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Performance of an ideal reheat–regenerative Rankine cycle power plant utilizing solid waste incineration

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Abstract

The performance of a100-MW reheat regenerative Rankine cycle steam power plant utilizing incinerator flue gas is reported in this paper.A computer model was created using mass and energy balances. The plant was simulated at three boiler pressures (15, 10 and 5 MPa). High- and low-pressure turbine stages were studied. Before reheating, steam was bled after high-pressure turbine and along the low-pressure stage for regeneration. Superheating the steam up to 1200°C increased thermal efficiency (based on heat input and turbine work, E_{th} and η_{th}) from 35 to 67% and 55 to 77%, respectively, and the total workdone from 1000 to 3,500 kJ/kg. Superheating reduced steam and fuel requirements from. Reheating the steam increased the boiler heat load and the total work done by the turbine, while the contribution of low-pressure turbine was increased from 50 to 83%, and increased E_{th} and η_{th} from 35 to 55%. Increasing the regeneration intensity (y and z from 8 to 28% and 12 to 15%, respectively) increased the boiler heat duty, lowered and increased the condenser duty, was but increased and lowered total work done, respectively. Similarly, regeneration increased E_{th} and η_{th} from 25 to 57% and 35 to 52.5%, respectively. Lower boiler pressure led to lower steam mass flow rate for the same power generation capacity, high heat requirements from the boiler, high total work done on the turbine, but lower contribution of low-pressure turbine to the total work done (50 to 72% compared to 65 to 83% at 15 MPa).

*Keywords:*Steam power plant, Boiler pressure, Reheat regenerative Rankinecycle, Feed water heater, high- and low-pressure turbines, Thermal efficiency.

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1. Introduction

1.1 A background of the study

Due to escalating environmental concerns and the pressing need for sustainable energy solutions, the exploration of efficient power generation methods has become paramount (IPCC, 2011; Lee *et al.*, 2014; Jouhara *et al.*, 2018).

As nations worldwide grapple with the challenges posed by climate change and strives towards decarbonization, the utilization of solid waste incineration for energy production has emerged as a promising avenue. The integration of solid waste management with power generation not only addresses the burgeoning waste disposal crisis, but also contributes to the diversification of energy sources and reduces reliance on fossil fuels (Tan *et al.*, 2020). With a growing emphasis on reducing carbon emissions and transitioning to renewable energy sources, the integration of solid waste management with energy production has gained significant attention. The RRRC stands out as a promising technology for enhancing the efficiency and environmental performance of waste-to-energy systems (Da Cunha *et al.*, 2017; Chantasiriwan, 2021). By incorporating principles of reheat and regeneration, this cycle offers a means to extract maximum energy from solid waste while minimizing environmental impacts (De Monte *et al.*, 2003; Karri, 2012). This study aims at investigating the performance of an ideal RRRC power plant utilizing solid waste incineration, shedding light on its feasibility, efficiency and environmental implications in the quest for sustainable energy solutions.

The need for sustainable electricity supply from environmentally friendly or green sources is increasing. The electricity supply is essential for supporting modern development and socio-economic progress. Generating electric power from fossil fuels, especially coal, leads to significant pollutant emissions and water consumption (Lee *et al.*, 2014; Li *et al.*, 2020; Zhen *et al.*, 2020). Because of uneven geographic distribution of the primary resources, that is, water and coal (Xiong *et al.*, 2017; Zhang *et al.*, 2020), electricity generation from coal is facing a challenge of transmission losses when located near coal deposits, but also coal transportation costs to reach the plants. This has driven a look into utilizing the calorific value of solid waste for electricity generation and feeding into a grid. Utilization of solid waste to generate electricity has multiple advantages, including environmental cleanliness of the cities, and waste treatment during incineration/combustion process in a boiler. Moreover, waste collection and transportation to the nearby power plants leads to employment.

A thermal power plant converts heat energy from the feed fuel, in this case, municipal and industrial solid waste, to electric power by generating steam in the boiler connected to a steam-driven turbine. Thus, initially hot water fed into the boiler turns into steam and drives a turbine connected to an electrical generator (Lee *et al.*, 2014; Marzouk *et al.*, 2022). After expanding through the turbine, the steam loses pressure and its kinetic energy and is condensed in the condenser. The condensate is recycled back to the boiler for re-evaporation. The water, therefore acts as energy or power carrier and goes through a cycle of evaporation and condensation, which forms the basic principle of the Rankine cycle. Further improvement of the basic Rankine cycle leads to different systems in in the power plants such as regeneration (preheating the feed water before entering the boiler) and reheat (sending the steam back to the boiler to raise its temperature after the first turbine stage).

In coal-fired power plants, water vapor and sulfur-laden flue gas at high temperature are discharged into the atmosphere (Tan *et al.*, 2020), resulting in significant water consumption and heat loss (Wang *et al.*, 2016; Ma *et al.*, 2017). Because of emissions around the power plants, formation and development of haze is inevitable (Lonsdale at al., 2012; Cheng *et al.*, 2016). This study focuses at recovering heat from solid waste so as to minimize emissions and also as part of sustainable solid waste management.

This study, therefore, presents the utilization of heat content in the flue gas from solid waste incineration, with temperature ranging from 900 to 1,300°C (Matee and Manyele, 2015). Using exhaust flue gas from a solid waste incinerator as a heat source is an essential source of energy (Ma *et al.*, 2018). Other sources of heat include coal, biomass, gas, diesel, etc. While aim is to transfer the heat in the exhaust gas to steam, a secondary aspect of such plants is to reduce emissions which necessitates use of air pollution control devices (APCDs) after the flue gases are cooled (De Monte *et al.*, 2003; Ogulata, 2004; Pettersson and Soderman, 2007; Liu *et al.*, 2008; Manyele *et al.*, 2008; Lee *et al.*, 2014; Mahir and Manyele, 2023).

1.2 Operational factors for steam power plant

In this paper, the key operational factors affecting the steam power plant efficiency, power generation and fuel consumption include steam superheating and reheating intensity, steam bleeding for regeneration via feed water heaters (FWH), degrees of superheat (DSH) for the steam, boiler pressure, number and types of FWH, the number of turbine steps, condenser duty and the heat input by boiler (Al-Taha and Osman, 2018). To enable effective operation of the steam power plant, it is critical to conduct establish the relationships between key performance parameters and operating conditions.

1.3 Steam power plant and the Rankine cycle

In addition to other auxiliaries, such as steam condenser, feed water heaters (FWH) and pumps, a steam power plant consists mainly of a boiler generating steam using solid waste or any other fuels, steam turbine and electricity generator (Söylemez, 2000; Gupta and Kaushik, 2010; Söylemez, 2011; Ma *et al.*, 2018). The role of the boiler is to generate saturated steam at high pressure and temperature (P_s, T_s). Once released into the turbine, the heat content and high pressure of the steam is converted into kinetic energy or mechanical energy which drives the turbine. Given the kinetic energy, the turbine transforms it into shaft work via high-speed rotation of the shaft. The latter is coupled to the rotor of the electricity generator, which generates electricity due to rotation. Thus, the generator is said to converts the mechanical energy into electric power.

A simple setup of the steam power plant is shown in Figure 1. The detailed setup of such power plant includes connections between different units via piping system, changes in phases or states of matter for water, (between liquid and gas) and changes in terms of pressure and enthalpy (heat content). This necessitates applications of material and energy balance equations, leading to a very complex mathematical model, which can be easily solved by computer software or program. In Figure 1, sections 1 and 2 of the power plant has been studied widely to determine the maximum possible temperatures for the flue gas that can allow for steam generation, reheat and regeneration (Mwaria *et al.*, 2021; Mahir and Manyele, 2023).

Thermal power plants are rated in MW (from 1 to 250 MW capacities), with capital costs expressed in \$/kW of electrical power generated. This variable increases faster until 20 MW capacities (Chowdhury *et al.*, 2022; Ponte and Poche, 2023) necessitating installation of larger plants to offset the costs. Thus, for industrial applications, combined cycle or combined heat and power modes are preferred (Al-Nasrawei *et al.*, 2022). Another means of justifying the initial capital cost is by using cheap fuel, such as municipal solid waste. However, where the fuel is of low cost (for example, wood waste, municipal solid waste (MSW), agricultural waste, etc.), then the thermal power plant gives an excellent choice. This is because , the higher capital costs can be offset by lower fuel costs when solid waste is used, which reduces the running costs. In the steam power plant, treated water is circulated in a closed loop to minimize corrosion of the mechanical parts with minimal loss, different from coal power plant (Wang *et al.*, 2016; Ma *et al.*, 2017). The circulating water losses are compensated by makeup. Worldwide, electricity demand is mostly provided by thermal power plants which use different heat sources to generate steam (Hu *et al.*, 2013; Marzouk *et al.*, 2022), different from diesel generators.

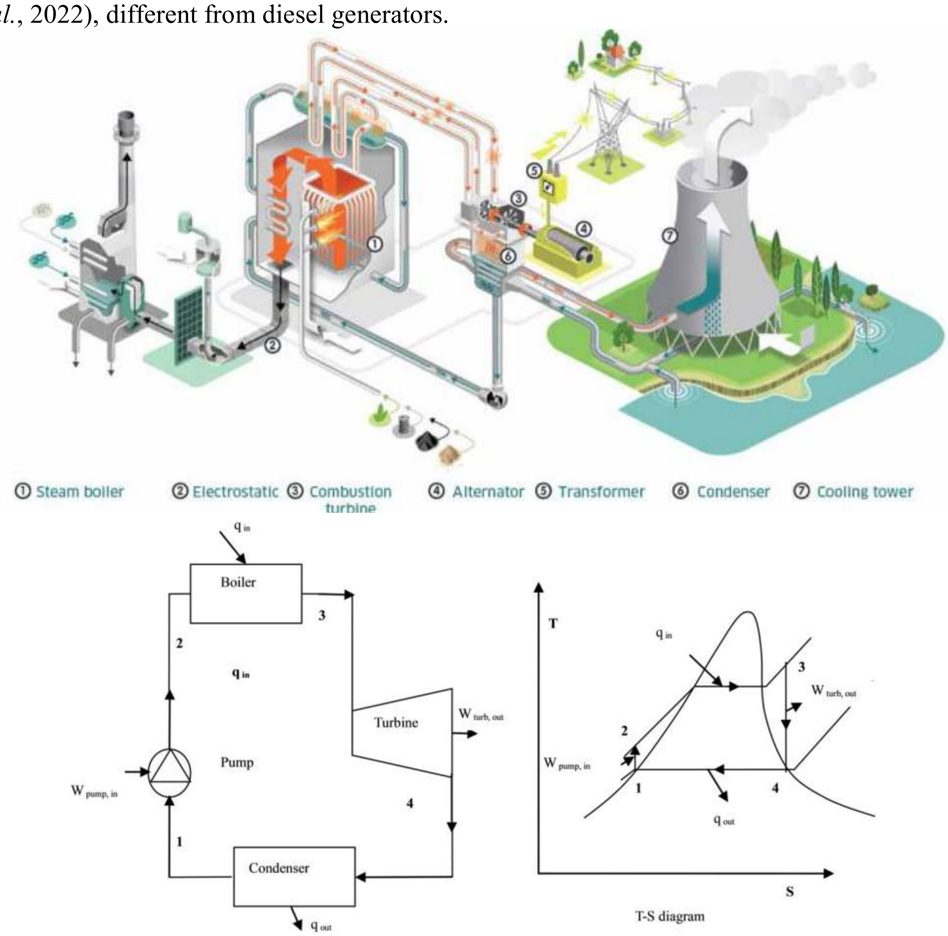

Figure 1: Schematic of Rankine cycle steam power plant with the corresponding simplified ideal process flow and T-s diagrams (Vundela *et al.*, 2010).

Almost all power plants are steam-electric, that is, the turbines are driven by steam (Kareem *et al.*, 2018; Marzouk *et al.*, 2022). When natural gas is used directly into a turbine, it is the flue gas, whose volume is several times that of the gas (leading to high pressure and kinetic energy) that drives the turbine (Ibrahim and Rahman, 2010; Omar *et al.*, 2017; Aderibigbe and Osunbor, 2019). In this case, the flue gas leaves the plant after turbine and there is no recirculation. The gas turbine generates electricity by burning fuel like natural gas and using the hot exhaust gases to turn a turbine (Franco and Claudio, 2002). The waste heat from a gas turbine can be recovered from the flue gas, be used to raise steam.This steam, generated as a by-product, is useful in generating steam to run a separate turbine in the RRRC system. This is known as a combined cycle power plant (CCPP), which in turn improves the overall efficiency (Hasan *et al.*, 2014).Thus, aCCPP uses both gas and steam turbines to produce electricity.This process increases the overall efficiency of the power plant compared to a traditional gas turbine power plant (Franco and Claudio, 2002; Hasa *et al.*, 2014; Martin *et al.*, 2014; Liu and Karimi, 2018; Gu *et al.*, 2021).

1.4 Power plant with steamboiler utilizing heat from solid waste incinerator flue gas

The novelty of this study is based on the use of heat from flue gas leaving a solid waste incinerator to generate steam. Electrical power is generated from a power plant as waste is treated contributing to volume, weight and toxicity reduction, also adding to environmental and energy sustainability. Sustainable boiler operation must meet local emission regulations. Advances in research and design has allowed operation of boilers in environmentally sound manner, partly due to installation of APCDs. The potential for energy efficiency and cost savings in the steam power plants exit, with reported efficiencies closer to 70% (Karri, 2012). Efficiency changes during operation need to be determined to allow for improvements. In this study, efficiency based on heat input and output (E_{th}) and also based on work done (η_{th}) has been studied in details at different operating conditions. Energy losses occur in the power plant, especially across the condenser (up to 39%) while exergy losses occur in the combustor up to about 42.73% as reported by Karri *et al.* (2012).

Given the need for sustainable energy solution, this study embarks on an innovative exploration of the performance of an ideal RRRC power plant fueled by solid waste incineration. While waste-to-energy systems have gained attention for their potential to address both waste management and energy generation challenges, the integration of advanced thermodynamic principles such as reheat and regeneration into this context presents a novel and crucial avenue. Unlike conventional Rankine cycles, the incorporation of reheat and regeneration facilities allows for the utilization of waste heat at multiple stages, enhancing the overall efficiency and optimizing resource utilization. The utilization of solid waste incineration and determination of waste feed flow rate as a function of power generation rate, adds on the novelty of the study. By examining the interplay between solid waste incineration and the RRRC power plant, this study offers fresh insights into the feasibility, performance and environmental implications of this innovative approach within the broader landscape of energy sustainability and waste management strategies.

2. Literature review

2.1 The detailed analysis of the RRRC

When steam is generated and used to run the turbine, condensed and fed back to the boiler, the cycle is known as a simple Rankine cycle (SRC). If the steam is bled from the turbine and sent back to the boiler for adding more heat, the plant is called reheat Rankine cycle (Dincer and Al-Muslim, 2001). When part of the steam in the turbine is bled and used to heat the feed water from condenser, before entering the boiler again, the cycle is termed regenerative Rankine cycle (Toledo *et al.*, 2014; Da Cunha *et al.*, 2017). A power plant comprising of reheat and regenerative sections is referred to as Reheat Regenerative Rankine Cycle (RRRC). This study presents the efficiency and performance of the RRRC power plant, which is more efficient than the separate reheat or regenerative designs. The studied system is also more complex, necessitating large number of detailed mass and energy balance equations.

Most of the thermo-electrical plants are designed to work according to Rankine cycle with a combination of superheat, reheat and regeneration (Toledo *et al.*, 2014). Many of these plants include a number of FWHs. Generally, a FWH works at a medium pressure,between 10 and 15 bars (1.0 to 1.5 MPa), with a temperature between 150 and 200°C. The temperature and pressure for FWHs are lower compared to the steam generated in the boiler (15, 10 and 5 MPa) and also lower than that of bled steam used in this study, that is, $P_{10} = P_{11} = 0.5$ MPa and $P_{12} = 1$ MPa. Such pressure is however, higher than the operating pressure of the condenser, $P_{13} = 10$ kPa = 0.01 MPa. Therefore, a pump connected to P_1 is used to move the condensate against steam pressure P_{12} to allow mixing without backflow. In this study, energy balances across an RRRC plant were conducted to predict its thermal efficiency under different operating conditions (with increasingly large number of equations compared to Toledo *et al.* (2014) and Da Cunha *et al.* (2017), who focused their studies on regenerative systems only. The steam power plant was studied using the first law of thermo-dynamics (i.e., energy analysis). Two FWHs were used as shown in

Figure 2, the first operating in open mode (OFWH) receiving hot water from the low-pressure turbine and pumped condensate, and the second operated in closed mode (CFWH), receiving steam bled prior to reheat and acting as a second stage water heater (Dincer and Al-Muslim,2001; Toledo *et al.*, 2014). The two FWHs require optimization in terms of fractions bled *y* and *z*, respectively (Söylemez, 2011; Toledo *et al.*, 2014). Studies on regenerative systems do not analyze the impact of reheated portion of the steam, i.e., (1-*y*). The optimum thermo-economic performance of the FWHs is based on the maximum savings in the thermal power plant. The FWHs must be designed close to this optimum point (Söylemez, 2011).

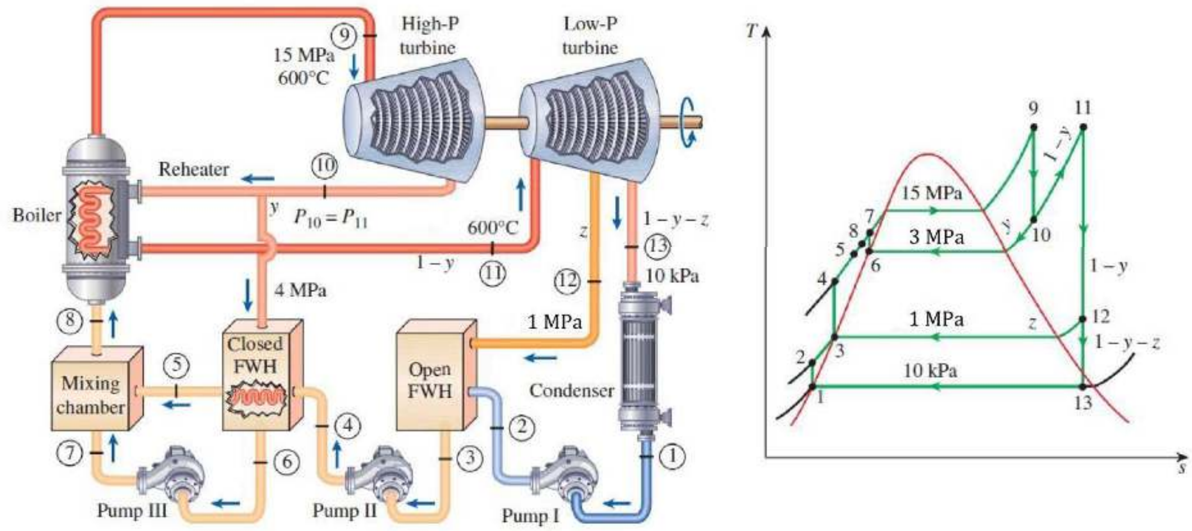

Figure 2. Schematic diagram of a double extraction regenerative reheat steam power plant with a T-s diagram.

The temperature and pressure ranges at which power was generated were considered similar to other literature reports (Hu *et al.*, 2013), but the ranges used in this study were higher after superheating to 600 to 1200°C, supported by flue gases from the incineration process. Table 1 summarizes the range of temperature and pressures for the power plants studied by other researchers. While the power plants reported in the literature were operated at lower temperatures compared to this study, the pressure range for P_9 (that is, 10 – 30 MPa) were higher than 5 – 15 MPa used in this work.

Table 1. Temperature and pressure ranges for steam entering HPT (stage 1) in steam power plants.

Source	Temperature range (°C)	Pressure range (MPa)
This study	600 – 1200	5 - 15
El-Wakil (1984)	538 – 600	20 – 25
Ziebik and Gladysz (2002)	480 – 620	10 – 28
Rao (2006)	540 – 620	15 – 30
Kehlhofer (2009)	565 – 620	22 – 25
Habib and Basha (2011)	538 – 566	24.1
Habib *et al.* (2021)	538 – 566	24.1
Tang *et al.* (2022)	540 – 620	15 – 30

The temperature and pressure data gives the unitary heat consumption, specific steam consumption (SSC) and fuel feed rate required to generate power (Toledo *et al.*, 2014). Other researchers have reported the SSC based on the live steam usage per unit of evaporated water (Pitarch *et al.*, 2017). The superheating adds extra heat to the steam, increasing its capacity to drive the turbine. Saturated steam is generated at a pressure of 15 MPa with saturation temperature of 342.16°C, and is then superheated to a temperature of 600°C, say, difference of which is denoted as degrees of superheat (DSH). The DSH was a variable affecting the plant efficiency, as presented in the results section, reported in this work by the superheating temperature, T_9(that is, DSH = T_9– 342.16), with T_9 ranging up to 1200°C. Former studies on the flue gas temperatures from the incinerator show that the flue gas has higher energy content to boil or generate the steam, superheat the steam and allow reheat of the steam between the high- and low-pressure steam turbine sections (Mwaria *et al.*, 2021). The superheated steam was used in a two-stage turbine, and expanded to a pressure of 4 MPa after the HPT. The steam was then reheated at the same pressure of 4 MPa to increase its dryness and improve the thermal efficiency of the cycle. The analysis was performed using mass and energy balances in each of the units. Other studies include the analysis of the number of FWHs required in the RRRC power plant.A simulation model for the power plant with two heaters and plants of up to seven FWHs have been reported (Toledo *et al.*, 2014). Only two FWHs were used in this study.

The regeneration section of the RRRC increases the feed water temperature flowing back to the boiler,thereby increasing the thermal efficiency of the process and lowers the fuel consumption for the same heating duty, q_{in}, as shown in Figure 1(Söylemez, 2011; Hu *et al.*, 2013; Da Cunha *et al.*, 2017; Chantasiriwan, 2021). Due to FWHs, the steam generation requires less fuel to generate the required power. This is because, while the boiler provides sensible heat to the feed water as its temperature approaches the boiling point,followed by the latent heat required to vaporize the water at its boiling point, regeneration provides part of the sensible heat before the liquid water reaches the boiler. In the RRRC, the steam at high pressure and temperature is fed into the turbine, where it expands in each of turbine stages.Part of this steam which is still at high temperature is then taken out and used to preheat the feed water. In this study, a two-stage turbine was implemented, with the first operating at high steam pressure (HPT), while the second was operated at a low pressure (LPT).

2.2 Performance of a Rankine cycle with regeneration

The efficiency of the RRRC was studied by varying the amount of steam bled for regenerative purposes into OFWH and CFWH, for different steam pressures entering the HPT (Omar *et al.*, 2017). When the condensate enters the boiler at low temperature it is said to cause greater irreversibility in the boiler which decreases cycle efficiency. To avoid this drawback for the studied system, the steam was extracted from the turbine at different stages (also referred to as, steam bleeding) and used in the FWHs. Thus, small fractions of vapor released by the turbine are used to reduce the irreversibility associated with the exchange of energy in the FWHs (Da Cunha *et al.*, 2017; Chantasiriwan, 2021). While the formers studies did not show the effect of the fraction of steam bled, but number of bleeding points, this study investigated the effect of y and z ranging from 8 to 28%, and 11 to 15%, respectively. In the RRRC, a specified quantity of energy supplied by solid waste or any other fuel, remains circulating within the physical cycle, minimizing irreversibility in the process. Mixing relatively cold water (in absence of FWHs) with hot boiling water in the boiler drum increases irreversibility of the process. The only portion of the energy leaving the fluid or the cycle is the power generated by the turbine (i.e., electrical energy) and the condenser duty.

The capacity of the turbines, can be divided in three categories based on the number of FWHs: a) Medium capacity turbines that do not use more than 3 FWHs; b) High pressure high capacity that does not use more than 5 to 7 FWHs and, c) Supercritical turbines that use between 8 and 9 heaters (Gupta and Kaushik, 2010; Söylemez, 2011; Oyedepo and Kilanko, 2014; Da Cunha *et al.*, 2017). In this study two FWHs were used indicating that a medium capacity power plant was modelled. The results of mathematical model simulation show that the maximum efficiency increases when the number of heaters and the superheating temperature increase. Other researchers reported that beyond four FWHs, the efficiency of the plant does not increase further (Oyedepo and Kilanko, 2014; Da Cunha *et al.*, 2017; Chantasiriwan, 2021).

There are several advantages of RRRC power cycle compared to a simple Rankine cycle. The heating process in the FWHs are nearly reversible, which allows for a more efficient heat transfer to the feed water and hence improve the efficiency, reduces fuel consumption and lowers emissions per kW of electricity generated. The thermal stresses or heat load requirements for the sensible heat are minimized in the boiler. The size of the steam condenser can be reduced because part of the steam which is used for FWH do not reach the condenser, but the nature of variation of Qc with y and z need to be established. The turbine efficiency increases especially the LPT, which is subjected to reheated steam (1-*y*), and bled vapors (z), thus decreasing the rate of turbine blade damage (Oyedepo and Kilanko, 2014; Da Cunha *et al.*, 2017; Chantasiriwan, 2021). The only disadvantage of RRRC is the fact that the plant become more complex in terms of layout and connectivity, while the complexity of interactions among the operating variables also increases (complex process dynamics). Moreover, the plant becomes more expensive in terms of equipment and piping. Eventually, the frequency and cost of maintenance, also increases. This paper, therefore, uses a less complex setup to build insight understanding of the process.

With the RRRC utilizing superheated steam, the irreversibility of the FWHs is derived from mixing coolcondensate with saturated or superheated fluid especially in the OFWH (Oyedepo and Kilanko, 2014; Da Cunha *et al.*, 2017; Chantasiriwan, 2021). In addition to the higher dryness of the superheated steam, superheating the steam increases the efficiency of the cycle. This is because the cycle receives heat at higher temperature. The effect of the DSH on plant efficiency is presented in this study.

When heat is transferred between the hot and cold streams during regeneration cycle, irreversibility occurs because the heat cannot be recovered, leading to reduced cycle efficiency (Durmayaz*et al.*, 2004; Da Cunha *et al.*, 2017; Chantasiriwan, 2021). The irreversibility during heat exchange is presented in the T-s diagram (Figure 2), by steps which deviate to right, such as 2-3, and 4-5 during regeneration via OFWH and CFWH, respectively, and step 5-8 across the mixing chamber. This is caused by inefficiencies in heat exchange causing entropy generation. The FWHs operate at different temperatures depending on the location of steam bleeding across the turbine stages, referred to as turbine distribution. Therefore, the way in which the temperature difference is distributed across the turbine affects the cycle efficiency. A large temperature difference between the hot and cold streams at the turbine inlet leads to higher efficiency, as more work can be extracted from the system (Kotas, 2007). This concept of temperature difference was taken into consideration. The temperature difference across the HPT, from $T_9$to T_{10} is expressed by a vertical distance from point 9 to 10 in the T-s diagram (Figure 2). In the LPT, the temperature difference is even higher, that is T_{11} (600°C) to T_{12} and T_{13}.

The performance of the FWH also affects the efficiency of the RRRC. The temperature difference between OFWH and the adjacent heater is a key factor in determining the efficiency. This is more pronounced when a large number of FWHs is used. A large temperature difference leads to more effective heat transfer and high efficiency (Incopera and DeWitt, 2002). To maximize the efficiency of a regenerative cycle, it is necessary to optimize the temperature difference and enthalpy difference as well, between a given FWH and the adjacent heater (that is, T_3 and T_5). This can be done by adjusting the flow rate of the hot stream (y and z) and cold stream (1-y and 1-y-z, respectively), and the heat transfer area of the heat exchangers (not investigated in this study). The distribution of the temperature difference across the turbine and heat exchangers must be uniform to achieve maximum efficiency in the RRRC (Holman, 2010). Thus, the terminal temperature difference (TTD) and drain cooler approach (DCA) must beincorporated in the energy balance across the FWHs. The aided system, where an external source is used as a heat source for the FWHscan become more efficient than the conventional regenerative Rankine plant (Oyedepo and Kilanko, 2014; Da Cunha *et al.*, 2017; Chantasiriwan, 2021), although the plant may become further complicated to operate.

2.3 Effect of reheat and its conditions on power plant performance

After establishing the steam conditions (P_9), several design issues must be addressed especially the choice of reheat pressure and temperature. Sufficient superheating is required to a higher dryness to minimize erosion in the pipe work and also in the LPT (Patel, 2015). The maximum wetness for the steam exiting the LPT is 12% (Patel, 2015). Hence, the reheat analysis was achieved by varying the resulting reheat temperature at constant pressure with flue gas in the boiler. It was assumed in this study that all processes take place without any friction (i.e., are reversible) and heat transfers take place across infinitesimal temperature differences. Reheating the steam improves the quality of the steam at the low-pressure end of the HPT (Patel, 2015). The reheat process is shown in the T-s diagram (Figure 2) as a route 10-11, at constant pressure (P_{10} = P_{11} = 4 MPa). Double reheating cycles exist in power plants. Efficiency improvement with single reheat starts from 5% while double reheat improves efficiency up to 8%, showing that using more than two reheat stages does not justify the economic gain (Oyedepo and Kilanko, 2014). Single reheat process was employed in this study.

In this study, a 100 MW plant is considered, which is a small plant (0.1 GW). Larger power plants up to 1 GW have been reported (Aderibigbe and Osunbor, 2019; Marzouk *et al.*, 2022). The final reheat temperature is limited by fuel used as the source of heat, heat transfer equipment design and materials heat resistance.

Da Cunha *et al.* (2017) studied the maximum thermal efficiency as a function of evaporating temperature (that is, $T_s$andP_9), superheating temperature (DSH) and the number of turbine steam extractions. They reported that increasing the number of extractions up to 3, increases the maximum efficiency. Similarly, when the evaporating temperature and superheating temperature were increased, the plant efficiency increased (Dincer and Muslim, 2001). The increasing of efficiency was, however, smaller when the number of extractions is increased beyond three. Large power plants can use up to two reheat cycles (Marzouk *et al.*, 2022), compared to only one used in this study.

2.4 The RRRC efficiency improvement approaches

While reheat is advantageous to the power plant operation, there is also a limit to the DSH that can be reached inone stage due to the metallurgical conditions (Patel, 2015). Moreover, a high pressure (15 MPa compared to 5 MPa used in this study) limits the achievable degrees of superheat for steam. Because of high momentum and kinetic energy, high pressure, on the other hand, accelerates carryover of liquid droplets which in turn, can cause heavy blade erosion (Wang *et al.*, 2021). Therefore, the reheating is essential in high pressure thermal power plants to protect turbine blades and increase the lifetime of the plant. The reheating cycle reduce 4 to 5% fuel consumption, with steam flow reduction of 15 to 20% (Patel, 2015). Lower steam pressures and temperatures and less expensive construction materials can be used to yield the needed thermal performance with a re-heat cycle. A reduction in the steam flow rate (by bleeding across the HPT and LPT) to the condenser can also be reduced by 7 to 8% (Patel, 2015; Wang *et al.,* 2021).

2.5 Design features of a steam turbine

The expansion of the steam through the turbine converts the heat (enthalpy) and kinetic energyinto turbine shaft rotation. The kinetic energy of the turbine is used to rotate buckets or blades, which are hit by steam and forced to rotate. With large number of blades, the whole shaft rotates at a high speed. This expansion process can be examined by plotting it on a T-s diagram, Figure 2. Therefore, the blades increase in size from the entry point of the steam towards the exit, resulting in the expanding shape shown in the process flow diagrams (Figures 1 and 2). Each stage of a turbine has two elements, that is, the stationary nozzle and the moving blade in a curved shape, also called buckets (Tan *et al.*, 2020). The design factors include entering steam temperature and pressure, exit steam pressure, shaft speedand steam flow rate. The work done on the turbine by steam is determined from the change in enthalpy of the steam between entry and exit states, and the mass flow rate. There are two different turbine designs and applications: back-pressure and condensing steam turbines. While the back-pressure design reduces the inlet pressure of the steam to the turbine's design value, they require a large amount of steam.

In this study, a condensing steam turbine was used because the RRRC was solely for power generation and does not provide process steam elsewhere. Therefore, the steam must be condensed prior to being returned to the boiler, for which a condenser was connected to the LPT turbine. In the condenser, the steam is condensed or cooled with heat being rejected from the steam to the surroundings, q_{out}, for steps 4-1 in the T-s diagram in Figure 1, and steps 13-1 in Figure 2, which happens at a constant condenser temperature. Thus, the steam in the LPT is expanded from 4 MPA to vacuum (10 kPa) in order to extract the largest amount of heat energy from the steam. A gain, inefficiency results from the waste steam being diverted from LPT to the OFWH (that is, z), further reducing the amount of waste steam entering the condenser (that is, 1-y-z), which reduces the condenser duty.

Two turbine stages or cylinders were suggested and used in this study to avoid excessive length of the turbine. This leads to HPT (which expands steam from 15, 10 and 5 MPa to 4 MPa) and LPT (which receives steam at 4 MPa and expands it to vacuum at 10 kPa). No intermediate turbine was used. Usually, the HPT has smaller blades compared to those of LPT, leading to large-diameter cylinder for the latter. To allow the two turbine stages to operate at different speeds, each stage was connected to a separate gear box, but leading to a common generator rotor speed. Turbines with higher HP speeds and lower LP speeds (referred to as tandem-articulated or cross-compounded turbines) are not directly coupled together.

2.6 Condenser and cooling tower for heat removal from the RRRC

The RRRC turns water into steam, uses it in the HPT and LPT before the condenser turns the steam into water again, forming a closed system (Bansal and Chin, 2003; Tahri *et al.*, 2009; Li and Wang, 2015). The purpose of RRRC is to produce maximum power at a highest efficiency, which is achieved via use of a condenser. The condenser releases q_{out}, creating enthalpy drop. The condenser increases enthalpy drop and turbine work output when the exit pressure is low, that is, P_{13} = 10 kPa (Bekdemir *et al.*, 2003). The condenser performs well if the turbine outlet pressure is low (Bekdemir *et al.*, 2003). The condenser operating conditions forms a critical part of the RRRC thermodynamic cycle, and greatly affects the economic performance of the system (Bansal and Chin, 2003; Tahri *et al.*, 2009; Li and Wang, 2015). To minimize the cost (and carbon footprint) of electricity generation and consumption, computer simulation was used to provide in depth understanding of the behavior of the power plant (Bansal and Chin, 2003; Karri, 2012; Fraidenraich *et al.*, 2013). Also, to improve plant efficiency, fouling and scaling inside the condenser heat transfer surface must be controlled, usually achieved by automatic brushing.

A continuous flow of cooling water is circulated between the steam condenser and the cooling tower. A low condenser pressure (10 kPa, well below P_{atm} = 101.3 kPa) was used in this model, because at this state, the latent heat of water, h_{fg}, is highest (and hence, q_{out} or Q_c is highest), and large water flow rate is required to cool the condenser with low temperature drop (Bekdemir *et al.*, 2003). As shown in Figure 2, the condensate extraction pump is installed for recirculating the condensate to the steam generator via the OFWH and CFWH.

A water-cooled surface condenser was used, with water circulating through the tubes, while the steam condenses outside the tubes (shell-and-tube condenser) and withdrawn from the bottom of the condenser. The condensate is re-used in the boiler, which means the circulating water/steam is not allowed to exit the cycle, and thus it is treated initially to reduce chances of corrosion. For best efficiency, the condenser temperature must be kept very low to achieve the lowest pressure in the condensing steam. Because the condenser works under vacuum (Bekdemir *et al.*, 2003), the possibility of non-condensable air to leak into the closed loop must be prevented.

2.7 Open and closed feed water heaters

The OFWH is a mixing chamber, where the steam bled from the turbine mixes with the condensate. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The OFWH receives steam at P_{12}= 0.5 or 1.0 MPa and T_{12}, and mixes this with condensate flowing at a pump pressure P_1 and condenser temperature T_1 (Figure 2). The advantages of OFWHs include: simple and low-cost, more efficient heat transfer due to direct contact (Gupta and Kaushik, 2010; Söylemez, 2011). The CFWHsare designed in the form of shell-and-tube heat exchanger, whereby, the feed-water is heated as the bled steam condenses outside the tubes. The two streams can be at different pressures (Gupta and Kaushik, 2010; Söylemez, 2011). The extracted stream from the exit of the HPT condenses in the CFWHat 4 MPa, while the feed water was at a pressure raised by the pump (P_4) in Figure 2. The disadvantages of CFWHs include complex and expensive equipment and lower the heat transfer efficiency. The design details of OFWH and CFWH were not used in the simulation, which attracts further.

Optimization of heat exchanger size for CFWH is important in order to get maximum savings (Söylemez, 2000; Ma *et al.*, 2018). The RRRC involve heat recovery, that is, from flue gas steam generation (Mwaria *et al.*, 2021) and bled steam in the FWHs. Thus, heat exchanger design features for heat recovery is an important aspect of the steam power plant (De Monte *et al.*, 2003; Pettersson and Soderman, 2007; Ogulata, 2004), which were not investigated in this study, since a thermo-economic analysis and direct measurements are required to estimate the optimum heat exchanger area for energy recovery application (Soylemez, 2000; Gupta and Kaushik, 2010; Söylemez, 2011; Ma *et al.*, 2018). The thermal efficiency of the RRRC increases with increasing number of FWHs. On the other hand, initial cost of the system increases due to large number of FWHs (Söylemez, 2011).

3. Methodology

3.1 Power plant layout and process description

In the simulated steam power plant, the solid waste is fed into the incinerator where it is burnt in the primary chamber to generate pyrolytic gases which move towards the secondary chamber due to increased temperature and pressure. In the secondary chamber, the gases burn further raising the flue gas temperature up to 1200°C. The flue gas from the secondary chamber passes into the boiler section causing water in the boiler drum to change into high pressure steam. From the boiler the high-pressure steam passes through the superheater where it is again heated to increase its temperature and dryness at high pressure, P_1. The maximum pressure for the inlet steam to the steam turbine was 15 MPa or 150 bar and a temperature of about 600°C,as set in this model. The pressure was however varied to study the plant performance at different pressure, that is, P_1 = 15, 10 and 5 MPa. The temperature and pressure were selected based on literature data, as shown in Table 1.

The heat contained in the flue gas is utilized in the generation, reheat and superheating of steam. The DSH was another parameter affecting the efficiency of the RRRC power plant. Once the steam leaves the first stage of the turbine, that is, HPT (it is at a low pressure of 4 MPa) a fraction, y, of which is bled and sent to the CFWH, and the remaining fraction (1-y), is sent back to the boiler for reheating to the temperature of 600°C, once again. The reheated steam enters the LPT and is expanded again, during which another fraction, z, of the steam is bled at a pressure P_4 = 0.5 MPa and fed to the OFWH, while the remaining fraction (1-y-z) at 10 kPa at the end of the LPT, is sent to the condenser. The steam entering the LPT expands until it becomes as wet (usually not more than 12% wet) as the final blades can tolerate. The condensate is then mixed with the bled steam in the OFWH and the mixed stream is heated again in the CFWH before feeding to the boiler via the mixing chamber.

3.2 Design features of the RRRC

The design features of the turbine stages are summarized in Table 2, while Table 3 shows the design features for the rest of the RRRC power plant units.

Table 2. Design features of the turbine stages (HPT and LPT).

Design parameter	High pressure turbine (HPT)	Low pressure turbine (LPT)
Operating pressure range	15 – 4 MPa 10 – 4 MPa 5 – 4 MPa	15 – 4 MPa 10 – 4 MPa 5 – 4 MPa
Final steam condition	Non-condensing turbine	Condensing turbine
Blade size	Smaller blades	Large blades
Diameter of turbine cylinder	Smaller diameter	Large diameter

Table 2 (cont'd). Design features of the turbine stages (HPT and LPT).

Design parameter	High pressure turbine (HPT)	Low pressure turbine (LPT)
Rotation speed	High speed	Low speed
Stem bleeding location	*y* kg/kg, after exiting HPT (8-28%)	*z* kg/kg,along LPT (11-15%)
Bled-steam pressure	P_{10} = 4 MPa	P_{12} = 0.5 MPa
Feed water heater type	Closed FWH	Open FWH
Turbine inlet temperature	600°C and DSH where applicable	600°C after reheat
Inlet steam flow rate (based on basis)	1 kg	(1-*y*) kg
Exit temperature	T_{10}	T_{12}, $T_{13} = T_{sat}$ (10 kPa)
Tolerable steam wetness	-	12% at T_{13}

Table 3. Design features of the RRRC power plant units (based on Figure 2)

Unit	Design features
Condenser	Shell and tube heat exchanger Water-cooled condenser Water cooled by a cooling tower Low pressure condenser (P_{12} = 10 kPa)
CFWH	Shell-and-tube heat exchanger Heated by bled steam at P_{10}= 4 MPa Steam condensed outside the tubes
OFWH	Mixing chamber Steam at P_{12}= 0.5 MPa is directly mixed with water condensate at a condenser temperature T_1
Boiler	Solid waste fired boiler Utilizes flue gas from the secondary chamber of the incinerator Flue gas temperature = 1200°C Comprises of a superheater Allows steam from HPT to recirculate back for reheat
Mixing chamber	Mixes hot water from CFWH at T_5 with pumped steam condensate at T_7 exiting the CFWH at T_6.

3.3 Mathematical model formulation

The key assumptions made in the model formulation are summarized in Table 4. These assumptions form a common idealization in the thermodynamic analyses of steam power plants. This approach is often used to simplify the analysis and to understand the upper limits of performance for the thermodynamic cycles (McGovern, 2015).

Table 4. Key assumptions in modeling reheat, regenerative Rankine cycle

Assumption	Justification	Associated challenges	References
No pressure loss in pipelines	Simplified the model by neglecting pressure drops in the steam and feed water pipelines.	Pressure losses can affect the cycle performance and need to be considered in detailed design	Lei *et al.* (2017)
Steady state operation	Assumes all states are steady and do not change with time.	Transient conditions are common in real power plants during startup, shutdown and load changes.	Cai *et al.* (2022)
No heat losses to the environment	Assumes perfect insulation, meaning no heat is lost to the surroundings.	Heat losses in real steam power plants can significantly affect cycle efficiency and performance.	Wang *et al.* (2016), Ma *et al.* (2017)
Reversible process	Assumes all processes are reversible, with no entropy generation.	Real processes have irreversibilities due to friction, heat losses and mixing.	Oyedepo and Kilanko (2014), Da Cunha *et al.*(2017), Chantasiriwan (2021)

Table 4 (cont'd). Key assumptions in modeling reheat, regenerative Rankine cycle

Assumption	Justification	Associated challenges	References
Isentropic turbine and pump efficiency	Assumes no entropy generation leading to maximum possible efficiency for turbines and pumps.	Real turbines and pumps have inefficiencies due to friction and other irreversibilities.	Omar *et al.* (2017), Da Cunha *et al.* (2017), Chantasiriwan (2021)
Perfect heat exchange	Assumed no heat loss from the steam and working fluid in the OFWH, CFWH and mixing chamber.	Heat losses and temperature gradients in real exchangers can reduce efficiency.	Gupta and Kaushik (2010), Söylemez (2011), Toledo *et al.* (2014), Ma *et al.* (2018)

The model simulated in this study was set to generate 100 MW although smaller power plants are still useful for powering residential areas. The simulation model of the RRRC was built using Excel platform (MS Excel 2019). The thermodynamic properties of water and water vapor were obtained from the steam tables. The energy balance across the quantities of bled steam to the FWHs can be presented as:

The energy balance across the CFWH leads to y, the fraction of steam diverted/bled from the steam exiting the HPT, that is:

$$y = \frac{(h_{10} - h_9)}{(h_2 + h_{10} - h_9 - h_{11})} \tag{1}$$

The energy and material balances around the OFWH leads to z, the fraction of steam bled from the steam expanding in the LPT, that is:

$$z = \frac{(h_8 - h_7)(1-y)}{(h_4 - h_7)} \tag{2}$$

The energy balance around the boiler given the total heat added to the fluid across the boiler, Q_{in}, that is:

$$Q_{in} = (h_1 - h_2) + (h_3 - h_2)(1+y) \tag{3}$$

The first term of Equation (3) gives the heat input to generate steam, while the second term is the heat input to the fluid during reheat of the fraction (1-y) of the steam. That is, an additional boiler heat is required to reheat the remaining fraction of steam after bleeding, (1-y), that is, h_3. The heat input supplied by the boiler can be divided into two parts, the heat required to vaporize the feed water, , and the heat utilized during the reheat, $Q_{in,r}$, based on which, the total heat input Q_{in} can be expressed as per Equation (4):

$$Q_{out} = (Q_{in,vap} + Q_{in,r}) \tag{4}$$

which is another form of Equation (3).

The total heat removed from the fluid via a condenser, Q_{out}, was estimated from Equation (5):

$$Q_{out} = (h_5 - h_6)(1 - y - z) \tag{5}$$

The thermal efficiency of the Rankine cycle, is defined based on the Q_{in} and Q_{out} as per Equation (6):

$$E_{th} = \left[1 - \frac{Q_{out}}{Q_{in}}\right] \tag{6}$$

The energy balance for the HPT, gives the work done by steam on the turbine, W_{t1}, that is:

$$W_{t1} = (h_1 - h_2) \tag{7}$$

On the other hand, the energy balance across the LPT leads to the work done on the turbine by the reheated low-pressure steam, as per Equation (8):

$$W_{t2} = (1-y)h_2 - zh_4 - (1-y-z)h_5 \tag{8}$$

Hence the total work done by the steam on the turbine, which is transferred to the electric generator (at a given thermal efficiency) can be expressed as per Equation (9):

$$W_{tt} = (W_{t1} + W_{t2}) \tag{9}$$

The plant has three pumps for recirculating the condensate back to the boiler. The work done by each pump, i = 1, 2, 3 is given as per Equations (10a, 10b, 10c):

$$W_{p1} = (1 - y - z)(h_7 - h_6) \tag{10a}$$

$$W_{p2} = (1 - y)(h_9 - h_8) \tag{10b}$$

$$W_{p3} = y(h_2 - h_1) \tag{10c}$$

The total work done by pumps is given by Equation (11):

$$W_p = (W_{p1} + W_{p2} + W_{p3}) \tag{11}$$

The thermal efficiency of the power plant based on work done on the turbine by the fluid on the turbine and work done on the fluid by the pumps is given as per Equation (12):

$$t^h = [(W_{tt} - W_p) / Q_{in}] \tag{12}$$

The mass flow rate of the steam was determined from the power to be generated in the turbine, M_w, the total work done by steam on the turbine, W_{tt}, and total work done by the pumps, W_p:

$$m_1 = \left[\frac{3.6 \times 10^6 M_W}{W_{tt} - W_p} \right] \tag{13}$$

The heat which must be generated by the boiler per hour to generate power in the turbine, Y_B, is given by Equation (14):

$$Y_B = (m_c \times Q_{in}) \tag{14}$$

4. Results and Discussion

4.1 Effect of superheating the steam on power plant performance

Figure 3 presents the effect of superheat temperature T_1 on the thermal efficiency of the power plant, E_{th} and η_{th}, at different steam pressures, P_1. The maximum reachable temperature was set to be 1200°C corresponding to the maximum flue gas temperature attained in the secondary chamber of the waste incineration. The minimum temperature of the superheated steam starts at the saturation temperature for the given pressure, that is, 263.9, 311.0 and 342.2°C corresponding to the saturation pressure of 5, 10 and 15 MPa, respectively. Results show that both E_{th} and η_{th} increases with superheat temperature for all steam or boiler pressures. The effect of steam pressure was, however, observed to be negligible at higher superheat temperatures.

Presented in Figure 3 are the final superheat temperature attained and at which steam enters the HPT at 15 MPa. It is, however, common to present the degrees of superheat (DSH), that is temperature above the saturation temperature at a given pressure, but since the saturation temperature at 15 MPa is known, the DSH can be easily deduced. It can be seen from Figure 3 that for each steam pressure studied, the starting points on the temperature scale are different, due to differences in T_{sat}. At T_1 = 1200°C, corresponding to the maximum temperature of the flue gas used for superheating the steam, the maximum possible DSH were 936.1, 890, and 857.8°C for 5, 10 and 15 MPa, respectively, being highest at a lower boiler pressure of 5 MPa. These temperatures can be attained via solid waste incineration (Mwaria*et al.*, 2021).

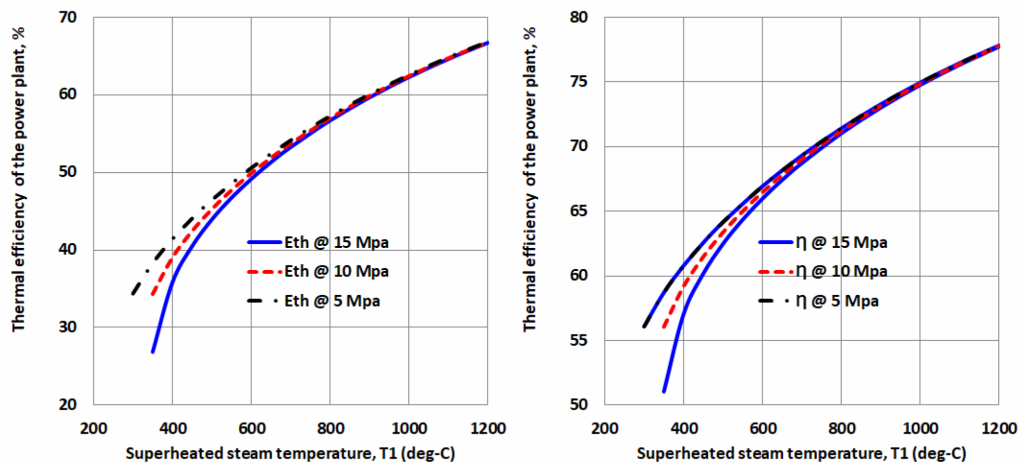

Figure 3. Effect of superheating temperature on the thermal efficiency of the steam power plant.

Figure 4 shows the effect of superheating temperature on the work done by steam on the turbine at different boiler pressures. The work done by steam on the HPT, W_{t1}, increases with T_1, but W_{t2} remains constant as T_1 was increased from 300 to 1200°C. The effect of boiler pressure is negligibly small, appreciable at lower DSH, for which highest work done was observed at 5 MPa. The total work done, however, increases with T_1. Figure 4 shows the variation of the total work done by steam on the turbine, W_{tt}, with steam superheat temperature, T_1. As T_1 increases, W_{tt} increases linearly for P_1 = 5 and 10 MPa for the whole range of T_1> 400°C. Below 400°C, W_{tt} shows faster increase with temperature. Both W_{tt} and W_{t1} increases linearly with T_1, while W_{t2} remains constant independent of T_1, attributable to the fact that superheated steam at T_1 expands in the HPT only.

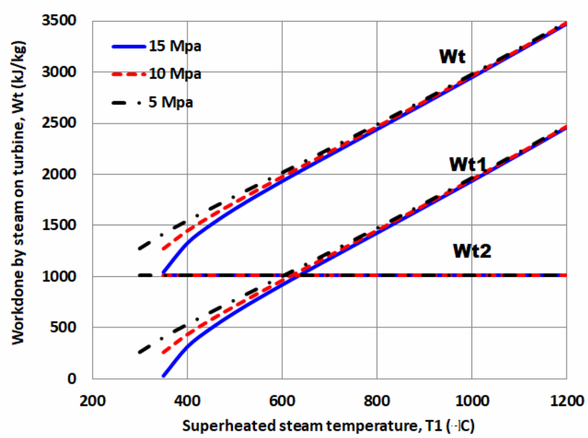

Figure 4. Effect of superheating temperature on the work done by steam on the turbine at different boiler pressures.

The enthalpy of steam is a function of P_1, T_{sat} and the DSH to a temperature T_1. At lower temperature, enthalpy will be low and both the work done by the turbine and the power plant efficiency will decrease. Hence, steam consumption for the required output will be higher. Figure 5 shows the effect of DSH applied to T_1 on the steam mass flow rate required to produce 100 MW of power. Figure 5 shows that by superheating the steam to a higher temperature, T_1, the steam consumption decreases. That is, higher steam inlet temperature leads to higher heat extraction by the turbine and hence reduced steam consumption for the same output. At higher steam pressure of 15 MPa, the steam mass flow rate was highest compared to 5 MPa, attributable to the decrease in h_{fg} as pressure increases. Figure 5 shows also that the required mass flow rate of the fuel also increases as steam temperature T_1 increases. This indicates that the heat released by the fuel is efficiently utilized when steam is superheated by utilizing the hot flue gases leaving the boiler. Results show further that at higher boiler pressure of 15 MPa, the mass flow rate of the fuel required to generate 100 MW of power is highest, attributable to high h_f and h_g but lower h_{fg} at higher steam pressure, necessitating high demand of heat from the fuel. Figures 4 and 5 present the effects of steam inlet temperature T_1 on turbine efficiency, steam consumption and fuel consumption, respectively, keeping all other factors constant in the steam power plant. The results presented in Figures 4 to 6 shows that the steam could be superheated to higher temperatures when using solid waste as the source of energy in the boiler (Patel, 2015).

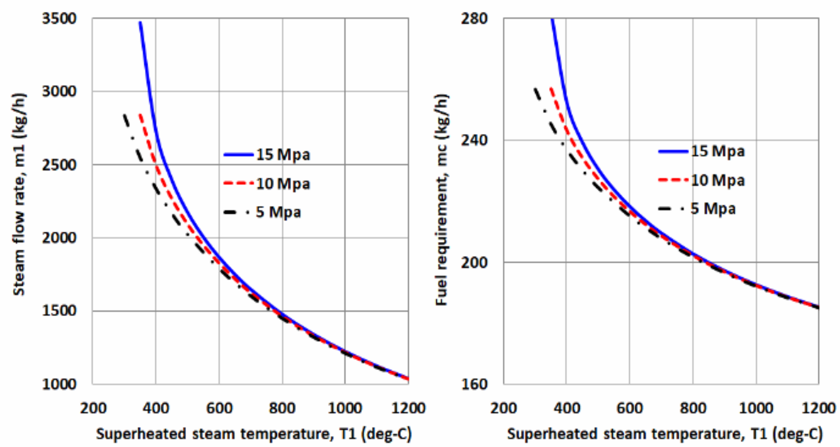

Figure 5. Effect of superheating temperature on the required steam flow rate into the high-pressure turbine and fuel requirements.

Superheaters and reheaters should be designed to increase the steam temperature and quality or dryness across the turbine stages. They are designed as single-phase heat exchangers with steam flowing inside and the flue gas passing outsidein the cross flow arrangement.

4.2 Effect of increasing reheat intensity

The effect of increasing reheat intensity on the plant performance was studied at three different boiler operating pressure of 5, 10 and 15 MPa (as shown in Figure 6). Increasing the reheat intensity linearly increases the heat required for reheat, $Q_{in,r}$ and also the total boiler heat requirements, Q_{in}. However, the heat required to vaporize the feed water remains constant, at the latent heat of vaporization and the required superheat, given the constant operating pressure and steam mass flow rate towards HPT. Figure 6 shows that the total heat required by the boiler, Q_{in}, increases also linearly as h_3 increases. This was observed for all boiler pressures from 5, 10 to 15 MPa. Figure 9 shows also that $Q_{in,r}$ used to reheat the steam to h_3 is independent of steam pressure P_1, since the settings of the process was to drop the steam pressure to 4 MPa after expanding in the HPT. It is also evident from Figure 9(a) that low pressure steam allows more reheat than high pressure steam. The heat required for vaporization, $Q_{in,v}$, however, depends strongly on steam pressure towards the HPT, P_1, being highest at lower steam pressure of 5 MPa.Assuming that the stages in both turbines are isentropic (adiabatic and reversible), and neglecting kinetic energy effects, the enthalpy of the inlet and exiting stream conditions were used to determine the combined work output of both turbines. The W_{t1} and W_{t2} were determined separately, based on which the total work done was determined, as shown also in Figure 6(b).

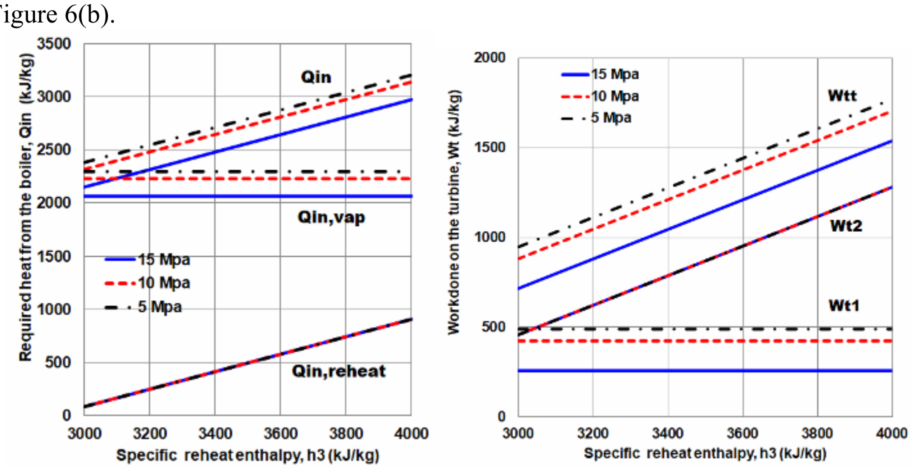

Figure 6. Effect of reheat intensity (heat input) on: (a) boiler heat requirement; (b) work done by steam on the turbine (W_{t1}, $W_{t2}$and W_{tt}).

The work done on the LPT, W_{t1}, remained constant despite the reheat, since reheat is applied to the steam after exiting the HPT.We notice that reheating the output of the HP turbine back to 600°C (process (2)-(3)), expressed by heat input to the reheated stream, h_3, allows both significantly more power output as well as increasing the quality at the LPT exit (state 4) to 98%, as shown in Figure 6(b). The reheat does not affect the work done on the HP turbine, and therefore, W_{t1} remains constant. The work done by the steam on the LPT, W_{t2} increased with h_3 independent of steam pressure P_1 (since the steam is reheated at a lower pressure, P_2 = 4 MPa in all cases). It was noted also that if the pressure ratio across the HPT (P_2/P_1) must be fixed, there will be a change in P_2 for a given P_1, not implemented in this study. The total work done on the turbine, W_{tt}, increased linearly with h_3, depending on steam pressure P_1. The highest W_{tt} and W_{t1} values were observed at a lower pressure of 5 MPa. It was further noted that the contribution of LPT to the total work done, W_{tt}, is higher than W_{t1} due to reheating. Again, W_{t2} is independent of P_1, since the increase in h_3 (the source of increase in W_{t2}) is done before LPT. The work done on the LPT, W_{t2} on the other hand, increases linearly regardless of steam pressure exiting the boiler (single line). The total work done, however, increases linearly, being highest at lower boiler pressure of 5 MPa that for 10 and 15 MPa, attributable to the fact that W_{t1} depends strongly on the pressure of the steam from the boiler, P_9.

Because W_{t2} increases with reheat, its contribution to the total work done on the turbine increases, as shown in Figure 7. It is interesting to note that the percent contribution of the reheated stream entering the LPT on the total work done was highest at P_1 = 15 MPa, as shown in Figure 7. This is because the W_{t2} values were constant for all P_1 values, while the total work done, W_{tt} were lower for P_1 = 15 MPa. Therefore, at higher boiler pressures, the percentage contribution of W_{t2} to the total work done W_{tt} is higher at higher steam pressure, P_1 = 15 MPa. Thus, reheat is more advantageous for higher pressure boilers for improving the plant efficiency than lower pressure boilers.

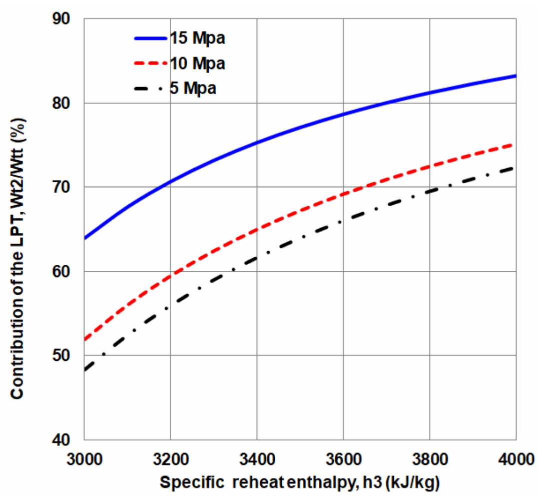

Figure 7. Variation of contribution of the low-pressure turbine on the total work done with increasing reheat intensity.

The thermal efficiency of the ideal Rankine reheat system, defined as the net work done (turbines, pump) divided by the total heat supplied to the boiler, is compared to the efficiency determined from the simpler method of evaluating the condenser heat transferred to the cooling water, values of which were determined at different reheat intensities. The efficiency based on the work done on the turbine was slightly higher than the values based on Q_c, attributable to underestimation of the pump work.

Increasing the reheat intensity (heat content of reheated steam, h_3) increases the thermal efficiency of the plant, especially at lower boiler pressure of 5 MPa. Figure 8 compares the thermal efficiency based on heat removed from a condenser, E_{th}, and efficiency based on work done by the turbine, η_{th}, (Equations (12) and (6), respectively).

The thermal efficiency of the power plant was however lower at higher boiler pressure of 15 MPa compared to 10 and 5 MPa. The higher efficiency of the plant at lower boiler pressure can be attributed to heat content for the steam at lower boiler pressure (h_1) compared to those observed at higher pressures. Increasing the enthalpy of the reheated steam from 4 to 20% led to an increase in E_{th} from 10 to 42% at 15 MPa, while the same increase in h_3 led to an increase from 7% to 28% only at 5 MPa. A 30% increase in h_3 leads to a 40%, 43% and 58% increase in E_{th} at 5, 10 and 15 MPa, respectively, indicating higher reheat effectiveness at higher boiler pressure. Such an increase in power plant efficiency has a critical impact in the industrial settings.

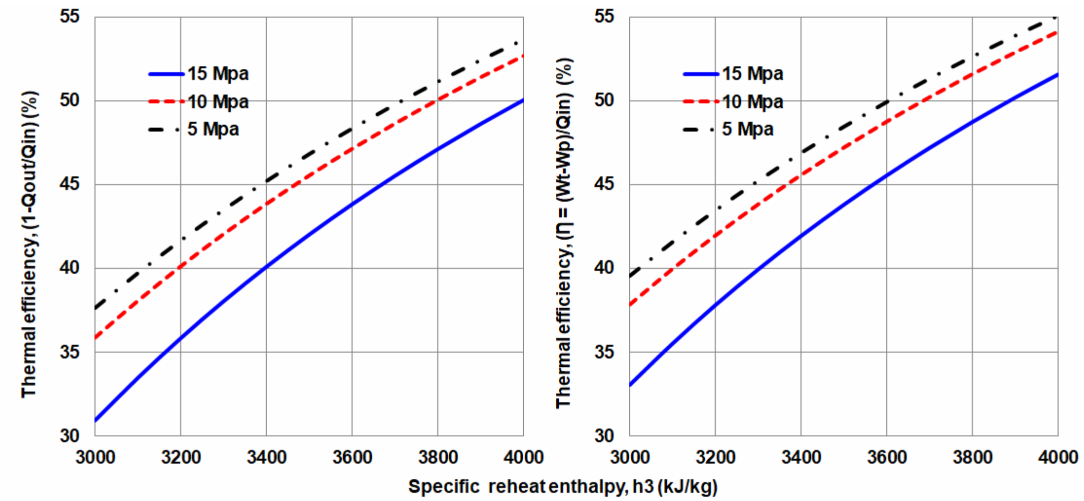

Figure 8. Effect of reheat intensity on the thermal efficiency of the steam power plant.

One of the main differences between the superheater and reheater is the steam pressure. The outlet pressure of the superheater in a subcritical drum boiler, for example, is for example at 15, 10 and 5 MPa, while the outlet pressure of the reheater is only 4.0 MPa, as it enters into the LPT (Tominaga, 2017). Despite the improved performance realized via reheating the steam after HPT, disadvantages of a reheat cycle exist in thermal power plants.

4.3 The effect of regeneration process on power plant performance

In a practical power plant, one may find various combinations of closed and open FWHs. In this study, one open and one closed FWHs were selected in order to illustrate the method of analysis and assessing their impacts. The same approach will apply to any combination of FWH. The required mass flow fractions y and z of the bled steam was also evaluated in order to bring the compressed water at the entrance to the boiler (state (13)) to the correct state. An enthalpy inventory and energy balance on both the CFWH and OFWH were carried out using mass and energy balance equations.

The CFWT was connected to the first steam extraction from HPT, to which a fraction y (kg/kg) was used to heat condensed water. The OFWH was supplied by a fraction z (kg/kg) of steam extracted from the steam after expansion across LPT. It can be generalized that high pressure steam is not directly mixed with water but the low-pressure steam from LPT. Figure 9(a) presents the effect of increasing both y and z on the total heat input required by the boiler, Q_{in}, at different boiler pressures. Increasing y increases the heat demand in the boiler, being highest at lower boiler pressure. On the other hand, increasing z decreases the heat demand in the boiler. The heat required in the boiler, Q_{in}, varies with y and z depending on the boiler pressure, P_1. Increasing y increases the heat demand, Q_{in}, while increasing z leads to a sharp decrease in Q_{in}. A horizontal line ABC in Figure 9(a) gives the values of y, z and Q_{in} for a given operation of the power plant.

Figure 9(b) shows the variations of the condenser duty, Q_c, with both y and z. Increasing z sharply increases Q_c, while increasing y lowers the Q_c. It can be noted that operating the steam power plant requires fixed settings for y and z, which can be presented by a horizontal line connecting y, z and Q_c in Figure 9(b), similar to Figure 9(a).

Figure 10(a) shows the effect of increasing both y and z on the total work done by the steam on the turbine, W_{tt}, and also on the work done by the pumps during regeneration process to run the FWHs, at different boiler pressures, P_9. The total work done, W_{tt} increases with increasing y and decreases faster with increasing z for all boiler pressures. Thus, to avoid power loss and lower electricity generation rate, values of z should be maintained lower around 13% only, corresponding to y = 25% for 15 MPa boiler pressure. For boiler pressures of 10 MPa, however, z = 13 (high W_{tt}) requires that y = 19% to maintain high W_{tt}> 1,300 kJ/kg.

Figure 10(b) shows the effect of both y and z on work done by pumps (W_{p1}, W_{p2}, and W_{p3}). The three pumps behave differently as y and z are varied simultaneously. The W_{p2}, for example, decreases with increasing y, but increases as z is increased. The W_{p1} on the other hand, increases with y and drops faster with slight increase in z. It can be concluded that the work done by pumps lead to a complex scenario when both y and z are varied, and affects the power plant performance in a complex manner. Therefore, regenerative power plant leads to complex power plant operation, necessitating modeling and simulation to build an insight of the scenarios encountered before actual operation.

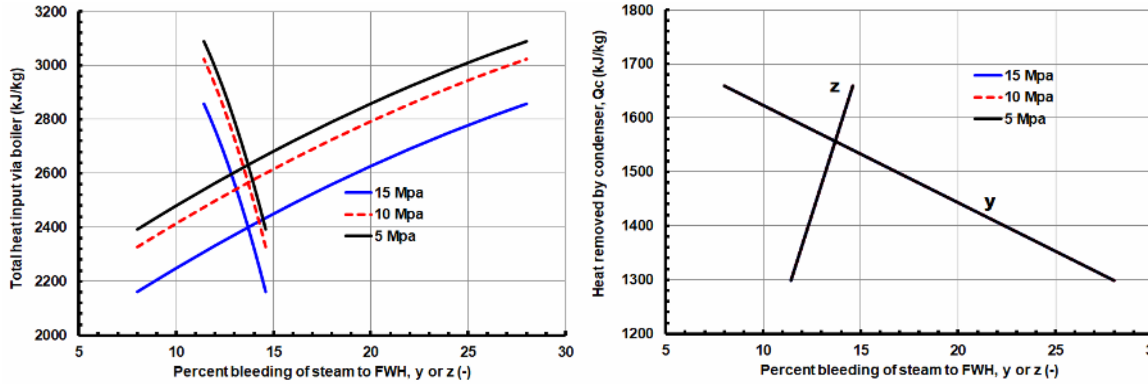

Figure 9. Effect of percent bleeding to the FWHs on: (a) the required heat input across the boiler, Q_{in}; (b) condenser duty, Q_c.

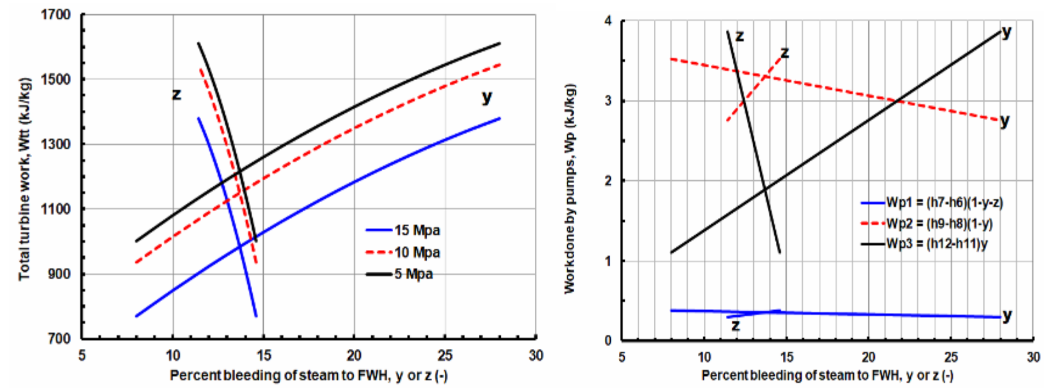

Figure 10. Effect of percent bleeding to the FWHs on: a) the total work done on the turbine; b) the work done by condensate pumps.

The power plant thermal efficiency is strongly affected by the regeneration cycles in different manners at various boiler pressures. Both E_{th} and η_{th} were observed to increase with y for all boiler pressures, while increasing z decreases both efficiencies, as summarized in Figure 11. Results in Figure 11 show that the power plant efficiency increases with increasing bleeding percent from the HPT. However, increasing the fraction of bled steam from the LPT lowers the efficiency. Thus, it is required to fix z very low, around 12%, while y is kept around 28% for all boiler pressures (15, 10 and 5 MPa) for which the power plant efficiency, η is around 47.5%.

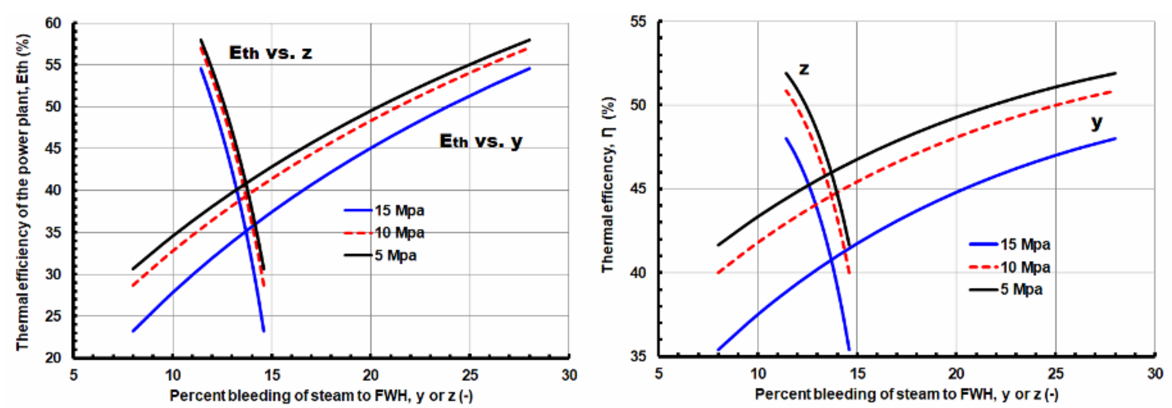

Figure 11. Variation of the steam power plant thermal efficiency (E_{th} and η_{th}) with percent bleeding to the FWHs, y and z.

A slight increase in z from 12to 13%, for instance, sharply lowers the power plant efficiency η to 42% while requiring new settings for y at 15% for P_1 = 15 MPa and approximately 14.5% for P_1 = 5 and 10 MPa. At 5 MPa, with z slightly increased from 12 to 13%, η_{th} is 52% while the y is required to be set at 27%. Thus, the plant efficiency becomes very sensitive to bleeding percentage when z is implemented.

Figure 12 shows the effect of increasing both y and z on steam mass flow rate required to maintain 100-MW power generation in the turbine. Increasing y decreases the steam mass steam mass flow rate, that is, lowers the steam generation load, while increasing z increases the steam generation rate for all boiler pressures. Increasing z, however, increases the steam generation rate. The required steam generation rate was the highest at a higher pressure of 15 MPa, attributable to lower h_{fg} at this high pressure. Thus, steam inlet pressure to the turbine strongly affects the turbine performance, as shown in Figure 12. All the turbines are designed for a specified steam inlet pressure range. Once the turbine pressure is known, the boiler capacity can then be determined. This is supported by results presented in Figures 5 and 12, based on which, the values of m1 were highest at P_1 = 15 MPa.

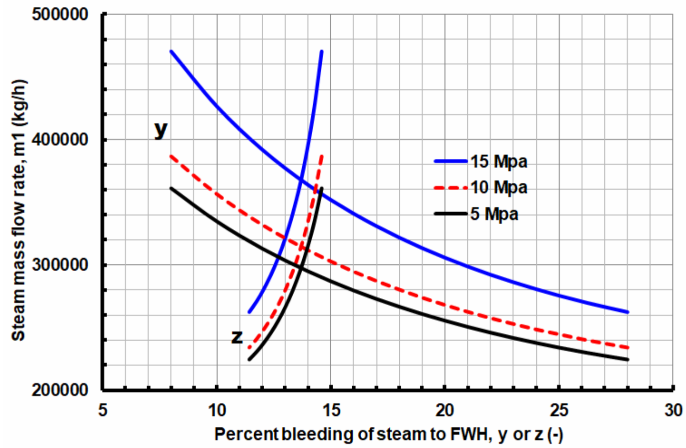

Figure 12. Variation of the required steam mass flow rate into HPT with percent bleeding to the FWHs.

5. Conclusion

Superheating the steam increases the power plant thermal efficiency and work done on the turbine up to a limit determined by the maximum flue gas temperature, metallurgical limitations of materials of construction and heat transfer efficiency. Both mass flow rates of steam and fuel are reduced by increasing the DSH of steam. Increasing reheat intensity for the steam after expansion across the HPT increases the heat requirements in the boiler while it increases the total work done on the turbine. Reheated steam expanding across the LPT contributes more than 50% of the total work done on the turbine for all boiler pressures.

Reheating the steam to increase its enthalpy increases the power plant efficiency, being highest at lower boiler pressure. Reheating of the steam, however, increases the equipment cost via piping and floor space requirements.

Regeneration set-up using the bled steam from HPT to CFWH, y, has stronger effect on the power plant performance. Increasing the percent of bled steam towards a CFWH increases heat requirements from the boiler, decreases the condenser duty, increases thermal efficiency and work done on the turbine, but increases work done by the pumps. Furthermore, increasing y lowers the steam mass flow rate required to maintain the same power generation. Increasing the fraction of steam bled from the LPT to the OFWH has different impact on the power plant performance. Increasing z lowers the heat requirements from the boiler, decreases thermal efficiency and work done on the turbine, but increases work done by the pumps. Furthermore, increasing y increases the steam mass flow rate required to maintain the same power generation.

Both y and z affect the work done by each pump in a different manner. The work done by the pumps is slightly affected by both y and z, while the work done by the third pump, W_{p3}, is strongly affected by y and z (increasing with y and decreasing with z). The work done by the second pump, W_{p2}, decreased with y and increased with z. The total work done by the pumps W_{pi}, should be minimized for higher plant efficiency. Lower boiler pressure shows high heat requirements from the boiler, high total work done on the turbine, but lower contribution of LPT to the total work done. Highest total work done on the turbine and thermal efficiency are realized at lower boiler pressure but at the expense of high heat demand from the boiler. In addition, lower boiler pressure corresponds to lower steam mass flow rate for the same power generation. Further studies are recommended on the heat exchanger design features impacting RRRC performance (area and orientation) for OFWH, CFWH and condenser.

References

Aderibigbe, D. A. and Osunbor, G. 2019. Performance of a 270 MW gas power plant using exergy and heat rate. *Energy and Power Engineering*, Vol. 11, pp. 15-34. https://doi.org/10.4236/epe.2019.112002

Al-Nasrawei, S.H., Radhi, R.M., Alhwayzee, M. 2022. Graphical prediction of optimum pinch point value with AEA software simulation and validation applied in closed cycle gas power plant. *International Journal of Heat and Technology*, Vol. 40, No. 5, pp. 1223-1233. https://doi.org/10.18280/ijht.400514

Al-Taha, W.H. and Osman, H., 2018. Performance analysis of a steam power plant: A case study. *MATEC Web of Conferences* Vol. 225, 05023. https://doi.org/10.1051/matecconf/201822505023.

Bansal, P. K. and Chin, T. C. 2003. Modelling and optimization of wire-and-tube condenser. *International Journal of Refrigeration*, Vol. 26, No. 5, pp. 601–613

Basha, M. A., & Habib, M. A. 2011. Supercritical and ultra-supercritical power plants. *Energy Conversion and Management*, Vol. 52, No. 7, pp. 3070-3078

Bekdemir Ş, Öztürk R, Yumurtac Z. 2003. Condenser optimization in steam power plant. *Journal of Thermal Science*, Vol. 12, pp. 176–178, https://doi.org/10.1007/s11630-003-0062-4

Cai, L., Hou, Y., Li, F. *et al.* Impacts of startup, shutdown and load variation on transient temperature and thermal stress fields within blades of gas turbines. *Journal of Thermal Sciences*, Vol. 31, pp. 727–740 (2022). https://doi.org/10.1007/s11630-022-1603-z

Chantasiriwan, S. 2021. Comparative thermo-economic analysis of regenerative Rankine cycles with two feed water heaters. *Case Studies in Thermal Engineering*, Vol. 28, 101476, https://doi.org/10.1016/j.csite.2021.101476.

Cheng, Y., Zheng, G., Wei, C., Mu, Q., Zheng, B., Wang, Z., Gao, M., Zhang, Q., He, K., Carmichael, G., Pöschl, U. and Su, H. 2016. Reactive nitrogen chemistry in aerosol water as a source of sulfate during haze events in China. *Science Advances.* Vol. 2, e1601530. https://doi.org/10.1126/SCIADV.1601530.

Chowdhury, S.A., Aziz, S. and Hossan, B.M. 2022. Cost efficiency evaluation of thermal power plants in bangladesh using a two-stage DEA model. *Economics of Energy & Environmental Policy*, Vol. 11, No. 1, pp. 145-163. https://doi.org/10.5547/2160-5890.11.1.scho

Colle, S., Vanderschuren, J. and Thomas, D. 2004. Pilot–scale validation of the kinetics of SO_2 absorption into sulphuric acid solutions containing hydrogen peroxide. *Chemical Engineering and Processing*, Vol. 43, pp. 1397-1402.

Da Cunha, A., Fraidenraich, N. and Silva, L. 2017. Optimum efficiency analysis of regenerative cycle with feed water heaters. *Journal of Power and Energy Engineering*, Vol. 5, 45-55. https://doi.org/10.4236/jpee.2017.58004

De Monte, M., Padoano, E. and Pozzetto, D. 2003. Waste heat recovery in a coffee roasting plant. *Applied Thermal Engineering*, Vol. 23, 1033-1044.

Dincer, I. and Al-Muslim, H. 2001. Thermodynamic analysis of reheat cycle steam power plant. *International Journal of Energy Research*, Vol. 25, pp. 727-739.6

Durmayaz, A., Sogut, O.S., Sahin, B. and Yavuz, H., 2004. Optimization of thermal systems based on finite-time thermodynamics and thermoeconomics. *Progress in Energy and Combustion Science*, Vol. 30, No. 2, pp. 175-217. https://doi.org/10.1016/j.pecs.2003.10.003.

El-Wakil, M. M. (1984). Power Plant Technology. McGraw-Hill.

Fraidenraich, N., Oliveira, C., da Cunha, V., Andre, F., Gordon, J. M. and Vilela, O. C. 2013. Analytical modeling of direct steam generation solar power plants. *Solar Energy*, Vol. 98, 511-522. https://doi.org/10.1016/j.solener.2013.09.037.

Franco, A. and Claudio, C. (2002). On some perspectives for increasing the efficiency of combined cycle power plants. *Applied Thermal Engineering*. Vol. 22. 1501-1518. 10.1016/S1359-4311(02)00053-4.

Gu, H., Cui, X., Zhu, H., Si, F. and Kong, Y. 2021. Multi-objective optimization analysis on gas-steam combined cycle system with exergy theory. *Journal of Cleaner Production*, Vol. 278, 123939, https://doi.org/10.1016/j.jclepro.2020.123939.

Gupta, M. K. and Kaushik, S. C. 2010. Exergy analysis and investigation for various feed water heaters of direct steam generation solar-thermal power plant. *Renewable Energy*, Vol. 63, No. 6, pp. 1228-1235. doi:10.1016/j.renene.2009.09.007.

Habib, M. A., & Basha, M. A. (2011). Supercritical and Ultra-supercritical Power Plants. ASME Journal of Power and Energy Systems, Vol. 23(3), 311-320.

Hasan, N., Rai, J. N. and Arora, B. B. 2014. Optimization of CCGT power plant and performance analysis using MATLAB/Simulink with actual operational data. *SpringerPlus*, Vol. 3:275, pp. 1-9, http://www.springerplus.com/content/3/1/275.

Holman, J. P. 2010. Heat transfer. McGraw-Hill Education.

Ibrahim, T. K. and Rahman, M. M. 2010. Effects of operation condition on performance of a gas turbine power plant. *Procedia Engineering*, Vol. 15, 4216 4223. http://dx.doi.org/10.1016/j.proeng.2011.08.791.

Incropera, F. P., & DeWitt, D. P. 2002. Fundamentals of heat and mass transfer. John Wiley & Sons.

IPCC Special Report on Renewable Energy Sources and Climate Change Mitigation. 2011. Cambridge University Press.

Jouhara, H., Khordehgah, N., Almahmoud, S., Delpech, B., Amisha Chauhan, A. and Tassou, S.A., 2018. Waste heat recovery technologies and applications. *Thermal Science and Engineering Progress*, Vol. 6, 268-289, https://doi.org/10.1016/j.tsep.2018.04.017.

Karri, V. 2012. A theoretical investigation of efficiency enhancement in thermal power plants. *Modern Mechanical Engineering*, Vol. 2 No. 3, 2012, pp. 106-113. doi: 10.4236/mme.2012.23013

Kehlhofer, R. H. 2009. Combined-Cycle Gas & Steam Turbine Power Plants. PennWell Books.

Kotas, T. J. 2007. The exergy method of thermal plant analysis. Butterworth-Heinemann)

Lee, Y., Bae, J., Kang, Y., Kang, Y. 2014. Korean atmospheric environmental impact assessment for thermal power plant. *Journal of Geoscience and Environment Protection*, Vol. 2, 25-27. doi: 10.4236/gep.2014.24004.

Lei, Biao & Wu, Yuting & Ma, Chong-Fang & Wang, Wei & Zhi, Rui-Ping. 2017. Theoretical analyses of pressure losses in organic Rankine cycles. *Energy Conversion and Management*. Vol. 153. 157-162. 10.1016/j.enconman.2017.09.074.

Li, Z., Zhang, H. and Chen, H. 2020. Application of transport membrane condenser for recovering water in a coal-fired power plant: A pilot study. *Journal of Cleaner Production*, Vol. 261, 121229. https://doi.org/10.1016/J.JCLEPRO.2020.121229.

Liu, Z. Karimi I. A. 2018. Simulation and optimization of a combined cycle gas turbine power plant for part-load operation. *Chemical Engineering Research and Design*, Vol. 131, pp. 29-40, doi.org/10.1016/j.cherd.2017.12.009.

Liu, S., Liu, J., Pei, L., Zhi–kizng, Y., Gao, J. and Cheng–hua, X. 2008. Simulation studies on limestone dissolution with organic acid additives in limestone–based flue gas desulfurization process Proceedings of the *2nd International Conference on Bioinformatics and Biomedical Engineering*, ICBBE 2008, Shanghai 2008, May 16–18, 2008, pp. 3899–3902.

Lonsdale, C. R., Stevens, R. G., Brock, C. A., Makar, P. A., Knipping, E. M. and Pierce, J. R. 2012. The effect of coal–fired power–plant SO_2 and NO_x control technologies on aerosol nucleation in the source plumes *Atmospheric Chemistry and Physics*, Vol. 12, pp. 11519-11531.

Ma, S., Chai, J., Jiao, K., Ma, L., Zhu, S. and Wu, K. 2017. Environmental influence and countermeasures for high humidity flue gas discharging from power plants. *Renewable and Sustainable Energy Review*. Vol. 73, 225-235. https://doi.org/10.1016/J.RSER.2017.01.143.

Ma, Y., Wang, Z., Lu, J., and Yang, L. 2018. Techno-economic analysis of a novel hot air recirculation process for exhaust heat recovery from a 600 MW brown-coal-fired boiler. *Energy*, Vol. 152, 348-357. https://doi.org/10.1016/J.ENERGY.2018.03.176

Mahir, S. and Manyele, S. (2023). Assessment of a multi-stage wet scrubber performance based on characteristics of scrubbing solution. *Tanzania Journal of Engineering and Technology*, Vol. 42, No.. 2, pp. 206-223.

Manyele, S. V. 2008. Toxic Acid Gas Absorber Design Considerations for Air Pollution Control in Process Industries using Computer Simulation. *Educational Research and Review*, Vol. 3(4), 137-147.

Martini, A., Sorce, A., & Traverso, A. 2014. Field measurement reconciliation for combined cycle heat recovery steam generator monitoring. *The Future of Gas Turbine Technology 7th International Conference*, 14-15 October 2014, Brussels, Belgium Paper ID Number (11).

Marzouk, O., Arman, A., Saadi, M., Al-Maqbali, A. and Al Sharji, S. 2022. Thermal power plant with 1 GW capacity for meeting future national electric demands. *Journal of Power and Energy Engineering*, Vol. 10, 1-11. doi: 10.4236/jpee.2022.108001.

Matee, V.E. and S.V. Manyele, 2015, Analysis of temperature profiles and cycle time in a large-scale medical waste incinerator, *Engineering*, Vol. 7, pp. 717-732. http://dx.doi.org/10.4236/eng.2015.710063.

McGovern, J. 2015. Thermodynamic Analysis of Closed Steady or Cyclic Systems. *Entropy*, Vol. 17(10), 6712-6742.

Moran, M. J., & Shapiro, H. N. 2006. Fundamentals of engineering thermodynamics. John Wiley & Sons.

Mwaria,P. R., Kaseva, M. E. and Manyele, S. V. 2021. Characterization of healthcare waste in Tanzanian zonal referral hospitals as a key factor for energy recovery. *African Journal of Environmental Science and Technology*, Vol. 159, pp. 349-365. https://doi.org/10.5897/AJEST2020.2917.

Ogulata, R. T. 2004. Utilization of waste-heat recovery in textile drying, Applied Energy, Vol. 79, pp. 41-49.

Omar, H., Kamel, A. and Alsanousi, M. 2017. Performance of Regenerative Gas Turbine Power Plant. *Energy and Power Engineering*, Vol. 9, 136-146. doi: 10.4236/epe.2017.92011.

Oyedepo, O.S. and Kilanko, O. 2014. Thermodynamic analysis of a gas turbine power plant modelled with evaporative cooler. *International Journal of Thermodynamics*, Vol. 17, pp. 14-20. https://doi.org/10.5541/ijot.480

Patel, D. K. 2015. Improve Steam Turbine Efficiency by use of Reheat Rankine Cycle. *International Journal of Engineering Research and Science and Technology*, Vol. 47, pp. 361-368.

Pettersson, F. and Soderman, J. 2007. Design of robust heat recovery systems in paper machines, *Chemical Engineering and Processing: Process Intensification*, Vol. 46, 910-917.

Pitarch, J. L., Palacín, C. G., de Prada, C., Voglauer, B. and Seyfriedsberger, G. 2017. Optimization of the resource efficiency in an industrial evaporation system. *Journal of Process Control*, Vol. 56, 1–12.

Ponte, J.G., and Rocha, P.A. (2023). Technical and financial viability of a 1 MW CSP power plant with organic rankine module: case study for a northeastern Brazilian City. *International Journal of Energy and Environmental Engineering*, Vol. pp. 1-27, https://doi.org/10.32388/NKKKOI

Rao, S. (2006). The Handbook of Turbine Design. Springer.

Söylemez, M. S. 2000. On the optimum heat exchanger siz-ing for waste heat recovery. *Energy Conversion and Management*, Vol. 41, No. 13, pp. 1419-1427, https://doi.org/10.1016/S0196-89049900181-8.

Söylemez, M. S. 2011. On the Thermo-Economical Optimization of Feed Water Heaters in Thermal Power Plants, 2011. *Smart Grid and Renewable Energy*, Vol. 2 No. 4, pp. 410-416. doi: 10.4236/sgre.2011.24047

Tahri, T., Abdul-Wahab, S. A., Bettahar, A., Douani, M., Al-Hinai, H. and Al-Mulla, Y. 2009. Simulation of the condenser of the seawater greenhouse. Part I: theoretical development., *Journal of Thermal Analysis and Calorimetry*, Vol. 96, no. 1, pp. 35–42.

Tan, H., Cao, R., Wang, S., Wang, Y., Deng, S. and Duić, N. 2020. Proposal and techno-economic analysis of a novel system for waste heat recovery and water saving in coal-fired power plants: A case study. *Journal of Cleaner Production*, Article 124372. https://doi.org/10.1016/j.jclepro.2020.12437

Tang, W., & Ge, Y. 2022. Optimal design of a dual-pressure steam turbine for rankine cycle based on constructal theory. *Energies*, Vol. 15, No. 13, Article 4854. https://doi.org/10.3390/en15134854

Toledo, M., Abugaber, J., Lugo, R., Salazar, M., Rodríguez, A. and Rueda, A. 2014. Energetic analysis of two thermal power plants with six and seven heaters. *Open Journal of Applied Sciences*, Vol. 4, 6-12. http://dx.doi.org/10.4236/ojapps.2014.41002

Tominaga, J. 2017. Steam turbine cycles and cycle design optimization: Advanced ultra-supercritical thermal power plants and nuclear power plants. In T. Tanuma, Ed., *Advances in Steam Turbines for Modern Power Plants*, Woodhead Publishing, pp. 41-56, https://doi.org/10.1016/B978 -0-08-100314-5.00003-8..

Vundela, S., Kaushik, S.C., Tyagi, S. and Panwar, N.L., 2010. An approach to analyze energy and exergy analysis of thermal power plants: A review. *Smart Grid and Renewable Energy*. https://doi.org/10.1016/10.4236/sgre.2010.13019

Wang, Zhengxian& Liu, Tong & Luo, Renda. 2021. Study on water erosion and preventive measures of last stage blade of steam turbine. *Journal of Physics: Conference Series*. 2076. 012071. 10.1088/1742-6596/2076/1/012071.

Wiener, M.S. and Salas, B.V., 2009. Erosion-corrosion in industrial steam turbines. *Materials Performance*. Vol. 48, No. 9.

Xiong, Y., Tan, H., Wang, Y., Xu, W., Mikulčić, H. and Duić, N. 2017. Pilot-scale study on water and latent heat recovery from flue gas using fluorine plastic heat exchangers. *Journal of Cleaner Production*, Vol. 161, 1416-1422. https://doi.org/10.1016/J.JCLEPRO.2017.06.081

Zhang, C., Yang, Y., Fan, L. and Huang, X. 2020. Numerical study on operating characteristics of self-driven total heat recovery system for wet-hot flue gas. *Applied Thermal Engineering*, Vol. 173, 115223. https://doi.org/10.1016/J.APPLTHERMALENG.2020.115223

Zhen, J. L., Wu, C. B., Liu, X. R., Huang, G. H. and Liu, Z. P. 2020. Energy-water nexus planning of regional electric power system within an inexact optimization model in Tangshan City, China. *Journal of Cleaner Production*, Vol. 266, 121997. https://doi.org/10.1016/J.JCLEPRO.2020.121997.

Ziebik, A., & Gładysz, P. 2002. Combined heat and power generation. *Applied Thermal Engineering*, Vol. 22, No. 10, pp. 1123-1133.

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