The effect of condenser backpressure on station thermal efficiency: Grootvlei Power Station as a case study

KM van Rooyen

20557167

Dissertation submitted in fulfilment of the requirements for the degree *Magister* in Development and Management Engineering at the Potchefstroom Campus of the North-West University

Supervisor: Prof JIJ Fick

November 2014

Executive summary

Grootvlei Power Station's thermal efficiency had been on a steady declining trend since it was re-commissioned in 2008, which had tremendous financial implications to the company at the time of writing. The main contributory factor to the thermal efficiency losses was identified to be the condenser backpressure losses that the station was experiencing. This loss was responsible for approximately 17% of the total efficiency losses. Therefore an investigation was conducted to determine the potential impact of the condenser backpressure loss on the thermal efficiency and the financial implications thereof. The deliverables were to determine the cause of the condenser backpressure loss and propose possible resolutions, to quantify the financial effect and to produce a cost benefit analysis in order to justify certain corrective actions.

Grootvlei Power Station is one of the older power stations in South Africa and it was used as the first testing facility for dry-cooling in South Africa. It consists of six 200MW units, two of which are dry-cooled units. In 1990 it was mothballed and due to rising power demands in South Africa, it was re-commissioned in 2008. Thermal efficiency has been playing a great role due to the power constraints and therefore it was deemed necessary to conduct this study.

The approach that was used was one of experimental and quantitative research and analyses, incorporating deductive reasoning in order to test various hypotheses of factors that could have been contributing to the backpressure losses. In order to do so, a logic diagram was designed which could be used to aid in the identification of possible causes of the condenser backpressure losses. The logic diagram was able to identify whether the problem had to do with the cooling tower or the condenser. It was able to identify which area on the condenser was defective i.e. whether the pumps were not performing, or whether the air ejectors were not performing. It was also able to indicate whether the inefficiency was due to air ingress or fouling.

Alongside the logic diagram, a condenser efficiency analysis was used in order to strengthen and improve on the investigation. This analysis was able to identify whether the condenser was experiencing fouling conditions, air ingress, passing valves or low cooling water flow.

After the investigation commenced, it was decided to focus on the two largest contributing units since the largest contributor was a dry-cooled unit and the second largest contributor was a wet-cooled unit, thus some comparison between the units was incorporated.

The condenser efficiency analysis on Unit 3 (wet-cooled unit) indicated a low cooling water flow, fouling as well as air ingress. The logic diagram indicated poor cooling tower performance, high air ingress as well as fouling. Further tests and analyses as well as visual inspections confirmed these phenomena and condenser fouling was identified to be the largest contributor to the backpressure loss on this unit.

The condenser efficiency analysis on Unit 6 indicated that air was entering the condenser. The logic diagram indicated that a segment of the backpressure loss was due to poor cooling tower performance. Inspection of the cooling tower indicated damage and leaks. A cooling tower performance test was conducted and the result of the test indicated that the tower was in need of cleaning. Further analyses according to the logic diagram indicated that the condenser was experiencing air ingress which concurred with the condenser efficiency analysis. A helium test, condensate extraction pump pressure test as well as a flood test was conducted on this unit and various air in-leakage points were identified.

The financial implications of the backpressure losses were investigated and found to be costing millions each month. The condenser backpressure loss was contributing more than 2% to the thermal efficiency loss. The cost benefit analysis indicated that the cost of cleaning the condenser on Unit 3 would be made up within six months and a return on investment of 16,6% was calculated. The cost benefit analysis motivates for extended outage times for the purpose of cleaning the condensers from a financial perspective.

Therefore, it was recommended to clean the condenser on Unit 3 and fix all known defects on the unit as well as on Unit 6. The cooling towers were recommended to be refurbished. Further investigation was recommended to determine the feasibility of installing an online cleaning system on the wet-cooled units' condensers such as a Taprogge system. Alternative investigation methods were suggested such as smoke stick analyses for air ingress determination. It was also recommended to review the maintenance strategies that were being used since many of the defects were found to be maintenance related.

If the identified problem areas are attended to, the condenser backpressure loss will decrease and the condensers transfer heat more efficiently which will lead to financial gains for Grootvlei Power Station as well as efficiency gains, plant reliability and availability gains.

Keywords: Condenser performance, thermal efficiency, temperature gradient, terminal temperature difference, condensate depression, temperature rise, condenser efficiency analysis, cooling tower performance.

Preface

Firstly I would like to thank Professor Johan Fick for serving as my study leader. Professor Fick's support, guidance and suggestions throughout the writing of this dissertation have been greatly appreciated and his extensive knowledge has constantly amazed me and he has taught me the value of hard work and endurance.

Special thanks go to Mr Nick Moolman, who acted as my mentor throughout this study and from whom I gained almost all of my technical knowledge and understanding that was needed for this research dissertation.

I would like to thank my manager, Mr Muzi Myeza, who has provided me with a lot of support and has allowed for me to attend many training sessions in order to complete this dissertation.

I would also like to thank our Almighty Father in Heaven, without whom I could do nothing, for I can do all things through Him that gives me strength as Paul wrote in his letter to the Philippians.

Last but not least, I would like to thank my family and friends who have supported me throughout the writing of this document, who have been there for me in stressful times and have encouraged me when I was discouraged.

Grootvlei, November 11, 2014

Kathryn van Rooyen

Table of contents

List of figures

List of tables

Definitions and abbreviations

Definitions

Condensate Depression

A condensate depression occurs when there is an air leak which leads to a loss of vacuum. The air leaking into the condenser causes a blanket of air around the condenser tubes which in prevents heat transfer.

Cooling Water Rise

The cooling water rise refers to the difference between the inlet temperature and the outlet temperature of the condenser.

Macrofouling in a condenser

Macrofouling is fouling that takes place due to large objects being present in the condenser and blocked the flow of water.

Outage

An outage is an appointed time that is allowed for a unit to be run down for maintenance and repair purposes.

S.T.E.P Factor (%)

The S.T.E.P factor is the ratio of the target station heat rate to the actual station heat rate, both averaged over the month, and is expressed as a percentage.

Terminal Temperature difference

The terminal temperature difference is indicated by the difference between the condenser outlet temperature and the hotwell temperature. The effect of dirty tubes or blocked tubes causes the temperature terminal difference to increase, due to the loss of heat transfer from the steam through the tube wall to the CW flow, and again affects the vacuum.

Thermal Efficiency of a Thermal Power Station (%)

The "thermal efficiency of a thermal power station", for a given period, is the quotient of the heat equivalent of 1 kWh and the average heat rate expressed in the same units. In Eskom, the term “overall thermal efficiency” means the heat rate was calculated using the net station production (kWh sent out or USO).

Abbreviations

Symbol	Description
A	Heat transfer surface area
ACF	Actual correction factor
ATCF	Acceptance test correction factor
BFPT	Boiler feed pump turbine
c/kWh	Cent per kilowatt hour
CD / C.D	Condenser depression
CEP	Condensate extraction pump
CV	Calorific value
CW	Cooling water
DE	Drive end
DO	Dissolved oxygen
EAL	Eskom's Academy of Learning
ECM	Engineering change management
Etc.	etcetera
HP	High pressure
kg/s	Kilogram per second
LMTD	Log mean temperature difference (ΔT_{LM})
LP	Low pressure
LPH	Low pressure heater
m^3/kg	Cubic metre per kilogram
mbar	millibar
MCR	Maximum capability running
NDE	Non Drive End
NPSH	Net Positive Suction Head
P_{sat}	Saturation pressure
Q	Heat duty

ROI	Return on investment
RTS	Return to service
SFPT	Steam feed pump turbine
STEP	Station Thermal Efficiency Performance
ΔT	Temperature driving force
T_1	Cooling water inlet temperature
T_2	Cooling water outlet temperature
T_c	Condensate temperature in condenser hotwell
$T_{v\,/}\,T_{sat}$	Backpressure equivalent saturation temperature
TR / T.R	Temperature Rise
TTD / T.T.D	Terminal Temperature Difference
U	Overall heat transfer coefficient
USO	Units sent out

Chapter 1 - Background, problem statement and deliverables

1.1 Background

Most of South Africa's electricity comes from thermal power stations fuelled by coal. However, the efficiency of a thermal power plant is typically varies between 30 and 50% (Roth, 2005) which means that less than half, and typically a third of the energy available in the coal, is converted into electricity. This fact justifies the importance of running a plant at maximum thermal efficiency. Energy security, as well as carbon dioxide emission reductions has become a concern and therefore the efficiency of a power plant plays a great role in this aspect also, since an efficient power plant reduces the consumption of coal (Leinster et al., 2013). A third factor that contributes to the importance of optimal power plant efficiency is that according to Hartnady (2010), South African coal reserves in Mpumalanga have been depleted extensively over the last decades and have been predicted to reach their peak in production rate in 2020 at 284 Mt/year in South Africa, at which stage the reserves will have been depleted by half of its total resources (±23 Gt).

Therefore, in order to maximize the utility of coal usage in power generation, a need exists to be able to monitor and control the thermal efficiency at which a power station operates. Eskom uses the Station Thermal Efficiency Performance (STEP) system for this purpose.

The STEP system is a tool for assessing power station thermal performance, i.e. to determine how efficiently the energy conversion process in the steam-generating cycle of a thermal power station takes place. Good thermal plant performance has both direct as well as indirect implications for Eskom. The direct implications include a potential saving in direct operating costs (mainly fuel), and the indirect implications include improved plant availability and reliability, as there is conclusive proof that poor thermal performance increases the incidence of plant outages (both planned and forced). Operating costs are also reduced when a power plant is optimized. (Moolman, 1980)

Grootvlei Power Station is one of Eskom's oldest power stations and was first commissioned in 1969 as a test facility for dry cooling in South Africa. Grootvlei has four traditional wet cooling units and two dry cooling units. In 1990 Grootvlei was mothballed due to the surplus of electricity generation capacity in South Africa at the time. Due to looming power shortages, re-commissioning of Grootvlei was started in 2006 and by 2008, two of the six units were back online. At the time of writing, all six units were online and the power stations had a maximum rating of 1200MW with each unit designed to deliver 200MW.

1.2 **Problem statement**

Grootvlei Power Station's thermal efficiency has been on a declining trend since it was re-commissioned in 2008. This decline in thermal efficiency had been costing Eskom millions each month as is indicated on the graph below.

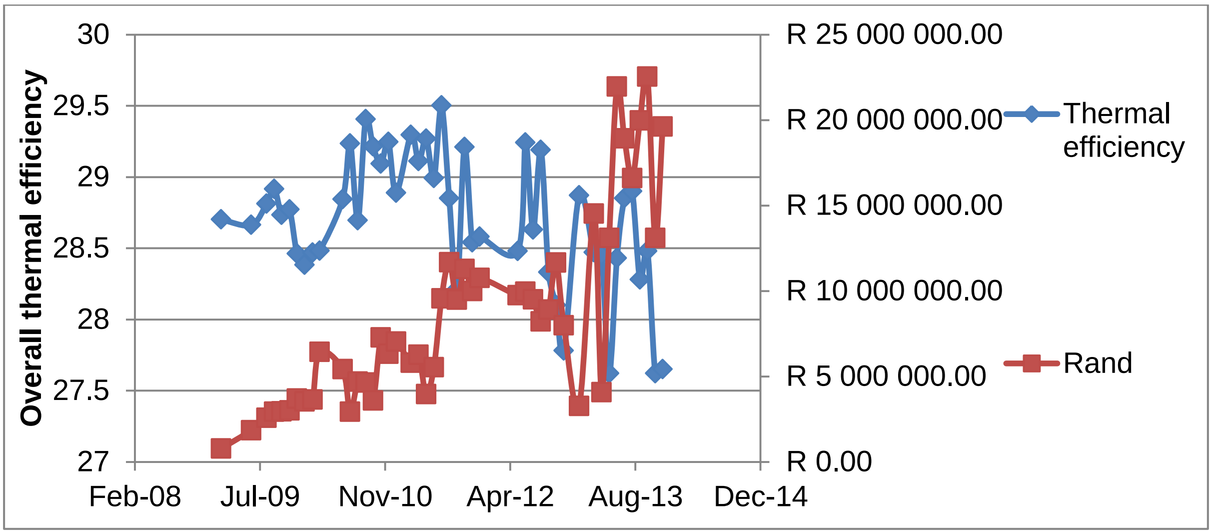


Figure 1 Graph displaying the steady decline in station thermal efficiency alongside the monetary impact thereof (2009 - 2013)

The STEP Losses were trended by a consultant for Eskom for the last year in order to establish which loss has the largest contribution to the efficiency problems. A STEP loss is the difference between the actual and the target loss of a measured variable.

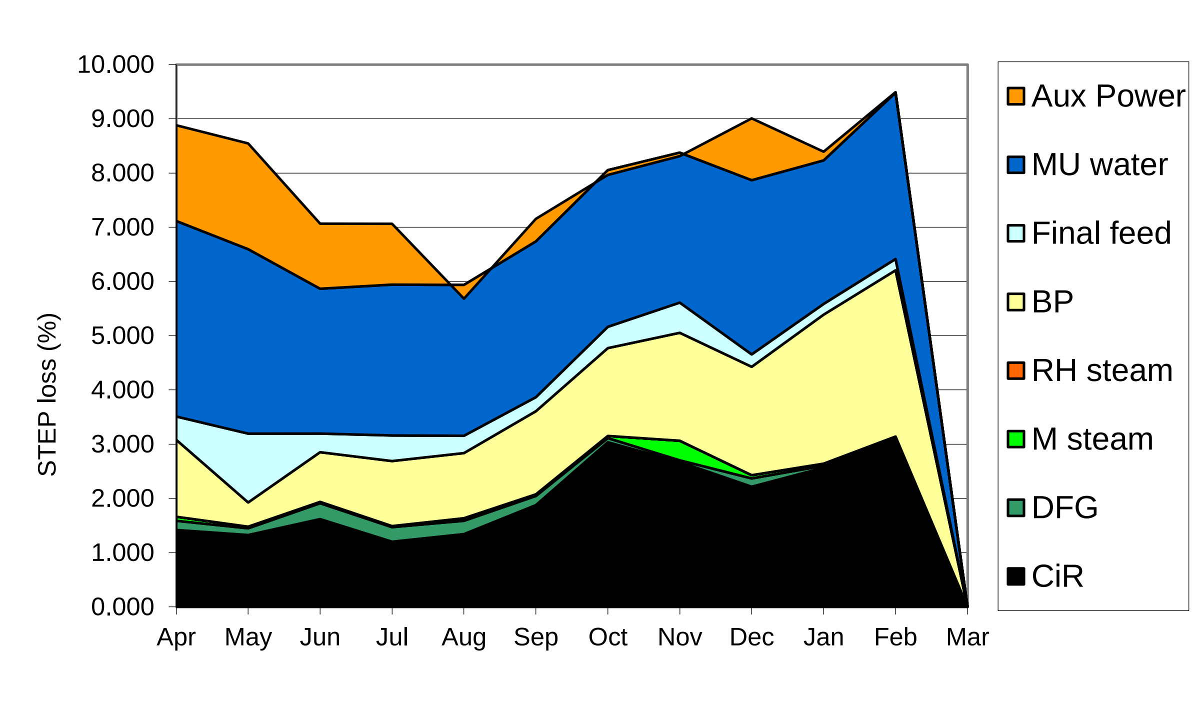


Figure 2 STEP loss trends indicating the three largest losses being experienced at Grootvlei Power Station: April 2013 – Feb 2014 (Zwiegelaar, 2014)

From analyzing Figure 2, the three main contributors to the STEP losses for the year 2013/2014 were:

- Make-up water (see section 2.5.3.2 for full description of this loss)
- Condenser backpressure (see section 2.5.2.2 for full description of this loss)
- Carbon in refuse (see section 2.5.1.1 for full description of this loss)

A closer examination of the losses indicated that the largest contributor to the thermal losses was the condenser backpressure losses as is indicated below.

Carbon in Refuse Loss	**15.8 %**
Condenser Backpressure Loss	**16.9 %**
Make-up Water Loss	**15.6 %**

Since the condenser backpressure loss was also the largest monetary contributor, and had been steadily increasing over the last year (Figure 2), it was decided to focus on the potential impact that this loss was having on the thermal efficiency of the power station and therefore, the planning of this research was based on the investigation of the condenser backpressure loss.

1.2.1 Research hypothesis and deliverables

1.2.1.1 *Hypothesis*

The thermal efficiency of Grootvlei Power Station is on a steady decline and the high condenser backpressure is postulated to be contributing substantially to this decline. If the cause of high condenser back pressure can be identified and the monetary implications quantified, then corrective action can be proposed and motivated via a cost benefit analysis.

1.2.1.2 *Deliverables*

The main deliverables of this research are listed below:

1. Determine what was causing high condenser backpressure.
2. Propose resolutions for the identified problem areas with the aim of minimizing the backpressure losses
3. Quantify the financial effect of minimizing the backpressure losses on Grootvlei Power Station
4. Quantify the effect of backpressure losses on the thermal efficiency
5. Produce a cost benefit analysis in order to motivate for corrective actions to be taken

1.3 Chapter division

Herewith the chapter division of this document

1.3.1 Executive summary and preface

The executive summary is a short summary of the report, acquainting the reader with the report without requiring the reader to read the entire report. The preface is where all relevant parties are thanked.

1.3.2 Chapter 1 – Background and problem statement

A brief introduction and some background information to the problem is given.

1.3.3 Chapter 2 – Literature survey

The literature survey discusses the problem areas that are usually related to condenser backpressure losses as well as the type of tests one can do to pinpoint such issues. It also provides a brief introduction to the Station Thermal Efficiency Performance tool that is used at all of the South African coal fired power stations to calculate thermal efficiency.

1.3.4 Chapter 3 – Methodology of investigation

The research was aimed at identifying opportunities for reducing the thermal efficiency losses experienced at Grootvlei Power Station, focussing on losses concerned with backpressure issues on the main condenser. This chapter describes the methodology used to investigate the causes of backpressure losses and the steps that were followed in order to investigate the stated hypothesis.

1.3.5 **Chapter 4 – Results and discussion**

The investigation results are recorded and discussed in this chapter.

1.3.6 **Chapter 5 – Conclusion and recommendation**

Following the discussion, a conclusion alongside some recommendations are made. A cost benefit analysis is included in this chapter in order to motivate the recommended changes that should be set in place.

Chapter 2 - Literature review

This chapter serves to provide some background to condensers as well provide insight into the STEP system, how it operates and which factors it takes into consideration when calculating the efficiency of a power plant. It also describes some of the work that has been done at Grootvlei Power Station in order to identify factors that are contributing to the vacuum decay in the condensers and the processes that were followed in order to illuminate some of the possible contributing factors.

2.1 **Basic overview of Grootvlei Power Station**

Grootvlei Power Station was first commissioned in 1969. In 1990 there were no power constraints and Grootvlei was mothballed. After this, the station was re-commissioned in 2008 due to rising power demands in South Africa.

Grootvlei was used as the first testing facility for dry cooling in South Africa and it consists of four wet-cooled 200MW units and two 200MW dry-cooled units.

Figure 3 A Google Earth image of Grootvlei Power Station (2014)

The initial design efficiency of Grootvlei at Maximum Continuous Rating was 32.9%.

2.1.1 Basic layout of the power station

Most of South Africa's electricity comes from thermal power stations fuelled by coal. The following is a basic description of the electricity generation process that takes place at Grootvlei Power Station:

Coal is obtained from nearby mines and delivered to the coal stockyard. From the coal stockyard it is conveyed to the mill bunkers where it is pulverised to a fine powder and then blown into the furnace. This is done with the aid of primary air which is heated to approximately 85°C. The primary air must be sufficient at all times to keep the pulverised fuel in suspension in the fuel pipe.

The coal is burnt and the energy that is given off is used to heat demineralised water in an array of boiler tubes. This generates high pressure steam which drives a turbine. The high pressure steam is at approximately 540°C. The unwanted gases formed from the combustion process are sent to the smoke stack where it is released into the atmosphere. The thermal energy is converted into mechanical energy by a steam turbine. A generator is coupled to the turbine shaft and converts the mechanical energy into electrical energy. From here, the generator produces an AC voltage which is stepped up by the transformers to a higher voltage to minimise transmission losses and then transferred to the grid. Hereafter it is transferred to the substation and lastly to the consumers. A basic diagram of the electricity generation process is shown in Figure 4.

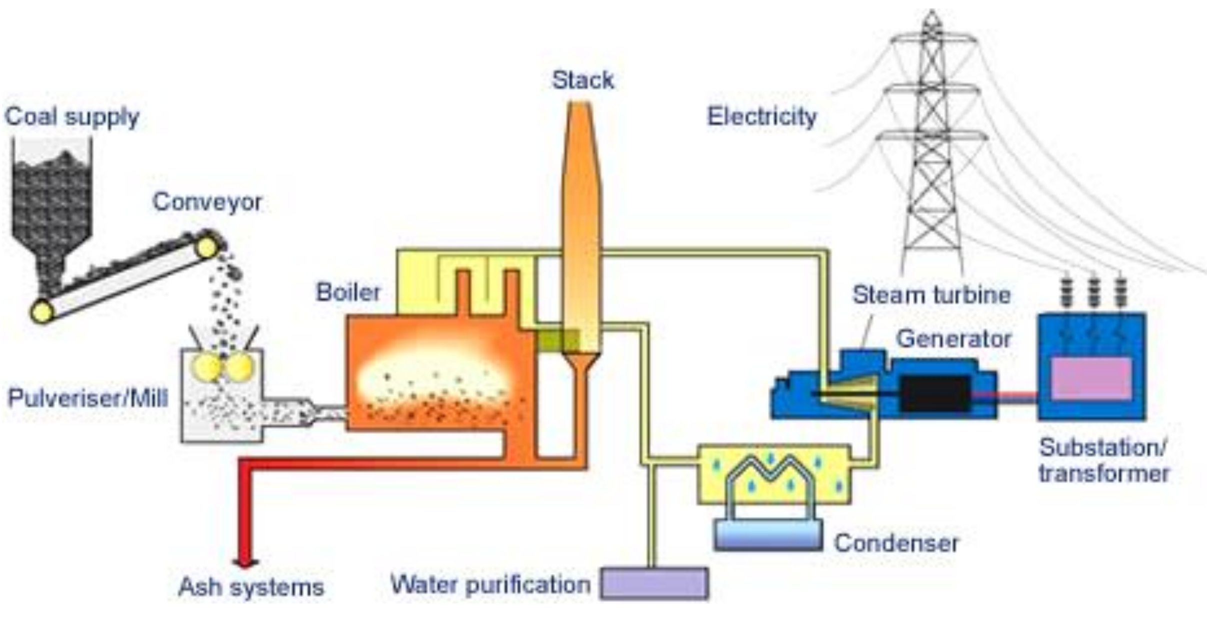


Figure 4 Basic diagram of the process followed to generate electricity

For the purpose of this research, the focus will be mainly on the condenser areas at Grootvlei Power Station, since the condenser backpressure loss was identified to be the highest contributor to the efficiency losses.

2.2 Condensers

The object of a condenser is to remove the latent heat of evaporation from the gas or vapour, which is to be condensed. In the case of a power station, the vapour is steam and therefore the condensate water. This condensing of the steam to water enables the water to be re-used over and over again in a closed system. The use of a condenser means that the turbine exhaust pressure may be reduced to a partial vacuum, enabling the steam to be expanded to a lower pressure, enabling more useful work to be extracted from the steam in the last stage of the turbine. (Moolman , 1999)

2.2.1 Types of condensers

There are two main types of condensers that are used at Grootvlei Power Station:

2.2.1.1 *Surface condenser*

In a surface condenser the condensing steam and the cooling water are prevented from mixing by means of tubes, i.e., one fluid will flow over the tubes whilst the other flows through the tubes. Grootlvei makes use of these condensers on Units 1 – 4 and 6. (McNaught, 2011)

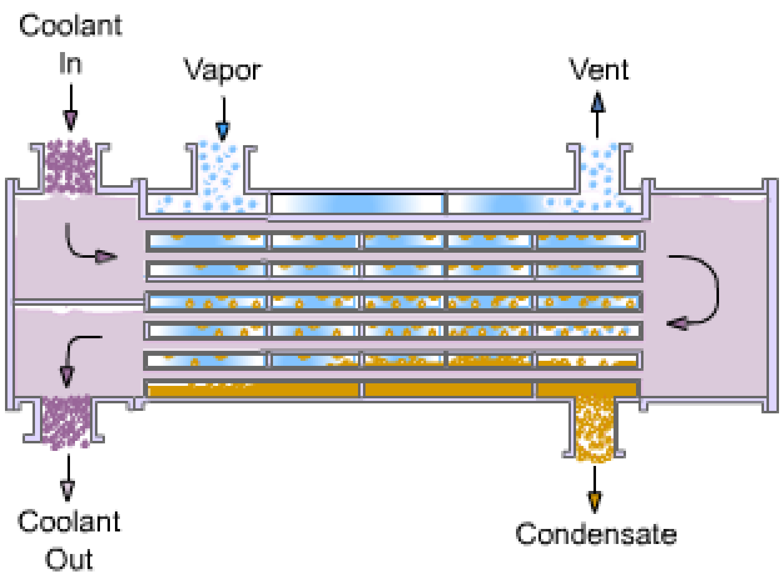


Figure 5 Schematic of a shell and tube type surface condenser (McNaught, 2011)

2.2.1.2 *Spray type condenser*

This is a direct contact condenser in which the cooling water is sprayed, using nozzles, into the steam, i.e., the two mix together. The spray should be of a very fine nature in order to maximize the interfacial area for heat transfer. For the same purpose, the residence time of the liquid should also be long enough. The advantages of this type of condenser are that it is low in cost and the simplicity of the mechanical design. (McNaught, 2011)

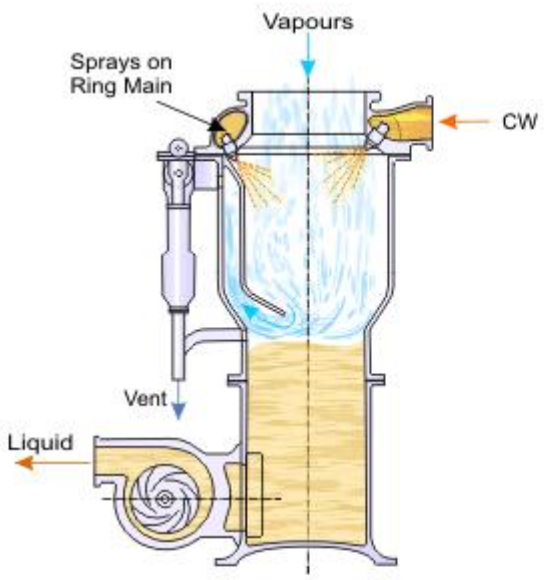


Figure 6 Visual of a spray type condenser (McNaught, 2011)

2.2.2 Separating Vacuum and Atmosphere

In order to fulfil the primary function of the condenser, the steam/condensate chamber must operate under sub-atmospheric pressure. To maintain vacuum and to avoid air contamination, a boundary must therefore be maintained between the condenser internals and the outside atmosphere. This boundary is quite large and must be maintained air-tight to avoid leakage into the system. (Gibbard & Terranova, 2010)

In practice, it is not realistic to totally exclude air from the steam space in the condenser. A vacuum pump or ejector is provided to continuously extract air from the steam space. The capacity of this extraction system is selected to avoid air accumulation i.e. the rate of extraction matches a design rate of air in-leakage. If the rate of air in-leakage rises above the design level, however, air will accumulate in the steam space and cause blanketing of the tubes and thereby inhibit heat transfer. (Gibbard & Terranova, 2010)

The condenser must therefore be monitored to detect the accumulation of air above design concentrations.

2.2.2.1 *Application of Dalton's law*

In the case of a shell and tube side condenser, Dalton's law applies. If, for example, the condenser shell is filled with steam alone and has a pressure of 10 kPa and if 2kPa of air is mixed into this steam; then the total pressure would be the sum of the partial pressures. It would be 12 kPa. In this case the corresponding saturation temperature would be 49,45 ºC. Therefore the condenser would show a condensate depression of 3,55 ºC as an indication of the pressure of air. Thus the temperature of the condensate depends on the partial pressure of the steam and not on the total pressure of the air and steam together. (Rathore, 2010)

From the above, it may be concluded that when air is present in a condenser, the heat transfer is impeded. This effect affects the backpressure negatively.

2.2.3 **Condensate Recovery**

The water used in the power cycle is treated in order to avoid fouling and corrosion, and as such is a valuable commodity. The efficiency of the condenser plant is therefore optimised by ensuring that as much as possible of the water used is retained within the cycle. The condenser has one primary and several secondary roles to play in this respect. Its primary role is to convert the uncondensed steam exhausting the turbine to condensate and collect it for re-use in the cycle. One of the condenser's secondary roles is to act as a collection point for water which exits the cycle at other points than the turbine exhaust. Examples of this are condensate drains and vents from feedwater heaters. The condenser is a convenient point to collect these vents and drains, since it is the lowest pressure point in the cycle, so flow towards the condenser is guaranteed. (Gibbard & Terranova, 2010)

2.2.4 **Condensate Reservoir**

The condensate generated by the condenser is pumped back around the steam cycle by the condensate extraction pumps (CEPs). The extraction pumps are so named because they must extract the condensate against the vacuum present in the condenser and create the differential head to send the condensate to the de-aerator. This is a difficult duty due to the relatively low NPSH available. The condenser assists the CEPs by providing a reservoir of condensate in the hotwell. This reservoir maintains a liquid level above the pump suction to ensure sufficient NPSH, and also acts as a buffer against flow variations. In order to fulfil this function, the condenser level control system must maintain the correct liquid level in the hotwell. If the level is too low the pump NPSH will be reduced and this will lead to pump cavitation. If the level is too high the bottom of the tube bundle may be flooded and the de-aeration function of the condenser may be compromised. (Gibbard & Terranova, 2010)

2.2.5 **De-Aeration**

Regardless of the oxygen level philosophy elsewhere in the plant, the oxygen level in the condenser is always maintained at very low levels (generally in the parts per billion ranges) in order to prevent air blanketing in the condenser. In order to achieve this, the condenser actually acts as a de-aerator, especially with regard to the make-up water which enters the condenser saturated with oxygen. The ability of the condensate to absorb oxygen is affected by the oxygen concentration in the steam space and the temperature of the condensate. In order to avoid high oxygen levels, the air concentration in the steam space must be minimised and sub-cooling of the condensate must be avoided. These conditions will generally be achieved by avoiding air in-leakage. (Gibbard & Terranova, 2010)

2.2.6 **Condenser performance**

A number of condenser operating parameters are routinely measured on an operating plant and the measured values provide valuable information regarding the condition of the condenser. In order to understand the significance of the measured values it is necessary to gain an understanding of condenser performance and behaviour. In essence, a condenser is a relatively simple heat exchanger whose performance can be described by the following equation: (Gibbard & Terranova, 2010)

$$Q = U \times A \times \Delta T \qquad (1)$$

Where:

Q = heat duty (MW)

U = overall heat transfer coefficient (W/m^2K)

A = heat transfer surface area (m^2)

ΔT = temperature driving force (K)

The heat duty of a condenser is almost entirely dependent on the steam flow coming from the turbine and is therefore a function of electrical generating load. If the steam flow is known, the duty can be calculated by multiplying the steam flow by the latent heat of vaporisation. Small additional loads arise from steam entering the condenser from other sources than the LP turbine (e.g. high pressure drains from feedwater heaters) but these loads are normally insignificant compared to the primary load. (Gibbard & Terranova, 2010)

The overall heat transfer coefficient represents the rate of heat transfer in the condenser and is affected by many parameters. The primary factors that affect the overall heat transfer coefficient are as follows (Rathore, 2010):

- Cooling water velocity
- Fouling
- Air ingress
- Tube material and/or coatings

The velocity of the cooling water affects the heat transfer rate with higher velocities giving higher rates of heat transfer. The water velocity is affected by the number of cooling water pumps in service and by the amount of water each pump delivers. The amount of water delivered by the cooling water pumps is affected by the flow resistance in the cooling water circuit, as indicated by the pump performance curve. (Rathore, 2010)

The flow delivered by the pump is indicated by the intersection of the pump curve to the system resistance curve and this is affected by fouling in the condenser tubes as shown in Figure 7. Although Figure 7 is drawn for the simple case of a single pump, the principle holds true for multiple pumps also. (Gibbard & Terranova, 2010)

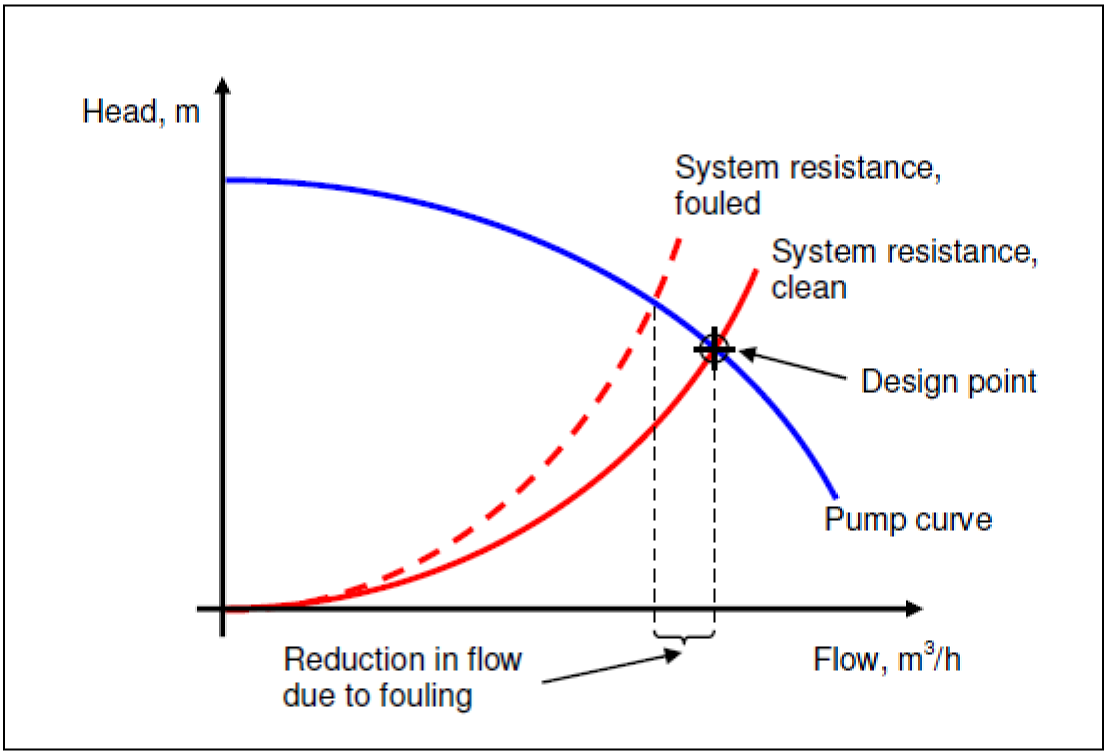


Figure 7 Pump performance curve and the effect of condenser fouling (Gibbard & Terranova, 2010)

The above mentioned factors that influence the condenser are discussed below (Gibbard & Terranova, 2010):

- Fouling also affects the overall heat transfer coefficient directly, by imposing an insulating layer on the inside surface of the tubes (external fouling of condenser tubes is extremely rare). The thicker the layer of fouling the lower the value of the heat transfer coefficient as the heat released by the steam has to travel through the fouling layer to reach the cooling water.
- Air ingress affects the overall heat transfer coefficient, as an increased concentration of air in the steam space will reduce the rate at which steam condenses on the outside of the tubes. In the absence of air, the steam condenses directly onto the film of condensate present on the outside of the tubes. If air is present in the steam space, the molecules of steam must diffuse through the air to reach the condensate film in order to condense. This slows down the condensation process, reducing the overall heat transfer coefficient. The higher the air concentration, the lower the value of U will be.
- Another consequence of condensing in the presence of air is that the condensate tends to sub-cool. Without air present the heat removal from the condensate layer is balanced by heat addition from the condensing steam. The net result is that the condensate remains close to the saturation temperature of the steam. In the presence of air, however, the rate of steam condensation is slowed such that the rate of heat removal exceeds the rate of heat addition from the steam. The excess heat removal capacity is met by removing sensible heat from the condensate, reducing its temperature. This effect on the condensate temperature provides a means of detecting the presence of air in the steam space.
- The heat transfer surface area is the total tube surface exposed to the steam, usually defined as the outside surface area of the tubes. On a day-to-day basis the area is a fixed quantity, but the effective surface area may reduce over time due to tube plugging. When a tube fails (or is close to failure) it will be plugged to prevent contamination of the condensate by cooling water.
- The temperature driving force is the difference in temperature between the steam and the cooling water. Although the steam condenses at a constant temperature (T_{sat}), the temperature of the cooling water varies in the condenser. This variation in cooling water temperature gives rise to a variation in temperature driving force internally within the condenser. In order to derive an overall temperature driving force

an average figure must be used, and this is the log-mean temperature difference (LMTD):

$$\Delta T_{LM} = \frac{(T_{sat} - T_1) - (T_{sat} - T_2)}{L_n\left(\frac{(T_{sat} - T_1)}{(T_{sat} - T_2)}\right)} \tag{2}$$

Where:

ΔT_{LM} = Log mean temperature difference (K)

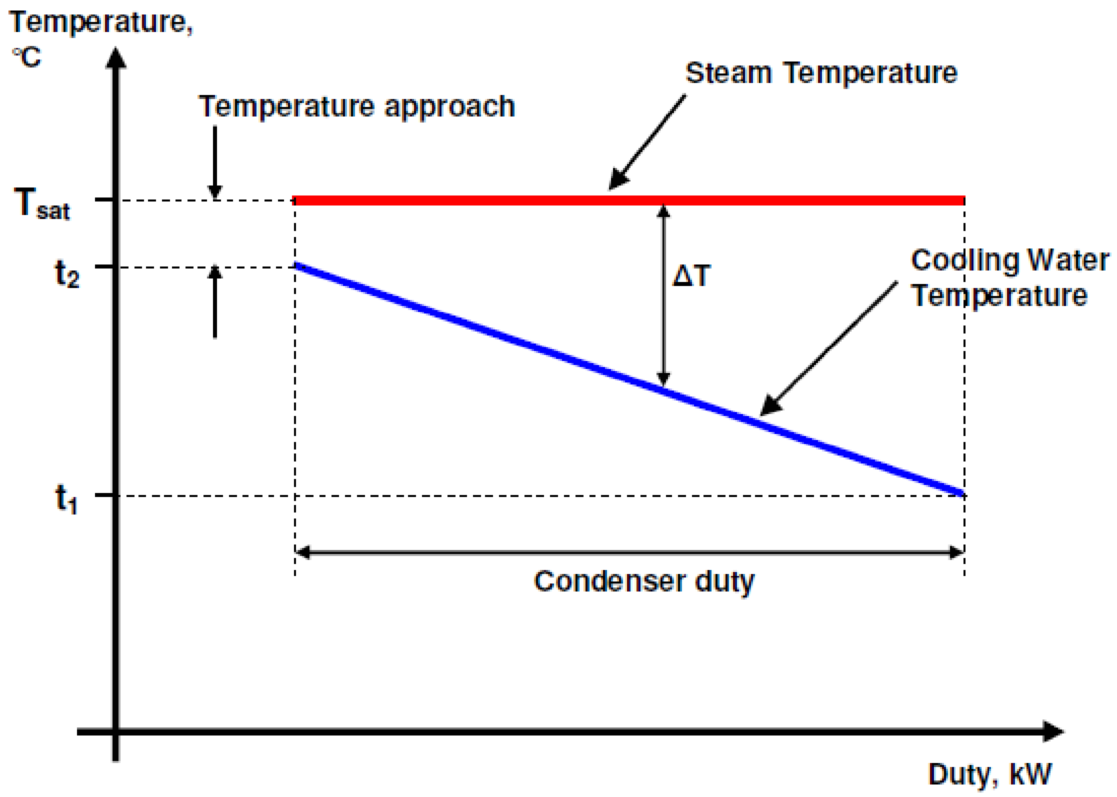


Figure 8 Temperature profiles in a condenser (Gibbard & Terranova, 2010)

Graphically, the temperature driving force is represented by the distance between the red and blue lines in Figure 8. For a large temperature difference the lines will be far apart, for a small temperature difference they will be close together. For heat to flow from the steam to the cooling water there must be a positive temperature difference, which means that the lines can never cross. In practice the lines will always maintain a minimum separation, defined as the temperature approach (or TTD, Terminal Temperature Difference). The bigger the surface area of the condenser, the smaller the temperature approach will be. For the lines to touch (zero temperature approach) the condenser would need an infinite surface area. (Gibbard & Terranova, 2010)

Applying principles outlined above, the behaviour of a condenser can be readily predicted. In particular, the response of the condenser to a change in operating conditions or fault can be anticipated and this information can be used to diagnose operating problems. Firstly, the response of the condenser to changes in operation can be predicted.

- The two operating parameters which vary according to plant operation are the heat load and the cooling water inlet temperature
- The heat load varies according to the generating output of the unit
- The cooling water inlet temperature varies according to ambient conditions

According to Equation (1), if the heat load reduces and U and A remain constant, then the temperature driving force also reduces. If the CW inlet temperature and the flow are also constant, then this duty reduction will affect the CW outlet temperature. This results in an increase in the temperature driving force. The net result is that T_{sat} must reduce to arrive at the correct temperature driving force. Therefore, a reduction in heat load will result in a reduction in T_{sat}, which is the same as saying that the turbine exhaust pressure will reduce.

Conversely, if the cooling water inlet temperature increases while holding Q, U and A constant, it is clear that T_{sat} must increase in order to keep the temperature driving force constant. Therefore, an increase in cooling water temperature will result in an increase in the turbine exhaust pressure.

The relationships described above are often incorporated into a condenser performance curve, which predicts the turbine backpressure as a function of plant load and cooling water inlet temperature.

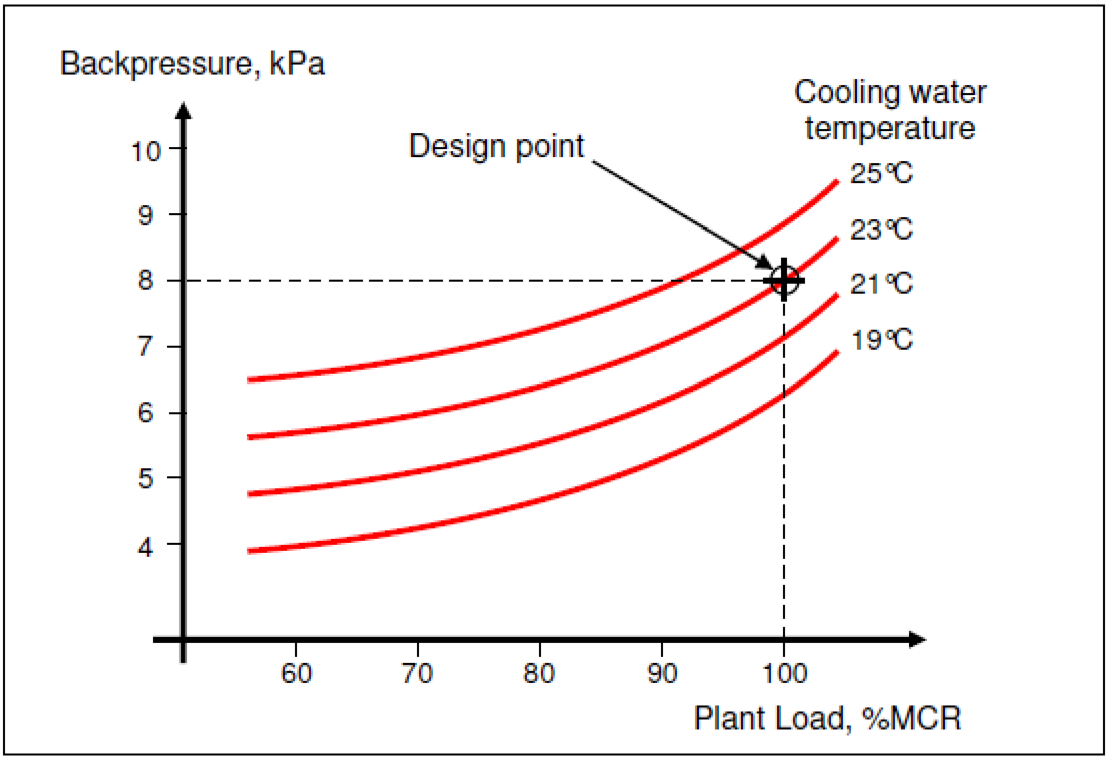


Figure 9 Representation of a typical condenser performance curve (Gibbard & Terranova, 2010)

The response of the condenser to a fault can also be predicted from an understanding of condenser behaviour. Faults will typically result in an increase in the turbine backpressure, so the presence of a fault can be detected with reference to a performance curve such as that illustrated above. The expected backpressure for the plant load and cooling water temperature can be determined, and if the actual backpressure is higher this indicates a fault. However, the performance curve cannot diagnose the type of fault present.

In order to do this we need to look at the detailed response of the condenser to the three typical faults (Gibbard & Terranova, 2010):

2.2.6.1 *Reduced Cooling Water Flow*

If the flow of cooling water is reduced at constant duty, the difference between the cooling water inlet and outlet temperatures will increase (due to the same amount of heat added to a reduced amount of water). An increase in backpressure does not in itself indicate a loss of cooling water flow, but when accompanied by an increase in the cooling water temperature rise this problem is identified.

2.2.6.2 *Fouling*

The primary effect of fouling is a reduction of the overall heat transfer coefficient value and this increases the backpressure in order to create a corresponding increase in temperature driving force. If it is assumed that the cooling water flow is constant (fouling will, in fact, result in a reduction of flow), the increase in driving force must be achieved by an increase in the approach temperature. Therefore, if an increase in backpressure is experienced at the same time as an increase in the approach temperature, this indicates fouling.

2.2.6.3 *Air Ingress*

The presence of excessive concentrations of air in the steam space will also reduce the overall heat transfer coefficient and have the same effect as fouling. However, air ingress has an additional effect which differentiates it from fouling. The effect of air in-leakage has a tendency to cause increased sub-cooling of the condensate. Therefore, if an increase in backpressure is experienced with an increase in approach temperature and significant condensate sub-cooling; this indicates air in-leakage.

Therefore it is imperative to constantly monitor the performance of a condenser in order to identify and rectify faults before they become critical.

2.2.6.4 *Backpressure Measurement*

The first question regarding the use of backpressure as a monitoring tool is whether the pressure is accurately measured. In many cases it can be readily demonstrated that the pressure measurement is significantly inaccurate and many plants operate with a consistently false pressure reading. Part of the reason for this is that it is quite difficult to measure vacuum accurately. For example, any method which measures vacuum relative to atmospheric pressure (such as a manometer) must be adjusted to reflect both the altitude of the measurement and variations in atmospheric pressure due to weather. Failure to do so will result in a false reading. There are a number of simple checks which can be used to verify the accuracy of the pressure measurement.

The first is to compare pressure measurements between condensers on the same side of the station at the same load. Two condensers receiving the same cooling water temperature and at the same load should (assuming no faults as well as similar levels of plugging) achieve the same backpressure. If the backpressure readings are significantly different this either means there is a fault on one of the condensers or that at least one pressure reading is inaccurate.

Another check is to calculate the backpressure corresponding to the measured condensate (hotwell) temperature, and compare it to the measured backpressure. Assuming that there are no significant air leaks, a large discrepancy means that either the temperature reading or the pressure reading is inaccurate. Generally speaking, temperature measurements tend to be more accurate, and the thermocouple can be readily calibrated if necessary.

A more effective way of monitoring the condenser is through temperature monitoring which is discussed in the next section.

2.2.6.5 ***Temperature monitoring***

A commonly used monitoring method is based on both pressure and temperature measurement. Four temperatures are used to create a graphical construction which allows the precise nature of any fault to be identified. The method requires a baseline measurement and full load acceptance test data is usually used for this purpose. (Gibbard & Terranova, 2010)

The four temperatures used are as follows:

- Saturation temperature (inferred from the backpressure)
- Cooling water inlet temperature (directly measured)
- Cooling water outlet temperature (directly measured)
- Condensate temperature (directly measured in hotwell)

A graph of the acceptance test values for these temperatures is constructed as shown in Figure 10. The four temperature points are plotted using an arbitrary spacing on the X-axis.

The saturation temperature can either be plotted with reference to the temperature axis or an alternative pressure scale can be used to plot the backpressure directly. Sometimes the order of the saturation temperature and hotwell temperature points is reversed. Provided that the graph is constructed the same way for both acceptance test and actual data, these variations do not matter. (Gibbard & Terranova, 2010)

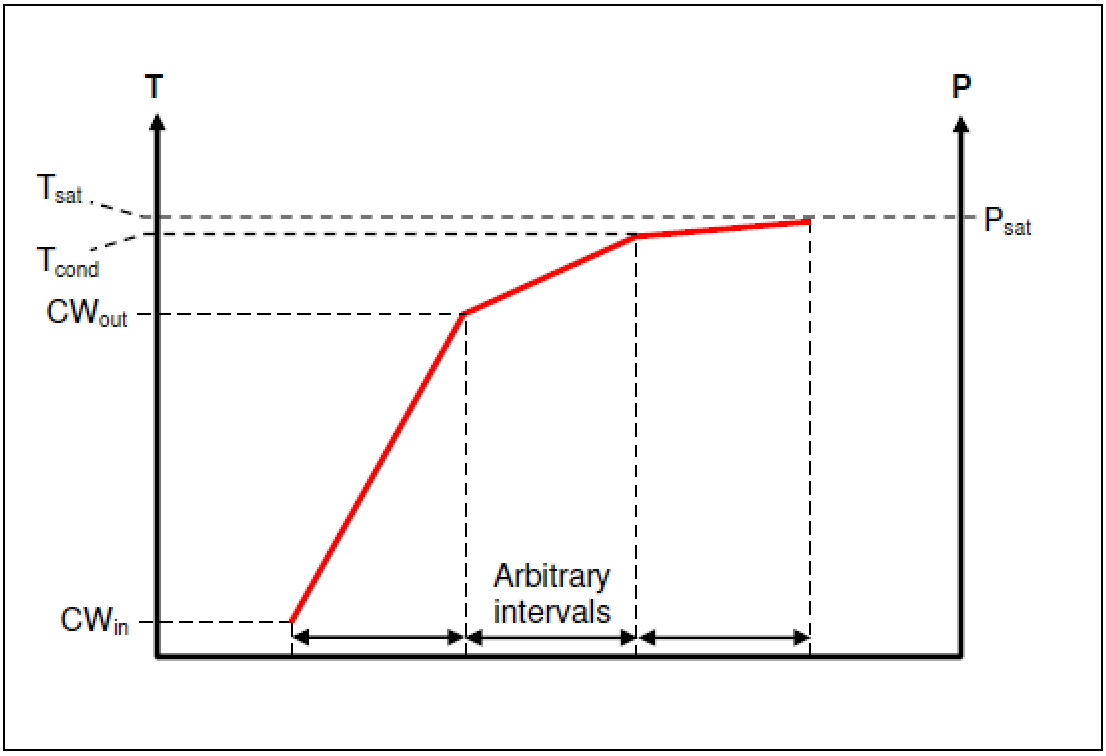


Figure 10 Baseline plot of acceptance data (Gibbard & Terranova, 2010)

Once the baseline plot shown in Figure 10 is constructed, plant measurements can be superimposed as shown in Figure 11. One advantage of the method is that it does not matter whether the cooling water inlet temperature is the same as in the acceptance test (although the plant does need to be at full load). If the condenser is working correctly the three line segments will have the same gradient in the acceptance test as in the measured data. Figure 11 shows a condenser which is working correctly. If the condenser has a fault it will be indicated as a difference in gradient between measured and acceptance test data for one or more of the line segments. The three possible faults are indicated graphically in Figures 11 - 13. Figure 12 shows the type of plot seen if there is a shortage of cooling water. The slope of the line between cooling water inlet temperature and cooling water outlet temperature has increased compared to the acceptance test data. (Gibbard & Terranova, 2010)

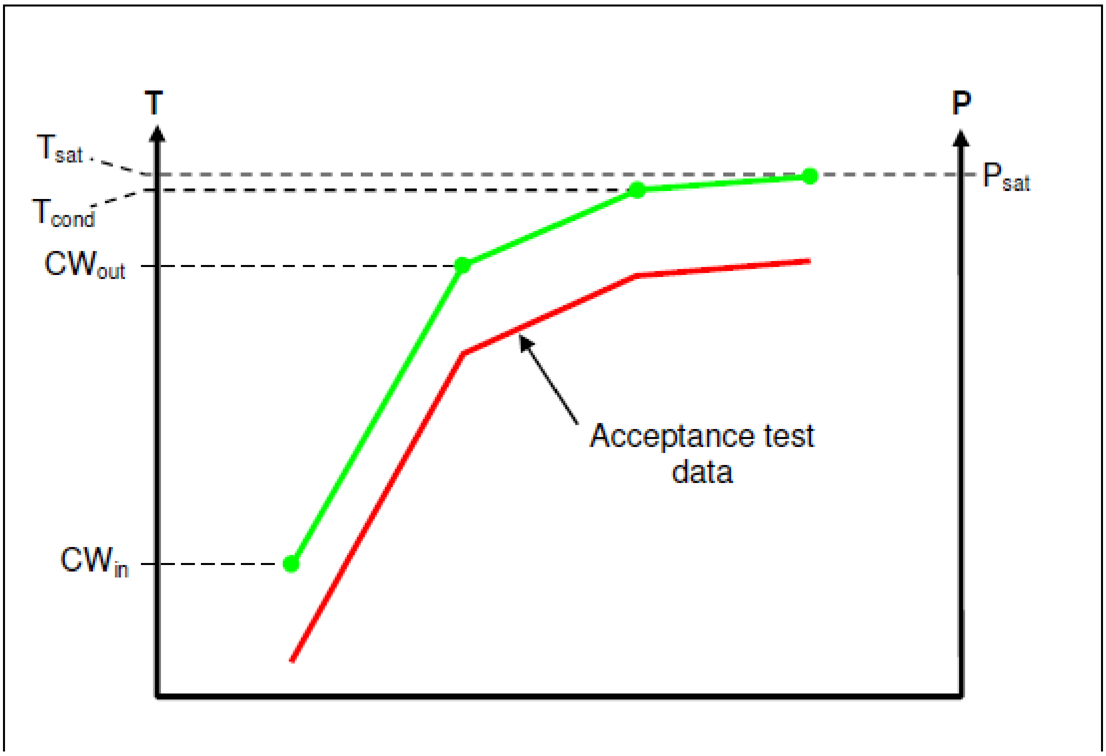


Figure 11 Performance test result of a condenser that is working properly (Gibbard & Terranova, 2010)

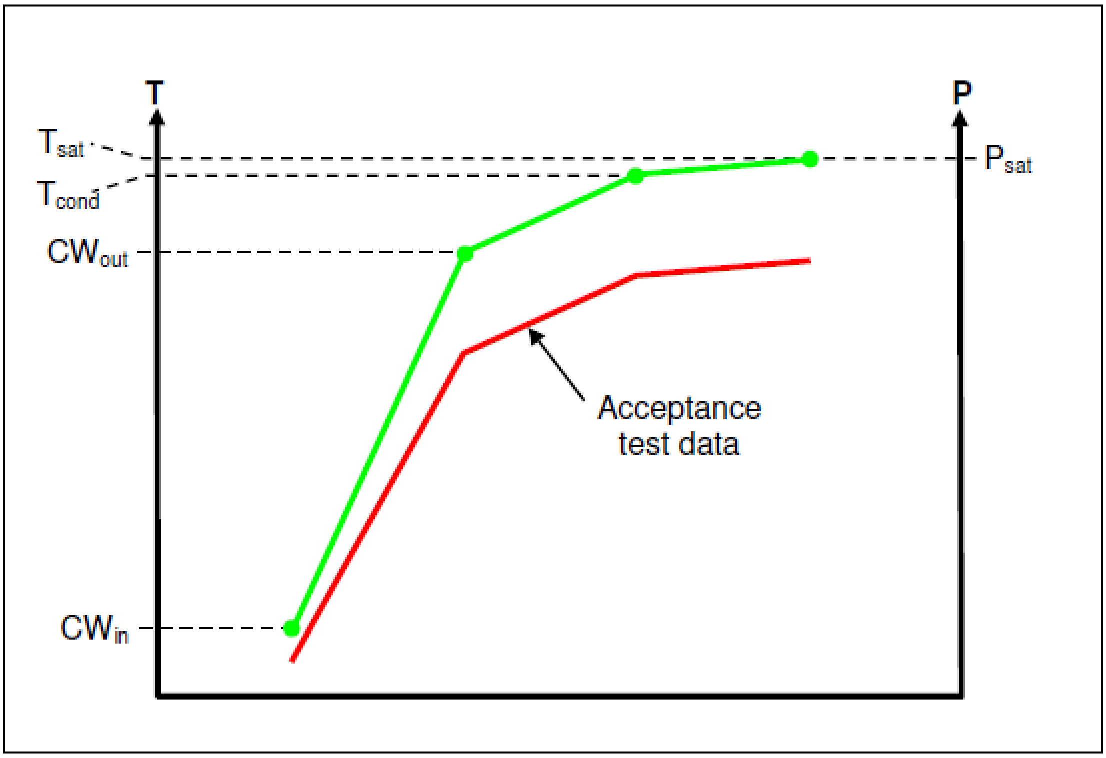


Figure 12 Performance test result of a low cooling water indication (Gibbard & Terranova, 2010)

If the slope of the line segment between cooling water outlet temperature and condensate (hotwell) temperature is increased, this indicates fouling. This type of characteristic is shown in Figure 13. Note that the test in this case was conducted in conditions where the cooling water inlet temperature was lower than in the acceptance test. (Gibbard & Terranova, 2010)

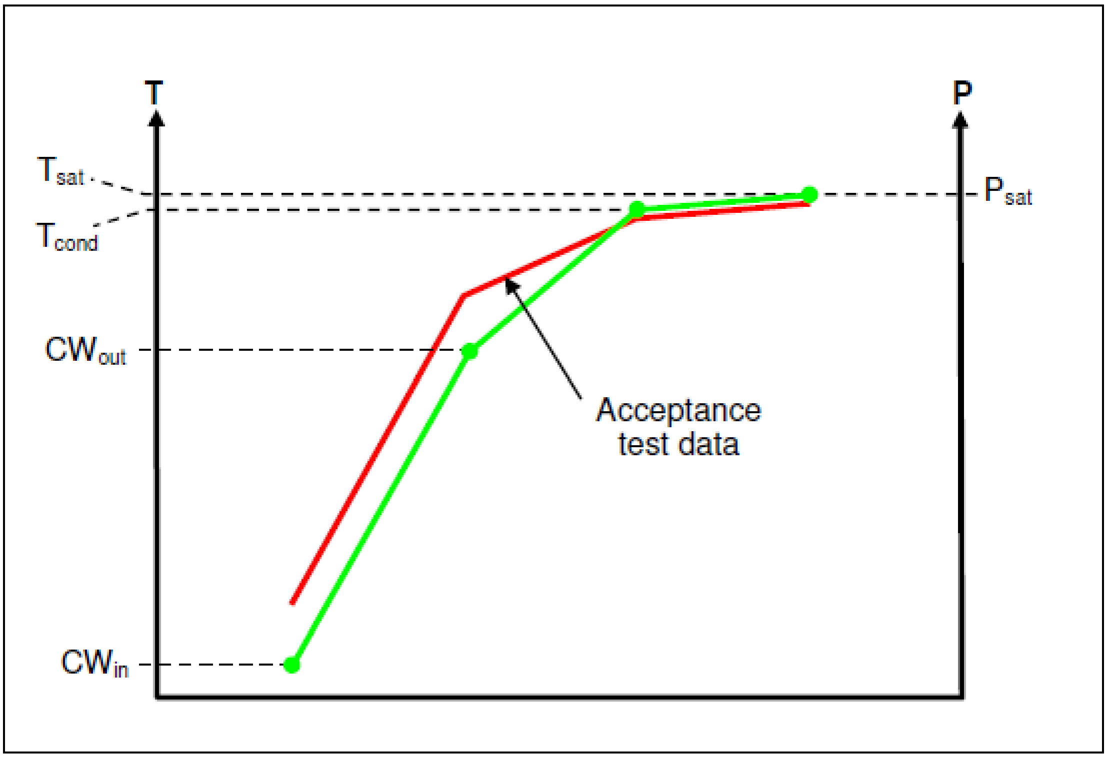


Figure 13 Performance test result of an indication of fouling

Figure 14 illustrates the type of characteristic seen if an air leak is present on the condenser. The third line segment shows an increase in gradient, indicating an increase in approach temperature and sub-cooling of the condensate. Again, this is an example of a performance test conducted at a cooling water temperature lower than that in the acceptance test. (Gibbard & Terranova, 2010)

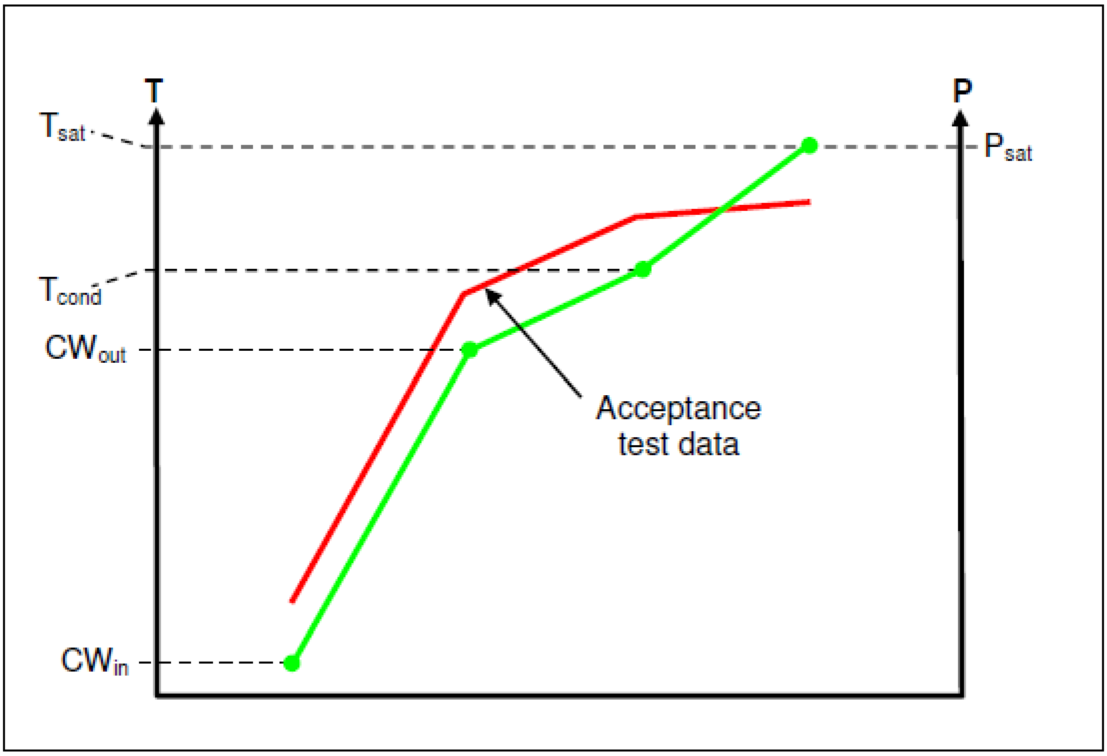


Figure 14 Performance test result of an air leak indication

One disadvantage of the temperature monitoring method is that it is dependent on the backpressure measurement to obtain T_{sat}. This means that it is subject to the difficulties in measuring the backpressure accurately. However, it is possible to detect whether an error in pressure measurement is present. If the backpressure reading is higher than the actual pressure, the results of the test will appear as Figure 14 and an air leak will be falsely indicated. The presence of a real air leak will typically be accompanied by an increase in the condensate dissolved oxygen measurement. The presence of an air leak can also be verified by vacuum decay testing. If the temperature monitoring method suggests an air leak which is not accompanied by any of the alternative indicators of air ingress, the pressure measurement should be checked. If the backpressure reading is lower than the actual pressure, the results of the test will have the appearance shown in Figure 15 below. This form of the chart shows an impossible situation where the condensate is hotter than T_{sat}. This is a clear indication that the pressure measurement is in error. (Gibbard & Terranova, 2010)

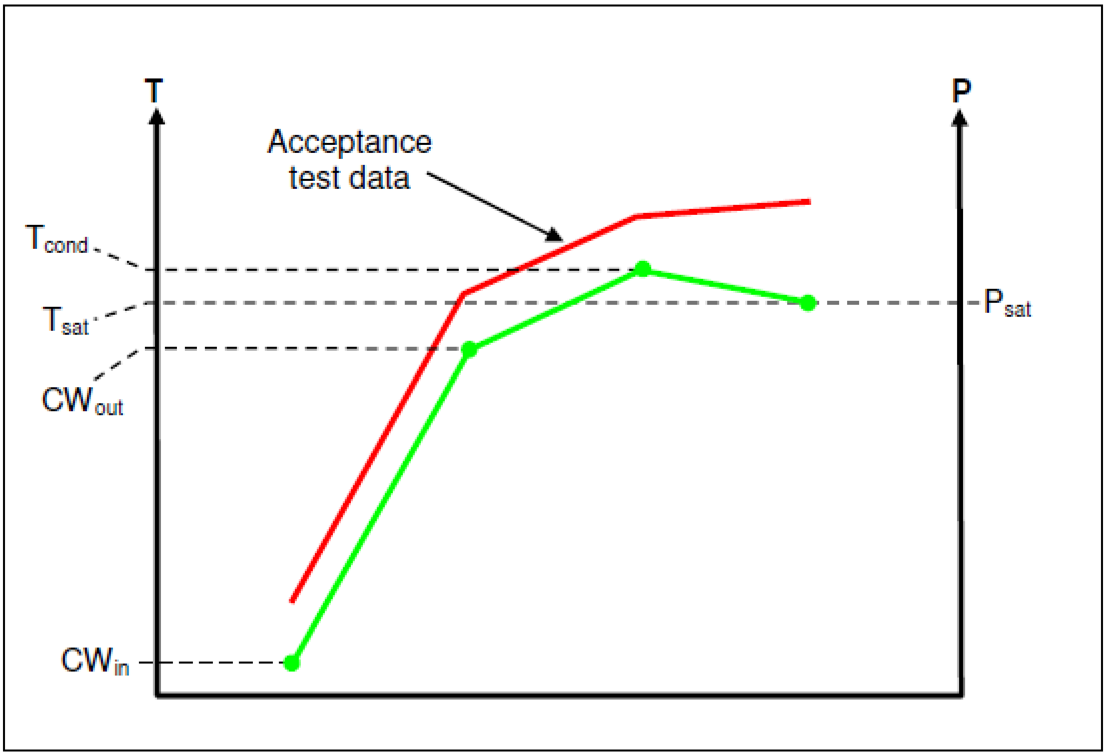


Figure 15 Performance test result which indicates a pressure measurement error (Gibbard & Terranova, 2010)

Another disadvantage of the method as presented above is that the graphical construction is designed to indicate the results of a single test and is not appropriate for the recording of trends over time. This can be easily addressed by plotting the data in a different form. Firstly, the data is converted into three temperature differences, and then these differences are plotted over time to highlight any trends. A typical trend plot is illustrated in Figure 16, showing the expected trend for a developing air leak. (Gibbard & Terranova, 2010)

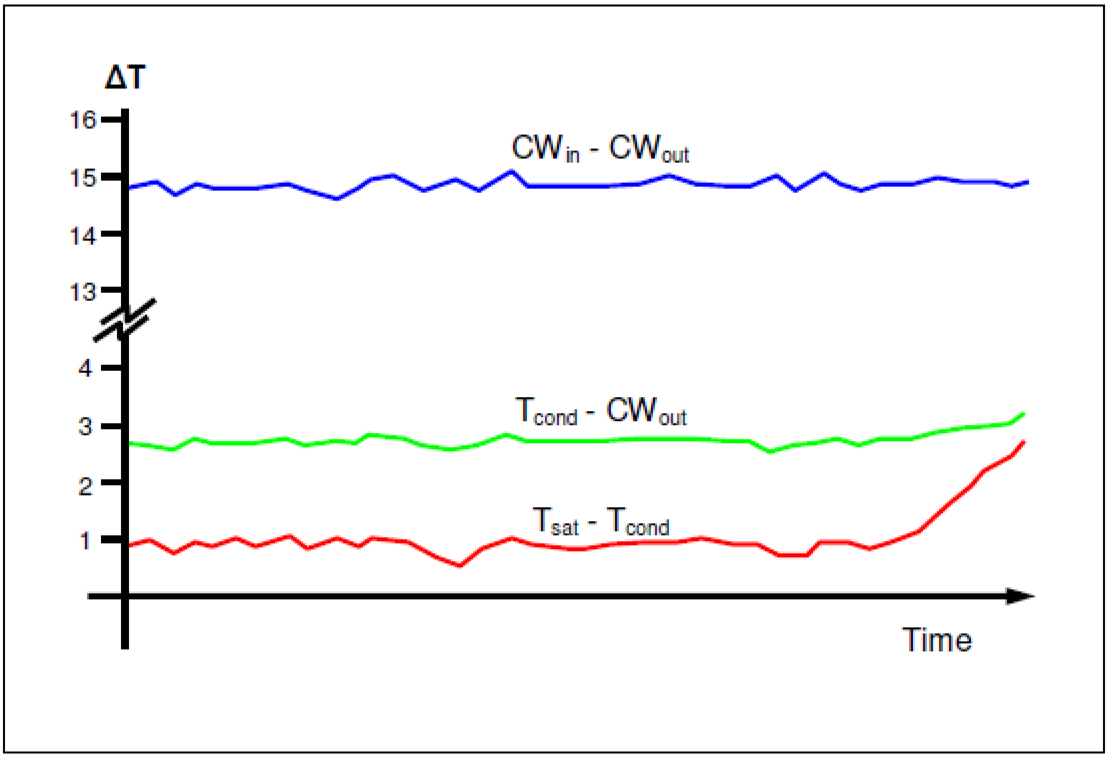


Figure 16 Temperature trend plot that indicates the development of an air leak over time (Gibbard & Terranova, 2010)

The recording of trends is recommended, as it makes it much easier to identify problems early and therefore to take appropriate remedial action. It also avoids the need to relate the temperatures back to acceptance test data, which may give misleading results (e.g. if the condenser is heavily plugged or if the condenser has been re-tubed in a different material). (Gibbard & Terranova, 2010)

2.2.6.6 *Estimation of condenser efficiency*

The efficiency of the condenser can be estimated by using the following equation:

$$\text{Condenser efficiency (\%)} = \frac{T_2 - T_1}{T_{sat} - T_1} \times 100 \quad (3)$$

This formula is derived from the LMTD formula and is only to be used as a quick and simple estimation of how efficiently the condenser is condensing the entering steam.

2.2.6.7 *Typical condenser tests that can be done (Moolman, 2014):*

2.2.6.7.1 Vacuum Decay Testing

Vacuum decay testing is a method specifically aimed at monitoring rates of air in-leakage.

The technique is simple to apply and can yield very useful information about the condition of the condenser, especially if conducted on a routine basis. The basic method is to isolate the condenser from its air extraction package (air ejector) and observe the change in backpressure over time. Isolating the vacuum extraction set allows air to accumulate in the condenser, reducing the overall heat transfer coefficient and therefore causing the backpressure to rise.

The rate of rise in backpressure (kPa/minute) indicates the magnitude of the air ingress to the system. A condenser in good condition will produce a slow rise in backpressure; a condenser with a substantial air leak will experience a rapid rise in backpressure.

2.2.6.7.2 Pressure drop test

A pressure drop test over the condenser can be done on-load in order to determine whether there is enough flow going through the condenser. For this test to be implemented; sampling points are required in the CW in and outlet ducts on the condenser. A portable manometer is attached to the sampling point and the pressure is then measured on the inlet and on the outlet CW duct of the condenser. From the difference in the pressure drop, one can derive whether there is adequate CW flow inside the condenser. A high pressure drop will most likely indicate that there could be macrofouling inside the condenser or a possible problem with the level in the waterbox.

2.2.6.7.3 CW flow test

The flowrate of the cooling water from the CW pumps to the condenser can be measured in order to determine whether there is enough flow going through the condenser and whether the pumps are performing.

2.2.6.7.4 Pressure reading test

The pressure indicator of the vacuum could be inaccurate and this transmitter reading should be verified from time to time. This can be done by inserting portable equipment in the place of the transmitter. This can however affect the backpressure since for a time period the condenser would be exposed to atmospheric pressure. A simpler method to verify the backpressure reading on the transmitter would be to compare the temperature corresponding to the backpressure to the neck temperature of the condenser. These values should be the same.

2.2.6.7.5 Helium testing

Helium tests can be done in order to determine where there is air in-leakage into the condenser. The helium is sprayed at various points outside the condenser and is used as a tracer gas while a helium sniffer is attached to the air extraction zone in order to detect any helium.

Helium is used since it is an inert gas and will not react with any of the materials on the power plant. This test must be done while the unit is on load.

Below are the areas that are recommended to be sprayed as a standard at Grootvlei Power Station:

2.2.6.7.5.1 Condenser

- Extraction pumps mechanical seals
- Condenser pressure transmitter impulse pipe work and standpipe.
- Extraction pump balance line
- Extraction pump suction valve
- Flanges on the outlet to condenser
- Atmospheric valve: drain to condenser, sight glass, valve bonnet and flanges.
- Condenser steam side inspection covers
- All flanges connecting to condenser including pipe work from HP drain vessel and condenser flash box.
- Air ejector primary drain lines to condenser (Check flanges).
- Rupture discs
- Inspection door next to rupture discs
- Vacuum breaker valve and isolating valve

2.2.6.7.5.2 Main Turbine

- LP turbine inspection doors (Usually referred to as LP Hood inspection doors)
- LP turbine rupture discs (on the condenser next to the atmospheric valve)
- Check main turbine glands 1 to 4

2.2.6.7.5.3 Heater Distillate and Steam piping

- All flanges, thermocouples, impulse lines and valves from LP heater 2 flash box to condenser
- All drains going into HP drains vessel
- Water extraction condenser shell vent valve
- LP heaters and vent lines to condenser
- Clean drains tank outlet line to condenser. (Check valve flanges and small vent valves on pipe work)
- Condenser, LP heaters local level gauges.
- LP heater bled steam pipe work.

2.2.6.7.5.4 Gland Steam

The following steps can be carried out on load to identify whether the gland steam is not sealing properly:

- The gland steam pressure can be increased one by one (up to 2 kPa), and the DO levels and condenser backpressure should be monitored closely while doing this.
- After the test is complete, the normal operating conditions should be returned to
- This test can be done on all four glands

2.2.6.7.5.5 Steam feed pump turbine (SFPT)

If SFPT is not in service, ensure that the gland steam leak off to LPH 1 and gland steam condenser is shut. The valve can be tested by spraying helium close to the BFPT glands.

2.2.6.7.5.6 Condensate system

- Flanges and valve glands on the line from the recirculation valve back to the condenser can be sprayed. It is easier to partially open the valve at a lower load (150 MW) and then to see if the DO levels/backpressure improves
- Flanges on the line to the hood spray system as this is also only pressurized up to the control valve during normal operation

2.2.6.7.6 Smoke sticks

Smoke sticks can be used to identify drafts and leaks on the plant. It is a small handheld theatrical fog machine that uses non-toxic fluid to induce the smoke.

Figure 17 A typical representation of a smoke stick (The Chimney Balloon, 2014)

2.2.6.7.7 CEP pressure test

The purpose of this test is to detect vacuum leaks on a unit while the unit is on load. The most probable area of air ingress is the CEP area between the pumps and the condenser and the line between the pumps and the suction valves can be tested by pressurizing the line on load. In order to do this test, the pump is stopped and the line is isolated by closing the suction valves as well as the valve to the balance line.

After this one can inspect for leaks. This test can only be done if only one CEP is in service as is the design at most power plants that use two CEP's.

2.2.6.7.8 Condenser flood test

In order to conduct a flood test, the unit needs to be offline. The condenser is then isolated and the steam space is filled with water. The water boxes can then be opened in order to inspect where water may be leaking out from the tubes. The area below and around the condenser must also be inspected for any water leakages. A flood tests should typically be done over a 24 hour period and be inspected at least twice during that period, once at the start of the test and once again after the 24 hours. After time has passed, the water has had time to seep into spaces where air could be entering during normal loading.

2.2.6.7.9 Air ejector flow test

A simple test can be conducted in order to determine whether the air ejectors are removing adequate air from the condenser by using an anemometer which measures windspeed.

Figure 18 Image of an anemometer (Test and Measurement Instruments C.C, 2014)

2.2.6.7.10 Cooling tower performance test

Condenser performance can be affected by cooling tower performance and therefore it is necessary to ensure optimum performance and operation of the cooling towers. Cooling towers are in general quite robust plant components performing reliable and requiring relative little attention. With time, however, deterioration in performance will occur. The most probable problem will be flow restrictions but the cooling process itself can also be impaired.

The measured parameters are:

- The cooling tower inlet and outlet temperatures
- The ambient air temperature
- The water flowrate
- The cooling water pump pressures and amperes
- The wind speed through the tower (For dry cooling towers)

A performance curve (attached in the Appendix) is used to determine the performance of the tower using the above parameters.

On the performance graph attached in the appendix (Figure 50), the ambient conditions (temperature and pressure) along with the temperature rise is used in order to determine the target cooling tower inlet temperature. However, this value is calculated for if the water flowrate is at 100%. If this is not the case, a correction factor needs to be incorporated.

The right hand side of the cooling tower performance graph is used to obtain this correction factor by first calculating the percentage water flow that is being experienced. Using this data and the temperature rise, the correction factor can be calculated. This correction factor should then be applied to the target cooling tower inlet temperature to obtain a more accurate target temperature.

2.2.6.7.10.1 CW Screens

An operational requirement is for the CW screens to be washed out and checked for damage regularly. The function of these is to stop dirt and debris from going through to the condensers without restricting flow in any way.

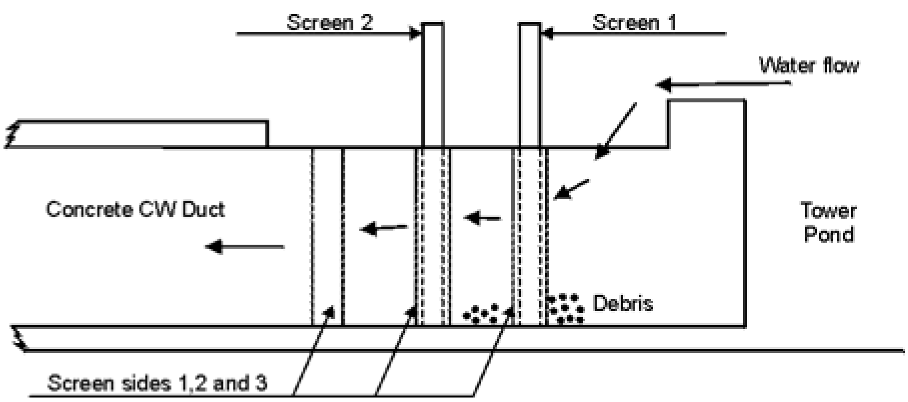


Figure 19 Typical layout of a decent screen replacement system (Moolman , 1999)

The screens should be washed off with a high-pressure water system to also clear it of all the algae that collects on it as the algae will also restrict flow to the condenser. There can also be algae in the condenser that will also cause flow restrictions in the tubes.

2.2.6.7.10.2 Outage inspections and actions:

- Spray nozzles:

Inspect during outage to replace / repair missing or blocked nozzles.

- End flaps:

Flow through the nozzles can be restricted due to water escaping through open end flaps. Some of these end flaps are occasionally opened in winter to reduce cooling thus preventing freezing in sections of the tower.

- Polygrids, or baffles below the sprays in the towers:

Must be in good condition and the structure intact to ensure ample surface area of water exposed to air for cooling to take place.

- Drift eliminators:

The baffles above the spray nozzles have the function of reducing the drift of droplets with the draft of air, which represents a water loss not contributing to the cooling process.

- Pipework and canals towards the outer perimeter of the tower:

This should be cleared of mud periodically since mud will restrict flow and affect cooling tower performance.

- CW System:

Condenser isolating valves checked to open fully and condenser water boxes cleared of all debris.

2.3 **Air ejectors**

Grootvlei Power Station uses both steam jet air ejectors and water jet air ejectors.

2.3.1 **Steam jet air ejectors**

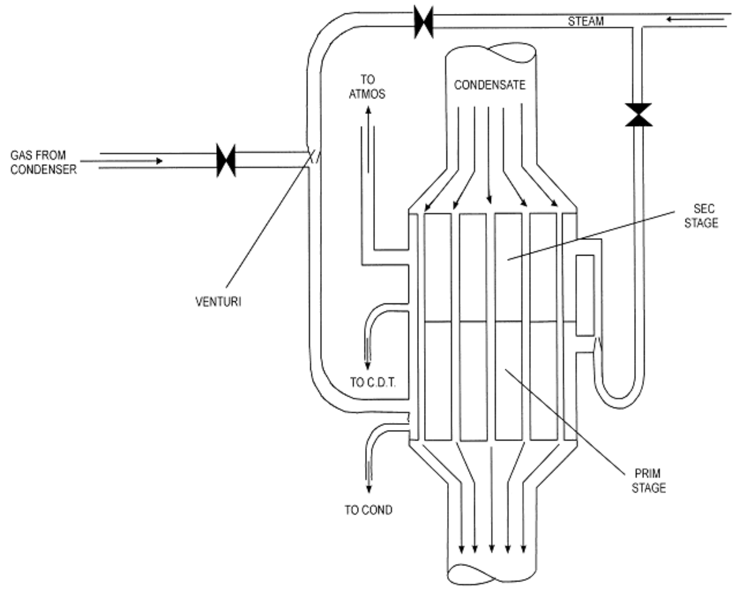


Figure 20 Schematic of a steam jet air ejector

Units 1 – 4 operate with steam jet air ejectors, and each unit has two of the two stage type of steam air ejectors installed for operational air extraction purposes. Figure 20 displays a schematic of such an ejector. Each steam jet air ejector is equipped with inter and after-condenser to which the first and second ejector stages are connected. The motive steam for both ejector stages is taken from the auxiliary steam system and turbine condensate is used as a cooling medium for the ejector condensers.

Steam is passed through a venturi and the pressure drop creates suction and therefore extracts any air that is present in the condenser. The ejectors condense the steam in each stage, however the first stage's steam is condensed and returned to the condenser whilst the second stage of the ejector condenses the steam and then delivers it to the clean drains tank. From the first stage, a mixture of steam and air passes through a second venturi into the second stage and from there the air is vented to atmosphere.

2.3.2 Water jet air ejectors

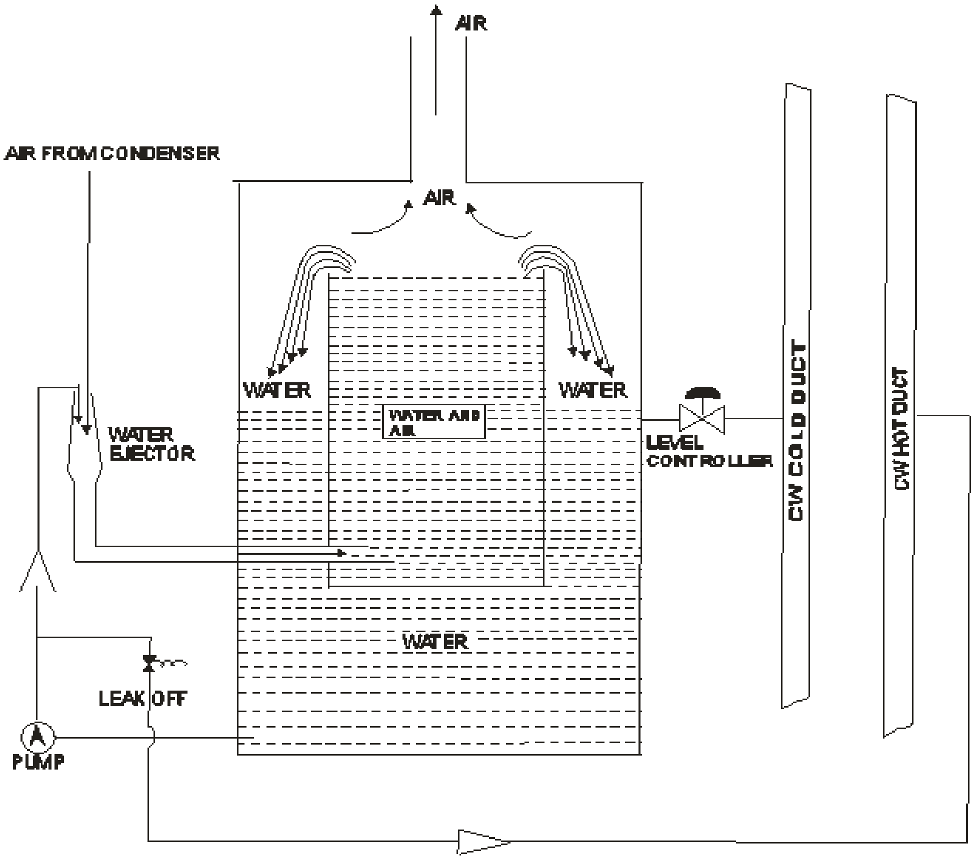


Figure 21 Schematic of a water jet air ejector

Units 5 and 6 have water ejectors installed. Each ejector system consists of three water jet air ejectors with one water jet pump each and a common air separator.

The water jet pumps deliver the motive water from the air separator with the requisite positive pressure to the water jet ejectors. These ejectors compress the air drawn from the spray condenser to atmospheric pressure. The motive water and the motive water/air mixture, led to the air separator, condense the steam mixed with the air. The air separator is designed as a tangential separator.

2.4 The Station Thermal Efficiency Performance tool

The Station Thermal Efficiency Performance program is a tool that is used in order to determine and monitor the thermal performance of the generating plant. The basic model used in this document is an Eskom based coal-fired power station, however much of the relevant material can be applied to other coal stations throughout the world. The core of good thermal plant performance is heat-rate control. If a station is operated with heat rates that are close to design, then the thermal performance of the generating plant will be satisfactory. Good thermal plant performance has both direct and indirect implications for Eskom. The direct implications include a potential saving in direct operating costs (mainly fuel), and the indirect implications include improved plant availability and reliability, as there is conclusive proof that poor thermal performance increases the incidence of plant outages (both planned and forced). Another derivative of good thermal plant performance is decreased emission of air pollutants from the stacks. (Moolman , 1990)

2.4.1 **Efficiencies**

Various efficiencies at station level and at individual component and unit level are calculated in S.T.E.P., to determine the performance of the generating plant.

An overview of some of the higher level power station efficiencies are shown in the following flow chart:

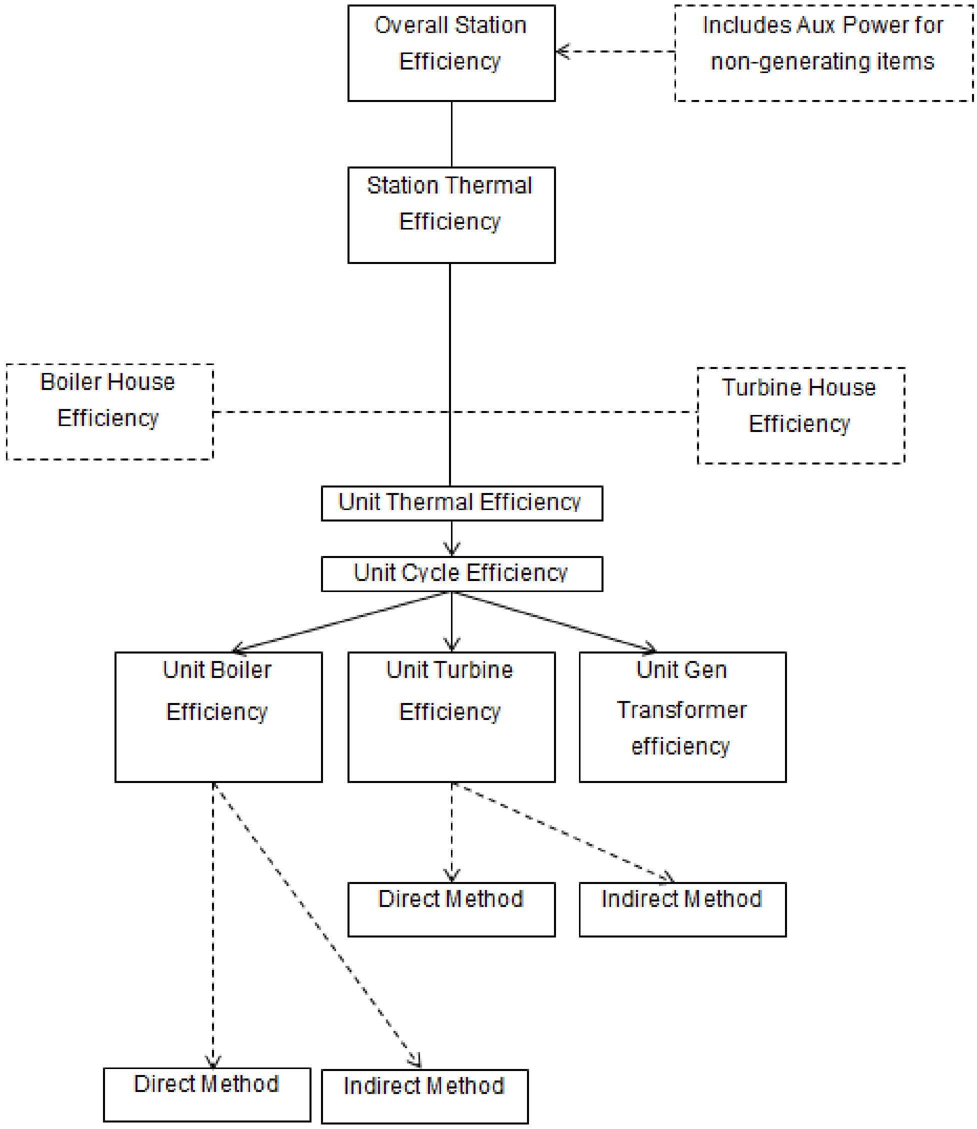


Figure 22 Power station plant efficiencies

For individual boiler and turbine efficiency determination, both the accepted direct and indirect methods can be used. Included in the turbine efficiency determination is the efficiency of the generator as well. Both the mechanical and electrical efficiency are included here.

2.4.1.1 ***Overall Station Efficiency***

The overall station efficiency is the overall thermal efficiency of the generating plant, but includes the auxiliary energy used for non-generating purposes, i.e., the works power not generating and the extraneous supply energy.

Because of the additional auxiliary power energy that is taken into account here, this overall station efficiency is slightly lower than the station average thermal efficiency, which only takes into account the auxiliary power of the generating plant.

2.4.1.2 ***Station Average Thermal Efficiency***

This efficiency differs from the overall station efficiency in that the performance of the generating plant only is considered. The total auxiliary power used here to determine the sent out power is the auxiliary power of the generating plant only, and does not include extraneous supply or works power not generating energy. This plant thermal efficiency therefore measures the effectiveness with which the energy conversion process in a power station takes place.

2.4.1.3 ***Station Average Boiler House Efficiency***

The station average boiler house efficiency is a summation of the individual boiler efficiencies using the indirect boiler efficiency method of calculation, derived from the total boiler losses.

2.4.1.4 ***Station Average Turbine House Efficiency***

The station average turbine house efficiency is determined from the individual turbine efficiencies using the target turbine heat rate, corrected for the total turbine STEP losses, and the power factor, and backpressure deviation from design.

2.4.1.5 ***Individual Unit Thermal Efficiency***

The thermal efficiency per unit is calculated on exactly the same basis as when determining the station average thermal efficiency. The only difference is individual unit energy sent out, together with coal burnt plus oil burnt per unit used.

Only the direct auxiliaries per unit are used to determine the sent out energy per unit in STEP. The coal burnt per boiler can be statistically determined from the station coal burnt.

The oil burnt per boiler can be directly measured as an input, and the average CV of the coal burnt per boiler is normally taken as the station average CV, when separate analysis per boiler is not carried out.

2.4.1.6 ***Individual Unit Cycle Efficiency***

The individual unit thermal efficiency as calculated under sub section 2.4.1.5 has to tie up with the individual unit cycle efficiency as calculated here, provided that USO is used in the determination of the turbine heat rate.

2.5 STEP loss calculations breakdown

The following is a summary of all the generating plant losses calculated in STEP. It is imperative that these losses are derived from the comparison of the actual results to the calculated targets.

2.5.1 Boiler losses

2.5.1.1 *Carbon in refuse loss*

The carbon in the refuse refers to the unburnt carbon in the fly ash as well as the bottom ash. This causes a loss of carbon since it is disposed of with the ash. If the value of unburnt carbon in the ash is high, the firing control is insufficient and this would be the cause of significant losses. This means that more coal will be needed per kilogram of steam produced. (Kaupp, 2012)

2.5.1.2 *Dry flue gas loss*

The dry flue gas loss accounts for the heat that is lost in the dry products of combustion sent to the stack i.e. carbon dioxide, oxygen, nitrogen, carbon monoxide and sulphur dioxide. The dry products carry away sensible heat whilst the wet products of combustion (mainly the moisture released from the combustion of hydrogen) carry away both sensible and latent heat. This loss is reduced by the economizer which heats the demineralised water that is to be sent to the boiler and also by the air heater which heats the air that is sent to the boiler. This way, the heat is transferred to the water and the air instead of to the atmosphere and less heat is used in the boiler to heat these components. Flue gas loss can also occur due to an over-feed of air, which causes the boiler to heat unnecessary air that is eventually lost to the atmosphere. Great care should be taken to ensure this doesn’t happen. (Anon, 2005)

2.5.1.3 *H_2 and fuel moisture loss*

This refers to losses due to the moisture from the combustion of hydrogen. The hydrogen component leaves the boiler as water vapour and carries with it sensible as well as latent heat. Certain fuels (like coal) also contain water and this water also carries latent heat in the flue gas. This can cause a significant efficiency loss if it is too high. Reducing the flue gas temperature can have an effect on this loss, but only a small effect unless a condensing heat exchanger is employed. (Anon, 2005)

2.5.1.4 *Radiation loss*

The boiler room “walls” are actually made up of tubes and superheated tubes. Radiation losses are caused by the difference in the temperature of the ambient air that surrounds the tubes in the boiler and the outer surface temperature of the boiler tubes. This difference in temperature causes radiation of heat from the tubes to the air and the heat from the tubes is lost to the atmosphere. Losses due to radiation are caused by poor insulation of the tube walls. Mineral wool or fibre glass is typically used to insulate these tube walls. Since one cannot insulate the boiler tubes completely, radiation will occur but it is imperative to ensure proper insulation to avoid great losses.

2.5.1.5 *Mill reject loss*

Heavy items in the coal; such as the tramp iron, iron pyrites and other unground items that are too heavy to be lifted up by the primary air velocity; drop out through the relief gate fitted within the throat plate of the mill. This is called the mill rejects and they then fall into a hot gas pass where brush ploughs sweep the unground material into the inner reject box. From here the material slides to the outer reject box. The mill reject loss depends on the amount of unground material as well as the mill operating condition. These rejects have a heat value, and this heat loss through the mill rejects is the accounted mill rejects loss. (Sathyanathan, 2010)

2.5.2 **Turbo-generator losses**

2.5.2.1 *Main steam temperature loss*

The main steam is the steam that leaves the boiler to go to the high pressure turbine. This steam travels through isolated but non-adiabatic pipes, couplings and valves. Radiation and convection heat losses occur through these components and this causes a main steam temperature loss.

2.5.2.2 *Condenser backpressure loss*

High condenser backpressure is directly related to the power output of the turbines and is a great cause of lost efficiency. There are many causes for the decrease in vacuum conditions of which the most important are (Intek, 2011):

- **air-in leakage**

  Since the condenser operates at near vacuum conditions, the ambient pressure is much higher so air will tend to leak inwards which decreases the vacuum conditions. These leaks can happen easily, especially on the turbine shaft and if the sealing water in the condenser is lacking. This causes excess backpressure, dissolved oxygen, corrosion as well as a low cleanliness factor.

- **Tube fouling**

Each condenser has 19 800 tubes of approximately 20mm in diameter. Tube fouling will occur when biological growth and/or material deposits obstruct the cooling circulating water flow through the condenser tubes. One can discover this by measuring the flow rates inside the tubes.

- **Reduced pump capacity**

  It is important to maintain exhaust pump capacity in order to ensure proper air removal from the condenser. Inadequate air removal can cause increased backpressure, a high level of dissolved oxygen as well as a low cleanliness factor.

- Dry cooling tower systems tend to deliver poorer vacuum conditions than wet cooling tower systems. This is because the water does not cool down quick enough and this will result in a reduction of station thermal efficiency and it is usually shown by the deterioration in the turbine vacuum.(Chizengeni, 1980)

2.5.2.3 ***Final feed water temperature loss***

Feed water to the boiler should ideally be at 218°C for Grootvlei Power Station so that not too much energy is spent on heating the water. If this water is less than 218°C, a final feed water temperature loss occurs. This water is usually heated through the economizer to 218°C, but can be lost through heat loss to the pipes leading to the boiler.

2.5.2.4 ***Turbine deterioration loss***

Superheated steam expands on the turbine blades which causes the turbine to turn. Deterioration of the blades could decrease this expansion which in turn causes the turbine to be less efficient.

2.5.3 **Station losses**

2.5.3.1 ***Work's power loss***

These are losses due to the electricity used to power the station. There are two types of work's losses i.e. unit auxiliary power and station auxiliary power. The unit auxiliary power consists mainly of the power needed for all the pumps. However, any auxiliary equipment necessary to power a unit falls under unit auxiliary power. All equipment outside of the plant fall under station auxiliary power such as the offices and the ash pumps.

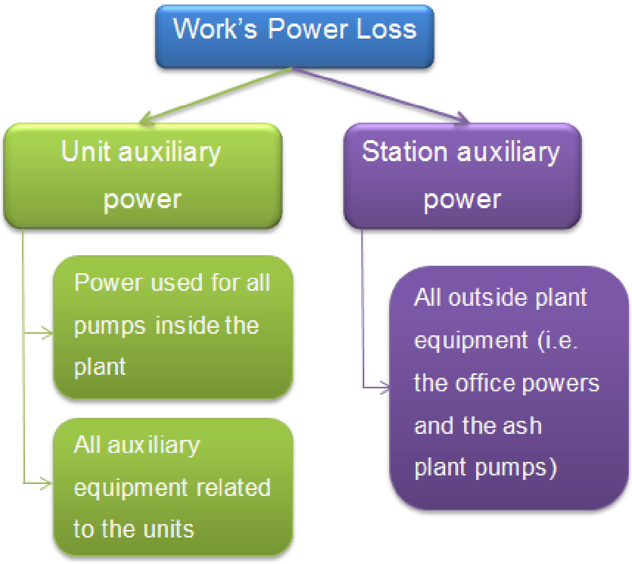


Figure 23 Works power loss breakdown

2.5.3.2 *Make-up water loss*

Make-up water is demineralised water used to supplement the water on the plant after some water is lost. Lost water can be caused by leakage or evaporation. Since water is bound to be lost, it must be continuously fed from elsewhere on the plant. At Grootvlei power station it is fed after the condenser. The make-up water that is fed comes from the water purification plant and is not at 218°C and must then be heated. This causes a loss called make-up water loss. This water contains a lot of dissolved oxygen and must first be sent through the de-aerator to remove the oxygen and the oxygen is then vented to the atmosphere.

2.5.3.3 *Unaccounted loss*

These are the losses that are not accounted for. Ideally this number should be zero; however this is not always possible since losses do tend to occur that one cannot account for.

2.6 Turbine Plant Losses

The main turbine plant losses accounted for in STEP are:

i. Condenser Backpressure
ii. Final Feed water Temperature
iii. Turbine Deterioration
iv. Main Steam Temperature
v. Reheat Steam Temperature

2.6.1 **Condenser Back Pressure Loss Theory**

The condenser backpressure loss is the heat equivalent loss of the steam entering the condenser at the turbine LP exhaust pressure and temperature conditions.

The higher the exhaust pressure of the steam entering the condenser, the greater the heat loss will be in this exhaust steam. In a condenser the exhaust steam changes state, i.e. from a gas (steam) to a liquid (condensate). In this way only latent heat is given up by the steam, and it must thus occur at constant temperature.

The higher the exhaust steam pressure entering the condenser the lower the condenser vacuum will be and hence the turbine efficiency will be lower, i.e. turbine heat rate increases considerably with a higher condenser backpressure.

This can be compared to a partially blocked motor vehicle outlet exhaust, which has a detrimental effect on engine performance, i.e. the engine is exhausting against a total or partial restriction. It is therefore extremely important for turbines to exhaust into reasonably high vacuum; otherwise turbine performance is adversely affected.

To illustrate the important contribution made to the work done by operating at a vacuum, consider

Figure 24. Steam is admitted to a turbine at a pressure of 11 bar absolute as shown by P_1. The volume of the steam is 0.177 m^3/kg. If, after expansion in the turbine, it is ejected at a pressure P_2 of 1 bar absolute, the volume will have become 1.7 m^3/kg and the work done will be represented by the area under the curve between the limits shown by P_1 and P_2.

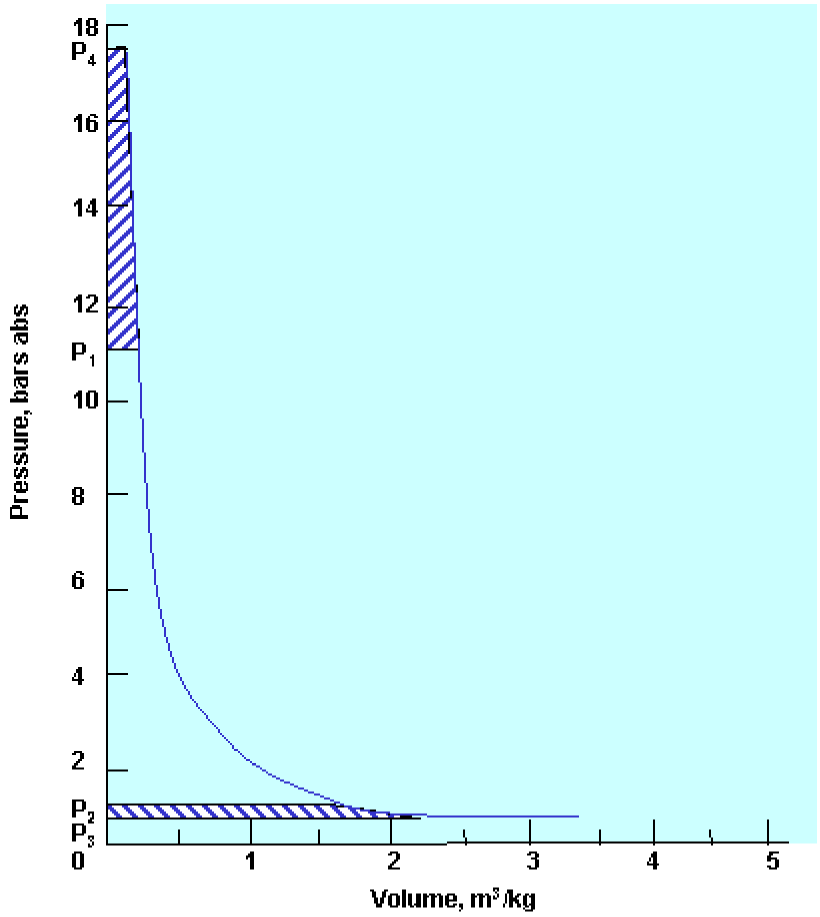


Figure 24 Graph indicating the relation of pressure to volume when considering steam

If, now, the final pressure is reduced to 0.5 bar absolute the expansion will continue to P_3 and the volume will be 3.3 m^3/kg. Thus, the extra work obtained per kilogram of steam is represented by the shaded area. This is a considerable amount of extra work, obtained by improving the back pressure by 0.5 bar. To achieve a comparable amount of extra work at the inlet to the turbine, the steam pressure would have to be lifted from P_1 to P_4, i.e. from 11 to 17.5 bars as shown by the cross-hatched area.

In practice it is, however, normal to operate with considerably lower back pressures than depicted in

Figure 24. Small changes in back pressure can cause considerable changes in the work done per kilogram of steam and there are over a million kilograms of steam entering the condenser per hour on large units. Therefore, it is important to the efficient operation of a unit that its back pressure is always maintained at the optimum level.

2.6.2 **Effect of Varying the Back Pressure**

From what has already been said it follows that a large amount of extra work is done by the steam when the back pressure is reduced. However, as the back pressure improves certain losses also increase. These losses are mainly:

- Increased CW pumping power
- Increased Leaving loss
- Reduced condensate temperature
- Increased Wetness of the steam

2.6.2.1 ***Increased CW Pumping Power***

Assuming that the CW inlet temperature is low enough the back pressure can be reduced by putting more and more CW through the condenser tubes. However, this will require more and more CW pumping power and the gain from improved back pressure must be offset against the extra power absorbed by the pumps. Therefore, CW pumps should be run only when the cost of running them is less than the resulting benefit from increased unit output. In other words, the pump operation should always be optimized.

2.6.2.2 ***Increased Leaving Loss***

Consider the last row of blades in a turbine. These present to the steam a fixed annulus through which it must pass to get to the condenser. The steam leaves the last row at a velocity which depends upon the conditions prevailing at that point. As this velocity is not utilized usefully it represents a loss of possible work. This is known as the "Leaving Loss". There is always a leaving loss but as the back pressure is reduced its magnitude increases rapidly. For example, if the back pressure is 60 mbar the loss would be a certain value. If the back pressure is reduced to 30 mbar the specific volume of the steam will be approximately doubled, and so the velocity of the steam through the fixed annulus must also double. But the leaving loss varies as the square of the velocity, and consequently will increase four times.

2.6.2.3 ***Reduced Condensate Temperature***

If the condensate in the condenser is at the saturation temperature corresponding to the back pressure, it will be 36^{o}C at 60 mbar. Reducing the back pressure to 30 mbar will cause the temperature to drop to 24^{o}C. Hence, when it enters first LP heater it will be cooler than before. Consequently more steam will automatically be bled to the heater because of the increased condensation rate of the steam. It follows that the extra steam being bled to the heater is no longer available to do work in the turbine downstream of the tapping point, and so the turbine will be deprived of some work.

2.6.2.4 ***Increased wetness of steam***

The lower the back pressure the greater the wetness of the steam. The extra moisture could result in damage to the moving blades. In addition the volume of steam is reduced. Thus at 30 mbar back pressure the volume of the steam without wetness would be 45.7m^3/kg. If there were 10% wetness the steam volume per kg would be reduced to 41.1 m^3. As a rough guide it can be assumed that every 1% wetness will reduce the efficiency of the associated stage by 1%.

The losses mentioned will eventually significantly affect the result. Continued reduction of the back pressure will result in the net improvement in heat consumption becoming progressively less until a point is reached at which the benefit due to improved back pressure is exactly neutralized by the losses, and this is the point of minimum heat consumption, as shown in Figure 25. Further reduction of the back pressure will cause the heat consumption to increase, and so there is no point in operating at a lower value. It should be noted, though, that the back pressure for minimum heat consumption varies with load and so the operations staff should be supplied with a curve such as shown in Figure 26 to enable them to determine the minimum back pressure for any loading for their particular machines.

Therefore it is imperative to operate the plant at the optimum back pressure.

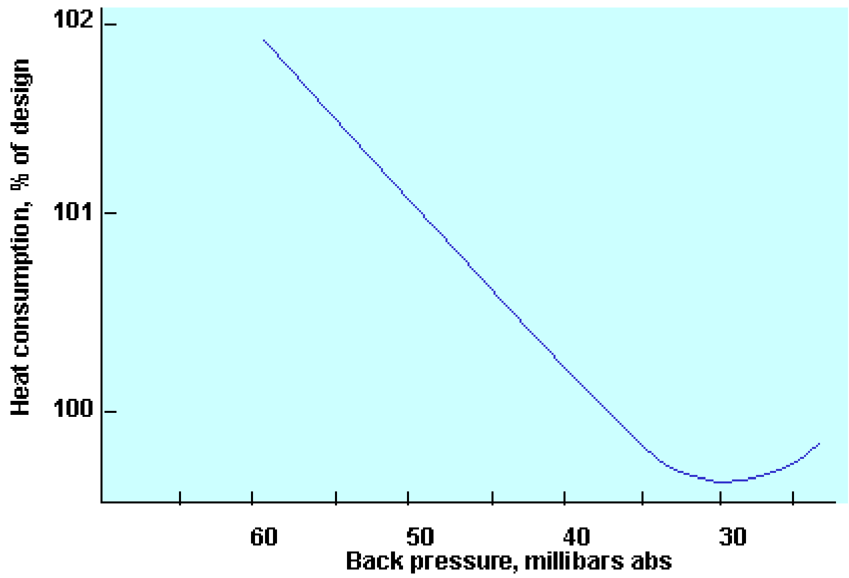


Figure 25 Backpressure correction curve

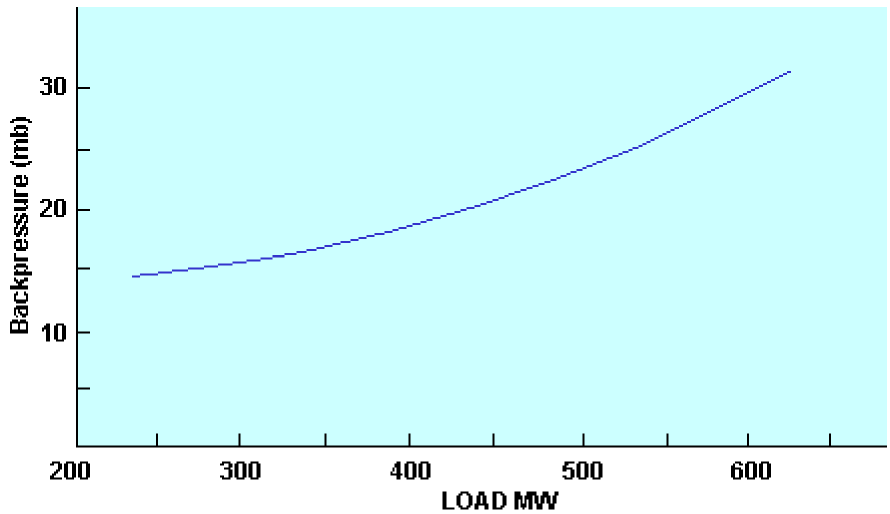


Figure 26 Graph displaying the minimum backpressure for various loads

The backpressure loss is mainly influenced and affected by one or more of the following:

a. CW inlet temperature different from design.
b. Ambient air temperature, wind speed and direction different from design.
c. CW quantity flowing through condenser incorrect.
d. Fouled tube end plates.
e. Dirty or leaking tubes
f. Air leakage into condenser
g. Turbine load

2.6.2.5 ***CW Inlet Temperature Different From Design***

All turbines are designed to perform at certain levels of performance if certain input parameters are met. One of these parameters as far as the condenser, is concerned is the cooling water (CW) inlet temperature (for wet cooled units).

A unit is guaranteed to produce a certain energy output at a certain heat rate, which is based on a certain condenser backpressure.

The condenser backpressure amongst other things depends on the temperature of the CW entering it. If the actual CW temperature entering the condenser is higher than design, condenser backpressure will increase and condenser vacuum will decrease, which will increase the heat rate of the unit, and hence lower the unit efficiency.

The reason for the higher condenser backpressure when the CW inlet temperature increases, is that the thermal conductance of the condenser decreases with an increase in CW temperature.

Thermal conductance is defined as the amount of heat (in this case latent heat) which the cooling water removes from the exhaust steam entering the condenser. The more effective the transfer of heat from the exhaust steam to the CW, the higher the condenser vacuum will be, and hence the higher the overall units' performance will be.

The effect of a CW temperature change on condenser performance is as given below Figure 27.

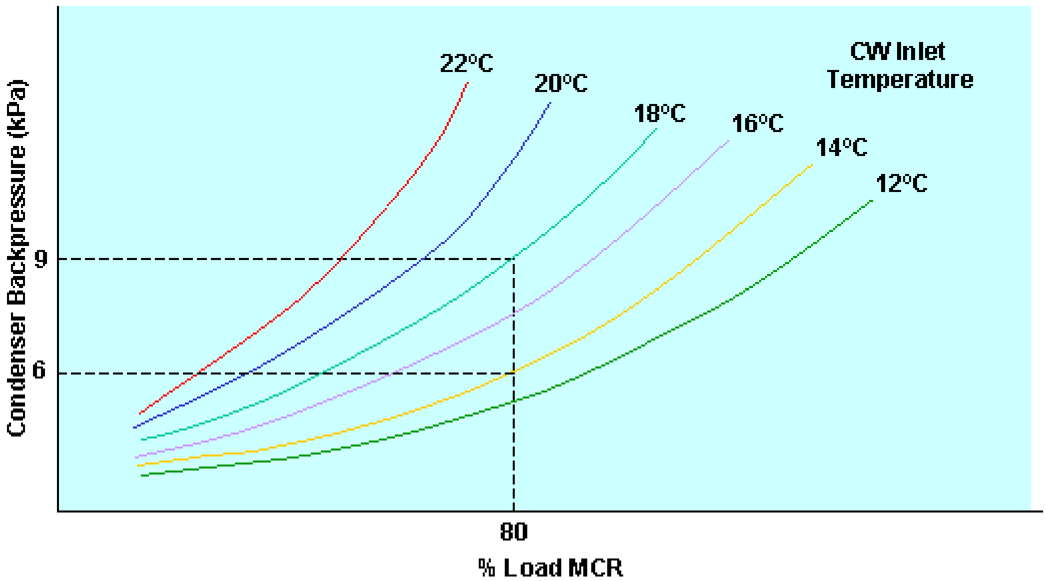


Figure 27 Target Condenser B/P v % Load (Wet Cooled Units)

Assuming a unit load of 80 % of MCR the condenser backpressure is 6 kPa with a CW inlet temperature of 14°C. With the same unit load, and assuming all other unit conditions remain the same, if the CW inlet temperature increases to 18°C, the condenser backpressure will increase to 9 kPa. From this, it can be seen that an increase in CW inlet temperature has an adverse effect on condenser performance, and hence overall unit performance.

2.6.2.6 ***Ambient Air Temperature, Wind Speed and Direction Different From Design***

These factors affect mainly the dry cooled stations where units are very susceptible to ambient air conditions. As with the wet cooled units where CW inlet temperature affects condenser performance, so with the dry cooled units ambient air temperature affects unit performance (See Figure 28 below).

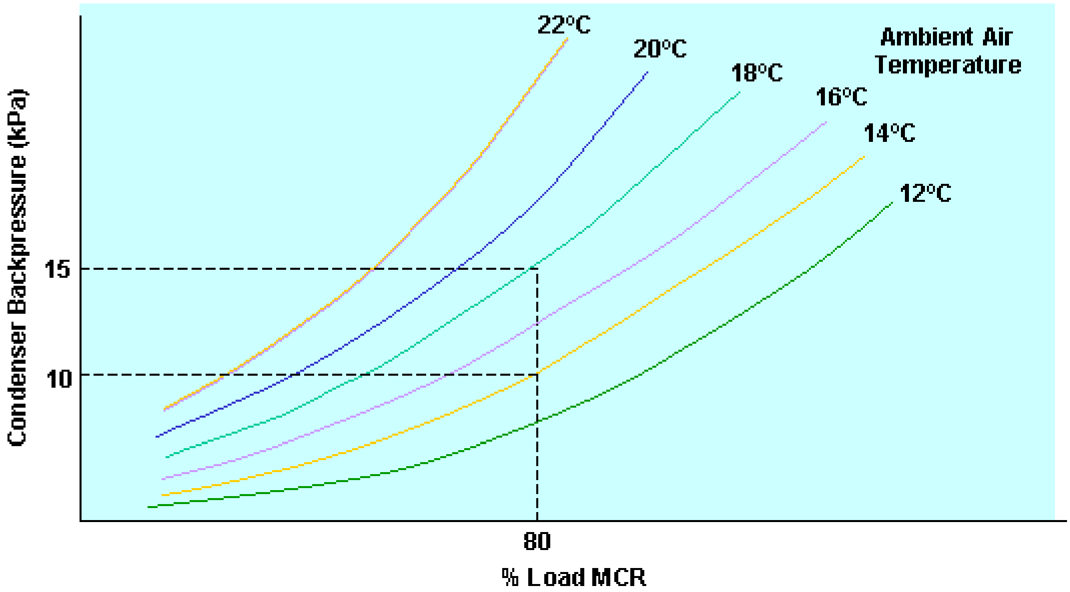


Figure 28 Condenser B/P v % Load (Dry Cooled Units)

If the ambient air temperature increases from 14 to 18°C, at a unit load of 80 %, and with all other conditions remaining the same, condenser backpressure increases from 10 to 15 kPa. From this it can be seen that any deviation of the ambient air temperature from design affects the condenser backpressure and hence overall unit efficiency.

Wind speed and direction also affects condenser performance. Both can cause inversion in and around the cooling towers. This is especially so with wet cooled towers and natural (indirect) dry cooled towers (such as Kendal), where wind speed can cause a substantial drop in the natural air flow through the towers. This occurs when the wind speed and direction affects the natural temperature gradient inside the tower, which gradient has cool air entering the tower base, which air increases in temperature as it moves up through the tower, ie the temperature difference between inlet and outlet of tower causes the natural draught flow through the tower.

If this temperature gradient is affected in any way, i.e. either through wind speed or direction which increases the air temperature at the base of the tower, a point can be reached where some of the air flow through and around the tower reverses direction and flows downwards. This affects the cooling effect of the ambient air and the tower performance drops, causing condenser backpressure to increase. This occurs as with a higher air temperature air density decreases causing fan throughput in kg/s to fall off. A lower air flow (kg/s) will have an adverse effect on the heat transfer across the radiators, as with a higher air temperature.

2.6.2.7 ***CW Quantity Flowing Through Condenser Incorrect (Wet Cooled Units)***

As mentioned above, all turbines are designed to perform at certain levels of performance if certain input parameters are met.

One of these parameters as far as the condenser is concerned is the cooling water (CW) flow quantity (normally given in m^3/s)

If the actual CW flow entering the condenser is different from design or is incorrect, the condenser backpressure is affected. What this means is that if the CW flow is lower than design, and assuming that all other parameters are the same, condenser backpressure will increase and condenser vacuum will thus decrease, which will increase the heat rate of the unit, and hence lower unit efficiency. The reason for the higher condenser backpressure with a lower than design CW flow, is that the thermal conductance of the condenser decreases with a decrease in CW flow, and vice versa.

What this shows is that the cooling effect of the CW within the condenser is much lower with a lower CW flow.

Correspondingly if a much higher than design CW flow exists in a condenser, the backpressure will be much lower than normal and correspondingly the vacuum much higher.

What must be borne in mind with a higher or lower CW flow, which will change the condenser backpressure, is that the feed pump auxiliary power consumption also changes with a change in flow. The ideal is to have a CW flow through the condenser which will give a reasonably low backpressure without excessively increasing the feed pumps power consumption. In the case of steam feed pumps the heat equivalent of the steam used for the steam feed pump must not be excessive just to maintain a low condenser backpressure by maintaining a higher than normal CW flow. CW pumps power also changes with a change in

CW flow. The way to do a quick check on the CW flow is to check the CW temperature rise across the condenser (T_2 - T_1). T_2 is the mean CW outlet temperature and T_1 is the mean CW inlet temperature. If this temperature difference is too high, then too little CW flow through the condenser exists, and vice versa. Normally this temperature difference should be of the order of 6 to 10^{o}C.

2.6.2.8 ***Fouled Tube End Plates***

If the end plates that house the tube nests are fouled on the water side (ie on the side within the condenser water boxes), then the entrance of one or more condenser tubes will be restricted or totally blocked. This will restrict the flow of cooling water through the tube nests which will affect the overall thermal conductance of the condenser. Condenser backpressure will correspondingly increase and vacuum will decrease.

As the cooling water entering the condenser water boxes is now untreated water, it is important to keep the water boxes and tube end plates as clean as possible, as the formation of scale and other deposits on these surfaces is much higher with untreated water. This is especially prevalent at the coast where sea water is used for cooling purposes in condensers (Koeberg, etc.).

2.6.2.9 ***Dirty or Leaking Tubes***

If condenser tubes have scale deposits on either the steam or the water side, or both, then the heat transfer process within the condenser is adversely affected.

On the steam side scale deposits (such as silica) can be carried over by the exhaust steam and deposited onto the outside surface of the condenser tubes, and on the water side, scale can form inside the tubes due to impurities in the cooling water flowing through these tubes.

The effect of dirty tubes is to increase the condenser terminal temperature difference (TTD). Terminal temperature difference is the temperature difference between the backpressure equivalent saturation temperature (T_v), and the CW outlet temperature (T_2).

Dirty or fouled condenser tubes affect the heat transfer capabilities within the condenser, hence condenser backpressure is increased. An increased condenser backpressure would increase the T_v.

Increased tube fouling also prevents the cooling water from removing the correct amount of latent heat from the exhaust steam, hence the CW temperature rise between inlet and outlet of condenser (T_2 - T_1) is lower than normal, hence the TTD difference is higher than design (T_v - T_2).

A word of caution needs to be said about using the TTD as a check on whether the condenser tubes are fouled or not. If the CW flow through the condenser is not correct, then the CW temperature rise (T_2 - T_1) across the condenser will be different from design.

To explain this further, assume the CW flow through the condenser is lower than specified, all other things being equal, the CW temperature rise through the condenser will be higher than normal, i.e. CW outlet temperature (T_2) will be higher than normal.

At the same time, because the CW flow through the condenser is lower than normal, condenser performance drops off and the condenser backpressure will be higher than normal, i.e. the backpressure equivalent saturation temperature (T_v) will be higher than normal.

From this example it can be seen that if the CW flow is lower than normal, both T_v and T_2 will be higher than normal. The result of this is that the terminal temperature difference (TTD), ie T_v - T_2 could give a result that appears normal if both T_v and T_2 increased by approximately the same amount. From this we can see that a normal TTD can be affected with a CW flow that is far below normal.

It is thus of extreme importance to ensure that the correct CW flow exists, otherwise the value of using TTD as a check on the fouling of condenser tubes etc., is diminished. If condenser tubes are leaking, i.e. CW flows from the water side of the condenser into the steam side, the CW that does leak into the steam side of the condenser will mix with the condensate collecting in the condenser hotwell.

This has the effect of not only affecting the heat transfer capabilities of the condenser, i.e. less heat is absorbed by the CW entering the condenser, which lowers the condenser vacuum and hence increases condenser backpressure, but also affects the purity of the condensate leaving the condenser, and via the heaters etc., entering the boiler economizer.

In a condenser the vacuum is created as a result of a change of state within the condenser, i.e. the exhaust steam changes state from a gas to a liquid (condensate). This change of state occurs with a great reduction in volume between the steam and the condensate. This reduction in volume is what creates a vacuum condition inside the condenser, i.e. vacuum condition being a condition that exists at a pressure that is lower than the prevailing atmospheric pressure.

A CW leak into the steam side of a condenser can be detected by carrying out regular conductivity checks of the condensate within the condenser hotwell.

2.6.2.10 ***Air Leakage into Condenser***

If an air leak exists in a condenser the condenser performance is adversely affected. Condenser backpressure will increase as the air inside the condenser affects the heat transfer between the exhaust steam and the cooling water. What happens is that an air film forms on the steam side of the condenser tubes, which restrict the heat transfer from the steam to the cooling water.

This in turn affects the condensation of the steam (i.e. the change in state from steam to water), which in turn lowers the condenser vacuum, which results in an increased backpressure. The air ejectors are designed to remove all air from the condenser. If however, there is a severe air leak, the air ejectors will not be able to do so.

An air leak can normally be detected by monitoring the difference between the backpressure equivalent saturation temperature (T_v), and the temperature of the condensate in the condenser hotwell (T_c). Because basically only latent heat is given up by the exhaust steam in a condenser when the change in state from a gas (steam) to a liquid (condensate) takes place, thermodynamic law states that this change in state takes place at constant temperature.

What this means is that the backpressure equivalent saturation temperature (T_v) should be the same as the condensate temperature (T_c) in the condenser hotwell. Daltons Law of partial pressure state that the total pressure within a vessel is equal to the sum of the partial pressures of all the gases within that vessel.

In the case of a condenser, the pressure within a condenser is equal to the sum of the partial steam pressure plus the partial air pressure (other gases such as CO_2 etc.) are so small in concentration that they can be ignored for all practical purposes.

As the exhaust steam entering a condenser flows downwards towards the condenser hotwell, an additional pressure drop occurs on this steam as a result of the air presence in and around the condenser tubes. This has the effect of lowering or suppressing the condensate pressure and hence equivalent temperature.

In this way a large difference between T_v - T_c occurs as a result of an air leak into the condenser.

The difference for air free condensers between T_v and T_c should be between 0 and + 1^{o}C maximum. Any larger difference suggests an excessive air leak and should be investigated

2.6.2.11 ***Turbine Load***

Turbine load has an effect on the condenser backpressure from the point of view that the backpressure is higher at higher loads. The backpressure STEP loss is not necessarily higher at higher turbine loads, as the target backpressure is automatically increased to compensate for the higher turbine load.

A typical backpressure curve showing variation in load for only one CW inlet temperature is as shown below.

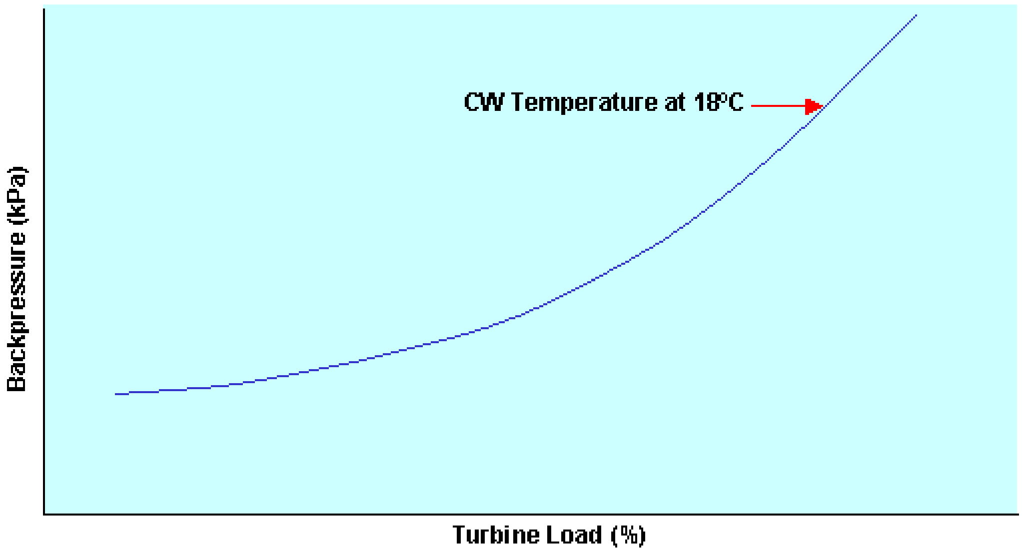


Figure 29 Typical backpressure curve

The only way that condenser backpressure could be reduced at the higher turbine loads would be to e.g. increase the cooling water flow, clean the condenser heat transfer surfaces, improve air ejector performance etc.

2.7 **Experimental design methods**

Various experimental methods were used during this research dissertation and a brief description of the methodologies that were used is documented in the sections below.

2.7.1 **Experimental research**

Research is collected and verified through a series of experiments (Blakstad, 2008). In the case of this research, experiments are conducted in order to prove or disprove a selected hypothesis.

2.7.2 **Quantitative research**

The following definition delivers an apt description of what is meant by quantitative research:

“Quantitative research is “Explaining phenomena by collecting numerical data that are analyzed using mathematically based methods in particular statistics’” (Aliaga & Gunderson, 2000)

2.7.3 **Deductive reasoning**

When applying deductive reasoning, it is assumed that a certain law or principle is true, and because it is true, it guarantees that a certain output is also true. Therefore if aspect A is present, then B will be the output. (Schechter, 2013)

2.8 Eskom Procedures

There are many procedures that have been put into place over time as Eskom has developed. This section describes the engineering change management procedure that had a direct influence on this dissertation.

2.8.1 **Engineering change management procedure**

Eskom defines an engineering change as and permanent or temporary change, deletion or addition to any system, equipment or structure; including permanent changes to operating, protection and control set points and software and technical documentation that results in any deviation from the established design base. The engineering change management procedure aims to ensure that all engineering changes are correctly prepared, motivated, reviewed, approved, controlled and recorded. The process of the Engineering Change Management procedure is indicated below:

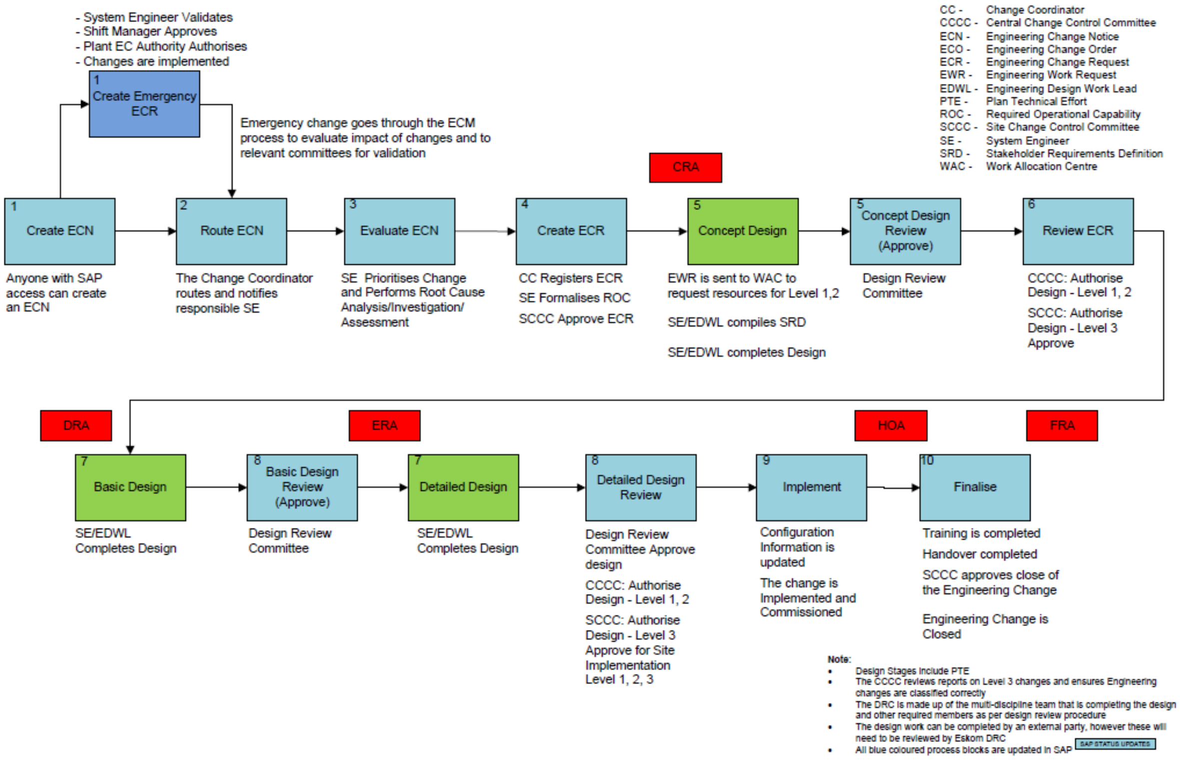


Figure 30 The Engineering Change Management Procedure utilized at Eskom in 2014

Figure 30 demonstrates the lengthy process that needs to be followed whenever an engineering change needs to be implemented. This process is very time consuming and requires a lot of effort to implement.

It is discussed here because certain simple modifications needed by the researcher, to facilitate taking measurements were abandoned due to the onerous demands of the ECM system.

2.9 Summary

To summarize, the backpressure inside a condenser can be affected by CW flow, CW temperature, heat load from the turbine, cleanliness of the condenser and air present inside the condenser (either due to leaks or poor ejector performance).

2.9.1 Effect of an air leak or poor ejector performance

If the heat load from the turbine is constant, and the CW flow remains constant, the following variables will increase if there is an air leak or if the ejectors are not performing:

- P_{sat} and therefore T_{sat} inside the condenser
- Condensate DO levels
- CD
- TTD

An extra air ejector or a quick start ejector might need to be run which will also affect the efficiency of the plant due to deviation from design operating philosophy.

2.9.2 Effect of an increased heat load

If the CW flow remains constant, the following variables will increase due to and increased heat load:

- P_{sat} and therefore T_{sat} inside the condenser
- TTD
- TR

The increase of heat load will not affect the CD.

2.9.3 Effect of a decrease in heat transfer or surface area

Fouled or plugged tubes will cause the following variables to increase if the heat load and CW flow remains constant:

- P_{sat} and therefore T_{sat} inside the condenser
- TTD

Fouled or plugged tubes will not affect the TR or the CD.

2.9.4 Effect of reduced CW flow

A reduced flow may be caused by blocked tubes/tubesheet, CW pump problems, valve complications or unvented waterboxes. If the heat load remains constant, the following variables will increase due to reduced CW flow:

- P_{sat} and therefore T_{sat} inside the condenser
- TR

A reduced CW flow will have no effect on the CD and a minute effect on the TTD.

2.9.5 Effect of an increase in CW inlet temperature

A higher than design CW inlet temperature may be caused by the cooling tower performance or high ambient temperatures. If the heat loads from the turbine and the CW flow remains constant, the following variable will increase:

- P_{sat} and therefore T_{sat} inside the condenser

The CW inlet temperature will not affect the TTD, TR or CD.

In the above summary, only one variable was changed each time in order to summarize the effect of certain circumstances on the condenser; however in reality it is probable that more than one of the above may occur at any given point in time. In such a case, plant measurements will reflect the combined effect of changes and the following chapter describes the methodology used to investigate and detect the causes of increased backpressure that was being experienced at Grootvlei Power Station and provide some insight into the effect of this loss on thermal efficiency. Potential gains from solving these problems are also documented in the following chapters.

Chapter 3 - Methodology of Investigation

The purpose of this chapter is to illustrate the methodology that was used to investigate the high backpressure losses, as identified in Figure 2, which Grootvlei Power Station was experiencing at the time of the investigation in 2014. Therefore this chapter provides a description of the experimental design that was followed to methodically isolate possible causes of the high backpressure losses in order to finally determine and find possible solutions to all of the contributory causes.

3.1 Appropriateness of the research design

The approach that was used was one of experimental and quantitative research and analyses, incorporating deductive reasoning (as described in section 2.7), in order to test various hypotheses of factors that could have been contributing to the backpressure losses.

3.2 Strategy and research design

The units that were contributing the most to the losses needed to be identified, and an analysis of the backpressure losses was taken from historic data for the period of the previous financial year in order to identify which units had the highest backpressure losses (2013-2014). Hereafter a logic diagram was designed to identify possible causes for the backpressure losses.

3.3 Condenser efficiency analysis

All the data from the STEP reports regarding condensers from the time that Grootvlei Power Station was fully returned to service in 2008 was trended for the identified units in order to determine when the backpressure started to diminish and when it was at its worst. From the STEP report data, a condenser efficiency analysis was conducted for the month of January 2014 in order to form hypotheses of what the main causes of the high backpressure losses could be. These causes were also compared to the logic diagram findings.

Four possible causes could be deduced from the condenser efficiency analyses (Figure 31):

3.3.1 Possible causes of condenser inefficiency

3.3.1.1 *Condenser cooling*

There could be cooling tower inefficiencies or inadequate cooling water flow through the condenser which will cause a lack of heat transfer across the pipes.

3.3.1.2 *Scaling/Fouling*

There could be scaling or fouling taking place inside the condenser tubes which also affect the heat transfer from the steam to the cooling water.

3.3.1.3 *Air ingress*

There could be air ingress into the condenser. The air forms a layer around the tubes (blanketing) which also inhibits the heat transfer through the tubes.

3.3.1.4 *Passing valves*

There could be valves that are in operation, which were meant to be closed during normal operation of the condenser, allowing excess heat in the condenser and therefore increasing the heat load on the condenser.

After identifying which of these possible causes were influencing the high backpressure losses, each one was investigated to determine whether it really was the case, and if so, what the contributory factors were. Figure 31 illustrates the factors discussed above, thus forming the framework of the investigation.

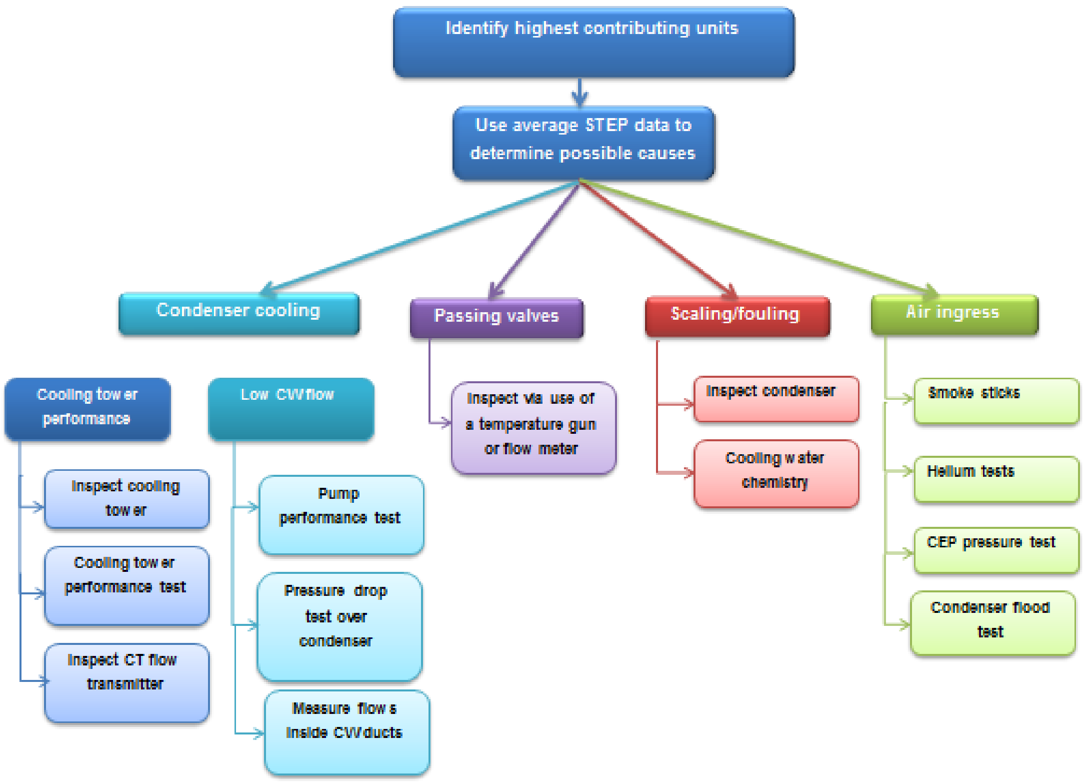


Figure 31 Structure of investigation

3.4 Logic diagram analysis

Figure 32 indicates that the logic diagram that was designed to methodically investigate various attributes that could be contributing to the backpressure losses. This diagram was used in conjunction to the condenser efficiency analysis to narrow down possible backpressure loss causes.

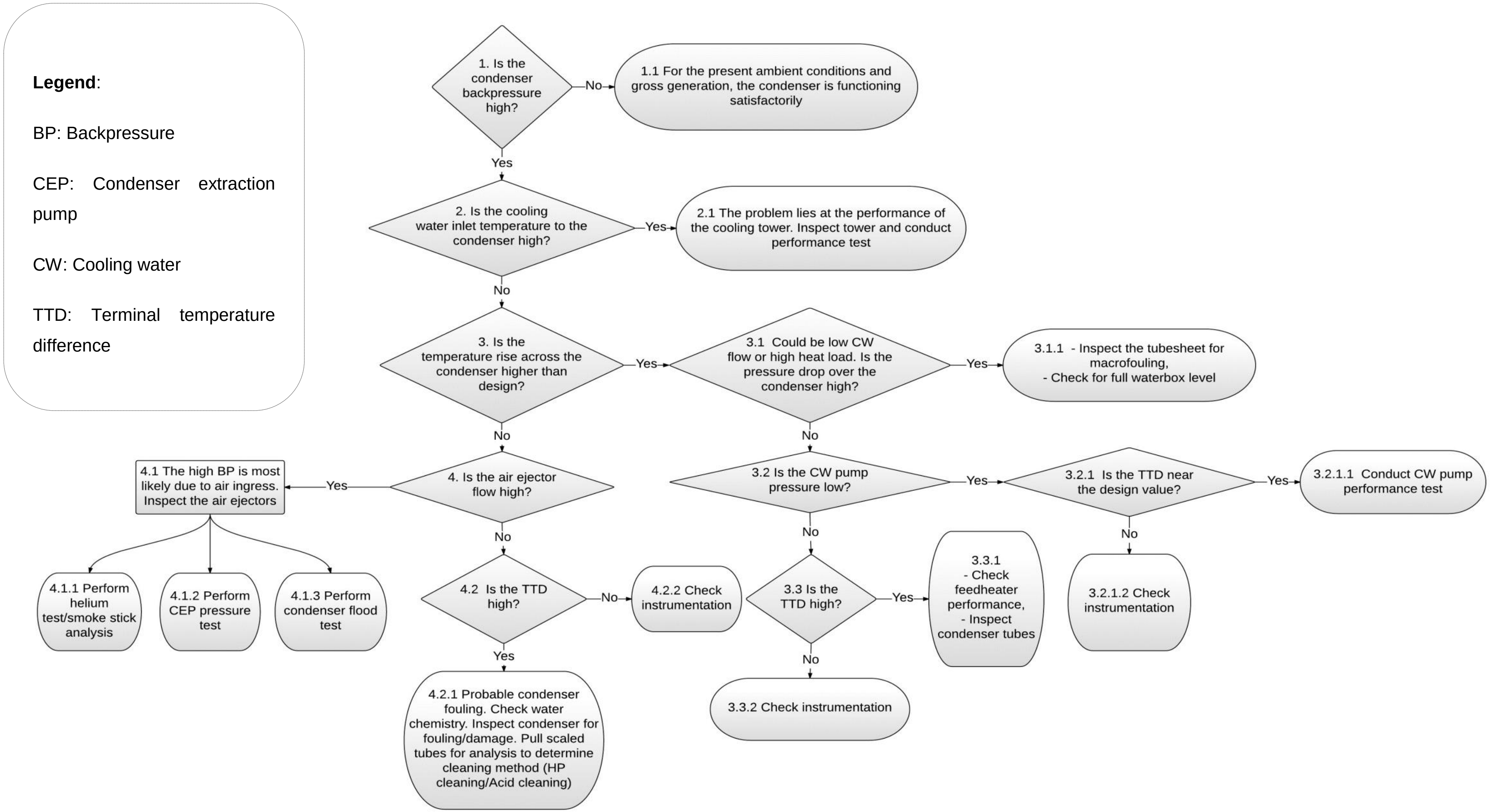


Figure 32 Logic diagram describing the methodology paths that were followed to determine the cause of the backpressure losses

3.4.1 Logic diagram description

If the condenser backpressure is high, the next parameter to look at is the temperature rise over the condenser. A temperature rise (TR) higher than design indicates that there is either low cooling water flow or a high heat load or both. Further investigation would be to conduct a pressure drop test over the condenser to calculate whether there is enough flow for the current circumstances through that condenser. A high pressure drop over the condenser will indicate that there could be macrofouling taking place inside that condenser or that the condenser waterbox is fuller than it should be which could be causing the indication of a low flow/high heat load.

If the pressure drop over the condenser is low, the performance of the pumps needs to be checked by checking the pump pressure along with the terminal temperature difference (TTD) data from the condenser efficiency analysis. A lower than design CW flow and higher than design TTD will indicate that the pumps are underperforming and an investigation of the performance of the pumps should follow by comparing the actual data of the pumps (the amperes and the pressure that they are operating at) to the design or acceptance test data.

If the TR is lower than design, the problem could be fouling inside the condenser, air ingress into the condenser or it could be an air removal issue. The air ejector flow can be checked in order to determine whether it is higher than design. If it is high, and the TTD is also high, then there is air ingress into that condenser which is inhibiting the heat transfer.

If there is no high air ejector flow, and a high TTD, there is most likely fouling taking place inside that condenser.

3.4.2 Summary of Investigation strategy

Below follows a detailed description of the inscriptions found on the logic diagram (Figure 32).

3.4.2.1 *Cooling tower performance not meeting design specification (view point number 2 on logic diagram)*

To ascertain whether the back pressure problem could be ascribed to under- performance of the cooling tower the following investigative logic was applied:

1. Compare the condenser cooling water inlet temperature with the acceptance test data. If found higher than design, the cooling tower is underperforming
2. Perform physical inspection of cooling tower, check for leaks and damage.
3. Conduct a cooling tower performance test

3.4.2.2 *Cooling water flow suspected to be below the design flow (view point 3.1 on the logic diagram)*

To investigate whether a high TR is caused by low cooling water flow, the following logic was followed.

1. If the pressure drop over the condenser is above design specification, then something is obstructing the flow and the tubesheet should be checked for macrofouling, and/or the waterbox level needs to be checked.
2. If the pressure drop over the condenser isn't above design specification, and the CW pump pressure is low: The problem most likely lies with the pumps. Conduct CW pump performance test.
3. If the pump pressure is within specification and the terminal temperature difference (TTD) is high, then the problem could lie by the feedheaters. It will also be wise to inspect the condenser tubes if a high TTD is observed.
4. Check whether the transmitter readings are accurate or physically measure the flow rates on the CW lines.

3.4.2.3 *Suspected scaling inside the condenser investigation strategy (view point 4.2 on the logic diagram)*

If, according to the logic diagram, there are indications of scale build up inside the condenser:

1. The water chemistry will be able to verify this phenomenon. Therefore the water chemistry should be checked for scaling potential circumstances.
2. If the water chemistry indicates that scaling could be taking place, the condenser should be inspected and cleaned at the next available opportunity.

3.4.2.4 *Possible air in-leakage investigation strategy (view point 4.1 on the logic diagram)*

If the logic diagram indicates that the backpressure loss could be due to air ingress, the following methods can be used in order to indicate air in-leakage.

1. Two methods to determine air leaks on a condenser were identified namely the smoke sticks method described in section 2.2.6.7.6 and the helium testing method as described in 2.2.6.7.5. Grootvlei Power Station only had access to a helium tester at the time that these tests needed to be done and therefore only this method was used. However, the use of smoke sticks would have been more beneficial to use for

the tests since much of the helium sprayed could be ejected through the de-aerators and this could cause certain leaks not to be detected.

2. Another method that can be used to determine where air could be leaking into the condenser is to conduct a flood test when the unit is offline as discussed in section 2.2.6.7.8.
3. A third method is to have a CEP pressure test done (section 2.2.6.7.7 describes these types of tests).

3.4.3 Further analyses

After identifying the main contributing factors to the condenser backpressure, a further analysis was done in order to indicate the impact of the backpressure loss on the thermal efficiency, the financial impact hereof and a cost benefit analysis in order to motivate the correction of the problems that were found.

3.4.3.1 *Thermal efficiency impact calculations*

The impact of the backpressure loss on the thermal efficiency was calculated by using the following method:

The monetary loss due to backpressure was taken from the monthly STEP report. From the monetary loss, the amount of coal that was over-burned was calculated by dividing the monetary loss by the price of coal:

$$\textit{Over-burned coal} = \frac{\textit{monetary loss}}{\textit{coal price}}$$

This amount of coal was then subtracted from the total amount of coal burned in order to determine how much coal would have been burned if there had been no backpressure loss.

The new thermal efficiency was then calculated by using the following formula:

$$\textit{Thermal efficiency (\%)} = \frac{USO \times 1000 \times 3.6}{\textit{coal burned} \times CV} \times 100$$

3.4.3.2 *Cost benefit analysis calculation methodology*

A cost benefit analysis was done in order to validate the cleaning of a condenser. The methodology of these calculations is indicated in this section.

3.4.3.2.1 Cost of running a unit with fouled conditions

Firstly the cost of running a unit at 100% load, with the condenser tubes fouled up was calculated. The data was chosen for a day when the unit was running at full load (MCR) whilst being fouled up.

The following plant data was necessary for the analysis to be done:

- Boiler steam flow at full load
- Boiler efficiency on standard fuel
- Calorific value of the coal used
- The cost of the coal used
- The mill rating at full load conditions
- The load and number of mills in service during the test

The cost of sustaining the fouled tube conditions at full load had to be calculated. The coal consumption in GJ/s was calculated.

3.4.3.2.2 Coal consumption per second in gigajoule (GJ/s)

Coal consumption (GJ/s)$= CV_{coal} \times$ *coal flow*

3.4.3.2.3 Correction factor calculations

A correction factor had to be calculated for the current circumstances. Therefore the correction curve polynomials were extracted from the STEP report and the acceptance test data and the actual plant data were inserted into these polynomials. The following data was needed for these calculations.

- Acceptance test backpressure
- Actual backpressure at the time of the test

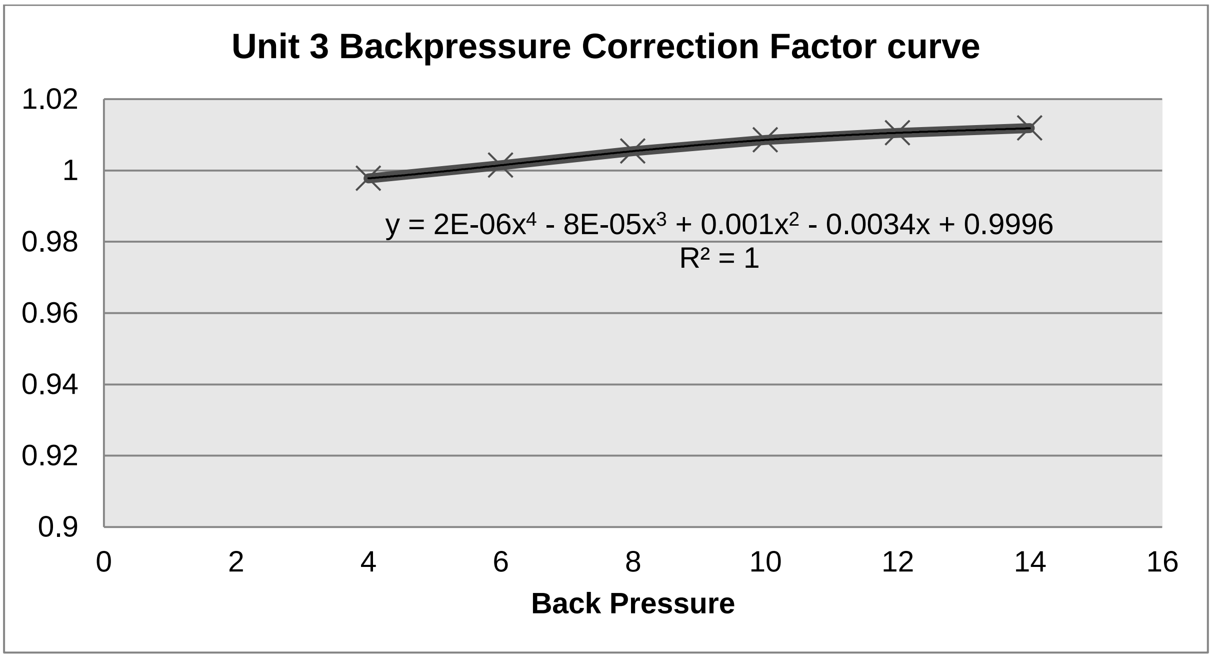


Figure 33 Unit 3 correction factor curve for backpressure at 100 % as taken from STEP

Figure 33 displays the extracted correction factor curve that was used to supply the polynomials. The backpressures of the acceptance test data and the actual data was to be inserted into this polynomial in order to obtain the respective correction factors for both parameters.

Therefore, the polynomial to calculate the correction factor was derived to be:

$$\begin{aligned} Correction\ factor \\ &= 0.22809\ exp^{-6}(\mathrm{BP}^4) - 0.79exp^{-4}(BP^3) + 0.1018exp^{-2}(BP^2) \\ &- 0.3379exp^{-2}(BP) + 0.999575 \end{aligned}$$

The correction factor for both the acceptance test data (ATCF) and the actual running data (ACF) was calculated and the variance between these two correction factors was found.

$$Correction\ factor\ variance = ACF - ATCF$$

3.4.3.2.4 Backpressure variance

The variance in backpressure was calculated:

$$BP\ variance = Actual\ BP - Acceptance\ test\ BP$$

3.4.3.2.5 Heat change correction factor

The correction factor for the change in heat was calculated:

$$Correction\ factor\ (Heat\ change) = \frac{Correction\ factor\ variance}{BP\ variance}$$

3.4.3.2.6 Extra heat at 100% load with fouled tubes

The extra heat for 100% load was then calculated

$$Extra\ heat = Full\ load\ heat\ consumption\ \times Correction\ factor\ variance$$

3.4.3.2.7 Extra heat consumption at 100% load

The extra heat consumption was calculated:

$$Consumption = \frac{Extra\ heat}{Boiler\ efficiency}$$

3.4.3.2.8 Cost of extra heat consumption

The cost of the extra heat consumption was calculated:

$$Cost\ of\ extra\ heat\ consumption = Consumption\ \times cost\ of\ coal$$

Chapter 4 - Investigation

This chapter describes the execution of the logic diagram indicated in Chapter 3 in combination with the condenser efficiency analyses.

In order to determine which unit had been contributing the most to the backpressure issues, the STEP reports for the duration of a year (dating from February 2013 to January 2014) were analyzed.

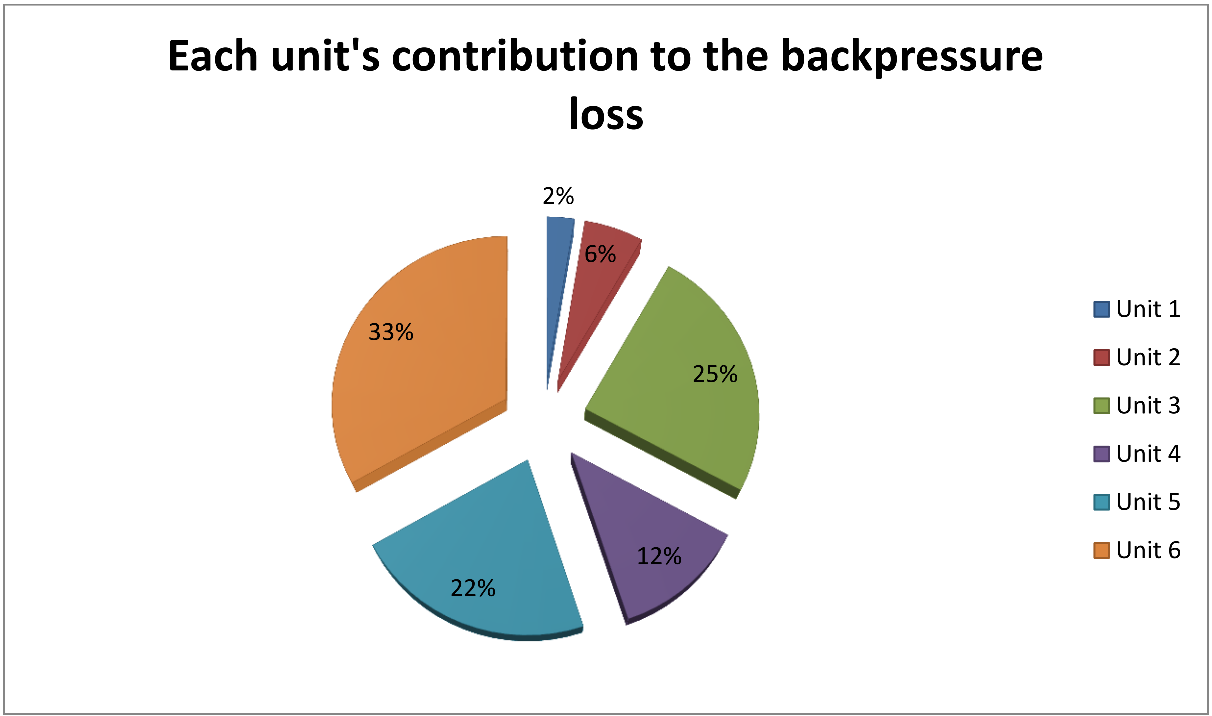


Figure 34 Indication of each unit's contribution to the backpressure loss at Grootvlei Power Station (2013-2014)

Figure 34 indicates that that Unit 3 and Unit 6 were responsible for over 50% of the power plant's condenser backpressure losses. It was therefore decided to limit the scope of investigating back pressure losses to these two units.

A further advantage of this choice was that Unit 3 was wet-cooled- and Unit 6 dry-cooled thus allowing some comparison between the systems.

.

After units 3 and 6 were identified to be the largest contributing factors to the backpressure losses, a condenser efficiency analysis, as described in section 2.2.6.5, was done for both units as indicated in Figure 35 and Figure 44 in the sections below:

4.1 Condenser efficiency analyses

A condenser efficiency analysis was conducted for each unit on the 8th of January 2014 at 11am. The results are recorded below:

4.1.1 Condenser efficiency analysis for Unit 3

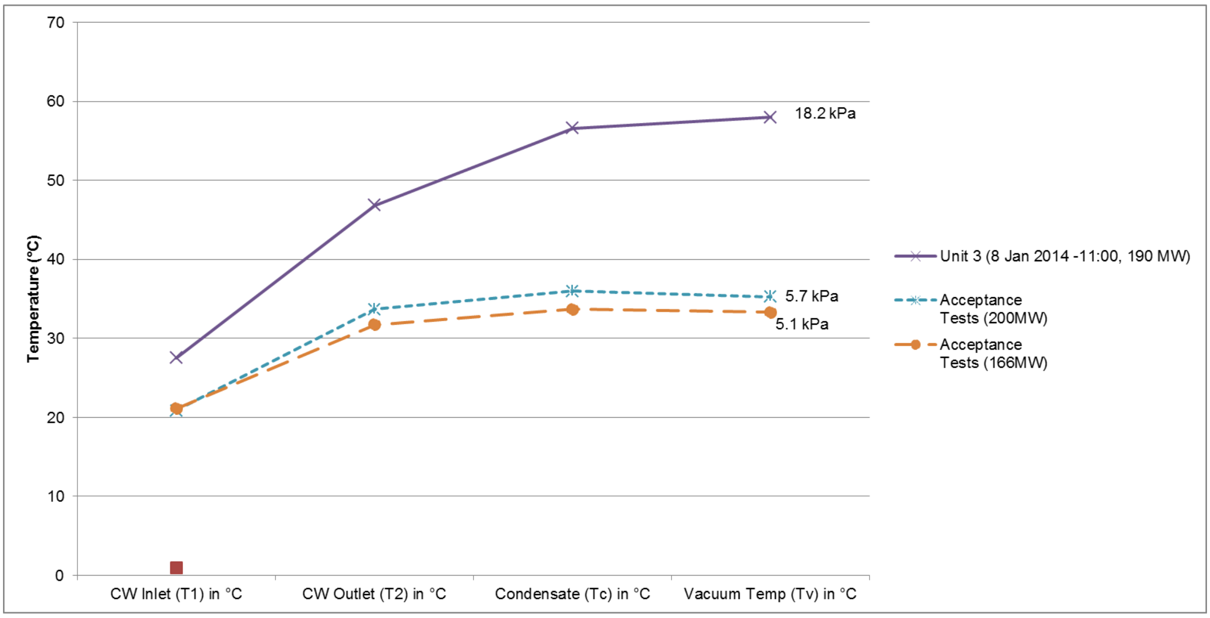


Figure 35 The condenser efficiency analysis graph for Grootvlei Power Station's Unit 3 as on 26 February 2014

Unit 3 was indicating a low cooling water flow, fouling as well as air ingress as per the discussion in section 2.2.6.5.

Table 1 Raw data used in the condenser efficiency analysis for Unit 3

LOAD	Acceptance Tests (200MW)	Acceptance Tests (166MW)	Unit 3 (8 Jan 2014 - 11:00, 190 MW)		
			A	B	
CW Inlet (T_1) in °C	20.9	21.1	27.4	27.7	27.55
CW Outlet (T_2) in °C	33.7	31.75	48.7	45	46.85
Condensate (T_c) in °C	36	33.7			56.6
Vacuum Temp (T_v) in °C	35.3	33.3			58.03

Backpressure (kPa)	5.7	5.1	[redacted]	18.2
TTD = T_v-T_2	1.6	1.55	[redacted]	11.18
CD = T_v-T_c	-0.7	-0.4	[redacted]	1.43
TR = T_2-T_1	12.8	10.65	[redacted]	19.3
Condenser Efficiency (%)	88.9	87.3	[redacted]	63.3

Table 1 indicates the raw data that was used for the condenser efficiency analysis. Additionally it indicates the calculations of the TTD, the CD, the TR as well as the condenser efficiency. The high TTD was indicating that there was low heat transfer taking place inside this condenser. This could be due to fouling or air ingress causing blanketing around the condenser tubes. The CD was also higher than it should have been which supported the suspicion that there could have been air ingress into this condenser. The TR was higher than the acceptance test values.

The TR is set by a combination of the heat load and CW flow. The heat load is fixed by the turbine. Therefore this higher TR could have been indicating that the CW flow was low through this condenser. However, this phenomenon could also have been caused by fouling inside the condenser which in turn impeded the flow through the condenser.

The logic diagram indicated in 3.4.1, Figure 32 was used for further investigation of the possible causes of the high backpressure.

4.1.2 **Unit 3 logic diagram investigation (wet-cooled system)**

The first parameter that was looked at according to the logic diagram was the condenser inlet temperature (point number 2 on the logic diagram). Table 1 indicated that the inlet temperature was approximately 7ºC higher than it should have been. According to the logic diagram, point number 2.1, this was an indication of poor cooling tower performance.

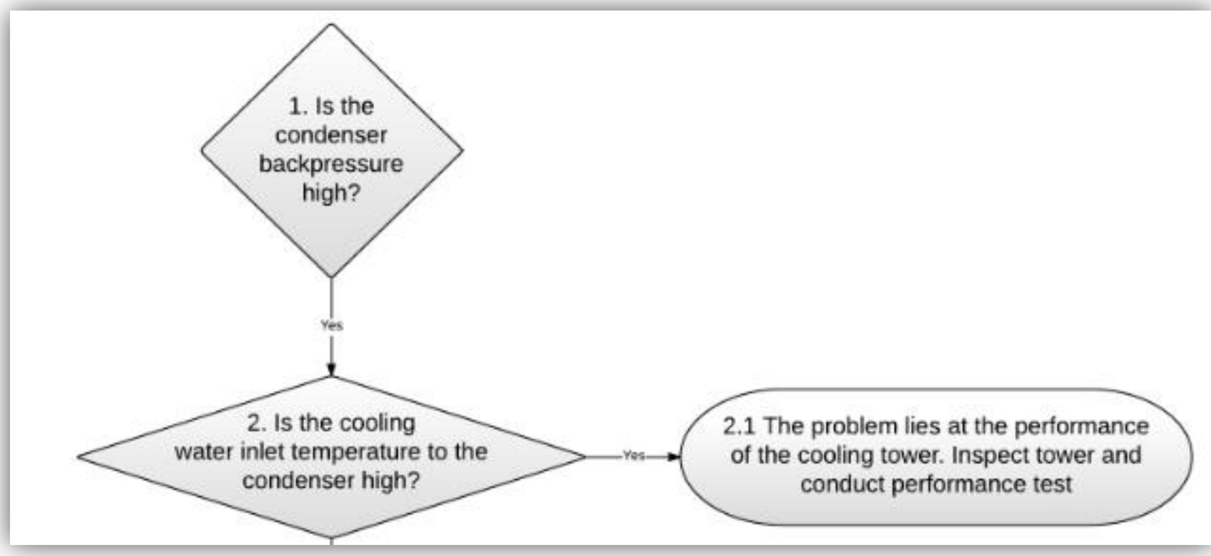


Figure 36 Logic diagram extract indicating poor cooling tower performance

Therefore, the cooling tower was inspected and found to be in very poor condition. There was broken splash packing, the spray water nozzles were blocked, the screens were blocked and many of the end caps had been blown off.

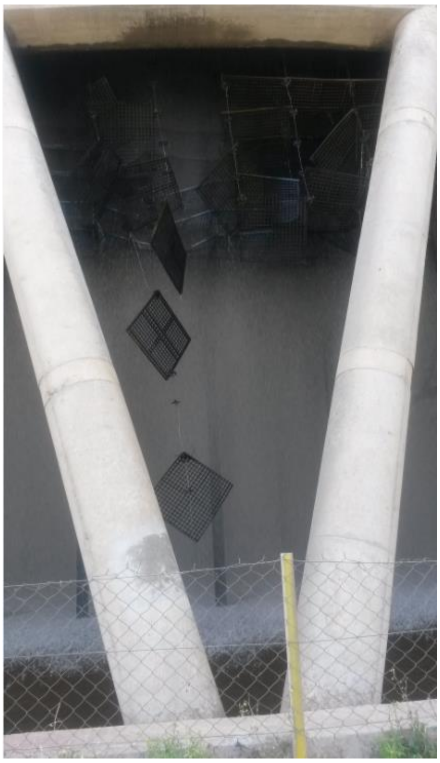


Figure 37 Broken cooling tower packing

During the winter months, the ambient temperatures are often below freezing point and this causes water to freeze inside the cooling tower and on the packing. This frozen water becomes heavy after accumulating and this is the main cause of cooling tower packing breaking off.

Figure 38 Blocked cooling tower screens due to broken off packing

The function of the cooling tower screens is to prevent large objects from passing through to the condenser. Three screens are in place for cleaning purposes. One screen can be removed at a time in order to clean off any objects and algae. These screens are generally cleaned once a month.

Therefore, a major contributor to the backpressure loss was identified to be the state of the cooling tower. It was decided to continue the investigation in order to identify other possible causes to the high backpressure losses on Unit 3.

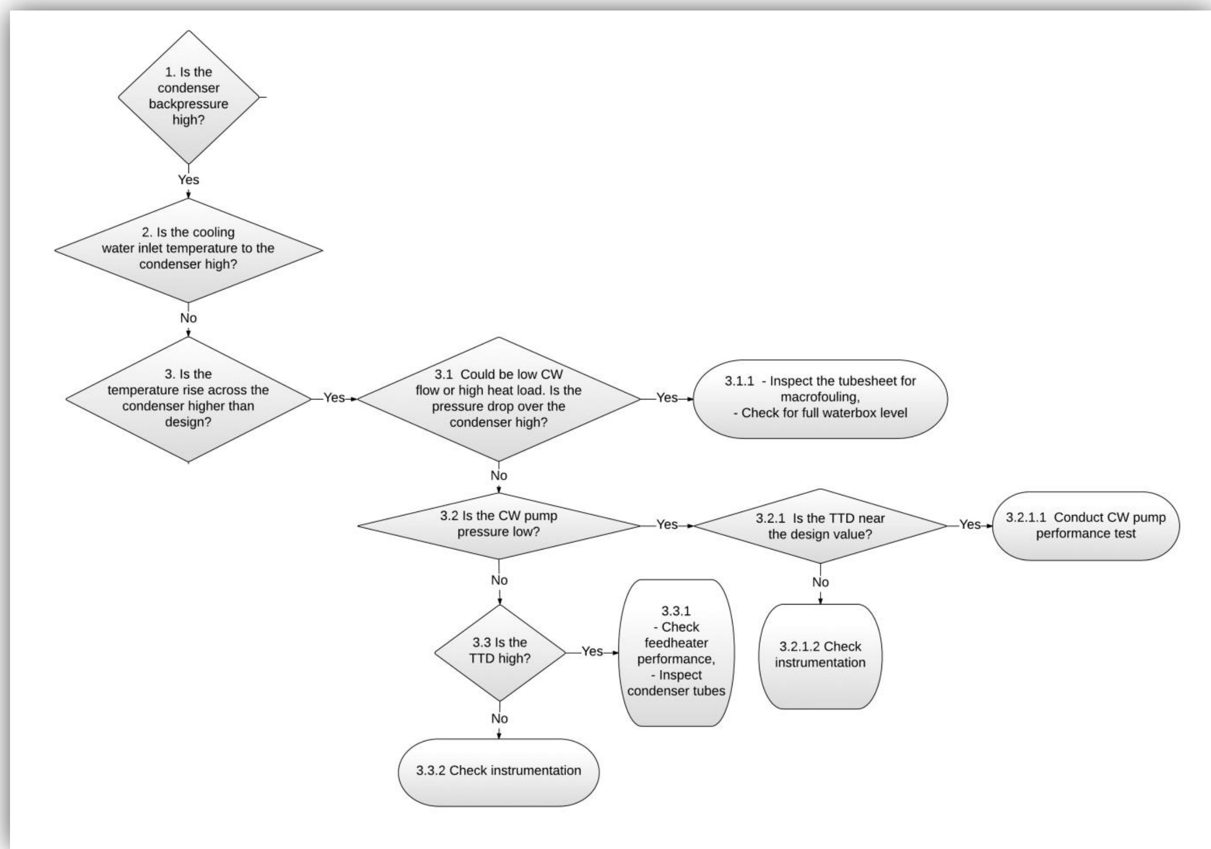


Figure 39 Logic diagram extract for high TR

The next parameter to look at according to the logic diagram (point number 3) was that of the temperature rise (TR). This unit was indicating a high TR according to Table 1. The performance test to conduct was to measure the pressure drop over the condenser as described in section 2.2.6.7.2 of this document. However, when this test was to be initialised, it was discovered that there was only one sampling point commissioned on the CW inlet duct. The other sampling points had not been re-commissioned with RTS. A quick scan of the other units indicated the same occurrence.

It was then suggested to have valves inserted into these points in order to have the points re-commissioned. The maintenance department was only willing to do this if the system engineer would follow the Engineering Change Management process as described in section 2.8.1.

However the system engineer did not agree on the fact that this was an aspect that needed to follow ECM, since it was assumed that these points had been commissioned at a point in time before Grootvlei was mothballed, and therefore the points were actually supposed to be re-commissioned at RTS. Therefore the opinion from engineering was that these points should be re-commissioned without the need of ECM. ECM is a lengthy process which takes a lot of time and effort. Due to disagreement from the two departments, these points, at the time of writing, had yet to be re-commissioned and this test could unfortunately not be completed.

To be able to continue with the investigation plan, engineering judgement was applied in making the following assumption:

It was decided to continue with the logic diagram as if the TR was not high since the condenser efficiency analysis indicated that fouling as well as air ingress were also playing a role in the backpressure loss.

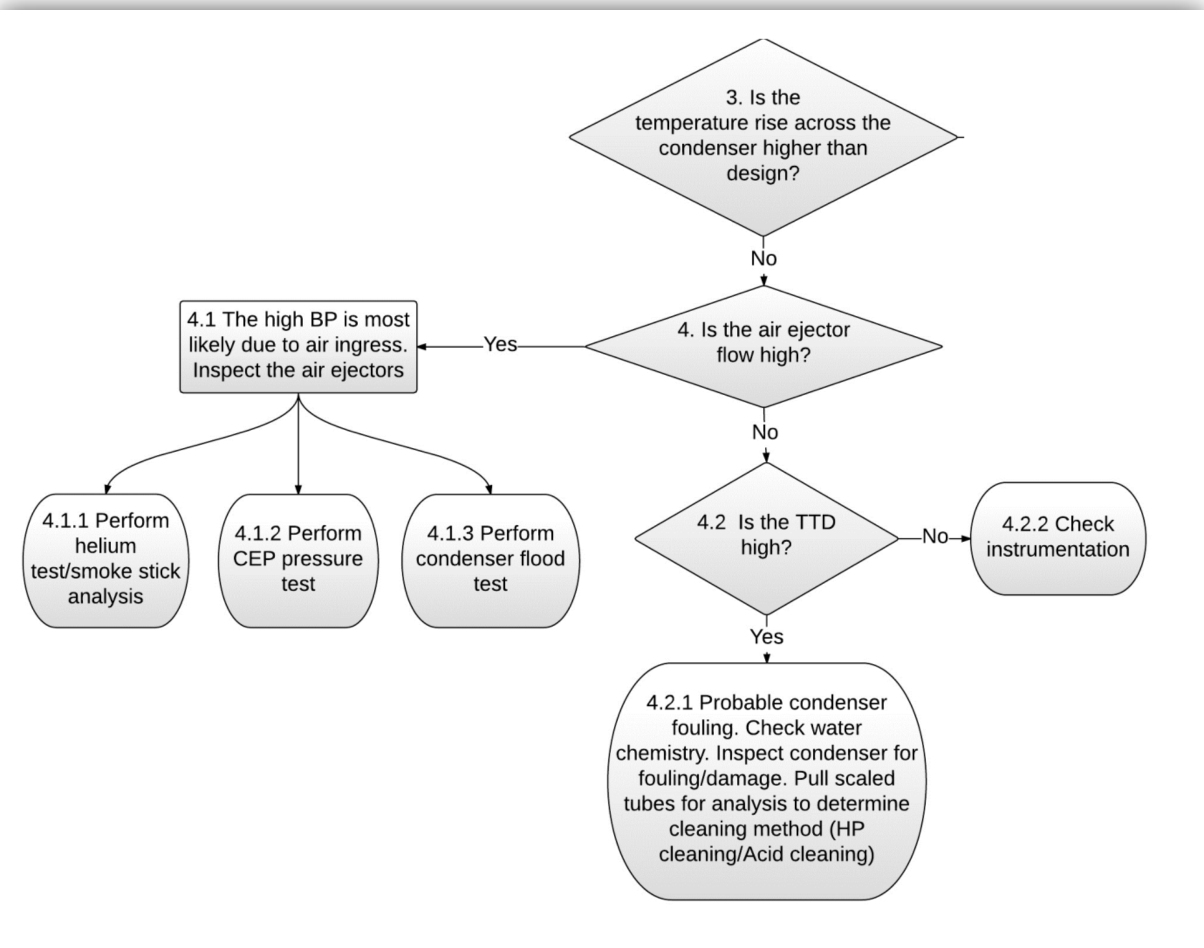


Figure 40 Logic diagram extract for fouling and air ingress

The logic diagram indicated (point number 4) that the air ejector flow had to be checked for high flows. The flows were taken from the outlet of the air ejectors and compared to the EPRI standard that was advised by Nalco (www.nalco.com/eu), a contracted company that had expertise in this field: “The air being ejected should be less than 150 litres per 100 megawatt”. Below is a summary of the measured flows:

Table 2 Air ejector flows on U3 as taken on the 3rd of September 2014

	Unit 3	
Ejector A	Load (MW)	150
	Air flow - ejector A (m/s)	3.81
	Diameter of pipe - ejector A (mm)	65
	Area of pipe A (m^2)	0.00332
	Volumetric flow (m^3/s)	0.01264
	Volumetric flow - ejector A (*l*/min)	758.565
	Volumetric flow - ejector A (*l*/min) per 100 MW	505.71
Ejector B		
	Air flow - ejector B (m/s)	2.64
	Diameter of pipe - ejector B (mm)	65
	Area of pipe B (m^2)	0.00332
	Volumetric flow (m^3/s)	0.00876
	Volumetric flow - ejector B (*l*/min)	525.62
	Volumetric flow - ejector B (*l*/min) per 100 MW	350.413

Table 2 illustrates that the air ejectors were removing up to three times more air than the stated benchmark noted above. This indicated that the air ejectors were being overworked and that they were probably not coping with the removal of all the additional air in the system. Therefore air was leaking into this system and the leaks needed to be detected.

A flood test was conducted on the 17th of September 2014 during a three day outage and there were no defects found on the condenser that could be the cause of the air ingress. The two CEP’s were being run at all times on this unit at that stage, and therefore the air could not have been entering through the CEP’s since both lines were pressurised. Therefore the leak had not been found yet, and a helium test was to be conducted on this unit. This test was conducted and four minor leaks were found.

During the outage, the condenser was opened for inspection. The condenser tubes were found to be filled with a slimy grey-brown sludge. This corroborates with the high TTD that the condenser was experiencing since this sludge would definitely have had an effect on the condenser’s heat transfer potential. Besides the sludge, there was also a lot of macrofouling found inside the condenser, most of which was broken packing that was coming from the cooling tower. The presence of the broken packing indicated that either the cleaning procedure of the cooling tower screens was not being followed suitably or that the screens had holes in.

Figure 41 Image of the grey-brown sludge found inside Unit 3's condenser tubes

A sample of this sludge was taken and sent for analysis.

Figure 42 Sample taken of the sludge found inside the condenser of Unit 3

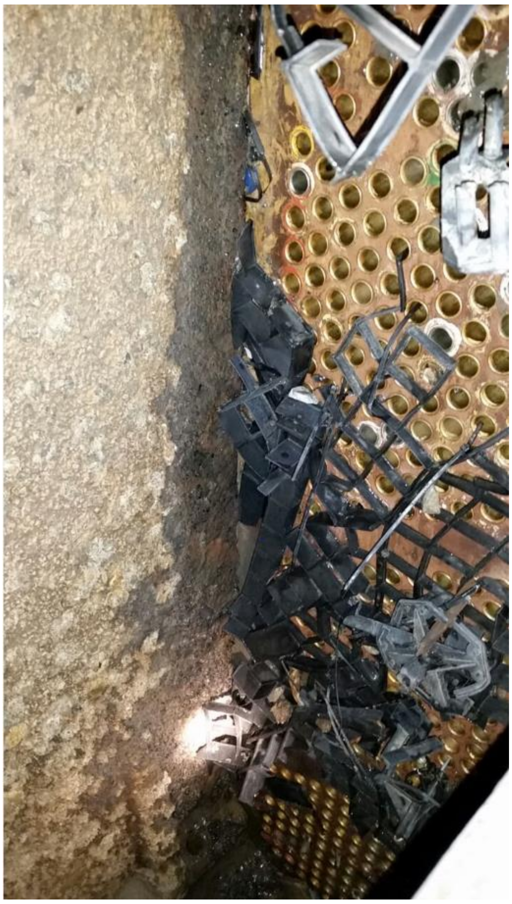


Figure 43 Macrofouling that was found at the inlet waterbox of the condenser

There was a lot of macrofouling found inside the condenser waterboxes. The broken packing was removed from the condenser. Two possible reasons for this packing to have passed through the cooling tower screens were identified:

1. The screens were broken and therefore allowing objects through
2. The persons responsible for cleaning the screens were not following protocol of removing the screens one at a time and cleaning them, but were for some reason removing all three cooling tower screens to clean them all at the same time. This would allow for the water to flow feely down that channel.

An inspection of the cooling tower screens disclosed that the screens were not broken and that therefore the passing of the large objects into the condenser was due to operator negligence.

No further investigation was done on this unit for the purposes of this dissertation.

4.1.3 Condenser efficiency analysis for Unit 6

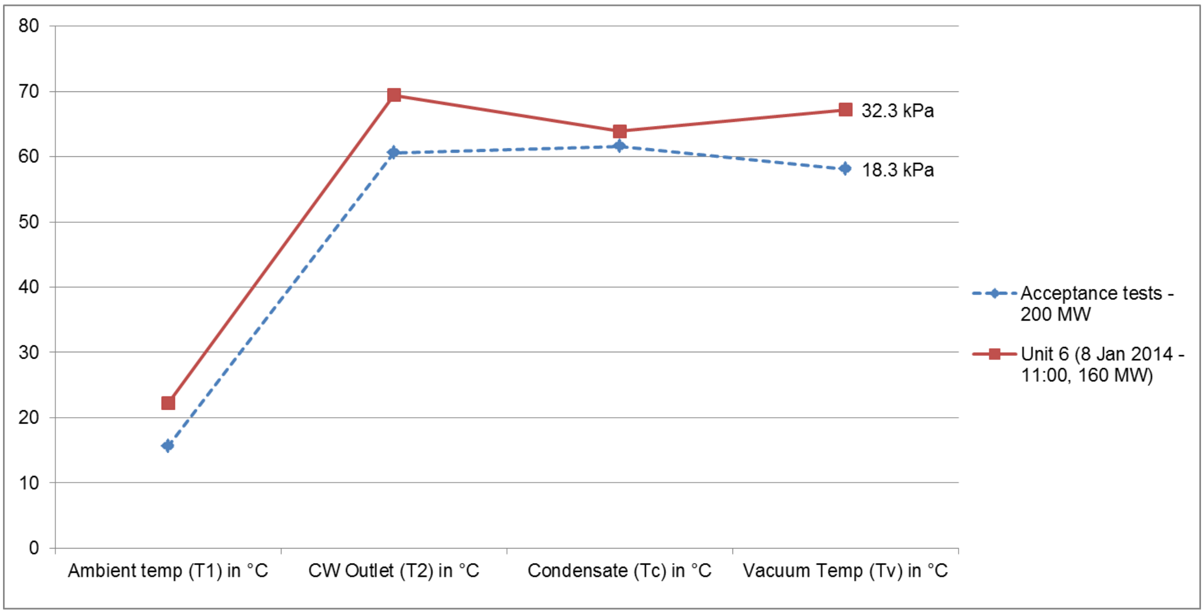


Figure 44 Condenser efficiency analysis for Grootvlei Power Station's Unit 6 as on 26 February 2014

Unit 6 was indicating an air in-leakage. The hotwell temperature was also lower than the T_2 temperature which indicated that this temperature was mostly incorrect. The hotwell temperature was therefore in need of verification.

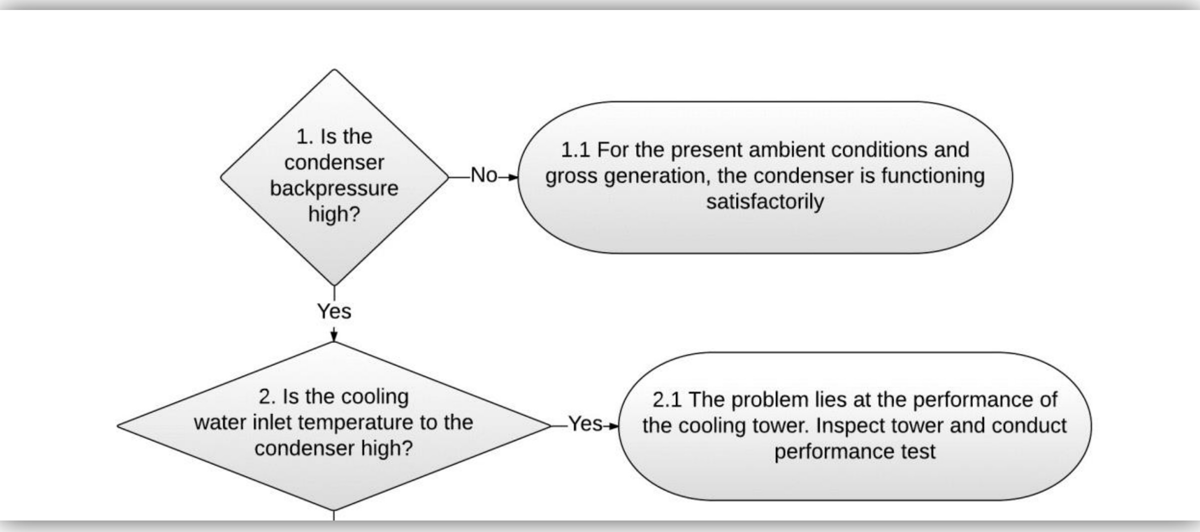
Table 3 Raw data used in the condenser efficiency analysis for Unit 6

LOAD	Acceptance tests - 200 MW	Unit 6 (8 Jan 2014 - 11:00, 160 MW)		
		A	B	
Ambient temp (T_1) in °C	15.6	22.2	22.2	22.2
CW Outlet (T_2) in °C	61.5	71.39	67.4	69.395
Condensate (T_c) in °C	61.6			63.9
Vacuum Temp (T_v) in °C	58.1			67.19
Backpressure (kPa)				70.8
CW inlet temp (T) in °C	42.85			53.5
TTD = T_v-T_2	-3.4			-2.205
CD = T_v-T_c	-3.5			3.29
TR = T_2-T_1	45.9			47.2

The TTD reading was close to the acceptance test values and therefore is not of concern. This unit uses demineralized water and therefore will rarely have fouling issues. However, this unit was indicating a high CD which indicated probable air ingress into the condenser. The temperature rise was slightly higher than the acceptance test values which were a possible indication of low CW flow.

4.1.4 **Unit 6 logic diagram investigation (dry-cooled system)**

The logic diagram (point number 1 and 2) was consulted for the purposes of further investigation for Unit 6. The condenser backpressure was higher than the acceptance tests indicated that it should be. Table 3 illustrates that the CW inlet temperature was approximately 10ºC higher than design. This indicated that the backpressure problem was partly due to poor cooling tower performance.



Therefore a cooling tower performance test was conducted and the cooling tower was inspected. During inspection it was found that some of the sectors were leaking water and therefore condensate was being lost in the cooling tower. This would lead to an added make up water loss.

Figure 45 Leaking sectors on Unit 6's cooling tower

This would have an influence on the performance of the cooling tower.

4.1.4.1 *Cooling tower performance test*

A cooling tower performance test was conducted (as described in section 2.2.6.7.10) using the cooling tower performance curve that was designed for this unit.

For the analytical inspection, the DCS parameters were trended for a stable part of the month of April 2014, and a snapshot point was selected at a steady state conditions.

Table 4 U6 condenser A and B side breakdown

	Condenser A		**Condenser B**	
CW_{in} (ºC)	*60PAB31CT001.DACA.PV*	47.21	*60PAB30CT001.DACA.PV*	47.32
CW_{out} (ºC)	*60PAB11CT001.DACA.PV*	66.57	*60PAB10CT001.DACA.PV*	61.32

Table 5 U6 parameters snapshot for April 2014 at steady state conditions

Parameter	KKS number	Value
Load (MW)	*60MKA10CE402.DACA.PV*	170
Average CW_{in} (ºC)	*A and B*	47.26
Average CW_{out}(ºC)	*A and B*	63.95
P_{sat} of vacuum (kPa)	*60MAG10CP901.DACA.PV*	29.93
T_{neck} (ºC)	*60MAG10CT001.DACA.PV*	67.74
$T_{condensate}$ (ºC)	*60MAG10CT002.DACA.PV*	58.80
Ambient T (ºC)	*60URC01CT001.DACA.PV*	14.45
CW flow (m^3/s)	*60PAB20CF001.DACA.PV*	4
T_{sat} of vacuum (ºC)		69.02
T_{sat} - T_{cond} (ºC)		10.22
T_{cond} - $T_{CW\ OUT}$ (ºC)		-5.14
$T_{CW\ IN}$ - $T_{CW\ OUT}$ (ºC)		16.68

The results from the cooling tower performance test are recorded in below. The breakdown of the exact methodology followed can be found in the Appendix (section 7).

Table 6 Tabulated results of the cooling tower performance test

Ambient T (°C)	14.5
Temp Rise (°C)	17
Target CW inlet from graph for 100% flow (°C)	57.5
Avg flowrate in ton/h	14400
Design flowrate (ton/h)	18000
% flowrate	90
Correction factor for T1 from graph	-3.12
Target T1 (°C)	54.38
Actual T1 °C)	63.95
Variance	9.57

A variance between the actual and the target cooling tower outlet temperature of 9.57°C was found. This indicated that the tower was not performing according to design and it was symptomatic of heat transfer prohibition. Therefore, it was suspected that the tower was dirty.

A visual inspection confirmed that this was the case. The dirt on the fins was deterring the cooling tower's ability to cool the water which led to warmer water entering the condenser and thus the condenser could not condense steam as efficiently as it was designed to which also contributed to the backpressure problems. The cooling tower performance curve that was used is also attached in the Appendix (Figure 50).

Since the condenser was also indicating air ingress from the condenser efficiency analysis, it was decided to follow the logic diagram (point 4.1) in order to determine whether the indication was correct. Unit 6 operated with water jet air ejectors instead of the traditional steam jet air ejectors and at the stage of investigation, all three air ejectors were in service and they were not able to remove all the air from the condenser. The operation of the air ejectors was to be investigated, however due to lack of sampling points; this investigation could not be performed at the time. Therefore (from the high condensate depression indication that was depicted in the condenser efficiency analysis) it was assumed that air was leaking into the condenser.

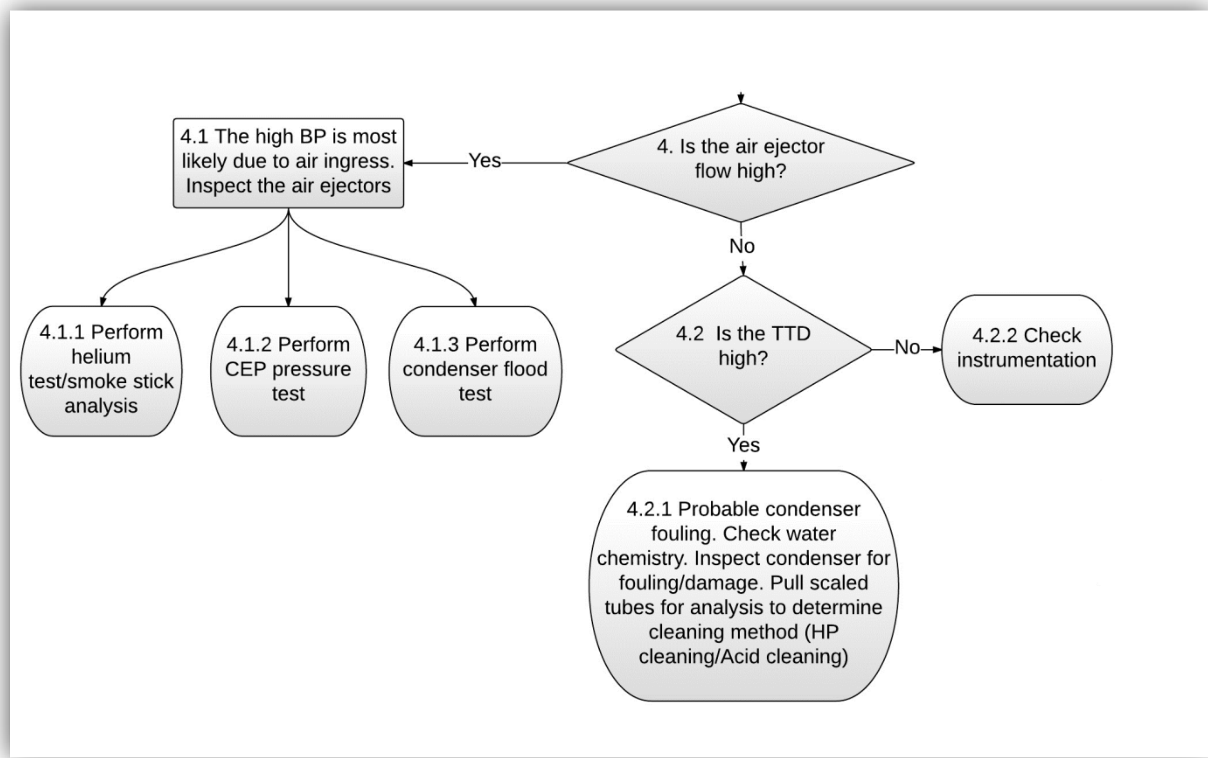


Figure 46 Logic diagram extract indicating possible air ingress

The logic diagram recommended that three tests be performed in order to determine possible sources of air ingress. All three of these tests were performed on this unit.

1. A helium test (described in section 2.2.6.7.5 was conducted on the 12th of February 2014 and numerous areas were found to be leaking a lot of air into the condenser. Notifications were loaded to the maintenance department to fix these leaks once an outage was granted.
2. On the 4th of June, a Condensate Extraction Pump (CEP) pressure test was conducted on CEP B on this unit for further leak detection analysis as described in section 2.2.6.7.7. Defects were found during this test and notifications were loaded to the maintenance department to fix these leaks once an outage was granted. At this stage, CEP A was in service which allowed for the opportunity to pressurise CEP B. Since CEP B was on standby during this period, air could have been entering the condenser through leaks on this CEP while CEP A was operating. A decision was made to changeover the CEP's and to observe the effect of the changeover on the dissolved oxygen (DO) levels. Three parameters were trended from 11pm until 3am, namely CEP pump A current, CEP pump B current and the CEP DO's. A graph displaying the three trended parameters can be seen below:

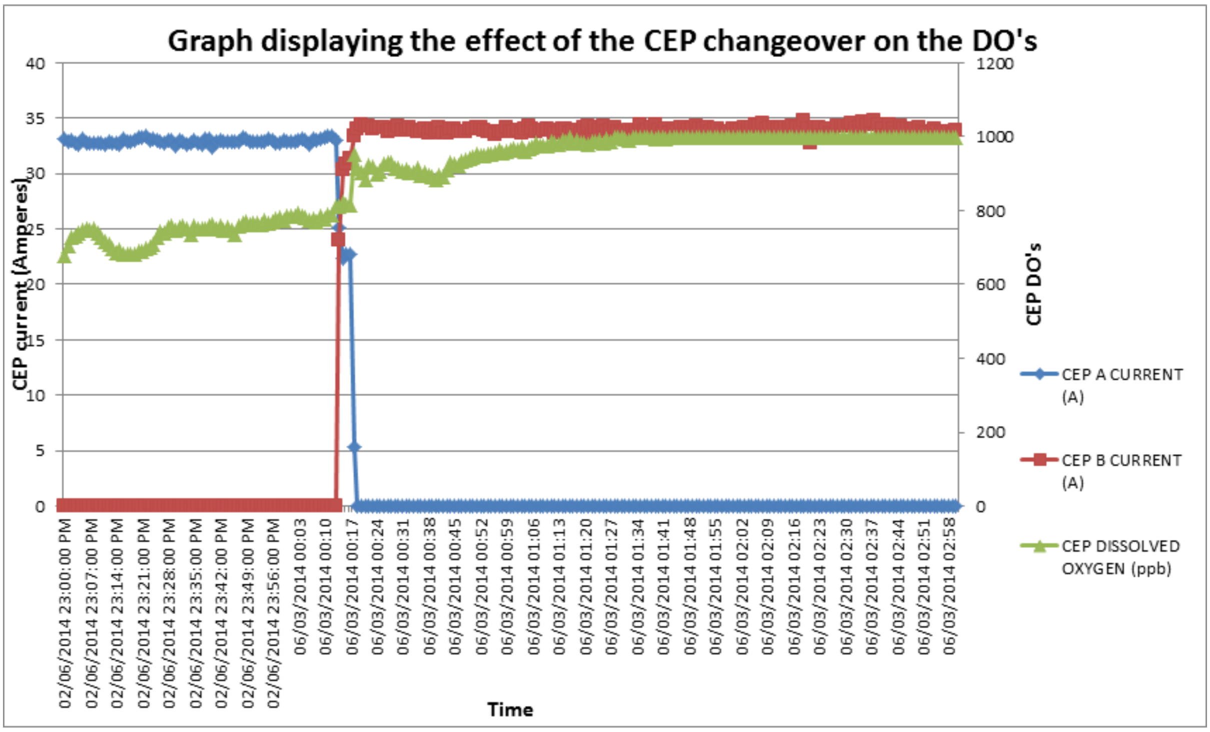


Figure 47 Graph displaying the effect of the CEP changeover on the DO levels

From the graph, one can see that the DO's increase as soon as CEP B is put in service which validated that air was entering the condenser through that CEP.

3. A flood test was done on the 17th of July, however no additional defects were found.

No further tests were done on this unit at the time.

4.2 **Financial implications and effect of condenser backpressure on thermal efficiency**

The financial implications of the high backpressure were investigated. Table 7 contains an extract of the monthly STEP report for the month of August 2014:

Table 7 STEP report extract for the month of August 2014

	Turbine Deficiency & Cost Analysis	Unit	Value
IDS072	Target Main Steam Temperature	°C	538.00
IDS073	Actual Mean Main Steam Temperature	°C	537.11
IDS074	Step Factor Main Steam Temperature Loss	%	0.03
IDS076	Cost Of Main Steam Temperature Loss	R	R 71 938.37
IDS077	Target Reheat Steam Temperature	°C	0
IDS078	Actual Mean Reheat Steam Temperature	°C	0
IDS079	Step Factor Reheat Steam Temperature Loss	%	0
IDS081	Cost Of Reheat Steam Temperature Loss	R	R 0.00
IDS082	Target Condenser Backpressure	kPa	8.251
IDS083	Actual Mean Condenser Backpressure	kPa	13.888
IDS084	Step Factor Backpressure Loss	%	6.021
IDS086	**Cost Of Backpressure Loss**	**R**	**R 15 512 687.57**
IDS087	Estimated Turbine Deterioration	%	0.192
IDS088	Step Factor Deterioration Loss	%	0.192
IDS090	Cost Of Deterioration Loss	R	R 384 658.94
IDS091	Target Final Feed Temperature	°C	211.70
IDS092	Actual Mean Final Feed Temperature	°C	201.04
IDS093	Step Factor Final Feed Temperature Loss	%	0.447
IDS095	Cost Of Final Feed Temperature Loss	R	R 747 969.93
IDS096	Total Turbine Step Factor Loss	%	6.69
IDS098	Cost Of Total Turbine Loss	R	R 16 717 254.81
IDS099	Target Make-Up Water Loss	%	1.171
IDS100	Actual Make-Up Water Loss	%	5.424
IDS101	Step Factor Make-Up Loss	%	3.862
IDS103	Cost Of Make-Up Loss	R	R 7 999 505.86
IDS104	Target Works Units On Load	%	9.312
IDS105	Actual Works Units On Load	%	9.524
IDS106	Step Factor Works Unit Loss On Load	%	0.212
IDS108	Cost Of Works Units On Load Loss	R	R 601 017.68
IDS109	Works Units Not On Load	%	0.001
IDS113	Total Accounted For Station Step Loss	%	11.56
IDS115	Cost Of Losses Accounted For	R	R 26 822 901.48
IDS116	Station Step Factor Loss Equivalent	%	19.111
IDS118	Cost Of Step Loss Equivalent	R	R 38 057 635.34
IDS119	Unaccounted Step Losses	%	7.55
IDS121	Cost Of Unaccounted Losses	R	R 11 234 733.87

The condenser backpressure loss for the month of August 2014 cost Eskom an amount of R15 512 687, 57. The condenser backpressure was contributing to 92.8% of the losses that were being experienced on the turbine side. It was by far the highest loss in comparison to the other losses.

The overall efficiency for Grootvlei Power Station for the month of August 2014 was **26.33%** and compared to the design efficiency of 32.9%, the thermal efficiency was very poor.

The effect of the condenser backpressure loss on thermal efficiency was calculated as described in section 3.4.3.1 and the results thereof are summarised below:

Table 8 Analysis of effect of condenser backpressure on thermal efficiency

Coal price	R 612.85
Total coal burned (ton)	346 395
USO (GWh)	511.51
CV (MJ/kg)	20.19
	Condenser BP
Monetary loss	R 15 512 687.57
Over-burned coal (ton)	25312.3
Coal that would have been burned if this loss was resolved	321082.6
Thermal efficiency if the backpressure losses were eliminated (%)	**28.4**

Therefore, if the condenser backpressure loss had been eliminated for the month of August 2014, the **efficiency would have been 28.4%, which is approximately a 2% increase in thermal efficiency.**

4.3 Cost benefit analysis results

A cost benefit was conducted and the results of the calculations applied are recorded below. A breakdown of the applied calculations is included in Appendix 0.

4.3.1 Cost of running the unit with fouled tubes per month

The cost of runn9ing a unit with fouled tubes per month was calculated

Table 9 Cost benefit analysis calculations summary

<u>Plant Data</u>

Description	Value	Units
Boiler steam flow at full load	214.2	kg/s
Boiler efficiency on standard fuel	88.7	%
Calorific Value of coal	19.29	MJ/kg
Cost of coal	418.58	R/ton
Mill rating at full load	25	Ton/hr
Load at the time that the cost benefit analysis was conducted	200	MW (100% load)

Condenser acceptance test results (at 200MW)

Description	Value	Units
Cooling water inlet temperature	20.9	°C
Cooling water outlet temperature	33.7	°C
Condensate temperature	36	°C
Saturation temperature	35.3	°C
Condenser backpressure	5.7	kPa

Condenser performance test results (12 June 2014, 6 pm, at 200MW)

Description	Value	Units
Cooling water inlet temperature	20.5	°C
Cooling water outlet temperature	42.7	°C
Condensate temperature	59.1	°C
Saturation temperature	35.3	°C
Condenser backpressure	19.7	kPa

Coal consumption for the boiler

Description	Value	Units
Coal consumption per mill for the boiler	6.95	kg/s
Coal consumption for 5 mills for the boiler	34.7	kg/s
Coal consumption in gigajoules	0.67	GJ/s

Correction factors (from correction curves)

	Polynomial result	Unit
Acceptance test correction factor	1.001166991	
Performance test correction factor	1.134201279	
Correction factor variance	0.133034288	
Backpressure variance	14	kPa
Heat change correction factor	0.009502449	/kPa

Extra heat consumption due to fouled tubes

Description	Value	Units
Extra heat at 100% load with fouled tubes	0.089105257	GJ/s
Extra heat consumption at 100% load	0.001004569	GJ/s
Cost of extra heat consumption	42.04	c/s

Cost of extra heat consumption per hour	R 1 513.77	/hr
Cost of extra heat consumption per day	R 36 330.55	/day
Cost of extra heat consumption per month	**R 1 089 916.38**	**/month**

Over burnt coal

Description	Value	Units
Amount of coal over burnt	R 2 603.84	ton/month

Therefore, to run Unit 3 with the tubes in a fouled condition was costing approximately R1 million a month. This is without taking the load losses into consideration. Load losses due to high backpressure were common on this unit and ranged between 10MW load losses to 50MW load losses. An assumed average load loss of 15MW was taken for the purpose of these calculations. The cost of losing 1 MW was R292/hr. at the time of writing. It was noticed that during the warmer part of the day, the backpressure would rise and the unit would not be able to reach full load conditions. Therefore, for the purposes of these calculations, it was assumed that the unit would run with a 15MW load loss for 9 hours a day. Therefore, the cost of the MW loss could be calculated to be:

$$Approximate\ ost\ of\ load\ losses\ per\ day$$
$$= Cost\ of\ MW\ loss\ \times amount\ of\ MW\ lost\ \times hours\ of\ load\ loss$$

$$= \ R292\ /hr\ \times\ 15\ MW\ \ \times 9\ hours$$

$= R$ 39 420 per day

The load losses were costing an additional R 1 182 600 per month. Therefore the total cost of running with fouled conditions was R 2 272 516.38 per month.

Hereafter the cost of cleaning the condenser tubes was calculated.

4.3.2 Cost of cleaning the fouled tubes

A summary of the calculations that were done in order to determine the total cost of condenser tube cleaning can be found below:

Cost to have a unit down per hour		
Cost of MW loss	R 292.00	/MW
Cost of losing 200 MW (unit off load)	R 58 400.00	/hr
Cost to have a unit down for condenser cleaning		

Days needed to clean a condenser	8	days
Cost to have a unit down for 8 days	R 11 212 800.00	
Fuel oil cost to shut down and light up a unit		
Cost of fuel oil	R 6.00	/liter
Burner usage	492	liter/hr
Amount of burners	4	burners/mill
Amount of burners for 5 mills in service	20	burners
Typical amount of hours of burner usage during shut down	1	hr
Typical amount of hours of burner usage during light up	6	hrs
Total fuel oil cost to shut down and light up a unit	R 413 280.00	
Approximate cost of cleaning a condenser	R 450 000.00	
Total cost for condenser cleaning	**R 12 076 080.00**	

The cost of running with a fouled condenser was then compared to the cost of cleaning in order to determine when Grootvlei would make up the large amount that it costs to clean a condenser.

4.3.3 Running with a fouled condenser versus cleaning

Table 10 Comparison between running with fouling and cleaning the condenser

Cost of running with fouled tubes for:				**Cost of cleaning fouled tubes**
Month	**1**	R 2 272 516.38		
	2	R 4 545 032.76		
	3	R 6 817 549.14		
	4	R 9 090 065.53		
	5	R 11 362 581.91		
	6	**R 13 635 098.29**		**R 12 076 080.00**

Table 10 indicates that after **6 months**, the cost of cleaning the condenser will be covered by the gain from running with a clean condenser.

4.3.4 **Return on investment**

The return on investment (ROI) can be derived by:

$$ROI = \frac{Money\ gained}{Cost\ of\ condenser\ cleaning} \times 100\%$$

$$= \frac{R\ 2\ 272\ 516.38}{R\ 13\ 635\ 098.29} \times 100\%$$

Therefore the ROI is calculated to be 16.6%.

4.3.5 **Annual savings on load losses**

Since it cannot be certain that the condenser will not become fouled up again, it was decided to determine the amount that can be saved by reducing the load losses only. Hence, if the condenser is cleaned and assuming the load losses are reduced to 10% due to the cleaning, then the amount that Eskom can save (90% savings) annually amounts to:

$$Amount\ saved\ annually\ due\ to\ reduced\ load\ losses$$

$$= 0.9\ \times Cost\ of\ load\ losses\ per\ month\ \times 12\ Months$$

$$= 0.9 \times R1\ 182\ 600\ \times 12\ months$$

$$= R\ 12\ 772\ 080$$

Typically, if the water chemistry is kept under precipitation potentials, then the condenser should be cleaned during scheduled overhauls which occur every three years at Eskom. Therefore, if it is assumed that after three years, the condenser will be fouled up to the same point that it is at the moment. Therefore, if it is assumed that the fouling factor is linear and fixed, the amount of fouling per month can be determined:

$$Fouling\ per\ month\ (\%) = \frac{Cost\ of\ fouling\big/_{amount\ of\ month}}{Cost\ of\ fouling} \times 100\%$$

$$= \frac{R\ 2\ 272\ 516.38\big/_{36\ months}}{R\ 2\ 272\ 516.38} \times 100\%$$

$$= 2.78\ \%$$

Figure 48 represents the condenser deterioration due to fouling over a three year time period assuming a linear deterioration.

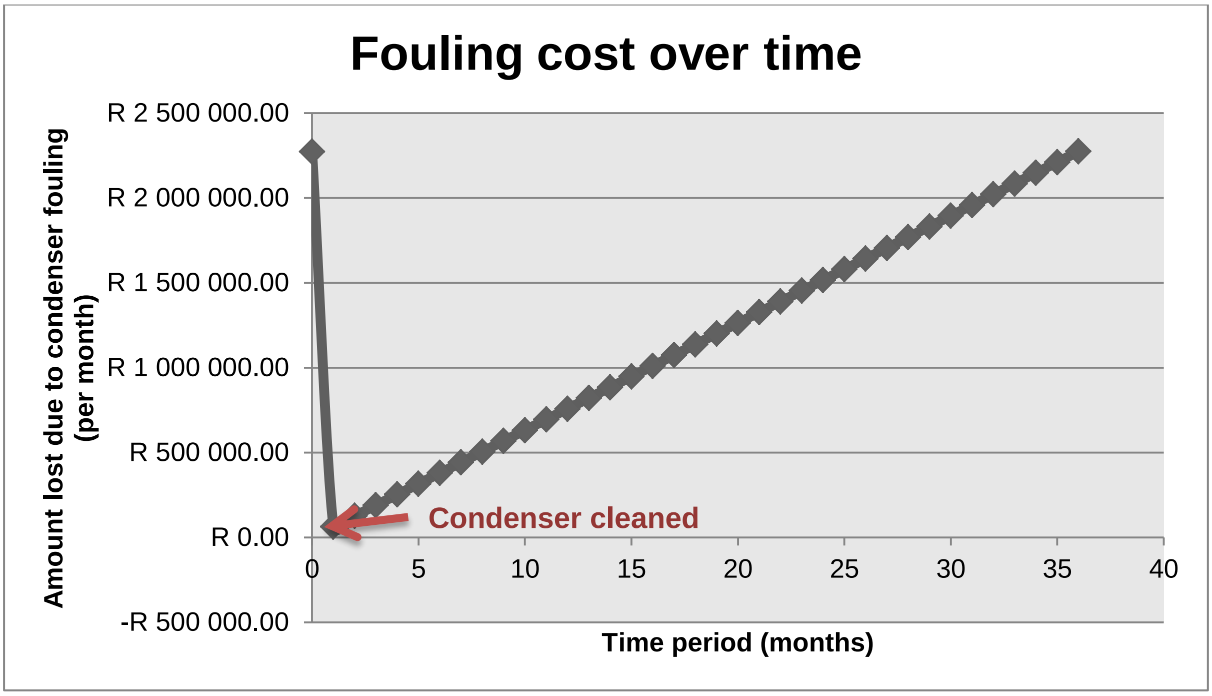


Figure 48 Condenser deterioration due to fouling over a three year times period

The return on investment will decline as the fouling inside the condenser increases up until the point is reached where the condenser will need to be cleaned again. This point will be determined by monitoring the TTD of the condenser.

Chapter 5 - Findings and deliverables summary

The deliverables that were stated in section 1.2.1.2 are recapped below:

5.1 Determine what was causing high condenser backpressure

On Unit 3 it was determined that the damaged cooling tower, air ingress, low CW flow and condenser fouling were all attributing to the poor backpressure. However, from the analyses it was determined that the fouled tubes were the main contributor to this unit's backpressure losses. (See section 4.1.2)

On Unit 6 it was determined that along with ambient temperature conditions, air ingress was playing a role in the backpressure losses. (See section 4.1.4)

5.2 Propose resolutions for the identified problem areas with the aim of minimizing the backpressure losses

Along with having known defects maintained, the main resolution found for Unit 3 was to have the condenser cleaned properly with HP cleaning.

To resolve the backpressure losses on Unit 6, the found defects would have to be fixed to prevent air ingress.

5.3 Quantify the financial effect of minimizing the backpressure losses on Grootvlei Power Station

The financial effect of the condenser backpressure losses at the time of writing was found to be approximately R15 million a month. Therefore, to minimize these losses would mean a gain of approximately R15 million a month. (See section 4.2)

5.4 Quantify the effect of backpressure losses on the thermal efficiency

It was determined that the backpressure loss was contributing more than 2% to the overall thermal efficiency loss. (See section 4.2)

5.5 Produce a cost benefit analysis in order to motivate for corrective actions to be taken

A cost benefit analysis was done in order to motivate for the corrective actions to be taken and it was found that by cleaning the condenser, Grootvlei Power Station would save a minimum of R12 million a year just from reduced load losses. Further analyses indicated that the return on investment of the cleaning of the condenser would be 16.6%. (See section 4.3)

Chapter 6 – Comments and recommendations

This chapter comprises of some comments from the author on the findings and the stumbling blocks encountered. Finally the recommendations are listed that will aid in reducing the backpressure loss that was being experienced.

6.1 Comments and basic discussion

In the problem statement, it was stated that Grootvlei Power Station was experiencing thermal efficiency problems. This investigation indicates that, should the recommendations be implemented, the thermal efficiency will increase by at least 2%.

There are many obstacles that were being experienced at this power station in 2014. Herewith follows a discussion of these obstacles.

Due to a high staff turnover, it was perceived that many of the staff did not have the required knowledge and experience to be able to identify contributory causes to the backpressure loss. Eskom had been cutting costs at the time and many contractors were retrenched, this added to the lack of knowledge problem since many of the contractors that were released had been at the power station even before it was mothballed. There is significant knowledge that gets lost when employees are let go and it was found that proper handover was imperative. It seemed that some documentation had also gotten lost during the mothballed years and therefore many tests could not be compared to design or acceptance test values. Even though Eskom offered in-house training at Eskom's Academy of Learning in Midrand, Gauteng; many of the facilitators were contractors and had been retrenched without replacement.

Eskom was experiencing financial problems at the time of investigation and this led to many stumbling blocks concerning the backpressure losses. Time was spent investigating the issue, but outage time and money were not always available for the reparation of plant defects. Larger defects generally had more detrimental effects on the backpressure, however these defects were usually much more costly to repair and thus were not always granted the money for repair (for example: one of the wet cooling towers was in desperate need of refurbishment).

Another stumbling block frequently experienced throughout the duration of this research was the lack of time. Most of the defects that could be repaired within budget were defects that needed time to be repaired. However, the general outage time that was usually given was three days (commonly over a weekend). To clean a condenser properly with HP cleaning, a minimum of 8 days is needed. Therefore any attempt to clean Unit 3's condensers within three days failed and the unit would come back on load with a dirty condenser and a high backpressure.

There appeared to be a maintenance challenge at Grootvlei Power Station. This station was old and needed perhaps even more maintenance than some of the newer stations would need. Even though an excellent system was implemented which dictated regular maintenance (reliability based optimisation), it did not seem to be assisting in keeping the station maintained. Another factor that would influence this was that these units were being run till they failed due to the rising electricity crisis, especially during the winter months. This caused more damage to the units which directly affected the amount of time that the units were able to be on load without tripping. Therefore this has a direct effect on the reliability as well as the availability of the units.

Grootvlei Power Station was one of the smallest stations in Eskom at the time of writing, with units of a maximum rating of 200MW, and therefore other stations often gained preference above Grootvlei when decisions were made regarding outage permissions. Most of the time this station was only granted extended outages when the units were so critically damaged that they could not run without certain repairs. These are called opportunity outages and this is when Grootvlei benefited the most concerning plant health; however this played a role when National Control had to decide which stations were allowed to bring a unit down for maintenance. Since Grootvlei had a reputation for units not coming on load at their appointed times, the units were not always considered safe to bring down for maintenance. Another factor that played a role was that spares were not always readily available and this caused long delays for machinery to be repaired.

Eskom had implemented a variety of procedures that need to be followed before any activity could take place. This is incredibly beneficial for documentation purposes as well as safety and adherence to laws. However many of these procedures are not always necessary for small activities, yet they were still required to be enforced. From the author’s perspective as an employee, the issue of the commissioning of sampling points that required an Engineering Change Management procedure was an example of unnecessary procedures that were required. It appeared that the requirement for a procedure to be in place before an activity may commence was extremely demotivating for the staff, and therefore many tended to opt to rather do nothing about a certain defect at hand.

6.2 **Recommendations**

Herewith are the recommendations to reduce the backpressure problems and therefore regain efficiency.

6.2.1 **HP cleaning**

HP cleaning may need to be done on all the wet cooled units. Since they all use the same raw water, the sludge that was found inside Unit 3 would most likely also be found in the other units and therefore inspection of these condensers is necessary.

This cleaning needs to be done properly and the correct amount of time needs to be allowed for this to occur. HP cleaning has the ability to remove hard scale as well and it is possible for pitting to be present beneath the scale. Therefore it is imperative that a flood test is done after HP cleaning has taken place in order to indicate whether tubes have started to leak after the cleaning.

However, care should be taken to keep the water chemistry under control since it plays a role in the fouling conditions of a unit. If the water chemistry is not kept under precipitation conditions, then cleaning the condenser will be in vain. Extra investigation is recommended to be conducted in order to determine the challenges surrounding the water chemistry at Grootvlei Power Station

From the cost benefit analysis, it can be seen that it is highly recommended to clean the condenser on Unit 3. The return on investment will be 18.8% provided the water chemistry is kept under control. Hence, the money to clean the condenser will be made up within a six month period.

6.2.1.1 ***Taprogge system***

A cleaning system exists that can be installed on a surface condenser that cleans the condenser tubes whilst the condenser is running. This is called a Taprogge system in which cleaning balls are constantly flowing through the condenser tubes, reducing scaling potentials. It is recommended that further investigation is done to determine the feasibility of installing such a system on Grootvlei Power Station’s condensers.

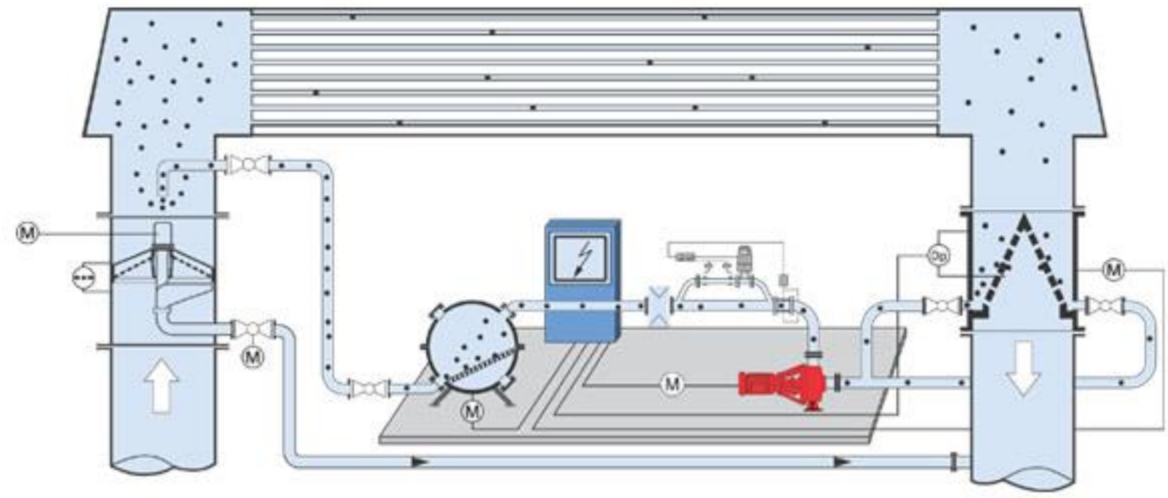


Figure 49 Schematic of a typical Taprogge system (http://www.taprogge.de/products-and-services/in-ta-ctR/monitoring/filteroptimizer/index.htm, 2014)

6.2.2 **Training**

Extensive training needs to be undergone by every new employee in which they are educated suitably in their respective job descriptions. Specialists should be employed for this purpose to share their knowledge and understanding before they leave Eskom.

6.2.3 **Outage times**

Outage times are one of the main problems at Grootvlei Power Station. It is recommended that outages are given that allow sufficient time for all the defects to be repaired.

6.2.4 **Maintenance**

It is recommended that the maintenance strategies are revised and the maintenance department restructured in order to enable the department to cope with the high volumes of work. This department is frequently shorthanded and therefore many problems arise, especially during an outage.

6.2.5 **Smoke sticks**

Smaller leaks that the helium sniffer doesn’t always pick up can be found by the use of smoke sticks as discussed in section 2.2.6.7.6.

6.2.6 **Wet cooling tower end caps**

During the winter months, it would be beneficial to open some of the end caps inside the wet cooling tower to reduce the possibility of water freezing. This will ensure that the cooling tower packing does not get destroyed during the winter.

6.2.7 **Sampling points**

In order to accurately perform assessments and tests, it is beneficial to have sampling points. Therefore it is recommended to install sampling points where they are required. This will provide for more accurate analysis techniques.

6.2.8 **Defects**

All defects need to be planned for to be repaired. If defects can only be repaired on outage, the maintenance department needs to ensure that they have the required spares on hand when that outage is allowed.

Therefore, if the above mentioned problem areas are corrected, the condenser backpressure loss will decrease and the condensers will transfer heat more efficiently. This will lead to financial gains for Grootvlei Power Station as well as efficiency gains, plant reliability and availability gains.

Bibliography

ALIAGA, M. & Gunderson, B. 2000. Interactive Statistics, 3rd Edition. Prentice Hall

ANON. 2005. *Measuring Boiler Efficiency*. Appendix 1

BLAKSTAD, O. 2008. Experimental Research. https://explorable.com/experimental-research Date of Access: 26 October 2014

Chizengeni, S. 1980. *Circulating water system and H_2 plant – Module 1: CW system. Student manual.* 34 p

Dowler, K.; Ferguson, D.W.; Hernen, B.; Hutchinson, G.P.; Leadbeater, A.A.; Scott, E.C.; Street, G.; Wright, V.J. 1981. *Modern Power Station Practice - volume 7. Operation and Efficiency.* Pergamon Press. 667 p.

Gibbard, I. & Terranova, T. 2010. Turbine Condensers – Maximising Performance and Reliability for Eskom, South Africa (Training Course)

Hartnady, C.J.H. 2010. South Africa's diminishing coal reserves. http://www.sajs.co.za/sites/default/files/publications/pdf/369-2707-5-PB.pdf Date of access: 21 May 2014

Intek. 2011. *Common Condenser Problems*. http://www.intekflow.com/ Date of access: 15 April 2014

Kaupp, A. *Energy Manager Training Lecture Notes.* Date of access: 4 April 2014

Leinster, M.G., Khan, M.M.K., Mahamud, R. & Rasul, M.G. 2013. Exergy Analysis and Efficiency Improvement of a Coal Fired Thermal Power Plant in Queensland. http://www.intechopen.com/books/thermal-power-plants-advanced-applications/exergy-analysis-and-efficiency-improvement-of-a-coal-fired-thermal-power-plant-in-queensland Date of access: 21 May 2014

McNaught, J.M. 2011. Condensers. http://www.thermopedia.com/content/654/ Date of Access: 10 June 2014

Moolman, P.N. 1980. *Thermal Efficiency Monitoring Program – Training Manual for STEP*

Moolman, P.N. 1999. *Thermal Power Plant Techniques course*

Moolman, P.N. 2014. Performance tests and engineering. 20 June, Grootvlei Power Station

Rathore, M.M. 2010. Thermal Engineering. Tata McGraw Hill Education Private Ltd. p 764

Roth, E. 2005. Why thermal power plants have a relatively low efficiency. https://docs.google.com/file/d/0BzBU0gQlsdocYmI1NDkyMDctY2RmYy00YzY0LTgwYmYtY2RjZGFhN2U1ZDk2/edit?hl=en_GB# Date of access: 21 May 2014

Schechter, J. 2013. Deductive Reasoning. http://fmri.neuro.brown.edu/Departments/Philosophy/onlinepapers/schechter/DeductiveReasoning.pdf Date of Access: 26 Oct 2014

Test and Measurement Instruments CC. 2014. Digital Anemometers. www.instrumentsgroup.co.za/index_files/anemometers.htm, Date of Access: 5 October 2014

The Chimney Balloon. 2014. Smoke Pencil Puffer Stick. www.smokepencil.com, Date of Access: 5 March 2014

Appendix

7.1 Cooling tower performance test

The methodology for the cooling tower performance test can be found in this section along with the cooling tower performance curve (Figure 50).

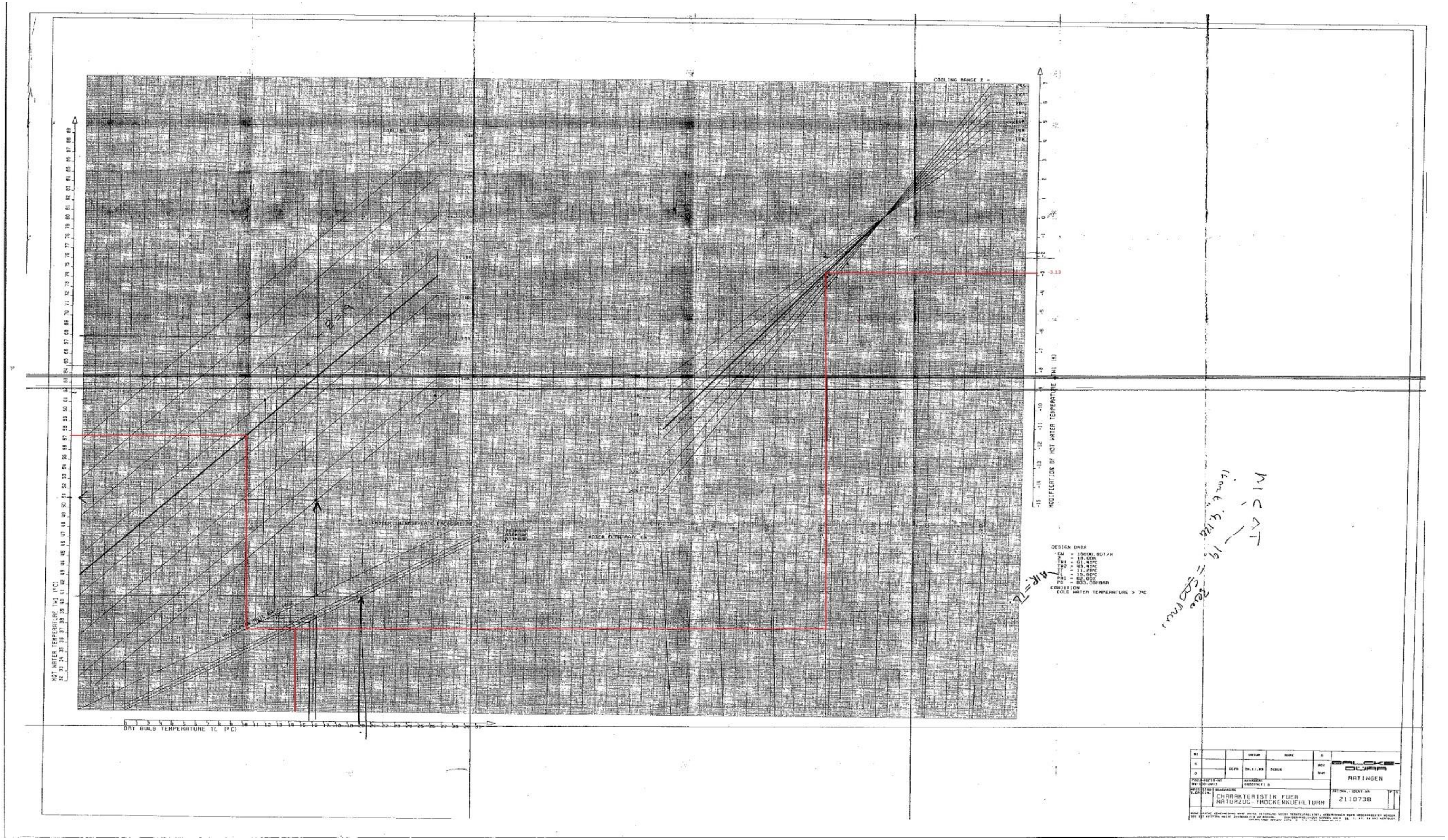


Figure 50 Unit 6 cooling tower performance curve

On the performance graph, one uses the ambient conditions (temperature and pressure) along with the temperature rise (±17°C) in order to determine the target cooling tower inlet temperature. However, this value was calculated for if the water flowrate was at 100% which was not the case at the time of the test and therefore a correction factor was necessary to be incorporated. The target cooling tower inlet temperature read from the graph for 100% water flow was 57.5°C.

The right side of the cooling tower performance graph was used to obtain the correction factor by first calculating the percentage water flow that was being experienced at the time. From the trended value, the average CW flow to the tower was 4 m^3/s for the month. The design flow was 16000ton/h. 4m^3/s was converted to ton/h and the relation to the design was 90% flow. Using this data and the temperature rise, the correction factor was found to be -3.12°C.

Therefore the target temperature at which water should have been entering the cooling tower was:

$$57.5^{\circ}C - 3.12^{\circ}C = 54.38\ ^{\circ}\mathrm{C}$$

However, the temperature of the water entering the cooling tower was found to be 63.95 °C which lead to a difference of 9.57°C. This means that the tower is not performing according to design and it was indicative of heat transfer prohibition.

7.2 Cost benefit analysis methodology and calculations

A cost benefit analysis was done in order to motivate for the cleaning of unit 3 condenser to be done.

7.2.1 Cost of running Unit 3 with fouled conditions

Firstly the cost of running a unit at 100% load, with the condenser tubes fouled up was calculated. The data was chosen for a day when the unit was running at full load (MCR) whilst being fouled up, this was on the 12th of June 2014 at 6pm, ambient temperature was 13 °C.

Table 11 Plant data used for cost benefit calculation (June 2014)

Description	Value	Unit
Boiler steam flow at full load (MCR)	214.2	Kg/s

Boiler efficiency on standard fuel	88.7	%
CV of coal	19.29	MJ/kg
Cost of coal	418.58	R/ton
Mill rating at full load	25	Ton/hr.
5 mills required for full load		
Load at test	200	MW (100% load)

First, the cost of sustaining the fouled tube conditions at full load was calculated:

7.2.1.1 ***Coal consumption per second for the boiler***

$$Coal\ consumption\ (kg/s) = \frac{25\frac{ton}{hr} \times 1000\frac{kg}{ton}}{3600\frac{kg}{s}}$$

$$= 6.95\ kg/s$$

Therefore, for 5 mills:

$$Coal\ consumption\ (kg/s) = 6.95\ kg/s \times 5$$

$$= 34.7\ kg/s$$

7.2.1.2 ***Coal consumption per second in gigajoule (GJ/s)***

$$Coal\ consumption\ (GJ/s) = CV_{coal} \times coal\ flow$$

$$= \frac{19.29\frac{MJ}{kg} \times 34.7\frac{kg}{s}}{1000\frac{MJ}{GJ}}$$

$$= 0.67\ GJ/s$$

7.2.1.3 ***Condenser acceptance test results (200MW)***

The acceptance test results for Unit 3 were drawn from the records:.

Table 12 Acceptance test data for Unit 3

Description	Value
Cooling water inlet temperature (°C)	20.9
Cooling water outlet temperature (°C)	33.7
Condensate temperature (°C)	36
Saturation temperature (°C)	35.3
Condenser backpressure (kPa)	5.7

7.2.1.4 *Condenser performance test results*

The running data of Unit 3 was drawn for the 12th of June 2014 at 6pm (full load running conditions)

Table 13 Unit 3 running data

Description	Value
Cooling water inlet temperature (°C)	20.5
Cooling water inlet temperature (°C)	42.7
Condensate temperature (°C)	59.1
Saturation temperature (°C)	35.3
Condenser backpressure (kPa)	19.7

A correction factor was calculated for the acceptance test backpressure and the actual backpressure using the correction curve equation taken from the STEP calculations page.

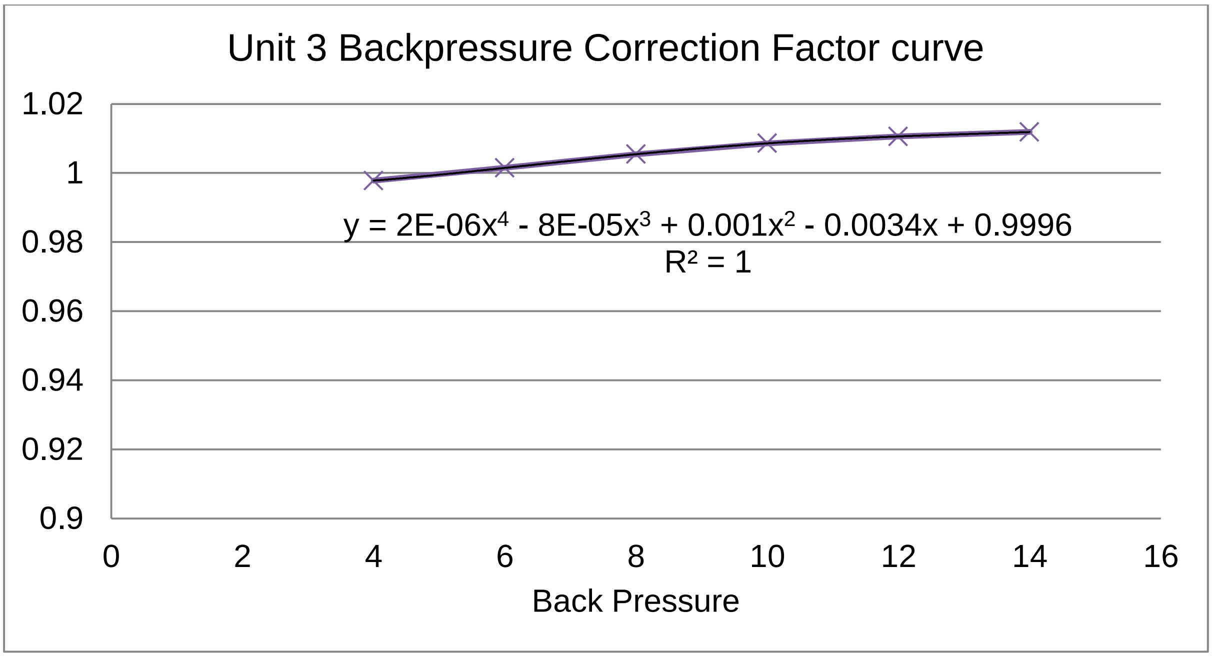


Figure 51 Unit 3 correction factor curve for backpressure at 100 % as taken from STEP

7.2.1.5 ***Acceptance test (AT) backpressure = 5.7 kPa***

The following equation is the STEP equation that was used to calculate the correction factor at 100% load:

$$Acceptance\ test\ correction\ factor\ (ATCF) = 0.22809\ exp^{-6}(\mathrm{BP}^4) - 0.79exp^{-4}(BP^3) + 0.1018exp^{-2}(BP^2) - 0.3379exp^{-2}(BP) + 0.999575$$

Substituting the backpressure into the equation:

$$ATCF = 0.22809\ \ exp^{-6}(5.7^4) - 0.79exp^{-4}(5.7^3) + 0.1018exp^{-2}(5.7^2) - 0.3379exp^{-2}(5.7) + 0.999575$$

$$= 1.001166991$$

7.2.1.6 ***Actual backpressure = 19.7***

$$Actual\ correction\ factor\ ACF = 0.22809\ exp^{-6}(\mathrm{BP}^4) - 0.79exp^{-4}(BP^3) + 0.1018exp^{-2}(BP^2) - 0.3379exp^{-2}(BP) + 0.999575$$

Substituting the backpressure into the equation:

$$ACF = 0.22809\,exp^{-6}(19.7^4) - 0.79exp^{-4}(19.7^3) + 0.1018exp^{-2}(19.7^2) - 0.3379exp^{-2}(19.7) + 0.999575$$

$$= 1.134201279$$

7.2.1.7 ***Correction factor variance***

The variance in the correction factors was then calculated as this is the correction factor that should be applied:

$$Correction\ factor\ variance = ACF - ATCF$$

$$= 1.134201279 - 1.001166991$$

$$= 0.133034288$$

7.2.1.8 ***Backpressure variance***

The variance in backpressure was calculated:

$$BP\ variance = Actual\ BP - Acceptance\ test\ BP$$

$$= 19.7\ kPa - 5.7\ kPa$$

$$= 14\ kPa$$

7.2.1.9 ***Heat change correction factor***

The correction factor for the change in heat was calculated:

$$Correction\ factor\ (Heat\ change) = \frac{Correction\ factor\ variance}{BP\ variance}$$

$$= \frac{0.133034288}{14\ kPa}$$

$$= 0.009502449\ kPa^{-1}$$

7.2.1.10 ***Extra heat at 100% load with fouled tubes***

The extra heat for 100% load was then calculated

$$Extra\ heat = Full\ load\ heat\ consumption\ \times Correction\ factor\ variance$$

$$= 0.67\frac{GJ}{s} * 0.133034288$$

$$= 0.089105257\ GJ/s$$

7.2.1.11 ***Extra heat consumption at 100% load***

The extra heat consumption was calculated:

$$Consumption = \frac{Extra\ heat}{Boiler\ efficiency}$$

$$= \frac{0.089105257}{88.7}\ (GJ/s)$$

$$= 0.001004569\ GJ/s$$

7.2.1.12 ***Cost of extra heat consumption***

The cost of the extra heat consumption was calculated:

$$Cost\ of\ extra\ heat\ consumption = Consumption\ \times cost\ of\ coal$$

$$= 0.001004569\ \frac{GJ}{s} \times\ 41858\ \mathrm{c}$$

$$=\ 42.05\ \ c$$

The cost per hour was calculated:

$$Cost\ in\ Rand\ per\ hour = 42.05c * 3600/100$$

$$= R\ 1\ 513.77\ /h$$

And per month:

$$= R1\ 089\ 916/\ \ month$$

Tons of coal over burnt:

= R1 089 916/ R418.58/ton

= 2 603.84 tons of coal over burnt due to the condenser fouling