.113.

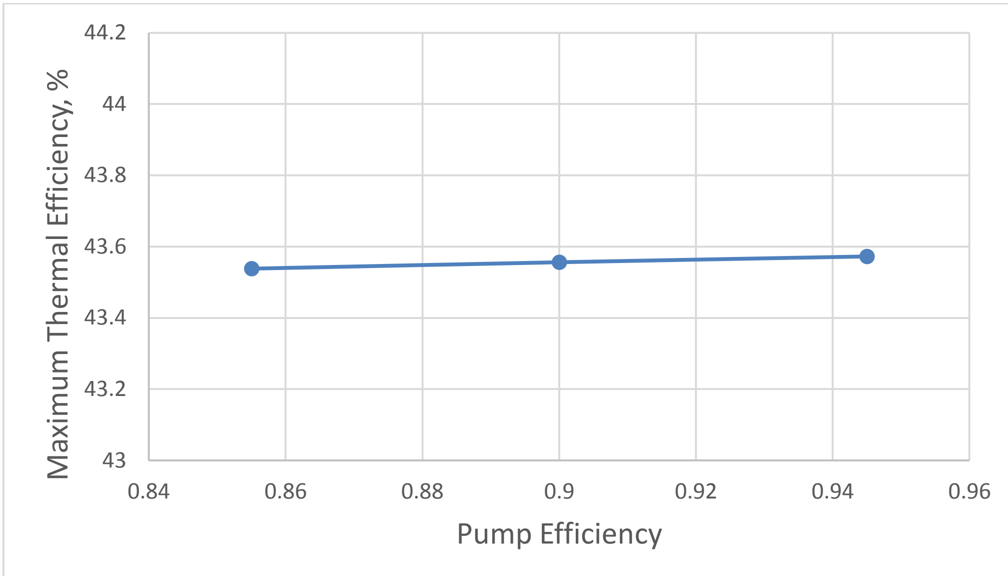

Figure 4.113. Maximum Thermal Efficiency vs. Pump Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

4.3.5 Turbine Efficiency

Figure 4.114 shows cycle thermal efficiency versus high pressure closed feedwater heater (HPCFWH) pressure in three different turbine efficiencies. Contrary to the pump efficiency, which has a slight effect on thermal efficiency, turbine efficiency has much effect on thermal efficiency. Figures 4.114 and 4.120 show that for turbine efficiencies from 0.855 to 0.945, the thermal efficiency increases from around 43 to 44 percent, moreover high pressure closed feedwater heater optimum pressures do not remain constant and they rise by increasing turbine efficiency. What mentioned is shown by figure 4.115.

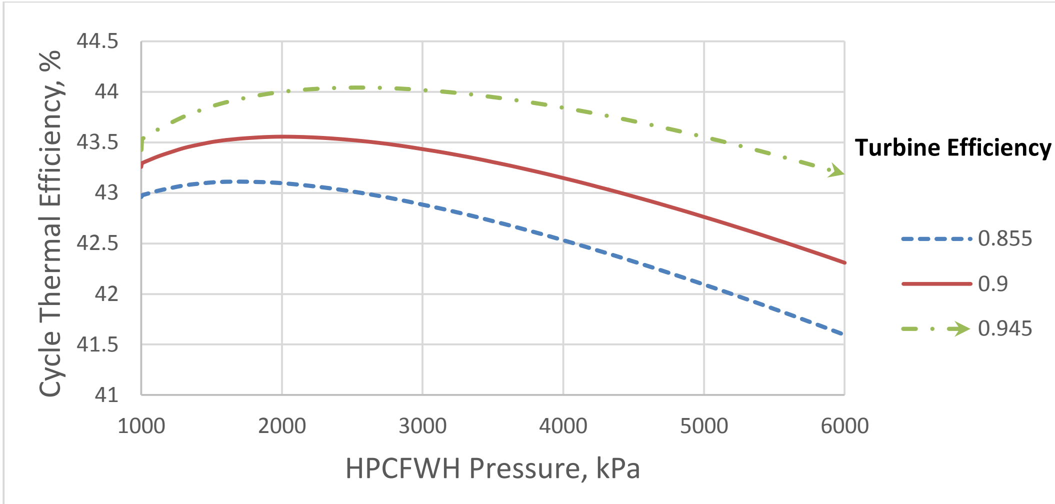

Figure 4.114. Thermal Efficiency vs. High Pressure Closed FWH Pressure at Different Turbine Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

By increasing turbine efficiency, high pressure closed feedwater heater optimal pressures rise from 1700 to 2550 kPa (figure 4.115).

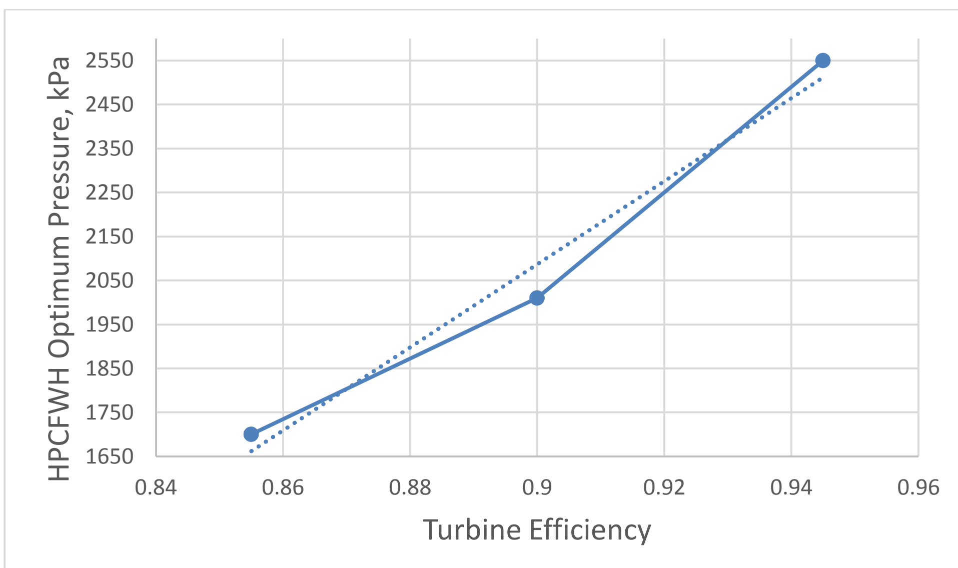

Figure 4.115. Optimum High Pressure Closed FWH Pressure vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

In figure 4.116 for three different turbine efficiencies, deaerator optimum pressure increases from 385.5 to 721.5 kPa, moreover thermal efficiency increases as well.

(Deaerator optimum pressure as mentioned before, is where the efficiency has the maximum amount on each curve.) This is completely observable in figure 4.117.

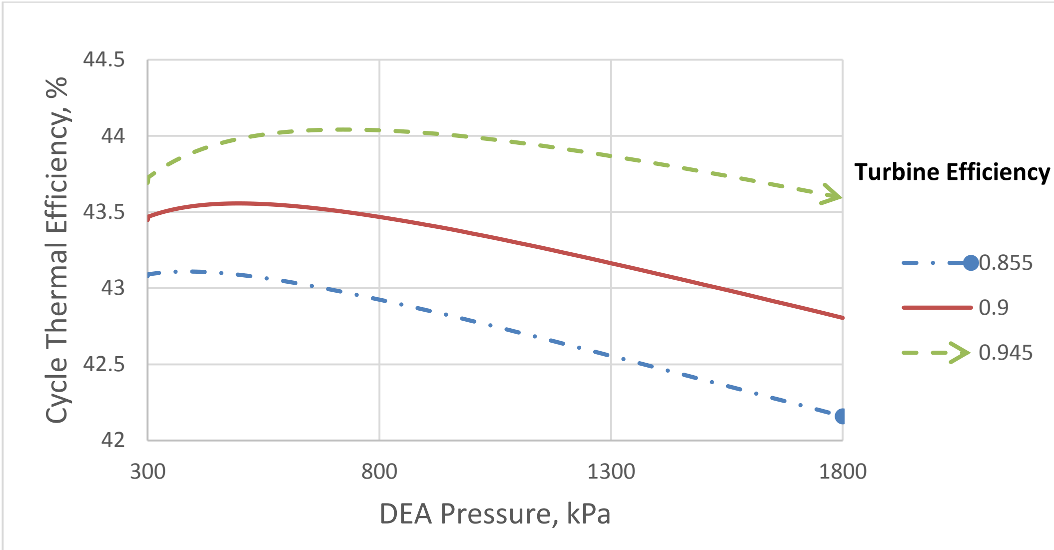

Figure 4.116. Thermal Efficiency vs. DEA Pressure at Different Turbine Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

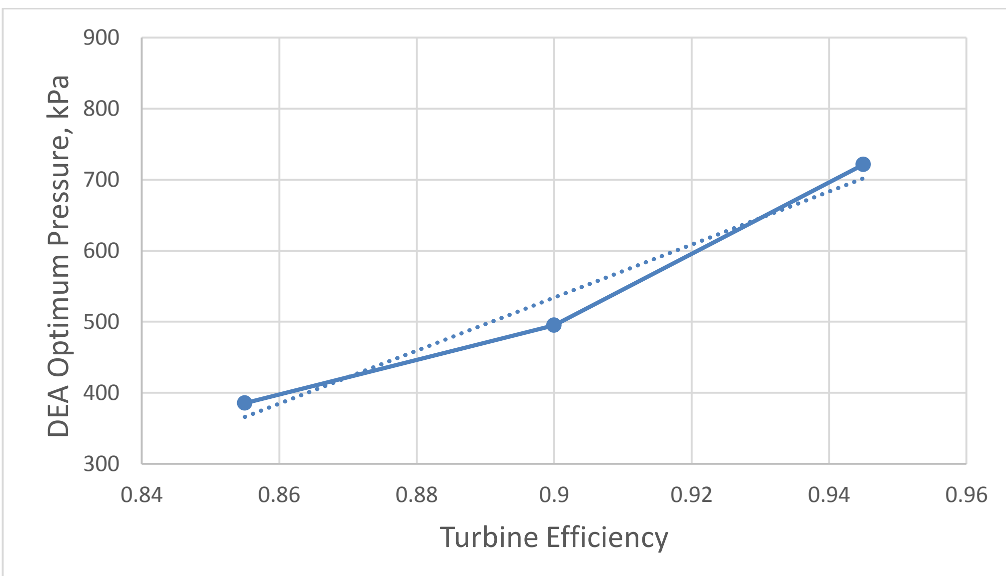

Figure 4.117. Optimum DEA Pressure vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

Figure 4.118 shows efficiency versus low pressure closed feedwater heater (LPCFWH) pressure in three different turbine efficiencies. It has been shown for turbine efficiencies from 0.855 to 0.945, the thermal efficiency increases, moreover low pressure closed feedwater heater optimum pressures (which is the peak on each curve as mentioned before) do not remain constant and rise by increasing turbine efficiency. This is shown by figure 4.119.

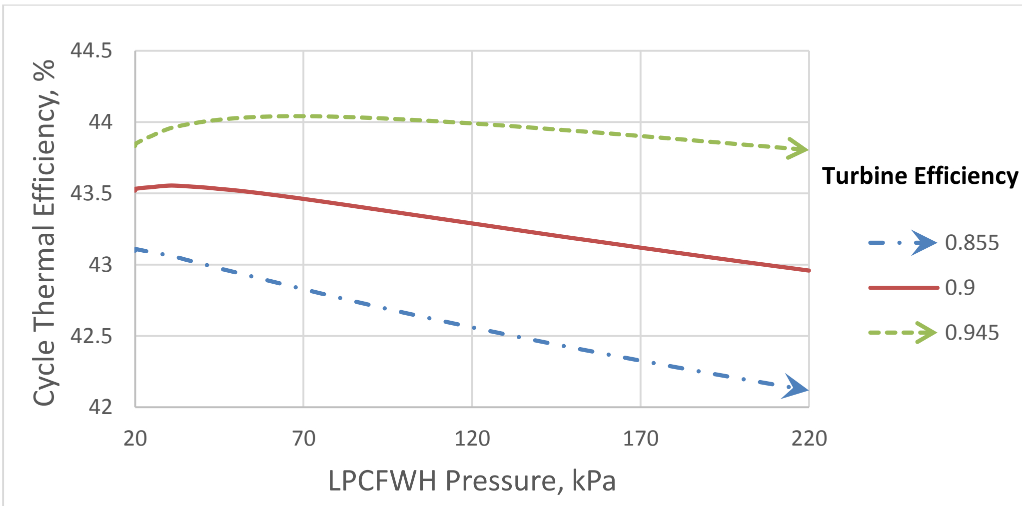

Figure 4.118. Thermal Efficiency vs. Low Pressure Closed FWH Pressure at Different Turbine Efficiencies for the Actual Regenerative Rankine Cycle with 3 FWHs

Low pressure closed feedwater heater optimum pressure increases from 20.2 kPa for 0.855 turbine efficiency to 68 kPa for 0.945 turbine efficiency (figure 4.119).

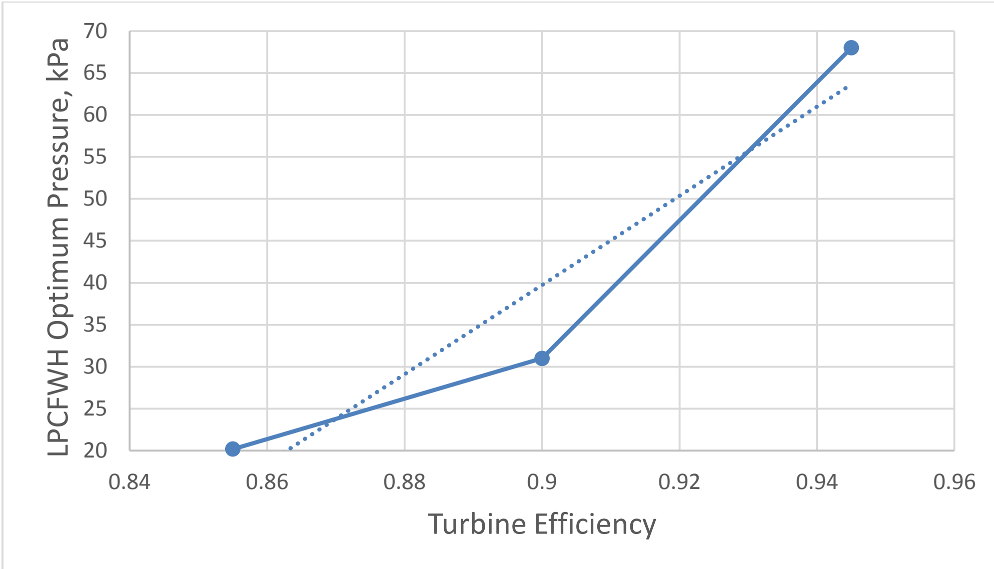

Figure 4.119. Optimum Low Pressure Closed FWH Pressure vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

Contrary to the pump efficiency, which has a slight effect on thermal efficiency, turbine efficiency has much effect on thermal efficiency. For three feedwater heater, efficiency has increased by almost 1 percent from 0.855 to 0.945 turbine efficiency. This is shown in figure 4.120.

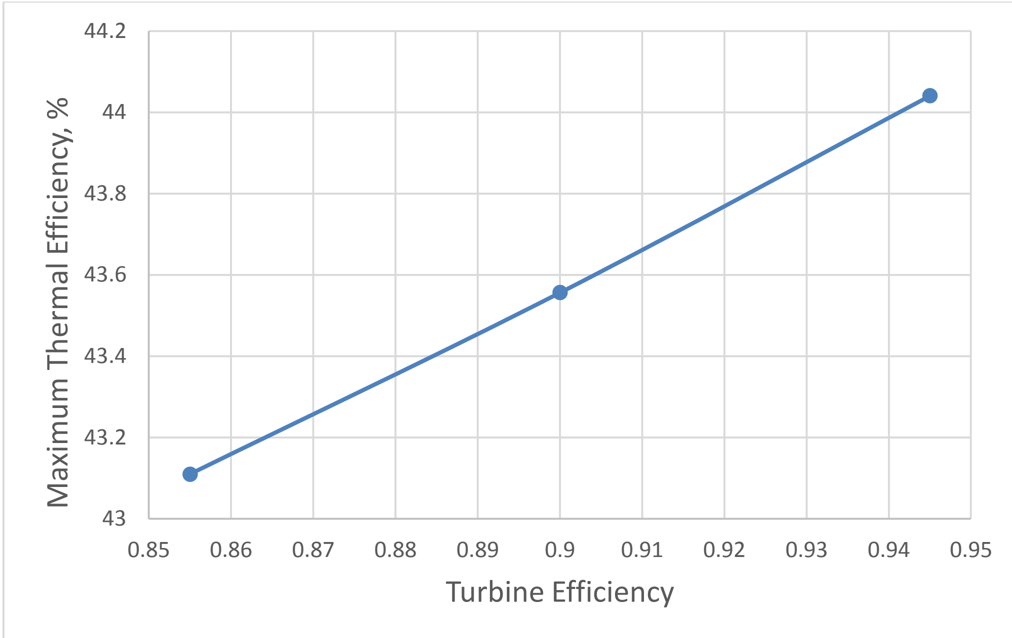

Figure 4.120. Maximum Thermal Efficiency vs. Turbine Efficiency in Actual Regenerative Rankine Cycle with 3 FWHs

The obtained values for three FWHs has been shown in table A.5 and A.6 for ideal and actual cycles respectively.

4.4 Cycle Thermal Efficiency Summary Results

Figures 4.121 and 4.122 and 4.123 show the change in cycle thermal efficiency by changing boiler pressure, boiler temperature and condenser pressure respectively in ideal and actual cycles for 1 FWH, 2 FWHs and 3 FWHs regenerative Rankine cycle. It is clear that, as the number of feedwater heaters increases, the actual cycle efficiency approaches to the ideal cycle efficiency. It is completely obvious that the efficiency of actual cycle for three FWHs Rankine cycle has higher efficiency than the ideal cycle which has one FWH.

Figures 4.121 and 4.122 and 4.123 illustrate that as the number of feedwater heaters increased the cycle thermal efficiency increases. For example, in base cases, efficiency of the 2 FWHs actual cycle is 2.10% more than 1 FWH actual cycle and the efficiency of 3 FWHs actual cycle is 9.76% more than 2 FWHs actual cycle.

It is clear from figures 4.121 and 4.122 and 4.123 that the cycle efficiency is directly related to the boiler pressure and boiler temperature; whereas, the impact of the condenser pressure has a reverse relation on the cycle thermal efficiency, (i.e., by raising boiler pressure and temperature, cycle thermal efficiency increases, however when condenser pressure rises, cycle thermal efficiency decreases.

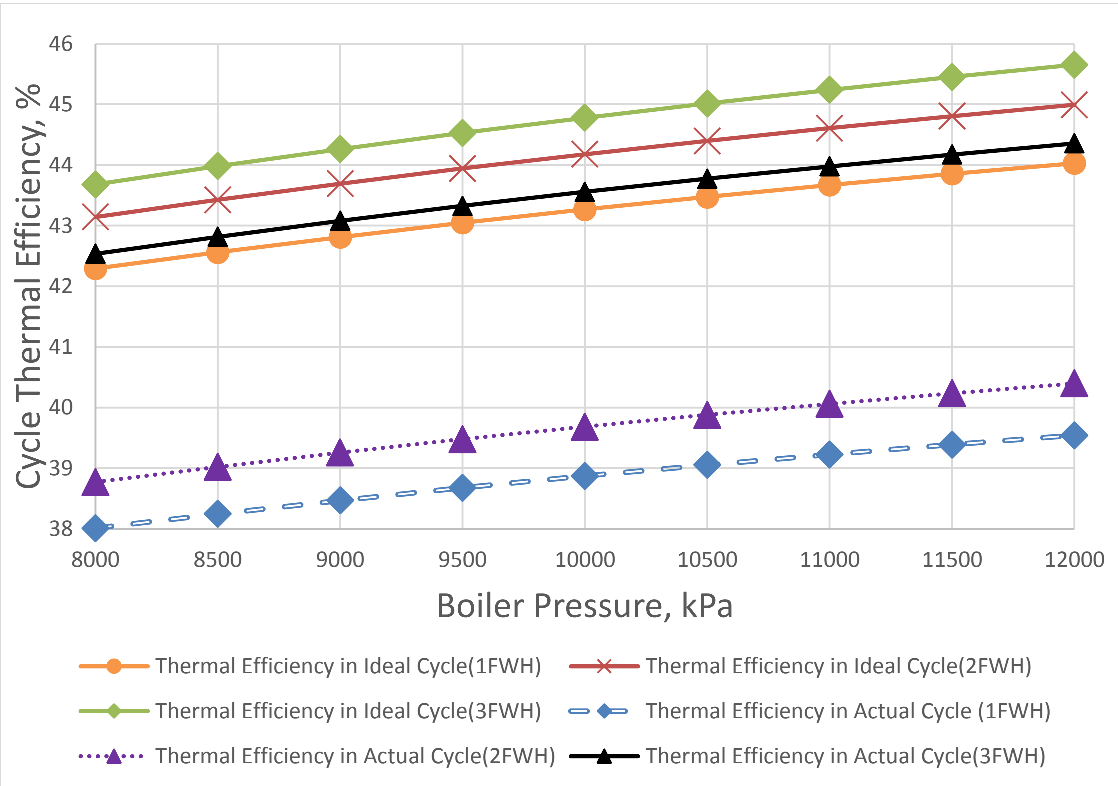

Figure 4.121. Change in Cycle Thermal Efficiency by Changing Boiler Pressure in Ideal & Actual Cycles for 1FWH, 2FWHs and 3FWHs Regenerative Rankine Cycle

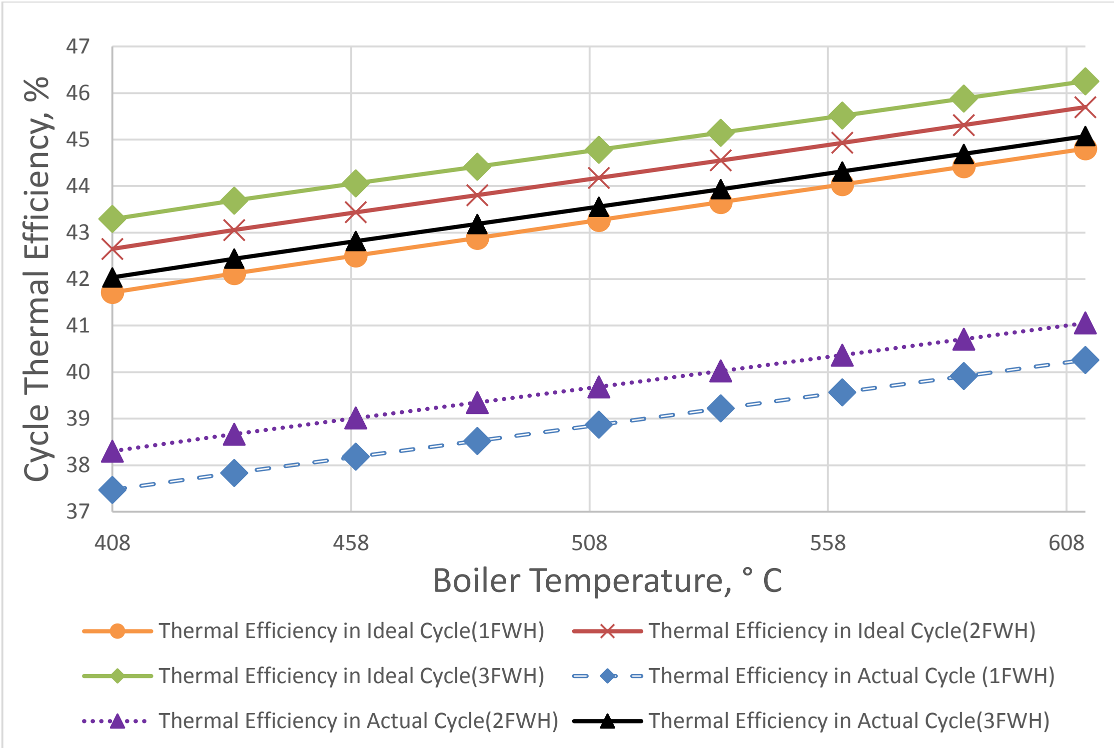

Figure 4.122. Change in Cycle Thermal Efficiency by Changing Boiler Temperature in Ideal & Actual Cycles for 1FWH, 2FWHs and 3FWHs Regenerative Rankine Cycle

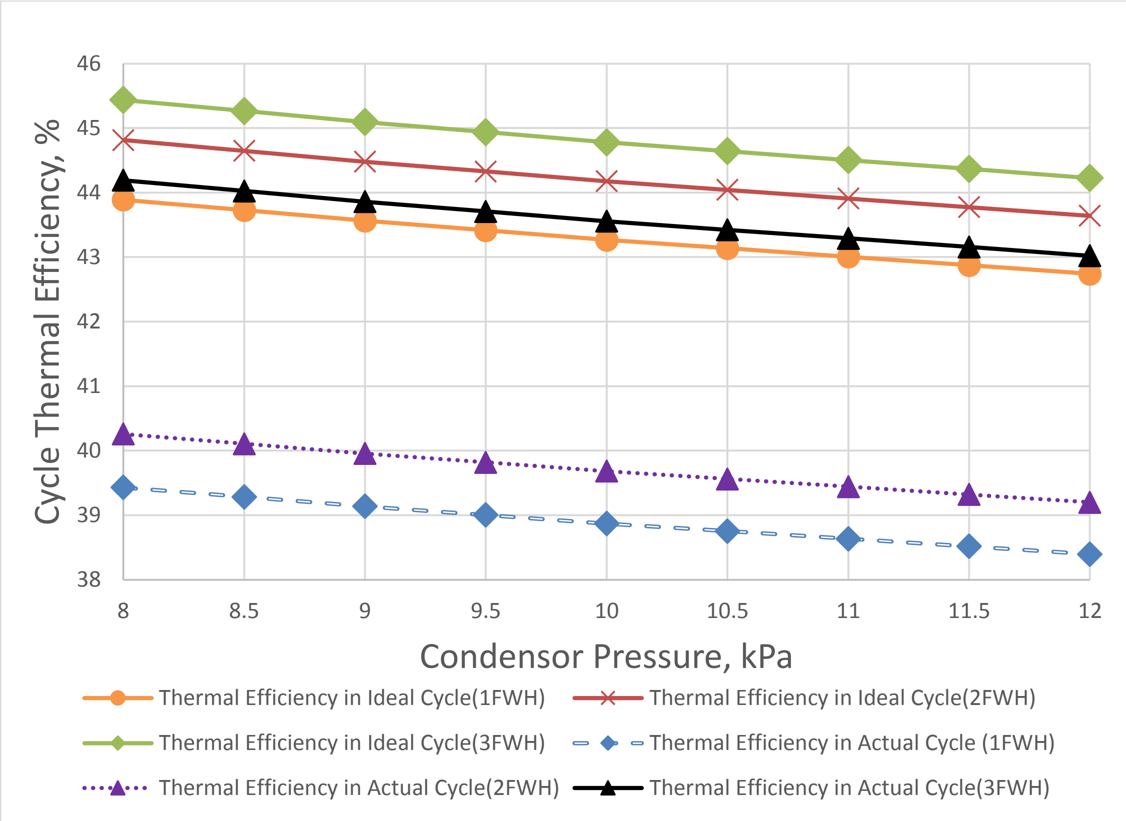

Figure 4.123. Change in Cycle Thermal Efficiency by Changing Condenser Pressure in Ideal & Actual Cycles for 1FWH, 2FWHs and 3FWHs Regenerative Rankine Cycle

Figure 4.124 and 4.125 show the changes in the cycle thermal efficiency as the pump and turbine efficiencies were changed. The impact of pump efficiency on thermal efficiency of the cycle is almost negligible so its impact can be neglected compared with turbine efficiency. The change in the cycle efficiency was about 0.04% in all the cycles (from 1 FWH to 3 FWHs) however turbine efficiency has a huge effect on the thermal efficiency. It is around 1.11% for 3 FWHs cycle and 5% for 1 FWH and 2 FWHs cycles.

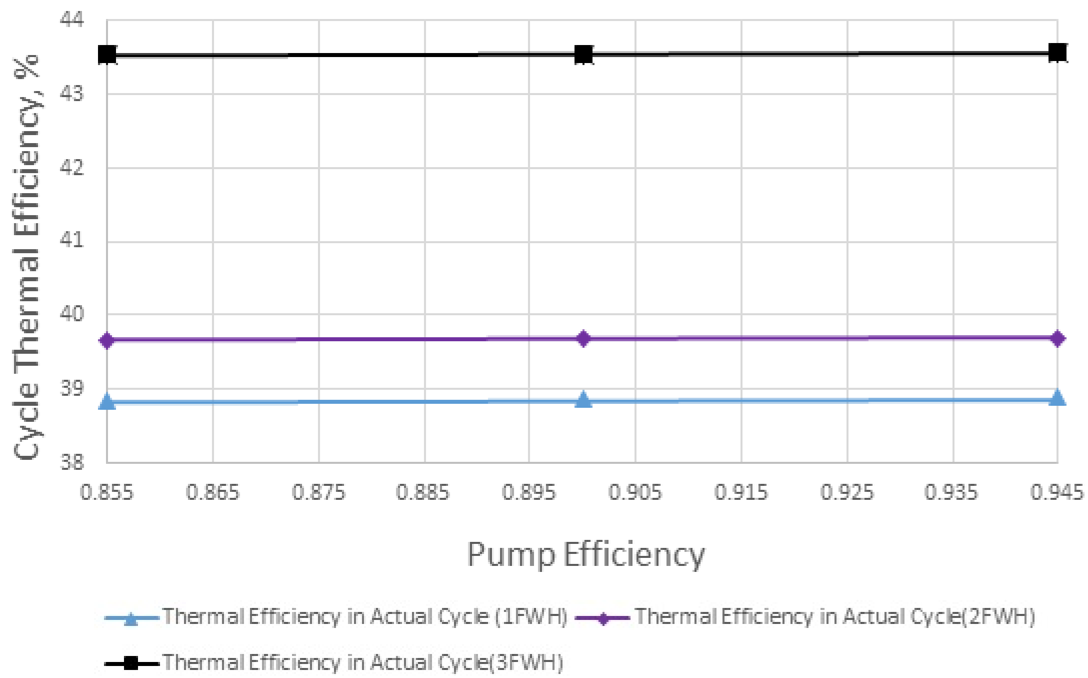

Figure 4.124. Change in Cycle Thermal Efficiency by Changing Pump Efficiency in Actual Cycles for 1FWH, 2FWHs and 3FWHs Regenerative Rankine Cycle

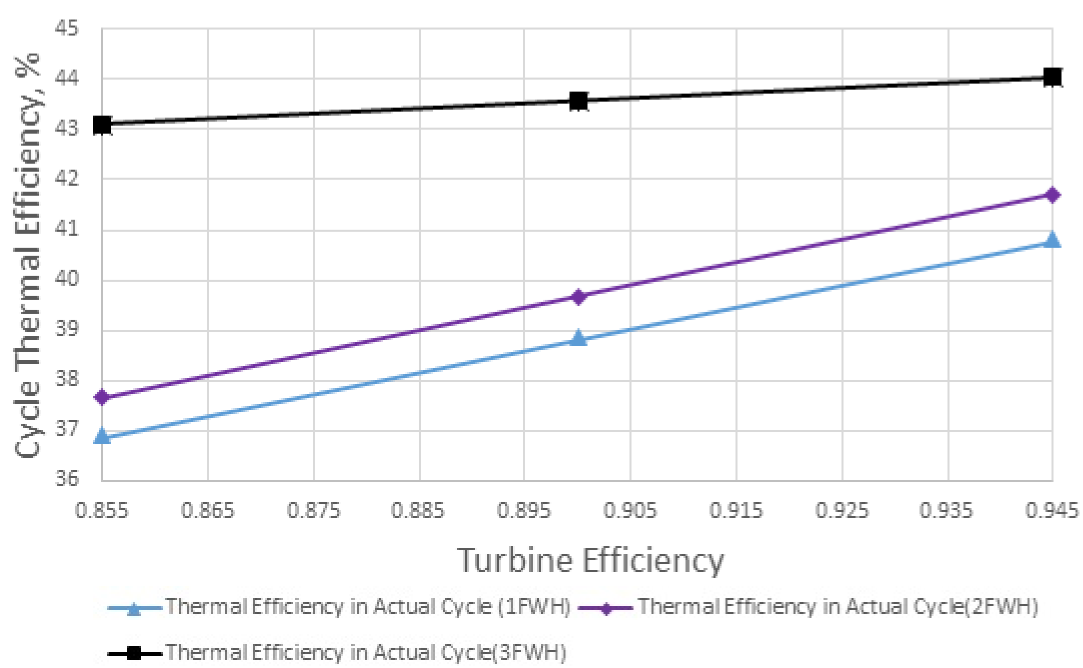

Figure 4.125. Change in Cycle Thermal Efficiency by Changing Turbine Efficiency in Actual Cycles for 1FWH, 2FWHs and 3FWHs Regenerative Rankine Cycle

Chapter 5

CONCLUSION

In this study, optimization of feedwater heaters pressure of single, double and triple-staged regenerative Rankine cycle has been investigated as well as thermal efficiency of the cycle by utilizing VisSim software. Simulation has been performed in both ideal and actual case for each of these three cycles. Actual cycles have less efficiency than the ideal one. Cycle with three feedwater heaters is more efficient than two feedwater heater cycle and the latter has higher efficiency than one feedwater heater cycle. Feedwater heaters optimum pressure has varied from single-staged cycle to triple-staged cycle. Although in one feedwater heater cycle the amounts of feedwater heater optimum pressure for ideal and actual could be considered almost the same as well as in two feedwater heater cycle for corresponding feedwaters. Nevertheless, in three feedwater heater cycle, feedwater heater optimum pressure in actual cycle is much less than those in ideal cycle for corresponding feedwater heaters.

By increasing the condenser pressure, efficiency decreases which illustrates that there is a reverse relation between them, however by rising all the other parameters like boiler pressure and temperature and pump and turbine efficiency, thermal efficiency increases. On the other hand, as the number of feedwater heater increased, thermal efficiency of the actual cycle approached to thermal efficiency of the ideal one. Based on the acquired data from VisSim software, in actual base one, two and three FWHs, the efficiency was 38.87%, 39.68% and 43.55% respectively and the change from ideal cycle was 10.16% decrease, 10.17% decrease and 2.73% decrease respectively. Even the thermal efficiency of an actual cycle with 3 FWHs (43.55% for actual base) has become 0.67% more than 1 FWH regenerative Rankine cycle working on an ideal cycle (43.26% for ideal base).

In base cases, the efficiency of 2 FWHs actual cycle is 2.10% more than 1 FWH actual cycle and the efficiency of 3 FWHs actual cycle is 9.76% more than 2 FWHs actual cycle. Therefore as the number of FWHs increased the cycle thermal efficiency increases.

Turbine efficiency (which changed from -5% to +5% of the initial base value) has the greatest effect on thermal efficiency in all the cycles. It causes 5% change (Table A.2 and A.4) in cycle thermal efficiency for 1 FWH and 2 FWHs regenerative Rankine cycles and 1.11% change (Table A.6) in cycle thermal efficiency for 3 FWHs regenerative Rankine cycles. Boiler temperature (which changed from -20% to +20% of the initial base value) is the second most significant parameter which affected thermal efficiency. It caused 3.5% change in thermal efficiency (for 1 FWH, 2 FWHs and 3 FWHs regenerative Rankine cycles). After that the other parameters are boiler pressure (which causes 1.80% change in thermal efficiency for 1 FWH, 2 FWHs and 3 FWHs regenerative Rankine cycles by changing it from -20% to +20% of the initial base value) and condenser pressure (which causes 1.40% change in thermal efficiency for 1 FWH, 2 FWHs and 3 FWHs regenerative Rankine cycles by changing it from -20% to +20% of the initial base value) in consecutive priority. The impact of pump efficiency (which changed from -5% to +5% of the initial base value) on thermal efficiency of the cycle is almost negligible so its impact can be neglected compared with turbine efficiency. The change in the cycle efficiency was about 0.04% in all the cycles (from 1 FWH to 3 FWHs)

It can be concluded that in one and two feedwater heaters cycles, the most important operating parameter which has the highest impact on the cycle efficiency is the turbine efficiency (which changed cycle thermal efficiency by 5%) nonetheless in three feedwater heaters cycle, changing of boiler temperature (which changed cycle thermal efficiency by 3.5%) take priority over the other parameters in affecting thermal efficiency of the cycle.

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APPENDIX

Appendix A: Efficiency and FWHs Optimum Pressure Values

Table A.1. Efficiency and 1 Open FWH Optimum Pressures in One FWH Ideal Cycle

	Efficiency (%)	OFWH Optimum Pressure (kPa)	Percent of Efficiency change to Ideal
Ideal Base Case (1FWH)	43.266759	950	0
Boiler Pressure - %20=8000	42.292751	798	-2.25116931
Boiler Pressure - %15=8500	42.560323	822	-1.63274536
Boiler Pressure - %10=9000	42.809793	874	-1.056159533
Boiler Pressure - %5=9500	43.046529	942	-0.509005077
Boiler Pressure =10000	43.266759	950	0
Boiler Pressure + %5 =10500	43.473554	974	0.477953525
Boiler Pressure + %10=11000	43.669373	1034	0.930538846
Boiler Pressure + %15=11500	43.855104	1126	1.359808346
Boiler Pressure + %20=12000	44.029675	1142	1.763284373
Condenser Pressure -%20=8	43.889251	934	1.438730366
Condenser Pressure -%15=8.5	43.726954	946	1.063622538
Condenser Pressure -%10=9	43.563254	950	0.685272035
Condenser Pressure -%5=9.5	43.415614	950	0.344040098
Condenser Pressure =10	43.266759	950	0
Condenser Pressure +%5=10.5	43.136879	954	-0.300184259
Condenser Pressure +%10=11	43.006115	962	-0.602411657
Condenser Pressure +%15=11.5	42.874467	970	-0.906682195
Condenser Pressure +%20=12	42.74192	1026	-1.213030539
Boiler Temperature -%20=408	41.716714	842	-3.582530875
Boiler Temperature -%15=433.5	42.121027	882	-2.648065227
Boiler Temperature -%10=459	42.504178	914	-1.762510106
Boiler Temperature -%5=484.5	42.882378	974	-0.88839795
Boiler Temperature =510	43.266759	950	0
Boiler Temperature +%5=535.5	43.652348	950	0.891189932
Boiler Temperature +%10=561	44.036429	950	1.778894509
Boiler Temperature +%15=586.5	44.419473	934	2.664202327
Boiler Temperature +%20=612	44.804084	926	3.553131863

Table A.2. Efficiency and 1 Open FWH Optimum Pressures in One FWH Actual Cycle

	Efficiency (%)	OFWH Optimum Pressure (kPa)	Percent of Efficiency change to Actual
Actual Base Case (1FWH)	38.870063	950	-10.161833 (change to ideal)
Boiler Pressure - %20=8000	38.008797	794	-2.215756635
Boiler Pressure - %15=8500	38.245922	818	-1.605711316
Boiler Pressure - %10=9000	38.466542	874	-1.03812798
Boiler Pressure - %5=9500	38.6756	930	-0.500289902
Boiler Pressure =10000	38.870063	950	0
Boiler Pressure + %5 =10500	39.052361	966	0.468993323
Boiler Pressure + %10=11000	39.224444	1034	0.911706781
Boiler Pressure + %15=11500	39.387191	1126	1.330401754
Boiler Pressure + %20=12000	39.540436	1118	1.724651179
Condenser Pressure -%20=8	39.430739	926	1.442436561
Condenser Pressure -%15=8.5	39.284518	934	1.066257598
Condenser Pressure -%10=9	39.137051	946	0.686873083
Condenser Pressure -%5=9.5	39.004104	950	0.344843794
Condenser Pressure =10	38.870063	950	0
Condenser Pressure +%5=10.5	38.753107	950	-0.300889659
Condenser Pressure +%10=11	38.635313	950	-0.603935219
Condenser Pressure +%15=11.5	38.516707	962	-0.90906979
Condenser Pressure +%20=12	38.397298	970	-1.216270218
Boiler Temperature -%20=408	37.467294	834	-3.608867318
Boiler Temperature -%15=433.5	37.833276	862	-2.667314946
Boiler Temperature -%10=459	38.180032	894	-1.775224805
Boiler Temperature -%5=484.5	38.522024	974	-0.895390882
Boiler Temperature =510	38.870063	950	0
Boiler Temperature +%5=535.5	39.218902	950	0.897449021
Boiler Temperature +%10=561	39.566303	942	1.791198538
Boiler Temperature +%15=586.5	39.91275	926	2.682493723
Boiler Temperature +%20=612	40.260443	926	3.57699446
Pump Efficiency -%5=0.855	38.854998	950	-0.038757334
Pump Efficiency =0.9	38.870063	950	0
Pump Efficiency +%5=0.945	38.883686	950	0.035047538
Turbine Efficiency -%5=0.855	36.90354	950	-5.059222569
Turbine Efficiency =0.9	38.870063	950	0
Turbine Efficiency +%5=0.945	40.836585	950	5.059219997

Table A.3. Efficiency and 1 Closed and 1 Open FWHs Optimum Pressures in 2FWHs Ideal Cycle

	Efficiency (%)	CFWH Optimum Pressure (kPa)	OFWH Optimum Pressure (kPa)	Percent of Efficiency change to Ideal
Ideal Base Case (2FWH)	44.176141	1876	405	0
Boiler Pressure - %20=8000	43.142277	1552	335	-2.340322121
Boiler Pressure - %15=8500	43.425971	1680	362.5	-1.698133841
Boiler Pressure - %10=9000	43.691935	1712	367.5	-1.096080348
Boiler Pressure - %5=9500	43.94182	1824	392.5	-0.53042433
Boiler Pressure = 10000	44.176141	1876	405	0
Boiler Pressure + %5 =10500	44.397347	1948	417.5	0.500736359
Boiler Pressure + %10=11000	44.606451	2048	440	0.97407784
Boiler Pressure + %15=11500	44.804708	2088	447.5	1.422865343
Boiler Pressure + %20=12000	44.991666	2188	470	1.84607569
Condenser Pressure -%20=8	44.814259	1824	375	1.44448561
Condenser Pressure -%15=8.5	44.64785	1848	387.5	1.067791322
Condenser Pressure -%10=9	44.480065	1848	390	0.68798223
Condenser Pressure -%5=9.5	44.328701	1852	392.5	0.345344787
Condenser Pressure = 10	44.176141	1876	405	0
Condenser Pressure +%5=10.5	44.043282	1880	407.5	-0.300748316
Condenser Pressure +%10=11	43.909556	1912	415	-0.603459229
Condenser Pressure +%15=11.5	43.775139	1948	430	-0.907734336
Condenser Pressure +%20=12	43.63976	1948	432.5	-1.214187088
Boiler Temperature -%20=408	42.650266	1740	385	-3.454070377
Boiler Temperature -%15=433.5	43.055587	1836	402.5	-2.536559271
Boiler Temperature -%10=459	43.436263	1852	402.5	-1.674836197
Boiler Temperature -%5=484.5	43.805204	1876	405	-0.839677237
Boiler Temperature = 510	44.176141	1876	405	0
Boiler Temperature +%5=535.5	44.552354	1876	405	0.851620335
Boiler Temperature +%10=561	44.930498	1880	407.5	1.707611808
Boiler Temperature+%15=586.5	45.310528	1880	410	2.567872554
Boiler Temperature +%20=612	45.694892	1940	420	3.437944025

Table A.4. Efficiency and 1 Closed and 1 Open FWHs Optimum Pressures in 2FWHs Actual Cycle

	Efficiency (%)	CFWH Optimum Pressure (kPa)	OFWH Optimum Pressure (kPa)	Percent of Efficiency change to Actual
Actual Base Case (2FWH)	39.68263	1852	395	-10.171805 (% change to ideal)
Boiler Pressure - %20=8000	38.768833	1544	332.5	-2.302763199
Boiler Pressure - %15=8500	39.020438	1600	345	-1.668720042
Boiler Pressure - %10=9000	39.25525	1700	365	-1.076995149
Boiler Pressure - %5=9500	39.476295	1748	372.5	-0.519963017
Boiler Pressure = 10000	39.68263	1852	395	0
Boiler Pressure + %5 =10500	39.877372	1944	415	0.490748723
Boiler Pressure + %10=11000	40.061147	2036	435	0.953860669
Boiler Pressure + %15=11500	40.235333	2072	442.5	1.39280839
Boiler Pressure + %20=12000	40.39906	2172	465	1.805399491
Condenser Pressure -%20=8	40.257881	1748	357.5	1.449629221
Condenser Pressure -%15=8.5	40.107492	1844	385	1.070649803
Condenser Pressure -%10=9	39.956367	1848	387.5	0.68981567
Condenser Pressure -%5=9.5	39.820032	1852	392.5	0.346252252
Condenser Pressure = 10	39.68263	1852	395	0
Condenser Pressure +%5=10.5	39.562862	1856	397.5	-0.301814673
Condenser Pressure +%10=11	39.442388	1876	407.5	-0.605408462
Condenser Pressure +%15=11.5	39.321089	1888	412.5	-0.911081246
Condenser Pressure +%20=12	39.199223	1944	430	-1.218182867
Boiler Temperature -%20=408	38.300406	1720	377.5	-3.483196552
Boiler Temperature -%15=433.5	38.667457	1816	395	-2.558230137
Boiler Temperature -%10=459	39.01239	1844	395	-1.689000956
Boiler Temperature -%5=484.5	39.346683	1852	395	-0.846584513
Boiler Temperature = 510	39.68263	1852	395	0
Boiler Temperature +%5=535.5	40.02312	1852	395	0.858032847
Boiler Temperature +%10=561	40.365276	1872	405	1.720264005
Boiler Temperature +%15=586.5	40.709043	1872	407.5	2.586554873
Boiler Temperature +%20=612	41.056535	1936	420	3.46223272
Pump Efficiency -%5=0.855	39.66511	1852	395	-0.0441503
Pump Efficiency = 0.9	39.68263	1852	395	0
Pump Efficiency +%5=0.945	39.698481	1852	395	0.039944429
Turbine Efficiency -%5=0.855	37.674827	1852	395	-5.059652044
Turbine Efficiency = 0.9	39.68263	1852	395	0
Turbine Efficiency +%5=0.945	41.690434	1852	395	5.059654564

Table A.5. Efficiency and 2 Closed and 1 Open FWHs Optimum Pressures in 3FWHs Ideal Cycle

	Efficiency (%)	HPCFWH Optimum Pressure (kPa)	OFWH Optimum Pressure (kPa)	LPCFWH Optimum Pressure (kPa)	Percent of Efficiency change to Ideal
Ideal Base Case (3FWH)	44.779609	3245	1069.5	160	0
Boiler Pressure - %20=8000	43.678972	2735	930	158.4	-2.457897745
Boiler Pressure - %15=8500	43.980781	2765	942	159.2	-1.783910172
Boiler Pressure - %10=9000	44.262502	2880	964.5	160	-1.154782303
Boiler Pressure - %5=9500	44.529542	3125	1035	160	-0.558439445
Boiler Pressure = 10000	44.779609	3245	1069.5	160	0
Boiler Pressure + %5 =10500	45.015188	3280	1083	161.8	0.526085433
Boiler Pressure + %10=11000	45.238925	3435	1150.5	174.6	1.025725794
Boiler Pressure + %15=11500	45.452355	3730	1255.5	183	1.502348982
Boiler Pressure + %20=12000	45.653078	3770	1267.5	184.2	1.950595415
Condenser Pressure -%20=8	45.436954	3245	1069.5	160	1.467956096
Condenser Pressure -%15=8.5	45.265747	3245	1069.5	160	1.085623593
Condenser Pressure -%10=9	45.09297	3245	1069.5	160	0.699785029
Condenser Pressure -%5=9.5	44.936936	3245	1069.5	160	0.351336252
Condenser Pressure = 10	44.779609	3245	1069.5	160	0
Condenser Pressure +%5=10.5	44.6426	3245	1074	163.2	-0.305962922
Condenser Pressure +%10=11	44.50494	3280	1122	174.8	-0.61337963
Condenser Pressure +%15=11.5	44.366429	3280	1125	177	-0.922696757
Condenser Pressure +%20=12	44.227019	3280	1125	181.4	-1.234021494
Boiler Temperature -%20=408	43.29169	3135	1042.5	160	-3.322760143
Boiler Temperature -%15=433.5	43.689085	3160	1047	160	-2.435313805
Boiler Temperature -%10=459	44.057639	3235	1069.5	160	-1.612274015
Boiler Temperature -%5=484.5	44.416829	3245	1081.5	161.8	-0.810145529
Boiler Temperature = 510	44.779609	3245	1069.5	160	0
Boiler Temperature +%5=535.5	45.146327	3245	1063.5	160	0.81893971
Boiler Temperature +%10=561	45.513742	3240	1054.5	160	1.639435932
Boiler Temperature+%15=586.5	45.882245	3230	1047	161.4	2.462361831
Boiler Temperature +%20=612	46.254316	3230	1041	162.8	3.293255642

Table A.6. Efficiency and 2 Closed and 1 Open FWHs Optimal Pressures in 3FWHs Actual Cycle

	Efficiency (%)	HPCFWH Optimum Pressure (kPa)	OFWH Optimum Pressure (kPa)	LPCFWH Optimum Pressure (kPa)	Percent of Efficiency change to Actual
Actual Base Case (3FWH)	43.556098	2010	495	31	-2.732295 (% change to ideal)
Boiler Pressure - %20=8000	42.535876	1700	426	30	-2.342317257
Boiler Pressure - %15=8500	42.815964	1750	435	30	-1.699266082
Boiler Pressure - %10=9000	43.078294	1890	468	30.8	-1.096985318
Boiler Pressure - %5=9500	43.325614	1960	484.5	30.8	-0.529165859
Boiler Pressure = 10000	43.556098	2010	495	31	0
Boiler Pressure + %5 =10500	43.773158	2055	507	31	0.498345834
Boiler Pressure + %10=11000	43.97806	2155	526.5	31	0.968778241
Boiler Pressure + %15=11500	44.172716	2265	556.5	31	1.415686961
Boiler Pressure + %20=12000	44.356179	2350	576	31.8	1.836897786
Condenser Pressure -%20=8	44.191078	1995	487.5	30	1.457844089
Condenser Pressure -%15=8.5	44.025656	1995	489	30	1.078053411
Condenser Pressure -%10=9	43.858828	1995	490.5	30.8	0.695034711
Condenser Pressure -%5=9.5	43.708086	1995	492	30.8	0.348947695
Condenser Pressure = 10	43.556098	2010	495	31	0
Condenser Pressure +%5=10.5	43.423799	2010	495	31	-0.303743921
Condenser Pressure +%10=11	43.290537	2010	496.5	31	-0.609698784
Condenser Pressure +%15=11.5	43.156301	2010	496.5	31	-0.917889844
Condenser Pressure +%20=12	43.021128	2025	504	31.6	-1.228232152
Boiler Temperature -%20=408	42.034985	1935	486	31	-3.492307782
Boiler Temperature -%15=433.5	42.439367	1985	492	31	-2.563891283
Boiler Temperature -%10=459	42.817704	1985	489	31	-1.695271234
Boiler Temperature -%5=484.5	43.185381	1985	489	31	-0.851125369
Boiler Temperature = 510	43.556098	2010	495	31	0
Boiler Temperature +%5=535.5	43.932761	2035	505.5	31	0.86477673
Boiler Temperature +%10=561	44.311912	2045	511.5	31	1.735265634
Boiler Temperature +%15=586.5	44.692615	2050	519	30.8	2.609317758
Boiler Temperature +%20=612	45.075198	2050	514.5	30.8	3.487686156
Pump Efficiency -%5=0.855	43.538124	1995	490.5	31	-0.041266323
Pump Efficiency = 0.9	43.556098	2010	495	31	0
Pump Efficiency +%5=0.945	43.572386	2010	496.5	31	0.037395453
Turbine Efficiency -%5=0.855	43.109629	1700	385.5	20.2	-1.025043612
Turbine Efficiency = 0.9	43.556098	2010	495	31	0
Turbine Efficiency +%5=0.945	44.040616	2550	721.5	68	1.112399922