SIMULATION OF A POWER STATION'S CONDENSATE PUMPING SYSTEM IN VIEW OF IMPROVING ITS EFFICIENCY

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DECLARATION

I, Pieter Leonardo Du Toit Meyburgh (Identity Number: 870706 5016 083), hereby declare that the work contained in this dissertation is my own work, and has not been submitted previously to any other university in order to obtain a degree. Some of the information contained in this dissertation has been gained from various journal articles, textbooks, manuals, and private consultations with industrial specialists, and has been referenced accordingly. The research in this dissertation is supported by the resources and funding of Eskom; and some of the information in the appendices is confidential. Confidential information can only be made available to the reader by way of a special request.

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ABSTRACT

With increasing environmental awareness, higher electricity production costs, and a shortage in electricity supply to the South African grid, increased focus has been placed on the efficiency of power plants, and the improvement thereof.

On the majority of the Condensate Extraction Pumps (CEPs) installed in power stations, the flow control is implemented by conventional valve throttling of the discharge valve of these pumps. This type of flow control is uneconomical, as more than 30% of the flow – and the pressure – that can be supplied by the CEPs, is throttled under normal operating conditions.

As an energy-efficiency improvement measure, many sources recommend retrofitting of this flow control mechanism with variable speed flow control; but as different applications have different requirements, a need was identified to develop a systematic approach that would assist in determining the feasibility of a variable speed drive (VSD) retrofit.

In order to develop this approach, a case study was conducted on an Eskom coal-fired power station – comparing different flow-control techniques with the aid of efficiency calculations, a simulated model, and an economic evaluation. In this case study, an electrical VSD was identified as a feasible retrofit, with an energy saving of 34.6% achievable in pump power consumption at full load conditions.
The power station investigated was operating at an average load output of 84.8%, which meant that the pumps would be utilized at lower than average flow rates. This increased the energy saving achievable to 49.2%; but a performance test on these pumps uncovered inefficiencies that increased the average flow rate through the pump again, reducing the savings potential to 39.1%.

By implementing this retrofit, an energy saving of 362 kW is achievable; and the Rankine efficiency of the plant would thereby increase by 0.018%; while the payback period for the initial investment would be 1 and a half years. More than R400 million would be saved in operating costs over the remaining lifetime of the plant.

In Eskom's fleet alone, six other fossil-fired power stations have the same configuration as in the case study, and similar savings can be expected in these power stations.

During the process of developing the evaluation approach; the simulated modelling, the load profiling and the performance testing were identified as essential steps in determining the actual pump duty and the actual pump performance. This provided a more accurate representation of the actual savings achievable, which would have a noteworthy effect on the results of the economic analysis.
As a result, the steps mentioned above were identified as being fundamental in the development of the systematic evaluation approach.

Keywords: Energy Efficiency, Condensate Extraction Pump, Flow Control, Coal-Fired Power Station, Variable Speed Drive, Feasibility, Economic Evaluation

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CONTENTS

List of Figures

List of Tables

Nomenclature

AC	Articulated Current
BEP	Best Efficiency Point
BFP	Boiler Feed Pump
CEP	Condensate Extraction Pump
COD	Coefficient of Determination
CPP	Condensate Polishing Plant
DC	Direct Current
DCS	Distributed Control System
DST	De-aerator Storage Tank
EE	Energy Efficiency
g	Gravitational Constant (9.81 m/s^2)
h	Entropy (kJ/kg)
H	Head (m)
HP	High Pressure
HR	Heat Rate
HVSC	Hydraulic Variable Speed Coupling
IP	Intermediate Pressure
LOP	Life of Plant
LP	Low Pressure
$\dot{m}$	Mass Flow (kg/s)
MCR	Maximum Continuous Rating (100% load)
MPCS	Master Process Control System
MTBF	Mean Time Between Failure
N	Speed (rpm)
NPSH	Net Positive Suction Head
$NPSH_A$	Net Positive Suction Head Available
$NPSH_R$	Net Positive Suction Head Required
P	Power (kW)
p	Pressure (kPa)
PLC	Programmable Logic Controller
s	Entropy (kJ/kg.K)

SMP	System Marginal Price
VSD	Variable Speed Drive
Q	Flow (kg/s or m^3)
$\dot{Q}$	Heat (kJ/s)
T	Temperature (°C)
$\dot{W}$	Work (kJ/s)
ρ	Rho (density in kg/ m^3)
η	Eta (Pump symbol for Efficiency)

1 Introduction

In an attempt to increase the electricity supply to the South African grid, as well as optimizing plant performance, Eskom, the largest supplier of electricity in South Africa and in Africa, invests time and money in plant efficiency and the improvement thereof[1]. The research proposed for this dissertation was identified as possible areas of efficiency improvement by a renowned international research and engineering company[2]. The results obtained from this research will be evaluated by Eskom management to determine whether this improvement will be implemented, or not.

1.1 Area of Investigation

This research is focused on the pumping system of the condensate and low pressure (LP) feed-heating system of a coal-fired power station; and it could be broadly applicable to all other electricity-producing power stations, or large industrial electricity consumers. The results would not be the same for all applications – due to different operating conditions and plant equipment; but the methods and practices used in this dissertation could be applied by other large electricity users to identify and quantify efficiency improvement with pumps.

The condensate and LP feed-heating system forms part of the electricity-generating Rankine cycle, which will be elaborated in section 2. The main objective of the Rankine cycle is to provide the steam required to drive the main turbine generators, which converts the mechanical energy provided by the steam turbine into electrical energy, which in turn is fed into South Africa's Eskom electricity grid.

Two areas of investigation have been identified for probable efficiency gain:

1. The condensate system's flow control (90% of the research focus)
2. LP drain-recovery system functionality (10% of the research focus)

1.1.1 Investigation Area 1: The condensate System's Flow Control

Two 100% flow-capacity vertical Condensate Extraction Pumps (CEPs) pump condensate from the condenser of a power station through the LP feed-heating stages to the De-aerator. The flow through these pumps is controlled by a valve on the discharge side of the CEPs.

When the turbine of the unit trips – due to an emergency, the turbine is isolated and steam is bypassed to the condenser. This steam has to be cooled down by condensate from the CEP discharge before it enters the condenser. During such an emergency condition, more flow

[1] The Energy-Efficiency Pledge of former Eskom CEO, Thulani S. Gcabashe
[2] As explained in a private consultation with Eskom's Energy-Efficiency Project Leader, Jason Hector

through the CEPs is required than under normal operating conditions. Consequently, one single pump is designed for a larger flow requirement during emergency conditions, as well as for normal operating conditions. This results in the valve on the discharge of the pump being throttled down to about 70% when the unit is at full load, and opened to 100% when an emergency condition forces the LP bypass system to open.

This throttling action is not an effective method for regulating the flow, as the large pressure drop over the valve is evidence of a loss of energy. The industry trend, when building modern power stations, is to rather focus on more efficient flow-control systems, such as variable speed control.

1.1.2 Investigation Area 2: LP Drain-Recovery System

As a subordinate study: the LP drain-recovery system of a power station is used to improve the overall efficiency of the cycle by re-introducing condensed bled steam from the LP heaters into the main feed heating line. Power station specific challenges cause this system to be taken out of service. An additional objective of this research was the quantification of loss in efficiency when this system is taken out of service, and to propose possible solutions to increase the reliability of this plant.

1.2 Problem Statement

Even though many sources identify variable-speed pumping (in investigation area 1) as the most efficient method of flow control currently available on the market (BPMA Website)[3], pumping applications are different from each other; and this type of control is impossible to implement in some cases. All pumps have different system requirements and capabilities, which means that every pumping application must be carefully investigated, in order to determine the feasibility of retrofitting the current control system with VSD technology rather than any other control system.

A need was identified for a systematic evaluation approach that could be applied by any pump user when determining the feasibility of a flow-control retrofit. Even though such an evaluation approach would not be applicable to all pumping applications, it should be a guide for large industrial pump users, outlining the most important considerations when determining the feasibility of a flow-control retrofit. This evaluation approach could only be developed through research and practical implementation. One of the most important deliverables of the research proposed is the development of such an evaluation approach through practical investigation, modeling, and performance testing of the CEPs, as identified in section 1.1.1.

[3] According to the British Pump Manufacturers Association (BPMA) website

1.3 Research Methodology

The proposed research was initiated by a literature survey; and all practical information was obtained from the case study of a 4110 MW coal-fired power station in South Africa with three wet-cooled 714 MW units. The literature survey was followed by modeling of the current plant – and the subsequent determination of the actual CEP duty point by load profiling and performance testing. This approach provided the tools for calculating the exact energy savings that could be expected by retrofitting the current plant with new flow-control technology.

Several different flow-control options may be investigated, but as will be explained in the literature survey and in the current study, these are limited to:

1. Traditional Mechanical Control
2. Variable Speed Controls
    a. Hydraulic Variable Speed Couplings (HVSC)
    b. Electrical Variable Speed Drives (VSD)

Simulated models were used to confirm and support the findings of the mathematical calculations. These calculations were followed by an economic evaluation that revealed which of the options would decrease the operating costs over the lifetime of the plant without diminishing its reliability.

2 The Literature Survey

The purpose of the literature survey is to present a comprehensive background of the systems, components, operation, technology, and all other stakeholders[4] involved in the power-generation industry, with specific focus on elements of the Condensate and Low Pressure Feed Heating Pumping System of a Power Station.

The literature survey will be initiated by an overview of Eskom's role in the South African power-generation industry.

2.1 Eskom as Power Utility in South Africa

Eskom is the largest producer of electricity in South-Africa, and is currently responsible for about 95% of all electricity produced in South-Africa, and approximately 45% of all the electricity produced in Africa (Eskom website). At present, Eskom is supplying the South African grid with just over 40,000 MW installed capacity – with the actual output depending on the demand during peak and off-peak times – while also being dependent on the supply available during any maintenance activities.

In the winter of 2013, the maximum demand for electricity reached a high of 35,303 MW; and was met with a supply of 37,106 MW[5], meaning that Eskom had less than 10% of the fleet unavailable for maintenance activities and other load losses – a number that decreases, when taking unplanned maintenance and load trips into account.

Together with the impact on the environment, the cost of production and the provision for maintenance opportunities, this shortage in electricity has led to an increased focus on energy efficiency in all Eskom power plants. An independent research company was tasked with identifying possible areas of improvement in energy efficiency; and, in turn, Eskom management tasked the Eskom Energy Efficiency (EE) and the Power Station Enhancement (PSEP) teams with seeing these projects through[6]. The research undertaken in this dissertation was recognised as prospective areas of energy-efficiency improvement by the aforementioned stakeholders. Any forthcoming recommendations might be implemented in the near future, depending on the conclusions reached in this research.

2.2 The Water-Steam Cycle of a Coal-Fired Power Station

A typical coal-fired power station has the following layout, as described in Figure 1[7], and is modelled from the Rankine cycle with regenerative feed heating, as displayed in Figure 2[8].

[4] Stakeholders like Eskom, large electricity users, subcontractors and the general public.
[5] According to Eskom's weekly System Status Bulletin on the 18'th of July 2013.
[6] Under Leadership of Jason Hector, Eskom Senior Engineer.
[7] Figure was developed using information from the as-built manuals supplied after plant construction (which is proprietary to Eskom and available at special request) and with the input of plant engineers
[8] Diagram compiled with the input of Eskom Chief Engineer, Professor Joe Roy-Aikins

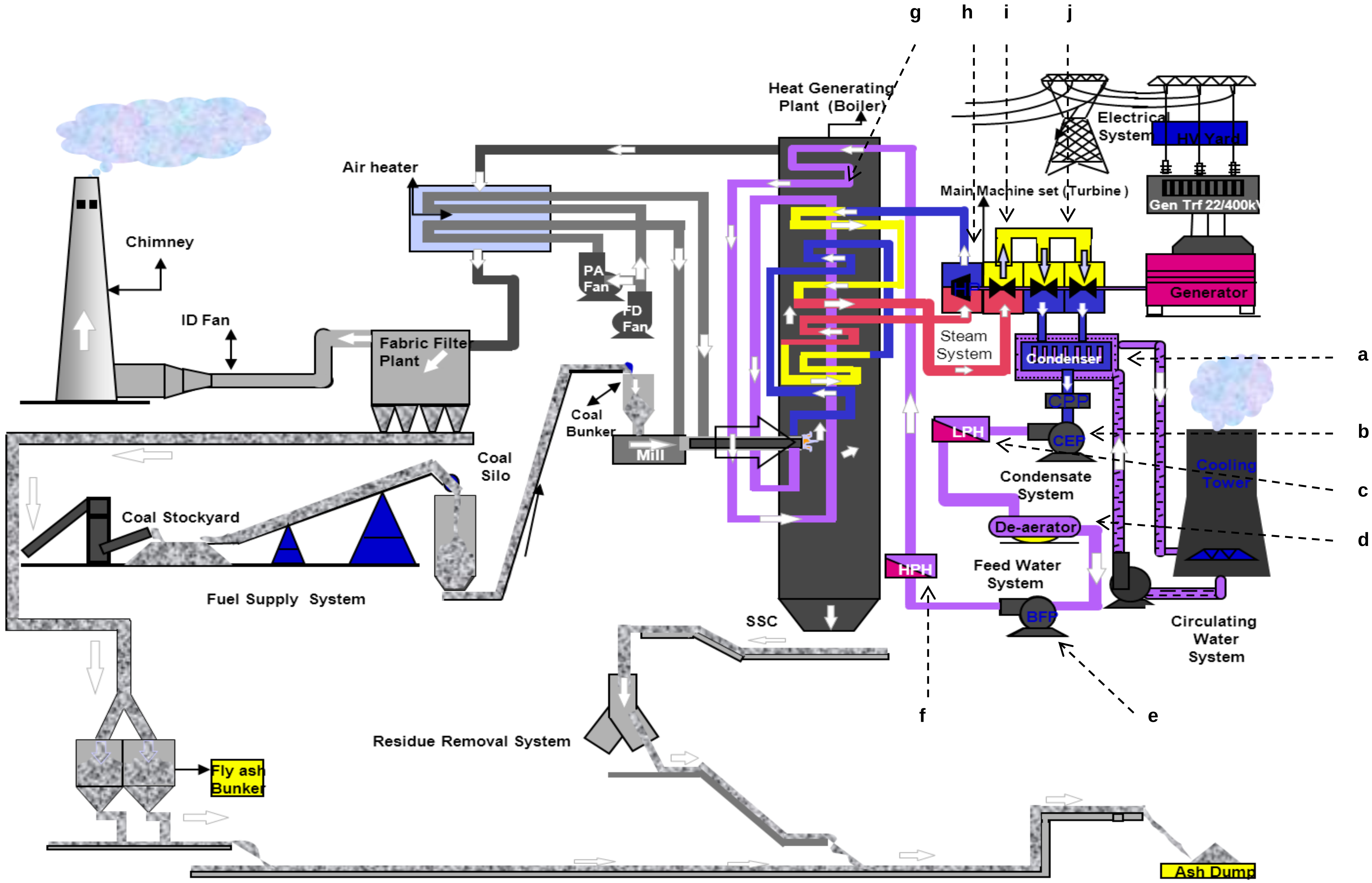

Figure 1: Basic Coal-Fired Plant Configuration

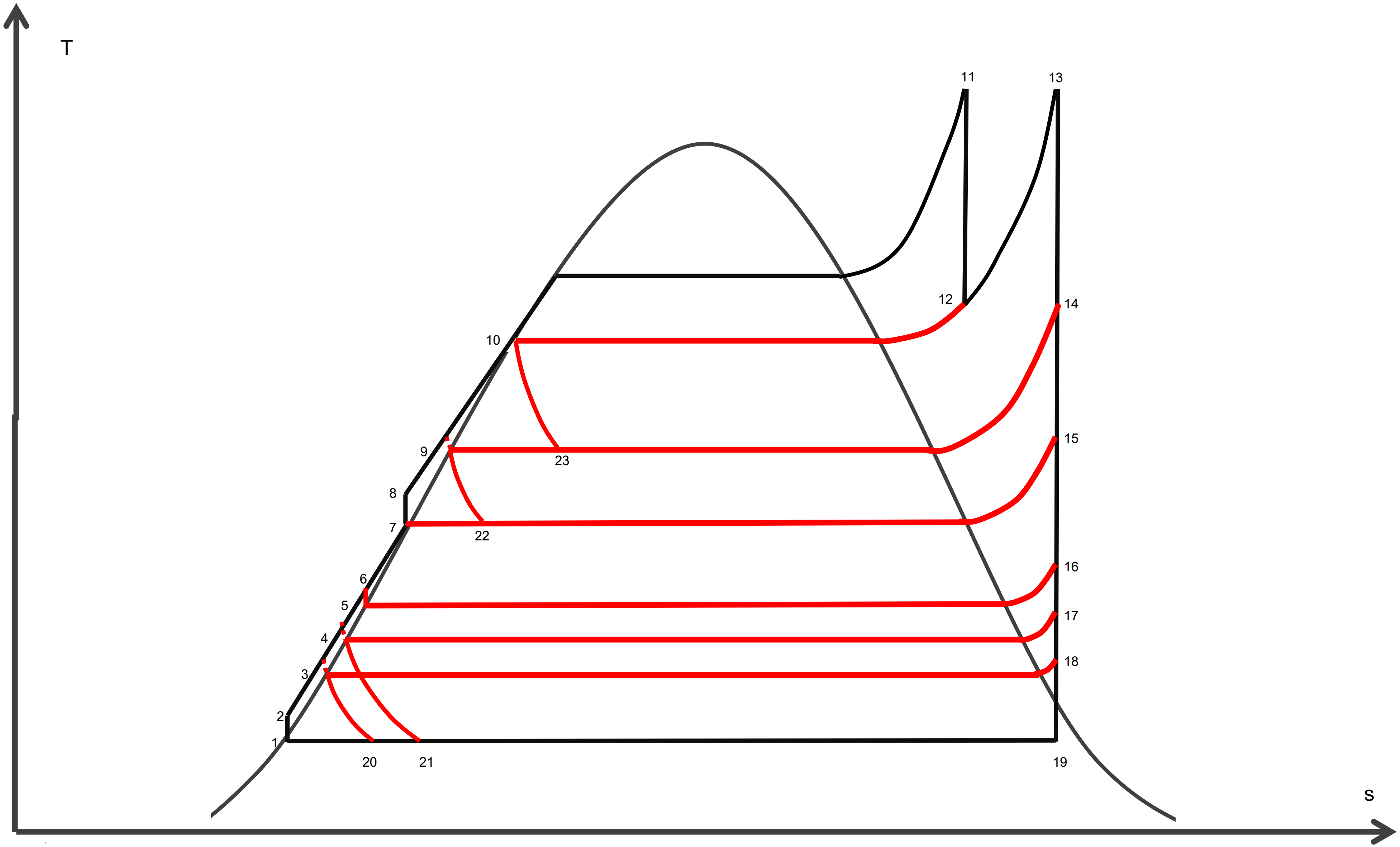

Figure 2: T-s Diagram of the Water-Steam Cycle

2.2.1 Basic Overview of the Water-Steam Cycle

From Figure 1, the following components make up the path of the water and steam, as they proceed through the water-steam cycle (in order of sequence):

a) Condenser
b) Condensate Extraction Pumps (CEPs)
c) Low Pressure (LP) Heaters
d) De-aerator Storage Tank (DST)
e) Boiler Feed Pumps (BFPs)
f) High Pressure (HP) Heaters
g) Boiler
h) HP Turbine
i) Intermediate Pressure (IP) Turbine
j) LP Turbine

2.2.2 The Rankine Cycle with Feed Heating Applied to a Power Station

The ideal Rankine Cycle (without losses taken into consideration)[9] displayed in Figure 2 (P. Ruestman, 1999)[10] shows more detail with reference to the power station investigated in the case study; and it can be broken down into the following sections (numbered sequentially):

- 19 to 1: **(Condenser)** Constant-temperature condensing of steam into condensate at the saturation temperature of the condenser pressure (5 to 15 kPa abs.);
- 1 to 2**: (CEPs)** Pumping of the condensate and increasing the pressure from 5 kPa to 1.7 MPa;
- 2 to 3: **(LP Heater 1)** Increasing of the condensate temperature by exchanging the heat from bled steam tapped from the LP turbine (point 18) in a heat exchanger and dumping it to the condenser (point 20);

[9] An ideal cycle with isentropic processes and no pressure losses in the individual components/pipelines or heat loss to the environment.
[10] The T-s diagram was developed further with the help of Eskom Chief Engineer, Prof. Joe Roy Aikins

- 3 to 4: **(LP Heater 2)** Increasing of the condensate temperature by exchanging the heat from bled steam tapped from the LP turbine (point 17) in a heat exchanger and dumping it into the condenser (point 21);
- 4 to 5: **(LP Heater 3)** Increasing of the condensate temperature by exchanging the heat from bled steam tapped from the LP turbine (point 16) in a heat exchanger; but instead of dumping the condensed bled steam, it is pumped back into the main feed water cycle (point 6);
- 6 to 7: **(DST)** Increasing of the feed water temperature by using bled steam from the IP turbine (point 15) in a heat exchanger;
- 7 to 8: **(BFPs)** Feed water pumped from the DST to the boiler by increasing the pressure from 1.7 MPa to about 21.5 MPa;
- 8 to 9: **(HP Heater 5)** Increase of feed water temperature by using bled steam from the IP turbine (point 14) in a heat exchanger, and dumping the condensed bled steam into the De-aerator (point 22);
- 9 to 10: **(HP Heater 6)** Increase of feed water temperature by using bled steam from the cold reheat line (point 12) in a heat exchanger, and cascading the condensed bled steam to HP Heater 5 (point 23);
- 10 to 11: **(Boiler)** Feed water is heated up in the boiler economiser, and the steam temperature is increased to about 535 °C in the superheater;
- 11 to 12: **(HP turbine)** Steam expands through the HP turbine to about 3.5 MPa, where the thermal energy is converted into mechanical energy;
- 12 to 13: **(Boiler Reheat)** Steam re-enters the boiler reheater, and is reheated to 535°C at the same pressure;
- 13 to 19: **(IP & LP turbine)** Steam expands through the IP turbine and from the IP turbine directly to the LP turbine, until it enters the condenser as totally exhausted steam (quality just below 1).

In Figure 2, the Condensate and the LP Feed-Heating System (investigated in the case study) starts at point 1 and ends at point 7.

2.2.2.1 The Importance of Regenerative Feed Heating in the Rankine Cycle Design

Regenerative feed heating is allowed into the Rankine Cycle design by the inclusion of feed-water heaters, more pipe networks, and more pumping stages.

With the addition of these plant components into the water-steam cycle, the following disadvantages can be identified:

1. Added plant complexity;
2. Increased maintenance costs, due to the increase in plant components;
3. Increased initial building costs, due to added plant complexity;
4. More pumping stages required.

The merit of the Rankine Cycle with regenerative feed heating is obvious when specifically considering the boiler heat input in the T-s diagram in Figure 2 and Figure 3. In the simplified diagram in Figure 3, the difference between the boiler heat input can clearly be seen when comparing a Rankine Cycle without feed heating and a single pumping stage to a Rankine Cycle with feed heating and two pumping stages.

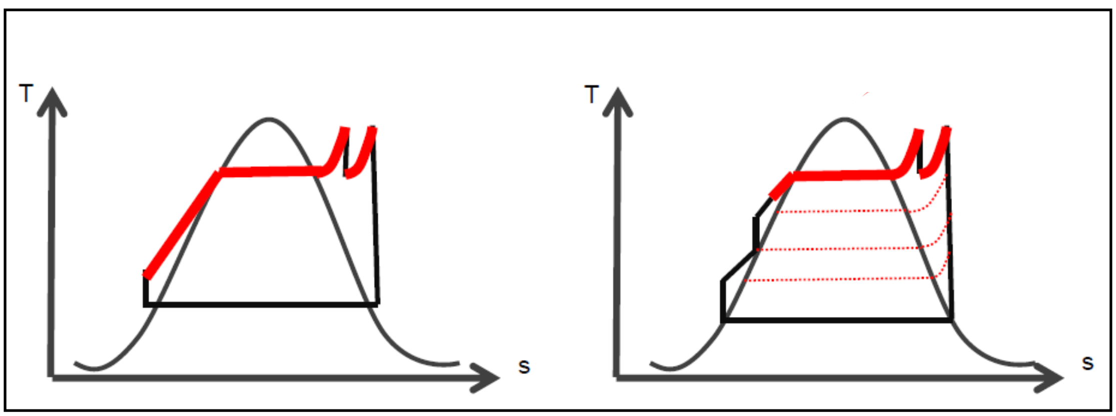

Figure 3: A Rankine Cycle without feed heating vs a Rankine Cycle with feed heating

This saving in heat input relates to a saving in coal burnt to turn the feed water into steam. Consequently, a smaller boiler is needed to generate the superheated steam. This results in reduced initial capital investment costs to build the boiler.

Efficiency is also expressed as energy produced divided by energy consumed, as may be seen in Equation 1 below (P. Ruestman, 1999)[11]. Since the work output by the turbines remains the same in both configurations in Figure 3, it is evident that the thermodynamic efficiency would be higher in the cycle, which utilises the feed-water heating.

$$\eta_R = \frac{W_{out}}{Q_{in}}$$

Equation 1: Rankine Cycle Efficiency Equation

[11] Pg. 6 of Section 4 in the training manual.

2.3 The Condensate and LP Feed-Heating System

The condensate and LP feed-heating system is a sub-system that forms part of the water-steam cycle, as presented in Figure 1. The system starts at the condenser hot-well, where the CEPs take suction; and it ends at the De-aerator Storage Tank (DST). The CEPs pump the condensate to the DST via the LP heaters. The following diagram (Figure 4) represents the Condensate and LP Feed-Heating System of a typical coal-fired power station[12]:

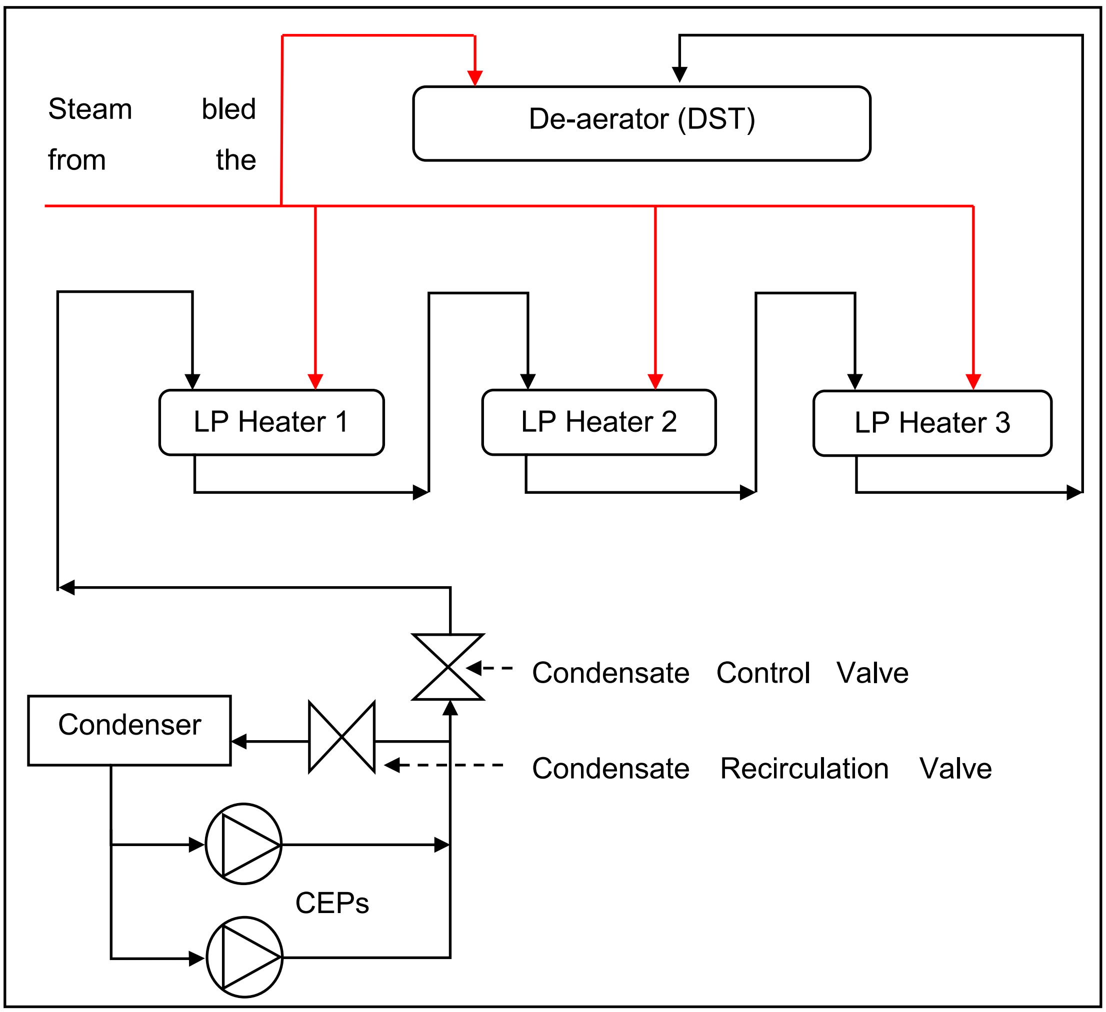

Figure 4: Condensate and LP Feed-Heating System

[12] This information was obtained from the original design manuals supplied by the company responsible for building the power station being investigated. Due to the confidential nature of these manuals, they are available only by way of a special request.

2.3.1 Condensate Extraction Pumps

The condensate extraction pumps (or CEPs) are two 100% duty pumps, with one of the two pumps running continuously, and the other on standby. Each of these pumps is able to supply the required condensate flow at all times. The following are the specifications of the CEPs at the power station being investigated:

Type:	400 mm - 700 mm Vertical Centrifugal 2–stages
Manometric Head:	146 m head
Flow Rate:	550 l/s rated
Medium:	Demineralised water
NPSH Required:	6.92 m
Efficiency:	77.3 % rated (83% throttled)
Power Absorbed:	1004 kW rated (925 kW throttled)
Medium Temperature:	+/- 55 °C
Rotation Speed:	1486 rpm
Pump Curve:	See APPENDIX C

The Original Equipment Manufacturer (OEM) designed vertical pumps, instead of horizontal pumps, for applications where $NPSH_A$ (Net Positive Suction Head $_{AVAILABLE}$) and floor space is not enough (M. Pugh, 2000)[13]. At newer and larger plants, the condenser is usually at ground level, and is not suspended in the air, which means that $NPSH_A$ is insufficient to install a horizontal pump. As a result of this, the CEPs are designed with vertical orientation and deep-set casings, where the pump suction impeller is situated on an extended shaft – well below ground floor – to increase $NPSH_A$ (M. Pugh, 2000)[14]. Figure 5 (M. Pugh, 2000) shows a sectional drawing of a typical condensate extraction pump.

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[13] Section 4.1
[14] Sections 4.1 and 4.2.3.

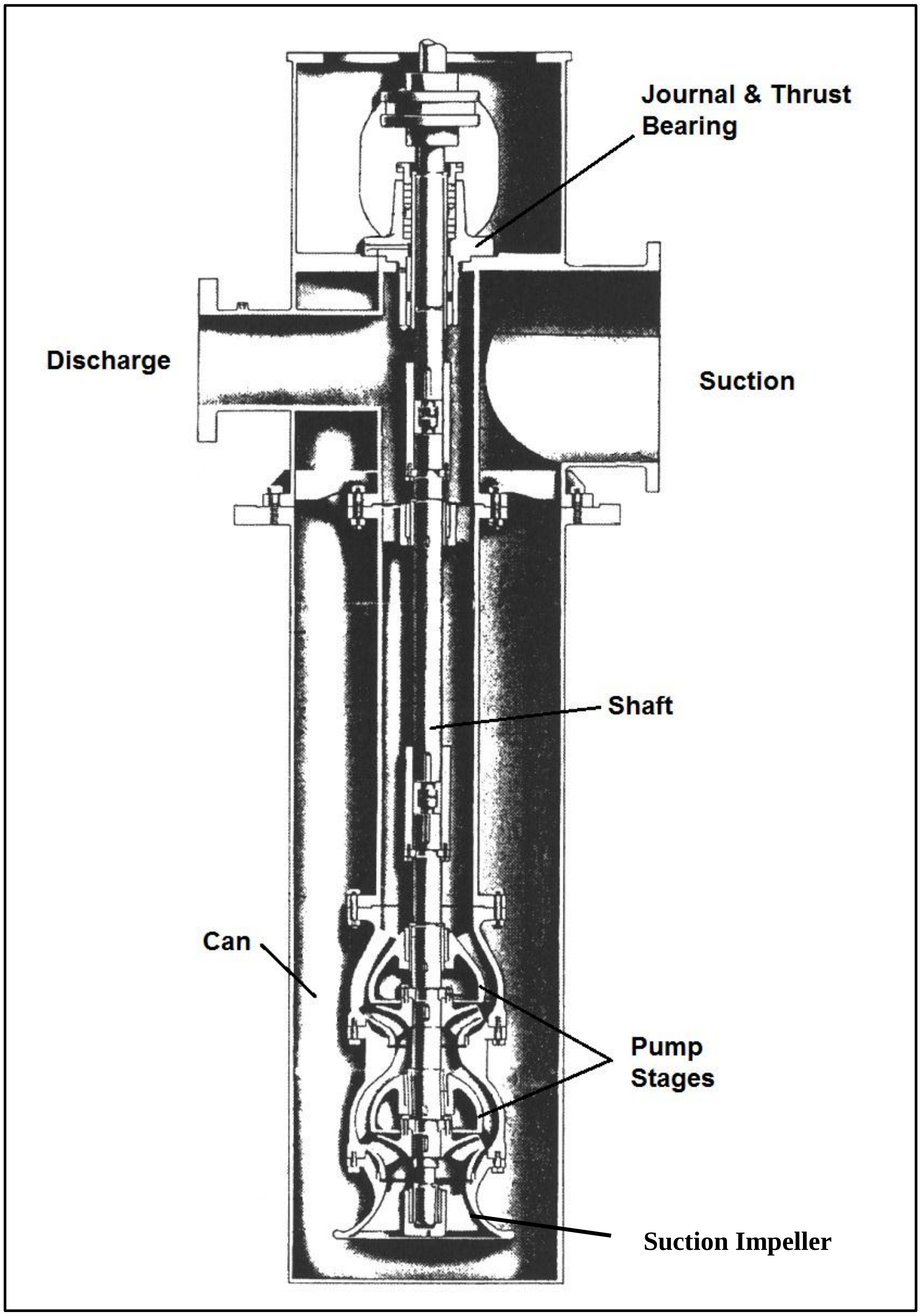

Figure 5: CEP Sectional Drawing

The pump impellers are suspended freely from the journal and thrust bearing, which is designed only to counteract the static weight of the impellers and shaft, as well as to absorb the added thrust during pump operation.

2.3.1.1 CEP Surplus Flow Predicament

A typical challenge with the overall water-steam cycle presents itself when leaks and excessive water use (resulting from valve passing, boiler tube leaks, pipes / vessels leaking, drain-recovery system not working, etc.) force more water through the cycle than is required. The extra flow and leaks are cascaded or dumped into the condenser, where the CEPs would then have to circulate this extra flow through the system.

With the CEPs pumping the flow required for normal operation, as well as the additional flow mentioned above, the discharge pressure of the CEPs would decrease, as anticipated from the pump performance curve in APPENDIX C (more flow = less head). An alarm would sound when the CEP discharge pressure drops to below 1.3 MPa; and the condensate system would stop operating if this pressure reaches a low of 1 MPa. The reason for this protection system is to ensure that there is always a pressure of more than 1 MPa available for the LP Bypass supply line. The LP Bypass spray water nozzles need this pressure to atomize the spray and thereby optimize the area of contact between the steam and the condensate spray water.

To prevent such a situation, the plant operators would start the second CEP – in order to increase the discharge pressure of the condensate line – whenever the pressure boundary alarm sounds. The extra power consumed when operating with two pumps in service has a noteworthy impact on the auxiliary power consumed, almost doubling this figure.

2.3.2 LP Feed Heating and Drain-Recovery System

The LP feed-heating system heats up the condensate pumped from the condenser to the DST, by making use of 3 LP heaters (Figure 4).

The LP Drain-Recovery System is designed to increase the overall efficiency of the cycle (section 2.2.1) by utilizing the energy still contained in the condensed bled steam from LP heater 3. Instead of dumping this warm and energy-rich condensed bled steam into the condenser, as is done with LP heaters 1 and 2 (as is evident from the heat balance diagram in APPENDIX A), it is pumped back into the LP feed water line by utilizing one of two 100% flow LP drain pumps. This heats up the feed water, and increases the overall efficiency of the Rankine Cycle.

2.3.2.1 The LP Drain-Recovery System Layout

The LP heaters are shell-and-tube feed-water heaters. LP feed water is pumped through the tubes inside LP heater 3 by the CEPs; while bled steam enters at the top of the heater, and is passed over these tubes, thereby heating up the LP feed water. The bled steam is condensed in the heat-exchange process, and drained from the heater via the condensate-drain branch (Figure 6). From here, the warm condensate flows to the LP heater flash box.

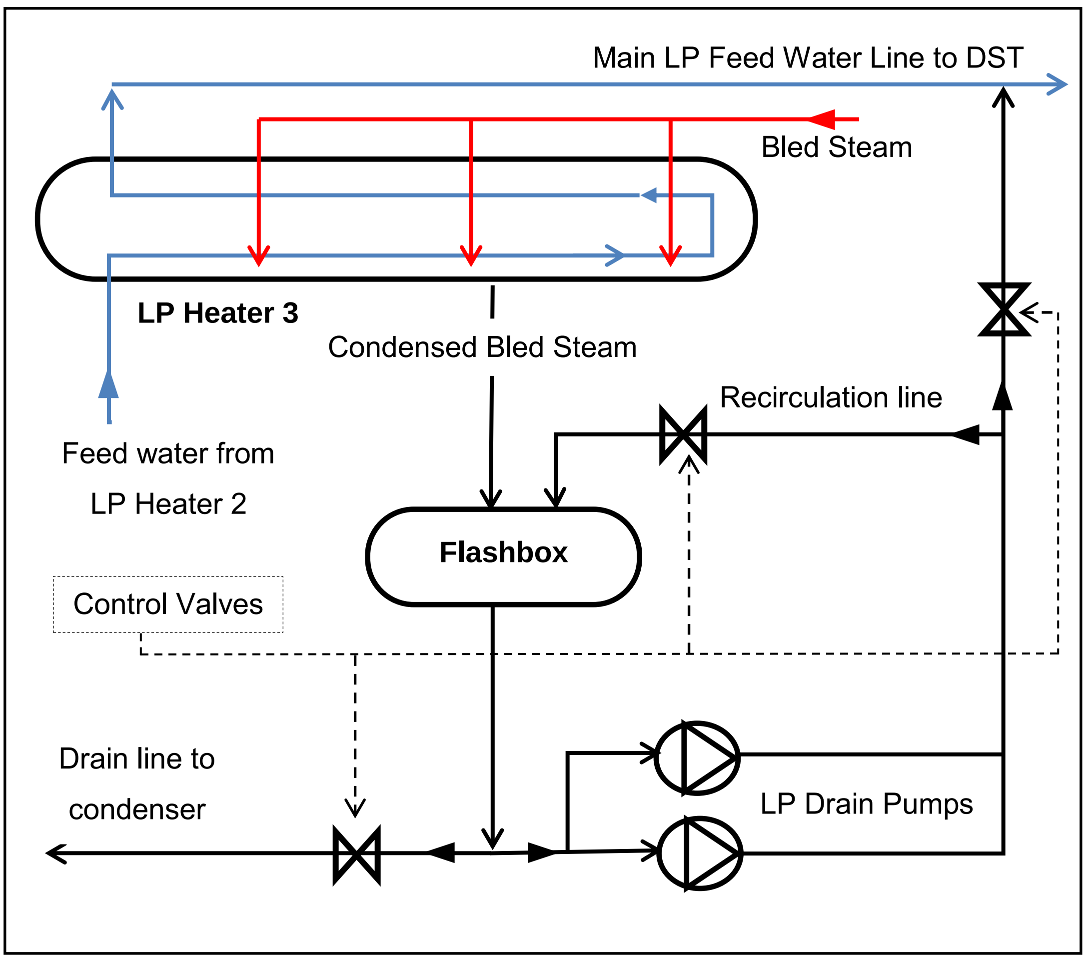

Figure 6: LP Drain-Recovery System

The three control valves, as shown in Figure 6, are used to control the level of water in the flashbox. These valves ensure that there is adequate $NPSH_A$ for the LP Drain Pumps to perform at optimal efficiency, while maintaining pump health.

The purpose of the flashbox is to provide the $NPSH_A$ for the LP Drain Pumps to pump the condensate back into the main feed heating line.

2.3.2.2 Fluctuating Flashbox Tank-Level Predicament

The largest challenge with the LP Drain-Recovery System design is the fact that the condensed water in the flashbox is very close to saturation pressure when the plant is in operation. This means that the saturation pressure of the demineralised water, as shown in the diagram below (Figure 7), would be very close to the pressure acting on the water (P_1); and that a sudden drop in this pressure (like a leaking drain valve) would result in the water flashing in the flashbox, significantly reducing the $NPSH_A$ for the pump. The minimum level alarm would initiate a LP drain-recovery system shut-down, meaning that the drain line would send the condensed steam back to the condenser.

Equation 2 (Sulzer Pumps, 1998) is used to determine the NPSH available for a pump:

$$NPSH_A = \frac{P_1}{\rho g} + H_{static} - H_{losses} - H_{vap}$$

Equation 2: NPSH available equation

Where,

$P_1 = Pressure\ acting\ on\ the\ fluid\ [Pa]$

$H_{static} = Head\ due\ to\ difference\ in\ height\ (z_1 - z_2)\ [m]$

$H_{losses} = Loss\ in\ head\ due\ to\ friction\ or\ other\ obstructions\ [m]$

$H_{vap} = Head\ at\ saturation\ conditions\ [m]$

$\rho = Density\ of\ the\ fluid\ [\frac{kg}{m^3}]$

$g = Gravity\ constant\ \left[9.81\frac{m}{s^2}\right]$

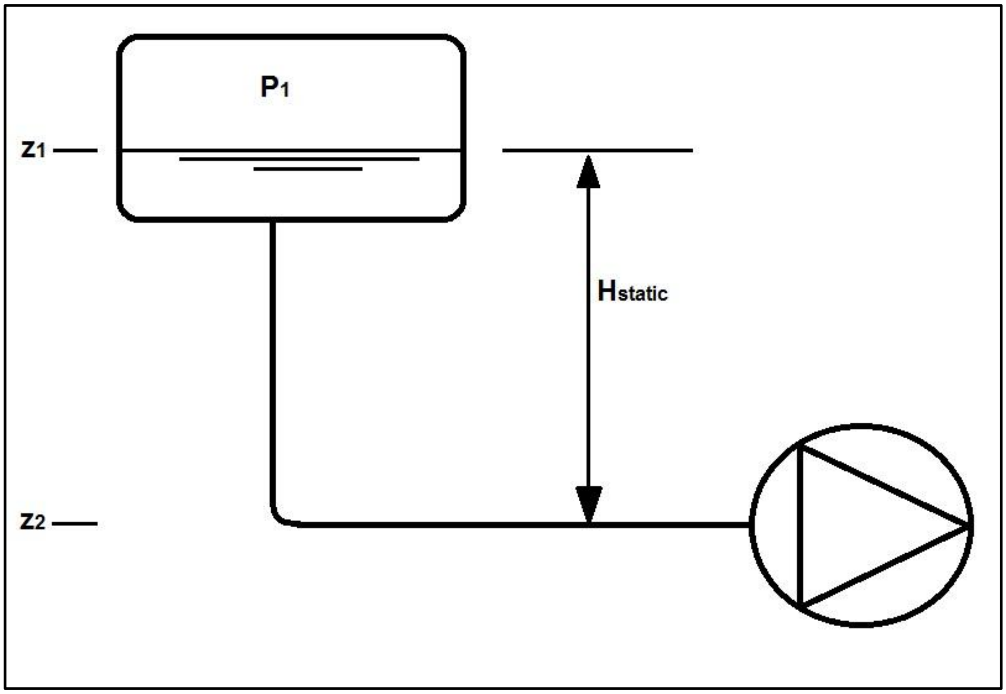

Figure 7: NPSH drawing

Large variations in the flashbox level, as discussed above, make this level very difficult to control; and in order for this control system to maintain the level of the flashbox in Figure 6, the valves – which are very intricate – frequently require calibration. The result is that malfunction of the LP drain-recovery system due to control issues is a common occurrence.

After an LP drain-recovery system shut-down, the circulation of feed water back to the condenser would result in the CEPs having to work harder to circulate water that could have been used to accommodate the forward feed requirement. Typically, the flow would increase from 430 kg/s to 458 kg/s, resulting in an increase in pump input power from 925 kW to 945 kW (values was taken from the pump performance curve in APPENDIX C). This 20 kW of extra power consumed increases the auxiliary power consumption, and decreases the overall plant thermal efficiency.

The increase in flow also results in operators starting the standby pump, as explained in section 2.3.1.1.

Another challenge encountered at many power stations – which is a maintenance function – is LP drain pump repair turn-around time. The LP Drain pumps are refurbished by the OEMs; however, they have too many pumps coming into their workshops; and as a result, they tend to prioritise larger pumps, because these entail larger penalties when not refurbished or repaired in time. LP drain pumps are considered small when compared with CEPs or Boiler Feed Pumps (BFPs) (48.5 kW in comparison with 1000 kW for CEPs to 16 MW for BFPs); and therefore, the refurbishment time is sometimes postponed until the maintenance of any larger pumps has been completed. As this is a function of maintenance, the efficiency of this process cannot be addressed by this research, and only recommendations can be made towards improving the pump repair turn-around time.

Redundancy is built into the LP drain-recovery system; and this ensures that a second pump is available when one of these pumps is sent for repairs (both are 100% duty pumps); however, there have been occasions when both pumps fail; or the second pump fails, while the first one has been sent for repairs; and this leads to the drain-recovery system being temporarily out of service.

The effect of the LP drains recovery system taken out of service is:

1. Increased CEP flow requirement, leading to the pump overloading;
2. Flow-Accelerated Corrosion (FAC: flow-induced corrosion most common at conditions close to saturation pressures). This reduces the drain line integrity and the also remaining Life of Plant (LOP); and
3. Decreased thermal efficiency, as energy-rich water is dumped into the condenser.

2.4 Pump Fundamentals

Pumps are estimated to consume about 20% of all the energy used by Industry (Cruz, 2009). And when energy efficient systems and a low carbon economy are desired, pumps are good candidates for investigation. The following section gives a background on pump characteristics, efficiency curves, and the key concepts with reference to pump design, maintenance, and operation.

2.4.1 Pump Definitions and Curves

Pump performance curves (Cruz, 2009) are typically used to illustrate a pump's performance with variation in flow. These curves are usually produced with the pump at procurement; and they look specifically at pertinent variables plotted against the flow (Figure 8). These variables are:

1. Flow
2. Head
3. Power
4. Efficiency
5. $NPSH_R$ (Net Positive Suction Head Required)

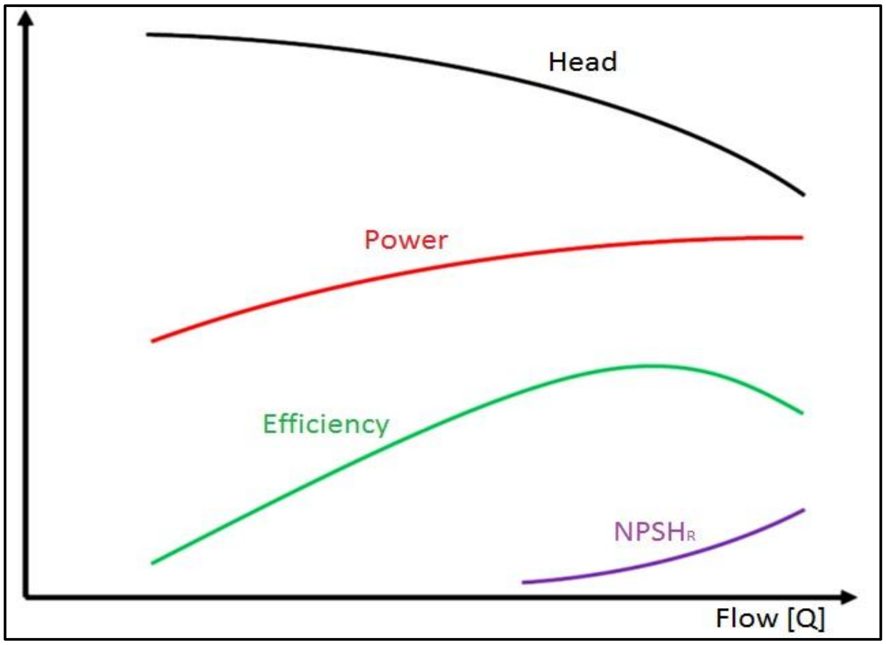

Figure 8: Typical Pump Performance Curve

Pump curves also usually specify different size impellers for the same casing on one graph to illustrate differences in flow, power, efficiency and $NPSH_R$ (Cruz, 2009). A pump curve with different size impellers would typically look like the curve shown in Figure 9 (Cruz, 2009):

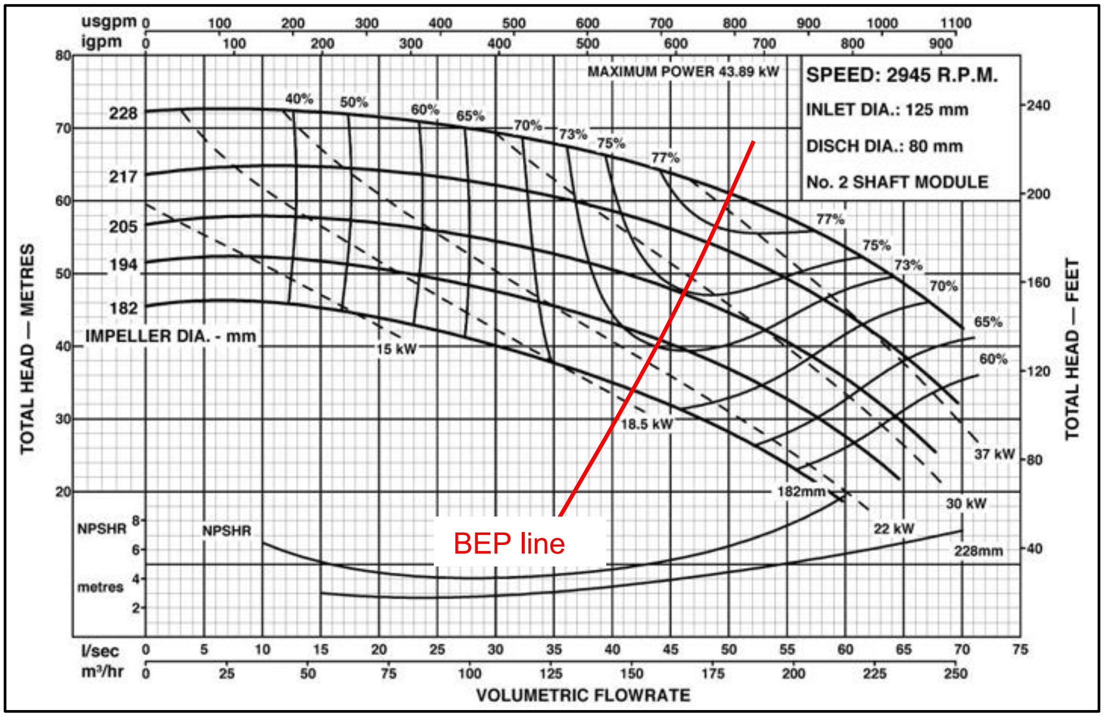

Figure 9: Pump Curve for Different Size Impellers

In the curve shown in Figure 9, the lines represent the Head (H) vs. the Flow (Q) for pump impellers sized from 182 mm to 228 mm diameter. The efficiency lines show that smaller-size impellers generally have lower efficiencies than larger-size impellers with the same size casing. These curves show, also for example, that a pump with a 228 mm size impeller would have to operate at a flow of 50 l/sec to operate at optimal efficiency. This point is called the Best Efficiency Point (BEP), which was drawn and superimposed on the curve in Figure 9 (shown in red).

2.4.2 CEP Flow Control

Two CEPs must be available at all times when the plant is in operation, with one running, and the second on stand-by. Each of the CEP motors is required to deliver a constant power supply of 1 MW to its pump at all times; and this is one of the largest consumers of auxiliary power at a power station.

The CEPs at the power station under investigation are designed to accommodate 100% flow at maximum continuous rating (MCR) or full load, as well as 143% flow at BYPASS conditions (which will be explained in the next paragraph), all achieved at a fixed speed. The 100% flow is the amount of flow that is necessary to sustain the full load requirements of the boiler; and the 143% flow is the required flow to the boiler plus the LP Bypass spray-water supply.

The LP bypass is a network of pipes designed to bypass the IP and LP turbines in the case of an emergency, such as an unexpected turbine shut-down; and it cooperates with the HP bypass to isolate the turbine from the water-steam cycle. The steam bypassed is sent to the condenser; but the steam (at 535°C and 3.5 MPa at the IP turbine inlet steam conditions) needs to be cooled down closer to the condenser temperature before it can enter the condenser. Spray water tapped from the CEP discharge is used to cool this steam down.

During normal operating conditions, the valves supplying this additional flow to the LP Bypass System are closed.

This means that the CEPs are designed to be able to operate at a discharge pressure higher than that which is required under normal operating conditions. This can be seen in Figure 10[15] below, depicting the hydraulic profile of the condensate and LP feed-heating system. In an ideally efficient system, there would be a minimal pressure drop over the control valve, which would mean that the CEP discharge pressure should ideally also be at a lower value.

[15] This figure was developed using the design manuals (classified material) and heat balance diagrams (APPENDIX A). Classified information is available to the reader by way of special request.

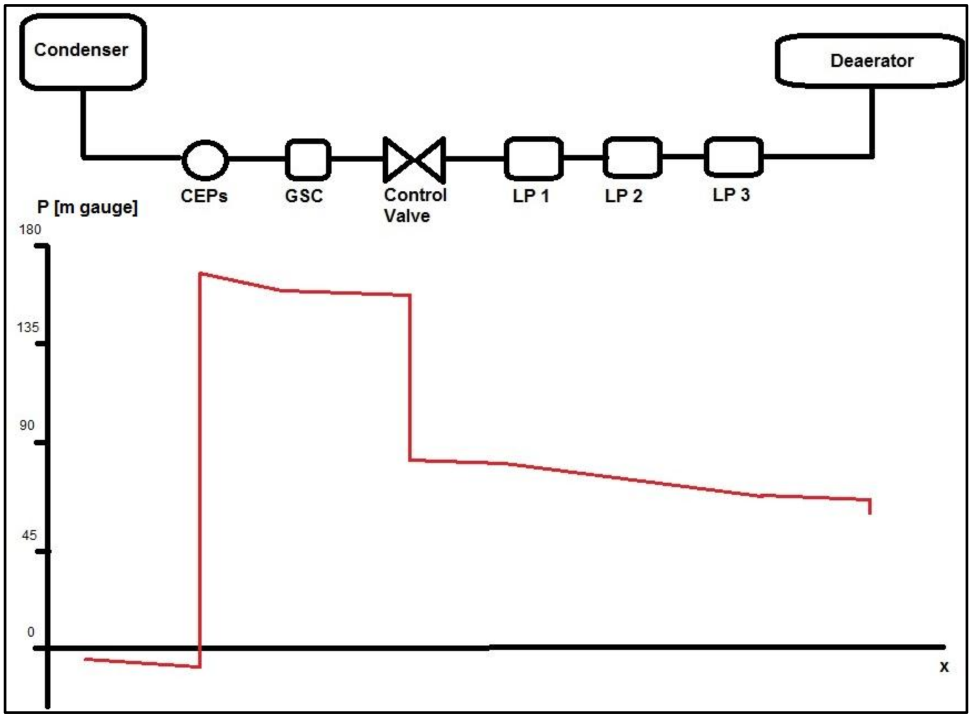

Figure 10: Condensate and Feed-Heating System Hydraulic Profile

A very important curve to consider during the design phase of a pump and system is the system-resistance curve (Europump, 2004)[16]. The system resistance, displayed in Figure 11, is the amount of resistance in the system against which the pump must perform; and this is brought about by restrictions (orifices, valves), pipes, bends, height difference, and other plant components. As is evident from Figure 11, the point where the pump is actually operating is, therefore, the point at which the system's resistance curve intersects with the pump performance curve, and this point is called the duty point of the pump.

If the pump and the pumping system are to operate at optimal performance, the system resistance must be so adjusted as to allow the pump to operate at optimal efficiency (as seen in Figure 8 and Figure 9). When this is achieved, the point where the system resistance curve intersects with the pump performance curve is referred to as the Best Efficiency Point (BEP). A typical method of controlling the system resistance is by opening and closing (throttling) a valve downstream of the pump (M. Pugh, 2000)[17].

[16] Section 2.2
[17] Section 7.2.3

The condensate control valve downstream of the CEPs are partially closed or "throttled", to ensure that only the 100% flow requirement is pumped to the DST. The dotted red line in Figure 11 depicts the increased system resistance, as a result of the valve being partially closed, or "throttled".

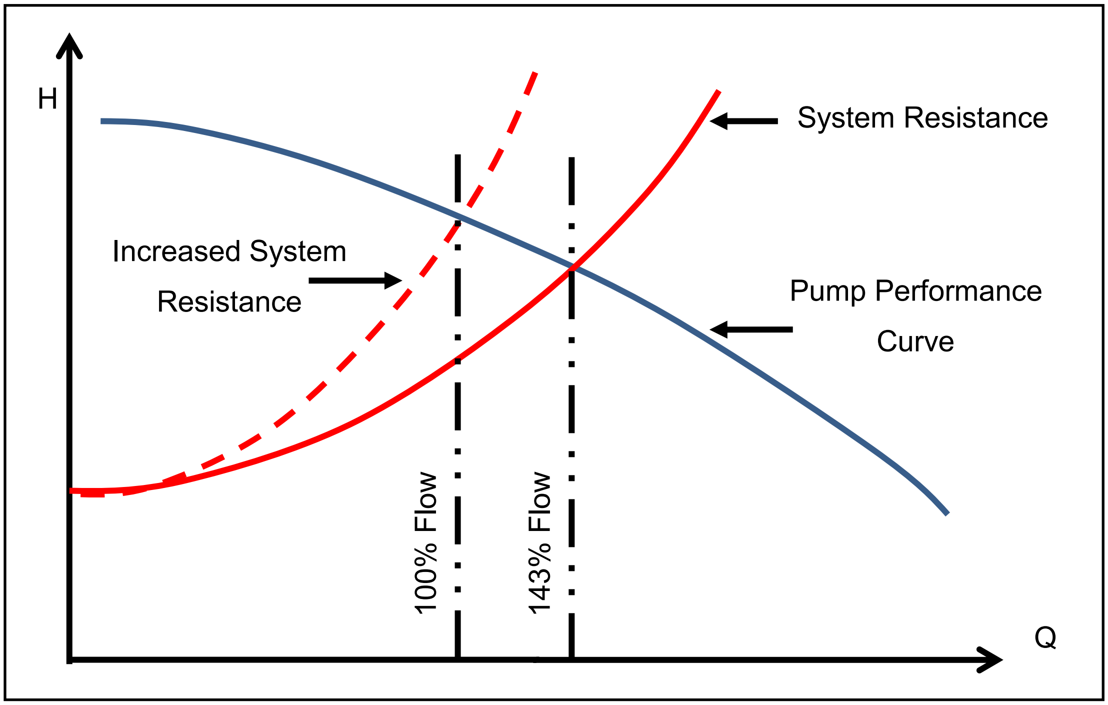

Figure 11: CEP Performance Curve

These CEPs were designed, so that the best efficiency is achieved at the 100% flow requirement (referred to as the duty flow), and not at the emergency 143% flow (referred to as the rated flow). This means that the pumps would operate at optimum efficiency when the valve is closed to allow only the 100% flow requirement through.

The throttling control is the current control system used to control the flow of condensate in all Eskom's power stations built so far; but it is however, not the only type of control that is currently used in industry. Pump experts consider this type of flow control as dated and inefficient; and reason that more advanced technological flow control techniques are available on the market, but the key is understanding them (Pemberton, January 2005).

2.4.3 Pump Flow Control Techniques

There are several methods of pump flow control (Europump, 2004)[18], but the most common is:

A. Valve Throttling Control;

B. Bypassing Valve Control;

C. On-Off Control;

D. Variable Speed Control.

Figure 12 below illustrates the four above-mentioned methods of flow control; and they are also numbered accordingly (ABB Drives Technical Guide Book, 2013)[19]:

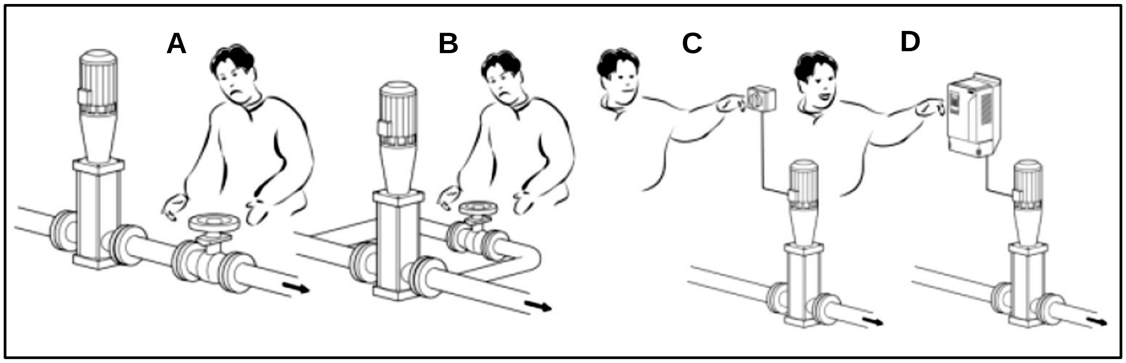

Figure 12: Different Types of Pump Flow Control

For the purposes of this study, *Bypassing Valve Control* and *On-Off Control* may be disregarded, for the following reasons:

1. This study is focused on energy efficiency, and the bypassing control would not significantly change the pump power consumed over different load ranges. The reason for this is that the flow through the pump would stay the same, as the required flow would be supplied to the main line: while the surplus flow circulates to the pump suction (Figure 8 in section 2.4.1 shows that the power consumption stays constant if the flow stays constant).
2. The on-off control is impractical, as the condensate system of a power station has to provide a constant flow and pressure, in order to protect the heat exchangers, to sustain the protection systems, as well as to maintain the levels in the De-aerator and the Condenser. High start-up torque and current

[18] Sections 8.1 and 8.2

[19] Page 22 in Section 4

are also harmful to the pump (Al-Bahadly, 2007)[20]. This is the reason for the pumps needing to be refurbished after a certain amount of starts-ups.

Therefore, the two methods of flow control that will be compared in this study are valve (or throttling) control, which is the current CEP flow control method, and variable-speed control.

2.4.4 The Pump Variable Speed Control

The two main types of variable speed drives available – utilizing speed variation – to control the flow through pumps are:

1. The Hydraulic Variable Speed Coupling (HVSC); and
2. The Electrical Variable Speed Drive (VSD).

By varying the speed at which the pump is operating, the flow and head of the pump can also be varied (M. Pugh, 2000)[21]. With HVSCs, the pump speed is varied by a coupling installed between the pump and the pump motor. In contrast, electrical VSDs vary the pump speed by changing the frequency (and ultimately the speed) of the motor.

Figure 13 illustrates the effect that variation of the motor speed has on the pump operating variables – with the solid line representing the conditions at 1485 rpm, and the dotted line at 1337 rpm. Since an HVSC requires the pump motor to be operating at a fixed speed, it is safe to assume that any changes in flow through the pump would have approximately the same effect on the power consumed by the pump motor, as with valve throttling control. The consequence is that when the present plant is retrofitted with HVSC flow control, the motivation to implement such a modification would need to be based on factors other than mere energy savings.

On the other hand, the electrical VSD control is the only type of control that would cause considerable changes to the load on the motor with speed variation (Al-Bahadly, 2007)[22]; and it is the only energy-efficient method of flow control in variable-flow pumping applications (ABB Drives Technical Guide Book, 2013)[23].

[20] Section 3.1 on pg.54
[21] Section 4.2.5.2
[22] Section 3.3 on pg. 55
[23] Page 23 of Section 4

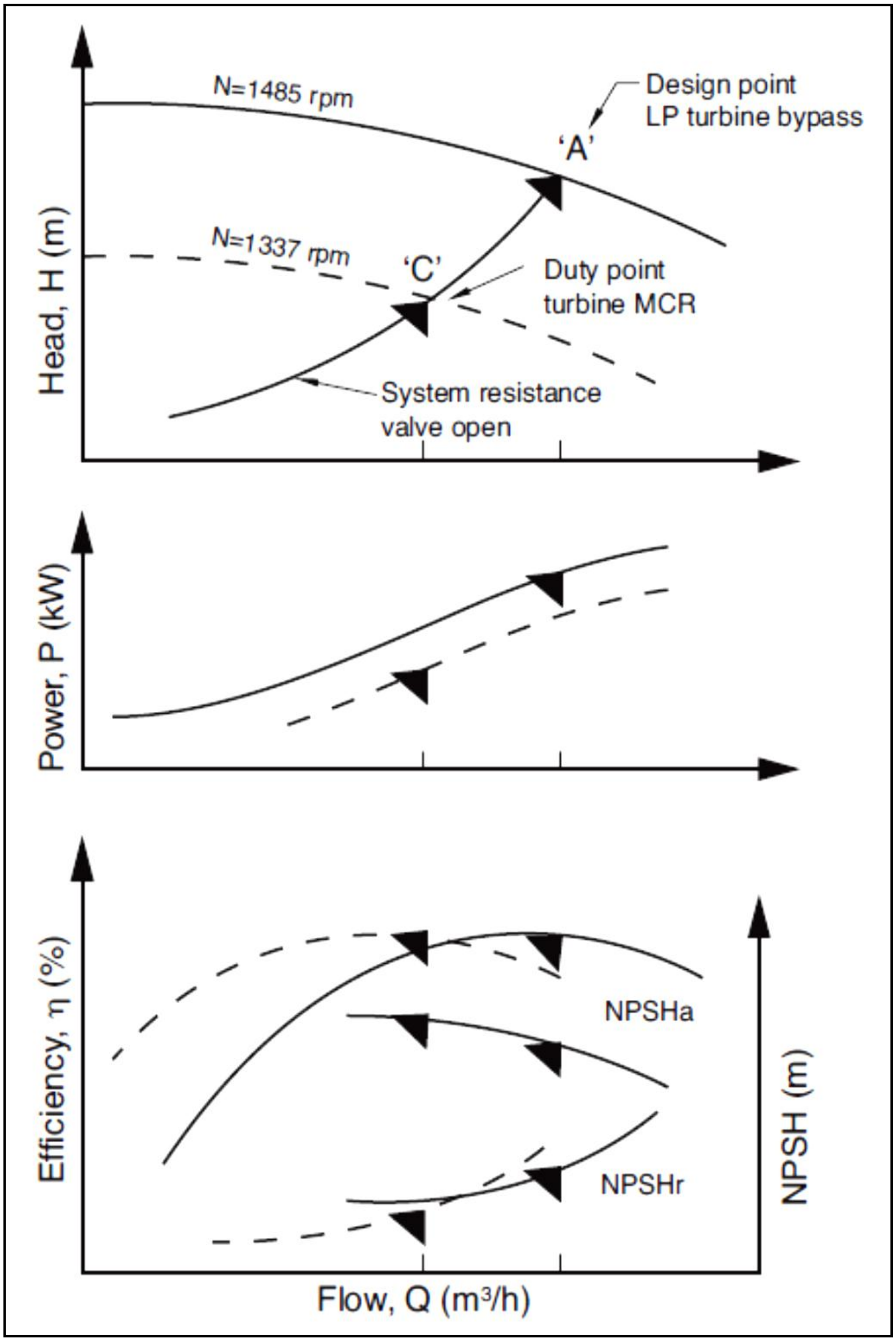

Figure 13: Pump Performance Curve for Different Operating Speeds

2.5 Hydraulic Variable Speed Couplings

This section explains the operation and characteristics of an HVSC[24] with the help of drawings and illustrations (Johannes Feuchter, 2012)[25].

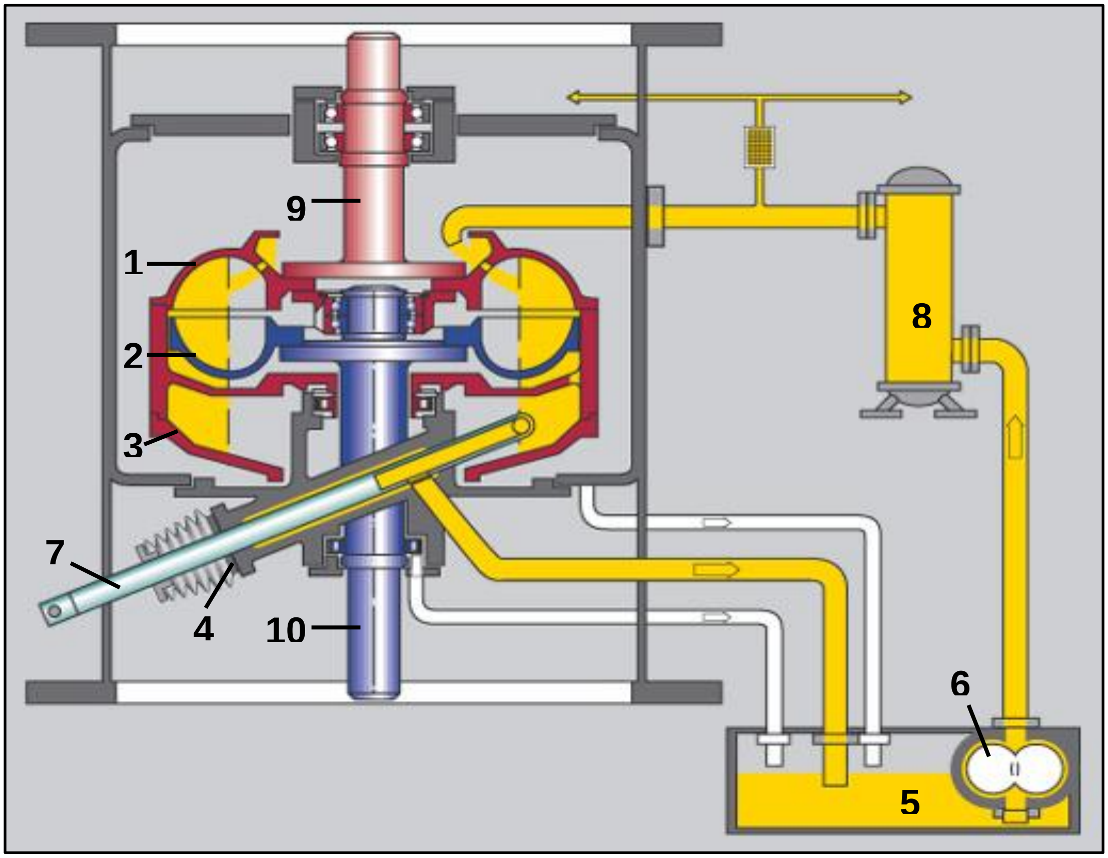

Figure 14: HVSC Sectional View

An HVSC has the following components, as depicted above (Figure 14):

1. Pump wheel
2. Turbine wheel
3. Shell
4. Scoop tube housing
5. Oil sump
6. Oil-circulation pump
7. Scoop tube

[24] An explanation of operation given from a training manual by HVSC supplier project sales engineer, Vickey Padayachee

[25] Pg. 5

8. Working oil cooler
9. Input Shaft (connected to the CEP motor)
10. Output Shaft (connects to the CEP)

The top input shaft is directly connected to the coupling pump wheel; and the bottom output shaft is directly connected to the coupling turbine wheel. The difference in speed between the two wheels is controlled by the amount of oil in the shell, which in turn, is controlled by the scoop tube and the oil-circulating pump. Variations in the amount of oil in the shell would cause a variation in the torque transferred from the pump wheel to the turbine wheel of the coupling. Excess heat generated by the friction between these two wheels is rejected in the oil cooler.

2.6 Electrical Variable Speed Drives

There are 3 different ways to change the speed of an electric motor (J.Tsou, 1998)[26]:

1. Change the number of poles on the motor;
2. Change the slip of the motor (wound motors); or
3. Change the frequency of the energy supply of the motor.

Changing the frequency of the energy supply is the only energy-efficient and practical option worth considering in this case.

The largest advantage that an electrical VSD has over any other type of flow control is the increased energy savings at partial load. Disadvantages of this type of control system are added plant complexity, more expensive maintenance, harmonics, etc. (J.Tsou, 1998)[27]. More advantages and disadvantages of this type of control system will be discussed later in this section.

[26] Pg. 10 & 11 of Section 7
[27] Pg. 11 & 12 of Section 7

2.6.1 Background

According to the Variable Speed Pumping handbook[28], pumps account for 22% (Figure 15) of the world's energy demand; and 20% to 50% of this power can be saved by installing an electrical VSD on these pumps (Pemberton, January 2005).

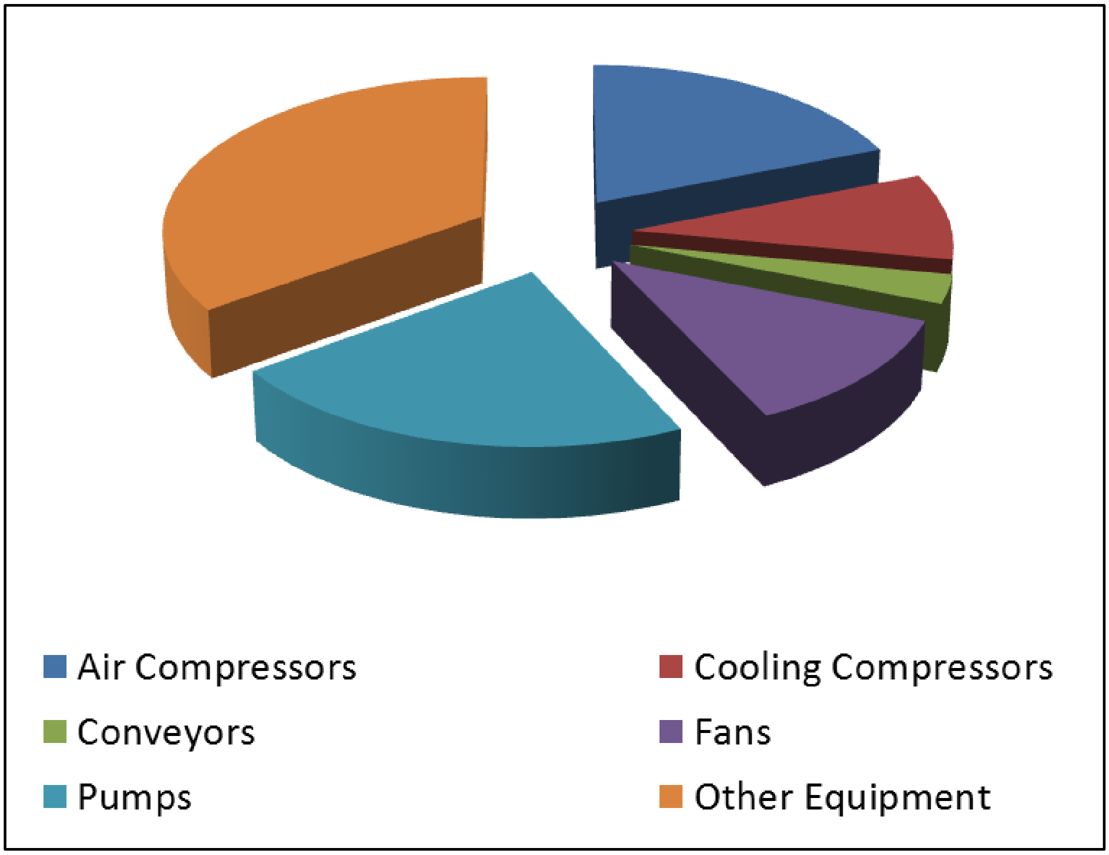

Figure 15: Global Electricity Demand

With the focus on reducing auxiliary power consumed, while still meeting production requirements, many companies have installed VSDs on their motors. According to the British Pump Manufacturers Association's website (BPMA Website), an estimated R1.2 billion has already been saved by using high efficiency motors and variable speed drives to date.

The application of VSD flow control not only reduces the life-cycle cost of running a centrifugal pump; but the greenhouse gas emissions can also be reduced by more

[28] Pg. 2 of Section 1

than 35% in applications where the pump runs for more than 2000 h/year (Ferreira, June 2011)[29].

VSDs have been internationally identified as the preferred technology when designing a new power station, because of their increased reliability, lower operating costs, and flexibility of control (Europump, 2004)[30]. In South Africa, Eskom engineers specify VSD flow control as the preferred technology (van der Westhuizen and Cattaert, 2009)[31] in the requirement specifications, when designing a new power plant.

The main focus of this research will not be to determine the feasibility of building new power stations with VSD flow control technology, but rather retrofitting of the existing plants with VSD flow-control technology.

2.6.2 Variable Speed Drives: Technical Information

VSDs provide for flexible and efficient motor and driver equipment control with higher precision, accuracy and response times, than those of a mechanical control system (Moncrief, 2001)[32]. Usually, a VSD consists of three functional blocks, a rectifier, a DC link, and an inverter (Figure 16) (Carrier Corporation, 2005)[33]. The rectifier converts AC voltages and currents into DC voltages and currents. The DC link filters the output of the rectifier and provides a DC source for the inverter. The inverter is a set of semiconductor switches that create AC voltage and current to drive the motor.

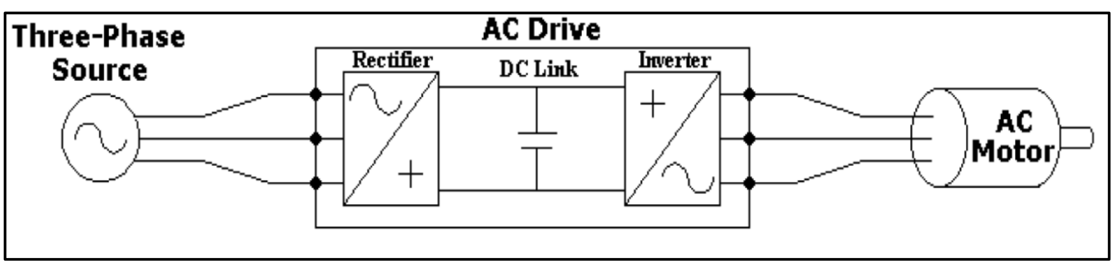

Figure 16: VSD functional blocks

[29] Pg. 2123 and 2124 of Vol. 58 of the IEEE TRANSACTIONS ON INDUSTRIAL ELECTRONICS, June 2011
[30] Pg. 3 of Section 1
[31] Pg. 18 & 19 of World Pumps Journal, December 2009
[32] Pg. 1 of Section 3
[33] Pg. 3

Most new VSDs have a Master Process-Control System (MPCS). An MPCS is the interface between the electrical drive and the rest of the system. The MPCS processes the information from transmitters, sensors, switches, etc., as well as the time response of system components (such as motors, drives, PLCs, process transmitters, sensors, etc.) to ensure that the VSD responds as quickly and precisely as possible. This, in turn, ensures that the system efficiency is highest and translates into less downtime, increased production, and decreased costs.

2.6.3 VSD vs. Fixed Speed Selection

The Electric Power Research Institute, or EPRI, has compiled a guide (Moncrief, 2001) in an attempt to help large electricity users to evaluate the possibility and advantages of retrofitting their motors with VSDs. This guide can be used as a tool in determining whether a CEP VSD retrofit would be a feasible modification.

2.6.3.1 VSD Installation Analysis

The *Guide to the Industrial Application of Motors and Variable Speed Drives* (Moncrief, 2001) emphasizes those key points that would guide the user in pre-qualifying the applications, where variable speed control is desired[34]. Addressing all issues in the planning phase of the project would ensure that no time and money are wasted.

According to this guide, the following four steps should be followed:

1. **VSD Analysis** – The calculation of savings expected, as well as the economic evaluation of the project. Once the return on the investment can be determined, and the modification proved successful, the project can be implemented.
2. **VSD Purchase Specifications –** Determining what the specifications of the VSD are a very difficult task – with modern technological advances presenting more options and extras that could affect the capabilities of the system.

[34] Pg. 1 of Section 4 in the *Guide to the Industrial Application of Motors and Variable Speed Drives*

3. **VSD Installation-Construction Information** – Site engineers should carefully examine the design and layout of the plant components, in order to ensure that no problems intervene with the performance of the VSD (such as high vibrations, difficult installation, performance-monitoring difficulties, etc.).
4. **Start-up and Training** – Before start-up, the equipment must be checked and calibrated, in order to guarantee the desired performance. Also, maintenance and operating personnel should be trained prior to commissioning.

These four steps should be addressed in chronological order, with the last three being the responsibility of site engineers and operating and maintenance personnel.

2.6.3.2 Screening for Energy Savings

With regard to the first step in section 2.6.3.1, and based on the research from EPRI technical publications and several world-renowned VSD experts, Moncrief (Moncrief, 2001)[35] focuses on some main characteristics, which are common to all successful and unsuccessful applications. These characteristics can be used to evaluate the CEPs at the power station under investigation, in order to determine whether a VSD retrofit would be a successful modification.

In some cases, the motivation for retrofitting to a variable speed drive is based the higher precision of control offered by this technology when compared to tradition mechanical flow control. In other cases where motivation is required, Moncrief uses the following characteristics to "screen" the applications:

1) **Rating the application:** Only variable torque load applications are economically viable for a VSD retrofit. The applications below are rated from good to poor, based on how much their load changes with variations in speed.
   a) **Good:** Centrifugal fans, *pumps*, blowers, compressors
   b) **Fair:** Other fluid-handling equipment (blenders, screw compressors)
   c) **Poor:** Conveyor belts, grinders, rollers, winders

[35] Pg. V (Report Summary) of the *Guide to the Industrial Application of Motors and Variable Speed Drives*

2) **The variability of the load:** If the equipment is required to operate at an average operating speed, which is lower than the rated speed, then a VSD retrofit would be a profitable option. Applications where the motor is required to operate at close to rated speed for large periods of time are poor candidates for a VSD retrofit.

3) **Motor Size:** The cost of acquiring a VSD increases with an increase in motor sizing; but auxiliary power saving is more important than any initial costs over the operating lifetime of the motor system. Thus, a larger motor would be a preferred option over a smaller motor.
   a) **Good:** Motors rated *over 55 kW*
   b) **Fair:** Motors rated between 22 and 55 kW
   c) **Poor:** Motors rated below 22 kW

4) **Operating Hours:** The longer the motor system is to be in operation, the greater the energy savings that could be expected from installing a VSD on the motor drive.
   a) **Good:** *More than 6000 hours annually*
   b) **Fair:** Between 2500 and 6000 hours annually
   c) **Poor:** Less than 2500 hours annually

5) **Other Considerations:** The following considerations could also influence the decision for incorporating a VSD into a motor system if further motivation for a VSD installation is needed:
   a) Motor life extension
   b) Reduced motor starting current
   c) Increased time between motor rewinds
   d) Reduced maintenance costs
   e) Better grade of process and product control
   f) Regulation and reduction of pollution
   g) Reduced audible load
   h) Reduced cooling load

i) Improved system reliability

If the screening characteristics above are evaluated, the decision tree (Figure 17) below can be used to determine whether retrofitting the plant under investigation with a VSD would be successful:

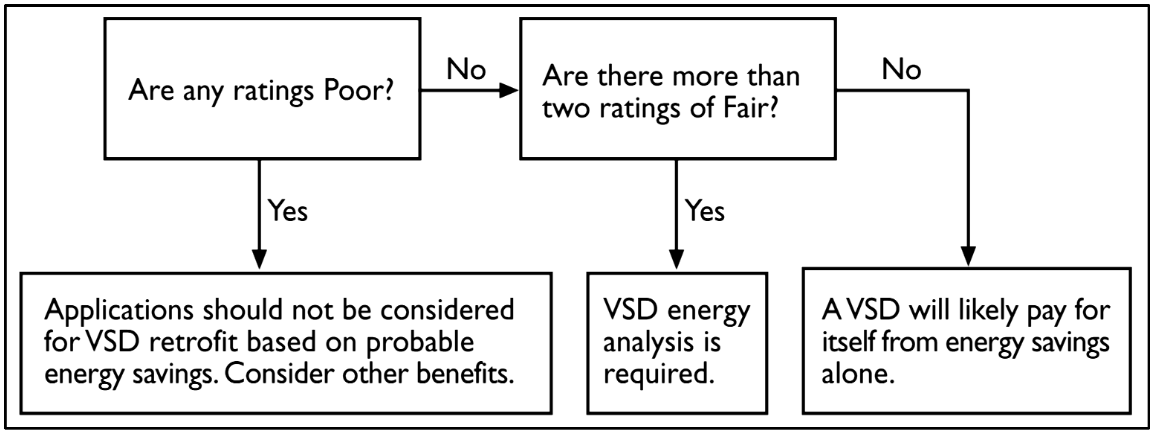

Figure 17: Decision tree for determining VSD application credibility

2.6.3.3 Further Information Concerning VSD Applications

The *Guide to Industrial Motors and Variable Speed Drives* (Moncrief, 2001) and the *Practical Speed Drives and Electronics* book (Barnes, 2003) will guide the user in terms of VSD technical specifications, system reliability, and power quality issues that must be taken into account when designing a retrofit.

2.6.4 Calculation of Energy Savings

To determine the savings when running the pump at a reduced speed, most sources recommend the use of the pump affinity laws (Sulzer Pumps, 1998)[36].

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$ - **Equation 3: Affinity Equation for Flow**

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$ - **Equation 4: Affinity Equation for Head**

[36] Pg. 55 of the *Centrifugal Pump Handbook* (Sulzer Pumps, 1998)

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$ - **Equation 5: Affinity Equation for Power**

Where N = pump speed

Q = flow

H = head

and P = power

The equations above support the argument for reducing the pump speed; and why this would have a significant effect on the head delivered and on the power consumed by the pump.

The decrease in power consumed has a significant impact on the operating costs of a plant, especially when considering the entire operating lifetime of the pumping system. Over the entire life cycle of a pump, the initial investment costs make up only 5% of the total costs (Europump, Hydraulic Institute, 2000)[37]; while the rest is spent mainly on energy consumption (about 50% for medium-sized industrial pumps); and the remaining is spent on maintenance.

However; the Affinity Laws are governed by Equations 3, 4 and 5; and they give a good indication of how performance curves change with speed; but they are not accurate in determining the exact savings achievable when implementing VSD control.

To further support the preceding statement, Equation 6 expresses the power consumed by the pump in terms of its efficiency:

$$P = \frac{\rho g H Q}{\eta}$$ - **Equation 6: Pump Power Consumed**

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[37] Pg. 1 of the executive summary of *Pump Life Cycle Costs*

From section 2.4.4, it may be observed in Figure 13 that any change in pump speed will result in a change in the pump's efficiency (van der Westhuizen and Cattaert, 2009)[38].

Also, since the Affinity Laws do not account for the changes in efficiency with changes in pump speed, they cannot give an accurate representation of the exact savings expected when operating the pump at reduced speed.

Rather than using the Affinity Laws, the pump performance curve (seen in APPENDIX C)[39] with different duty points is used to find the actual power consumed. From these curves, the flow and head values were determined for every operating condition (if the pump was performing at design conditions). Substituting these values into Equation 6 would give the power required to drive the motor.

The Affinity Laws are accurate enough to use when the pump operates at zero static head; and this is due to the system resistance curve and constant efficiency curves having the same orientation on the pump performance curve when pumping to a zero static head (Europump, 2004)[40].

2.7 Merits and Alert to Consider with Variable Speed Drives

Most of the advantages and disadvantages of implementing electrical VSD speed control are listed in section 11 of the Variable Speed Pumping handbook (Europump, 2004).

2.7.1 Merits Specific to Electrical VSD control

This handbook identifies electrical VSDs as the most energy-efficient method of controlling pump speed in applications with high friction loss – that require flow or pressure control.

[38] Diagram on Pg. 38 of the Literature Survey
[39] The performance curves was supplied by the pump OEM
[40] Pg. 6 & 7 in Section 2

Listed below are some of the advantages (Europump, 2004) to gain in implementing electrical VSD control:

- Energy and maintenance savings, pump and process reliability improvements, better process control, and less fugitive emissions (Ferreira, June 2011).
- Pumps are typically oversized to sustain the flow requirements at reduced performance; and this relates to increased throttling, energy losses, and pump wear. Since VSDs eliminate the necessity of throttling the flow though the pumps, cost savings are achievable.
- VSDs reduce hydraulic transients, as they can be soft-started/stopped, thereby reducing pressure leakages in the discharge line. The pumps are controlled with greater precision.
- Wear between bearings and rubbing surfaces decrease to the seventh power, when the operating speed of a pump is reduced (Europump, 2004).
- The Mean Time between Failure (MTBF) for VSDs has increased over the last couple of years – by up to 20 years for some drives.
- Easy to retrofit.

2.7.2 Alerts Specific to Electrical VSD control

The following alerts might increase the modification cost of a VSD retrofit; and they must be considered when implementing electrical VSD control with pumps[41]:

- Motors need to be re-insulated, as they usually do not have “inverter-duty” insulation.
- Some undesirable harmonic distortion may be expected; but this could be eliminated by specifying the maximum harmonic distortion limits[42].
- Pulse width modulated (PWM) waveform switching (especially at a high rate) can lead to electromagnetic disturbances. These can be eliminated with screened output cables, rate-of-rise filters, and sinusoidal filters.

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[41] Also in Section 11 of the *Variable Speed Pumping* Handbook
[42] Refer to Appendix A3.3 of the *Variable Speed Pumping* Handbook

- Voltages may be induced on larger motor shafts. This leads to circulating currents, which destroy the bearings. The risk may be eliminated by installing insulated bearings on the non-drive end of the motors.
- A VSD needs a ventilation source to reduce excessive heat. This has a direct influence on the efficiency loss and the life expectancy of the drive.
- As VSDs pose arc risks and electronics are not corrosion and vibration resistant, VSDs require installation in a safe, clean and controlled environment. A separate room is often needed to house the drive units.

2.7.3 Merits Specific to HVSC control

Even though no significant energy savings can be achieved by implementing HVSC control, the following merits are worth mentioning, as being specific to this type of control[43]:

- HVSC MTBF is typically 48 years, which is 3 times more than the average electrical VSD. However, careful consideration must be taken with regard to the date of installation, as a replacement of the entire drive after 48 years could have a detrimental effect on the lifecycle cost of the plant if there are only a few years left before the power station is due to be decommissioned.
- Parts will not reach obsolescence soon, as most of the parts are mechanical, and are available off-the-shelf, or by special manufacture (if obsolete).
- Efficiency loss through the drive is lower than that of an electrical VSD.
- An HVSC can run continuously for 8 years between maintenance intervals.
- A HVSC can be used in hazardous areas (a.k.a. dirt, moist, vibration, heat, etc.) without major risk.
- The speed range over which a HVSC can operate is higher than that of an electrical VSD (exceeding 20000 rpm).

Other advantages not mentioned above are shared with the advantages of electrical VSD control, such as improved process control, reduced hydraulic transients, and a decrease in the wear of the pump's bearings, and other moving parts.

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[43] Information in this chapter was supplied by HVSC project sales engineer, Vickey Padayachee, as well as brochures from the product supplied.

2.7.4 Alerts Specific to HVSC Control

The following alerts could increase the costs of an HVSC retrofit; and they must be considered before the project is initiated:

- The HVSC unit for a pump is mounted between the pump and the motor. Most condensate extraction pumps (also applicable to this case study) are vertically mounted – with the motor bolted directly unto the pump. Installing the coupling between the pump and the motor would increase the inertia of the complete unit for this study and the CEP mentioned in section 2.3.1. The HVSC weighs 2 tons and is 1.6 m high, and the motor weighs another 5 tons. The pump would not be able to handle this extra weight. Consequently, a dynamic vibration analysis, and a support structure would have to be included in the design proposal for such a retrofit. Extra costs include the engineering design, the manufacture, and the installation of the support structure.
- The HVSC's oil cooler needs cooling water at about 25 m^3/s and 30 °C to reduce the heat during operation. Demineralised water from the power station's auxiliary cooling plant would have to be relayed to the oil coolers[44]. This would mean that extra costs would have to be allocated to the design, manufacture, and installation of such a pipeline.
- HVSCs are very common in horizontal applications, but not so much in vertical applications. The result is high initial capital investment costs that would not be redeemed by the subsequent energy savings.

2.7.5 Important Limitations to Speed Variation in General

An important limiting factor that must be considered when changing the speed of a pump is the minimum allowable speed of the journal and the thrust bearings[45]. The bearings are designed to carry the weight of the rotating element of the pump, and to absorb the pump's axial hydraulic thrust. These bearings are lubricated by oil; and they make use of "tilting pads" to push the bearing surfaces away from each other.

[44] Information received from various quotations supplied by HVSC suppliers
[45] This was identified during a meeting with Eskom Corporate Pump Specialist, Willem van der Westhuizen.

To keep a hydrodynamic oil wedge between bearing surfaces, the bearing operates above a specified minimum speed, which is dependent on the viscosity of the oil. The biggest influence on the oil viscosity is temperature, which means that full-load (or maximum temperature) conditions should be used to determine the minimum speed.
The bearing Original Equipment Manufacturer (OEM) was contacted to aid in the calculation of this minimum speed; and they calculated[46] the minimum speed to be 390 rpm at hot oil bath temperatures.

Also important to consider is the cooling requirements of the motor[47]. The motors driving the pumps at the power station investigated are air-cooled (Europump, 2004)[48], and cooling fans are directly coupled to the motor drive-shaft. This means that reducing the shaft speed of the motor would also reduce the speed of the cooling fan. The OEM did, however, confirm that the motor cooling fans are designed to have sufficient cooling capacity to cool down the motors at speeds above half the rated speed, which is equal to 750 rpm for the case study.

An important consideration when running a pump at a reduced speed is the natural frequency of the pump. According to the pump OEM, a typical value on the first natural frequency of this specific pump is 1.5 times the operating speed. This means that vibrations would be at their highest at a speed of 2250 rpm; and this would not be a concern, as the pump will never be required to run above the rated speed. According to the pump performance curve, the maximum flow demand can be met by running the pump at full speed with the discharge valve open (APPENDIX C).

[46] This report is confidential and available to the reader by special request.
[47] This was identified in a meeting with the motor OEM Commercial Manager, Brian Lindsay.
[48] And also according to the confidential original design manuals, which is available to the reader by special request.

2.8 Conclusion of the Literature Survey

In retrospect, the following challenges were discussed in the literature review:

1. A background was given on the Condensate and LP Feed-Heating system, and where it forms part of the water-steam cycle of a power station and the Rankine Cycle.
2. The relevant pump theory was discussed, with a specific focus on pump flow control.
3. From the information supplied on flow control, it is clear that mechanical control is dated and inefficient; and more large electricity users are considering variable speed drives when building or retrofitting a plant.
4. The merits of an HVSC retrofit were discussed; but as was evident from sections 2.7.3 and 2.7.4, installing these drives with a pump is only beneficial if the pump is operating at extremely high speeds, in hazardous locations, or where Mean Time between Failure (MTBF) is more important than energy efficiency.

In conclusion, an electrical VSD retrofit was identified as the most energy efficient and advanced technological option when flow is controlled through pumps, and up to 50% savings in auxiliary power that can be expected (section 2.6.1). However, careful consideration must be given to the minimum operating speed of the pump, emphasizing the importance of installing minimum speed protection.

Given the information supplied by the literature survey, the actual energy savings can now be calculated if the pumps at the power station investigated are retrofitted with electrical VSD flow control.

3 Evaluation Approach

Using the information provided by the literature review, the first step in determining the feasibility of a VSD retrofit was to develop a systematic evaluation approach. The following evaluation approach was developed by trial-and-error throughout the research period:

1. Pre-qualify the pumps under investigation[49] to determine whether they are suitable for a VSD retrofit. This will eliminate unnecessary expenditure if the pumps are not suitable candidates to bring under investigation.
2. Create a mathematical model of the water-steam cycle from the power station heat balance diagrams for the plant under investigation.
3. Create a simulated model of the plant and components – simulating the current design of the plant under investigation.
4. Conduct a performance test on the pumps under investigation.
5. Evaluate the load profile of the pumps under investigation.
6. Calculate the savings expected when utilizing VSD pump flow control, instead of conventional throttling control.
7. Verify the calculations with the simulated model developed above.
8. Conduct an economic evaluation to determine the capital costs and payback period upon implementation of a VSD retrofit.

[49] This is done by following the guidelines in Section 2.6.3.2 of the Literature Survey

4 Pre-qualifying the Pumps of the Power Station under Investigation

In paragraph 2.6 of the literature survey, the first step in qualifying the existing plant is screening the plant to determine how well it is suited for a retrofit.

Table 1 below contains the results of the screening process:

Table 1: VSD Screening Results

	Screening Measure	Application	Poor/Fair/Good
1	Rating the application	Pump	Good
2	Variation of the load	High Load Variation	Good
3	Motor size	More than 55 kW	Good
4	Operating hours	More than 6000h annually	Good

Using the information in Table 1, and the other considerations in section 2.6.3.2; and then applying this information to the decision tree in Figure 17 on pg. 46, the screening process has identified the CEPs investigated ideal for a VSD retrofit, with energy saving most likely paying for the investment costs – depending on the remaining Life of Plant (LOP).

Based on the estimated cost of a VSD retrofit, and the expected influence of deteriorating plant performance; it was decided to still conduct a comprehensive study to determine the feasibility of the project.

This will also help Eskom to budget for the required capital cost necessary to implement the CEP flow control retrofit.

5 Modelling of the Plant

5.1 Power Station Heat-Balance Modelling

The heat-balance diagram shown in APPENDIX A illustrates what thermodynamic properties can be expected for the water and steam at the different points on their path through the Rankine Cycle. These properties are under design conditions at 100%, 90%, 80%, 60% and 45% load (Figure 36 and Table 13 in APPENDIX A).
In APPENDIX B, EES (Engineering Equation Solver) software was used to determine all the thermodynamic properties of the water/steam at the different sections in this diagram at 100% load (Figure 35). To easily distinguish between different points in the flow path, the points in this diagram are numbered from 1 to 35.

5.1.1 Rankine Efficiency Calculation

To determine the efficiency of the Rankine Cycle, and to calculate the work done by the pumps, the heat load on the boiler, and the work produced by the turbine, the thermodynamic properties of all the points in the water-steam cycle should be known. This includes the temperature (T), pressure (p), enthalpy (h), and entropy (s) values, as well as the mass flow ($\dot{m}$) through all the individual components.

Using the steam tables, EES calculated the thermodynamic properties for all the points in APPENDIX A; and then, to guarantee the accuracy of heat balance calculations, a mass and energy balance was done over all the heat exchangers in the system to determine whether the mass flows are correct.
For example:

$$\dot{m}_{in} = \dot{m}_{out}$$

and when considering the De-aerator mass balance in Figure 35 (APPENDIX A), the following equation should be equal to zero:

$$\dot{m}_{DA.b} = \dot{m}_{17} - \dot{m}_{16} - \dot{m}_{20} - \dot{m}_{19} - \dot{m}_{18} = 0$$

The mass balance calculations were checked and all proved to be using the correct mass-flow values (Figure 37 in APPENDIX B) – with the exception of the LP gland steam and the gland steam condenser flows, which were balanced and corrected (LP1, LP2, and LP3 in the EES Code). All the mass flows in the heat-balance diagram are key variables in calculating the pump work, the turbine work, the boiler heat input, the condenser heat reduction, the efficiency calculations, etc.; and since small deviations in these flows would ultimately result in inaccuracies in the Rankine efficiency calculation.

The formulas for calculating the pump work, the turbine work, and the boiler heat absorbed, as well as the condenser heat reduced are very similar:

$$\dot{\boldsymbol{W}}_{\boldsymbol{turbine/pump}} = \dot{\boldsymbol{m}}_{\boldsymbol{turbine/pump}} \times \Delta \boldsymbol{h}_{\boldsymbol{turbine/pump}}$$ - **Equation 7: Work Equation**

$$\dot{\boldsymbol{Q}}_{\boldsymbol{boiler}} = \dot{\boldsymbol{m}}_{\boldsymbol{boiler}} \times \Delta \boldsymbol{h}_{\boldsymbol{boiler}}$$ - **Equation 8: Heat Equation**

The Rankine efficiency of the cycle is determined by dividing the Net Work produced by the total Heat Input into the boiler, as described by equation 1:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{boiler}}$$

From the equations, the plant Rankine efficiency was calculated as being 44.85% (Figure 37 in APPENDIX B).

5.2 Flownex Modelling

The Flownex model variable inputs were obtained from the power station heat-balance diagrams (APPENDIX A), supplied by the company awarded the turn-key contract for building the power station under investigation. The heat-balance diagrams were subject to acceptance tests during the commissioning.

Five different heat-balance diagrams display the thermodynamic properties of the water and steam at five different loads (100%, 90%, 80%, 60% and 45%), as can be seen in APPENDIX A, where the Condensate and the LP Feed-Heating System's values are displayed in Table 13 and Figure 36.

The physical layout of the model was designed from the detailed arrangement drawings (APPENDIX E) of the Condensate and the LP Feed-Heating system. These drawings are the Master Copy drawings; and they are the original design drawings. Any changes to the plant layout would necessitate an immediate revision of these drawings.

Enthalpy (h), temperature (T), pressure (p) and mass flow ($\dot{m}$) values are all indicated on the heat-balance diagrams at points between all "property changers" (heat exchangers, pumps, boiler, etc.). These points are numbered in chronological order from the condenser outlet back to the inlet (as in APPENDIX A) to easily identify where each property is situated on the T-s diagram (Figure 2). The results from the EES programming were also used in the Flownex modeling, where any gaps in the property values existed.

5.2.1 Modeling Procedure

Every pipe in the drawings in APPENDIX E was included in the Flownex model, and due to the platform of the individual components complicating the model, it was divided into two subsections: the Condensate System and the LP Feed-Heating System. In order to model the pressure drop over the Condensate System, all the variables had to be specified from back to front, meaning: from the De-aerator to the Condenser. The first step was to import all the physical components into the model from the plant layout drawings – before assigning any thermodynamic properties to the individual points. The detailed modeling procedure is shown in APPENDIX F.

Information in the power station design manuals was missing; and as a result, there was no sizing specified for the orifice labeled C16 in Figure 40 (in APPENDIX E). This orifice was initially installed in the line at this exact location, in order to force a certain percentage of the main feed water to flow through the gland steam condenser. Due to this shortage of information, a flow measurement was conducted at the power station to determine the exact flow though the gland steam condenser.

The resulting flow measured at full load was 112 kg/s, which was 25% of the main feed water flow at the time of the test. It is, therefore, safe to assume that the gland steam condenser was designed to accommodate 25% of the main feed water flow, and the orifice was sized to supply this flow.

The designer function in Flownex was used to determine the diameter of the orifice, and Flownex calculated this diameter to be 310 mm.

6 Calculation of Energy Savings Achievable by Implementing VSD Flow Control

6.1 OEM Research Report 2003

A rudimentary study was done by the OEM in October 2003 to investigate how a variable speed drive installed on the CEPs would affect the power consumed by the pump[50]. The following duty points were used to determine their results (available from the performance curve in APPENDIX D):

Table 2: Pump OEM Data Points

Parameters	Operating Points at Fixed Speed			
	MCR	Rated	Bypass	Run-out
Flow (kg/s)	425	542	615	635
Pressure (m)	184	146	114	110
Power (kW)	924.3	1004.25	1074.66	1123.33
Efficiency (%)	83	77.3	64	61

Using a software package that the OEM developed to predict pump performance at reduced speed, the OEM obtained the following results (Table 3):

[50] This report contains confidential information and is available the reader by special request

Table 3: Pump OEM Results 2003

Parameters	Operating Points in 2003	
	MCR Throttled	**MCR Reduced Speed**
Speed (rpm)	1485	1258
Head (m)	184	114
Efficiency (%)	83	80
Power (kW)	924.3	594.12

The study by the OEM concluded that a saving of 330.2 kW in pump-power consumption is achievable. The OEM did, however, recommend that a feasibility study and an economic analysis be completed by the power station's site personnel, in order to evaluate the retrofit based on plant performance, operation and all other plant-specific factors.

6.2 OEM Research Report 2013

Currently, the pump OEM has a long-term partnership agreement contract with Eskom, which benefits both parties, by keeping an open channel of information and the sharing of expertise.

The OEM was approached to supply the pump characteristic curves and the torque speed curves that would be used by the drive supplier(s).

The performance curves at reduced speed were plotted by entering all the pump data into the OEM's software (an upgraded version of the software used in the study of 2003). This software can also predict the performance of the pump, when operating at lower speeds. The curves plotted by the OEM are shown in Figure 18; and the results are displayed in Table 4 below:

Table 4: Pump OEM Results 2013

Parameters	Operating Points in 2013	
	MCR Throttled	**MCR Reduced Speed**
Speed (rpm)	1485	1260
Head (m)	184	113
Efficiency (%)	83	79.5
Power (kW)	924.3	605

This means that the saving in auxiliary power calculated by the OEM in 2013 amounts to 319.3 kW.

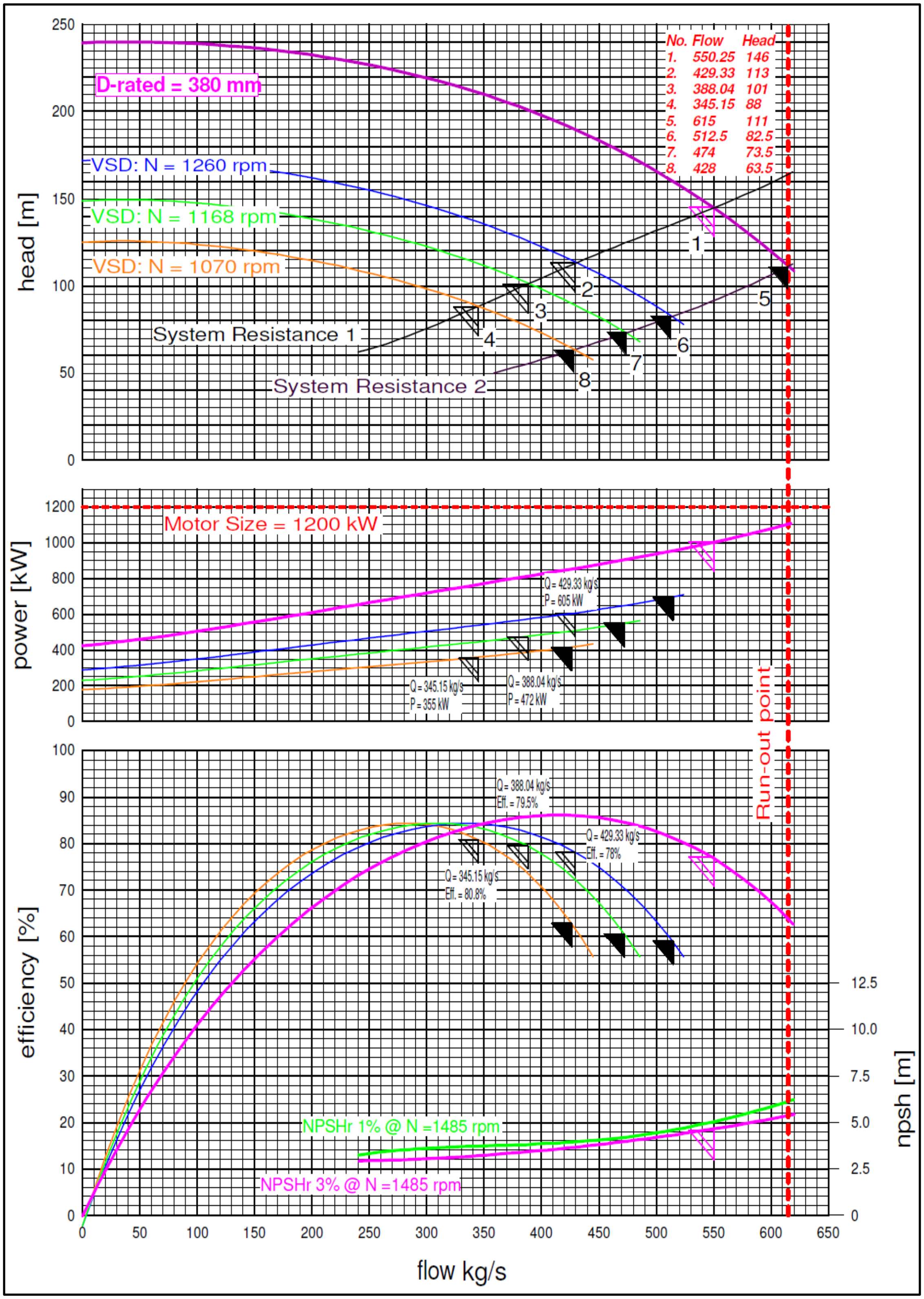

Figure 18: Pump OEM Predicted Performance Curve at Reduced Speed

6.3 EES Calculation of Energy Savings Expected

In APPENDIX D, the Affinity Laws (see Equations 3, 4 and 5) were used to calculate the energy saving when running the pumps at a reduced speed utilizing electrical VSD control. From Table 2, the conditions at the RATED CEP duty point (as seen in Figure 19) will be used as the conditions that the pump experiences when not running with increased system resistance (meaning that the throttling valve is open).

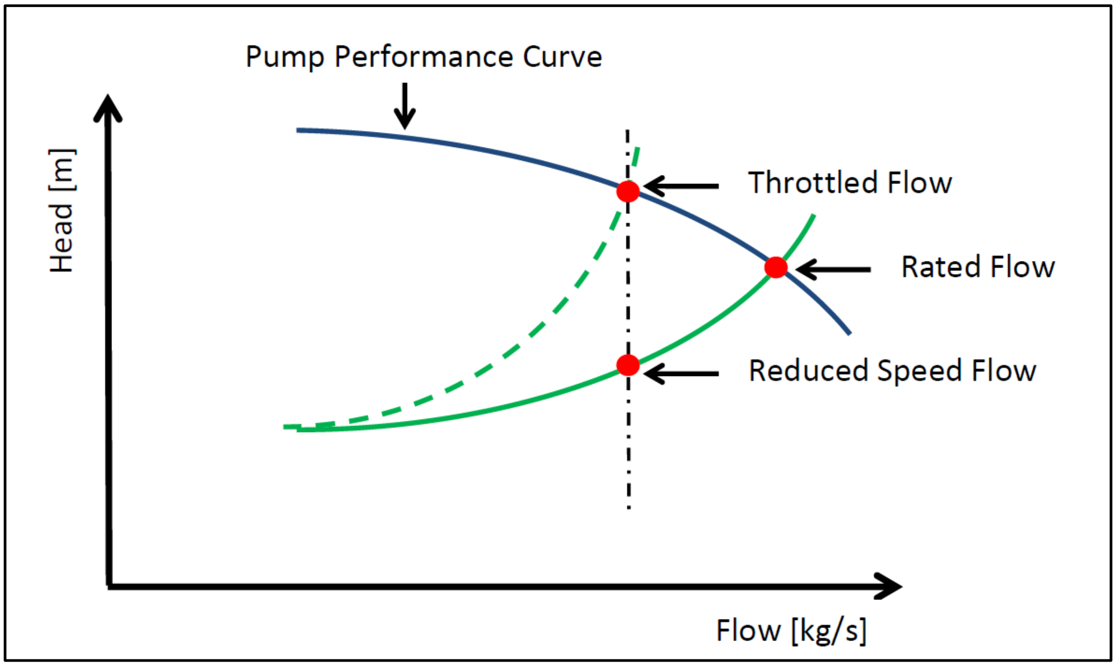

Figure 19: Pump Operating Points

The RATED duty point flow conditions were specified as “Condition 1”, and the reduced speed flow conditions as “Condition 2” (Figure 19). All the variables for “Condition 1” can be read from the pump performance curve (APPENDIX C), which means that one variable has to be specified to calculate all the other variables at “Condition 2”.

Because the flow through the pump is the most important variable for maintaining the level in the condenser and the DST, this will be the first variable to specify. Flow is normally throttled to 429.3 kg/s, according to the 100% load heat-balance diagram (APPENDIX A).

The results from EES as calculated from the Affinity Laws are tabulated in Table 5:

Table 5: EES Affinity Laws Results

Parameters	Operating Points	
	Rated Duty Point (542 kg/s)	**MCR Reduced Speed (affinity laws)**
Speed (rpm)	1485	1176
Head (m)	146	91.6
Efficiency (%)	77.3	77.3
Power (kW)	1004.25	499

However, in section 2.6.4, the Affinity Laws were found to be inaccurate in calculating the exact change in pump head delivered and power consumed after a change in speed. To support this, APPENDIX C shows how two system resistance curves intersect with the pump performance curve. System resistance curve 1 represents the resistance the pump would experience when the plant is running under normal operating conditions with the LP Bypass spray water valves closed. However, when the LP Bypass spray water valves are open, the system resistance curve drops down to curve nr. 2. This means that at the design flow of 429.3 kg/s under normal operating conditions, the head delivered by the pump should be more or less 115m, which is far from the 91,6m head delivered, as calculated by the Affinity Laws (Table 5).

As a consequence, it was decided to rather use Equation 6 on pg. 47, and to specify the variables that are known for the operating conditions.

$$\boldsymbol{P = \frac{\rho g H Q}{\eta}}$$ - Equation 6: Pump Power Consumed, where:

$$\dot{m} = \rho Q$$ - and this gives:

$$\boldsymbol{P = \frac{\dot{m} g H}{\eta}}$$ - **Equation 9: Applied Pump Power Consumed**

The mass flow of the water (429.3 kg/s) and the manometric head (115 m) the pump delivers is available from the pump performance curve. The only unknown is the efficiency of the system.

Substituting the head and mass flow values above into Equation 9, and estimating the efficiency at 79.5% (0.5% lower than the efficiency expected by the pump OEM in the previous section), the new pump power consumed is tabulated in Table 6 below:

Table 6: EES results from Efficiency Calculations

Parameters	**Operating Points**		
	MCR Throttled (429.3 kg/s)	**Rated (542 kg/s)**	**MCR Reduced Speed (429.3kg/s)**
Speed (rpm)	1485	1485	1257
Head (m)	184	146	115
Efficiency (%)	83	77.3	79.5
Power (kW)	924.3	1004.25	609.2

The new value for the power consumed is closer to the value that was predicted in the study done by the pump OEM in 2013, with the small difference being attributable to the differences in the efficiency values. The speed of the pump was calculated, using the Affinity Laws, and specifying the power consumed above, as the second condition variable.

This means that an auxiliary power consumption of 315.1 kW can be saved by using a VSD to control the flow of condensate through the pumps.

This energy saving calculated is a lot less than the 425.3 kW calculated by the Affinity Laws in Table 5. The difference in these two values is evidence enough that the Affinity laws are inaccurate in this case study; and that careful heed should be taken when using these equations in a feasibility study.

7 The Plant Performance Test

A good understanding of the performance of the pumps is essential before a recommendation can be made for the savings to be expected by implementing a VSD retrofit. The savings potential was identified in the previous chapter; but reduced pump performance could have a negative impact on the savings potential. Deviations from the original pump performance would have an effect on the pump efficiency, speed and power consumed – as will be demonstrated in this chapter.

7.1 Unit 4 Pump Performance Test

A pump performance test was conducted on one of the CEPs installed on the 4th unit at the power station under investigation.

The following test procedure was set up to do the performance test:

1. Arrange for measuring equipment and a qualified person to operate the equipment on the test day;
2. The unit is to be kept at constant load for the duration of the test;
3. Condenser level is to be kept constant (to keep the suction pressure constant) for the duration of the test;
4. The pump with the longest running hours (lowest efficiency) is to be put into service before the test.
5. Installation of measuring equipment:
    a. Pressure Gauge on the discharge of the pump;
    b. Ultrasonic flow-meter on the suction or discharge line of the pump;
    c. Second ultrasonic flow-meter for verification.
6. Pressure and flow readings are to be taken for a time period of 1 hour in 5-minute increments.

To ensure that the performance test results were as accurate as possible, and to verify local performance indicators, an external contractor was used to do the performance test of the pump with his own equipment and skilled personnel[51].

[51] External Contractor, Peter Bekink, CEO of Tuthukani Plant Performance conducted the performance test.

A second ultrasonic flow-meter was also installed by the power station's plant-performance department – to verify the first measurement; and skilled personnel from this department were tasked with operating the equipment[52].

The ultrasonic flow-meters require a pipe length of at least three and a half times the pipe diameter in both directions of the installation point, in order to eliminate any deviations in flow readings due to turbulent flow. Due to a lack of sufficient pipe length downstream of the pump, the flow-meters were installed on the suction side of the pump. The flow-meters were also installed in close vicinity to each other, to eliminate different readings due to external factors (like a blockage or large amounts of deposits in the pipe line), but also not close enough to cause any interference with the results.

Unit 4 was chosen for the test because it was taken off Automatic Grid Control (AGC), due to maintenance issues; and it was kept at constant full load. Both pumps had more or less the same running hours (+/- 40,000 hours), so they could be expected to have similar performance results.

The conditions and variables were used by the flow-meter to calculate the distance between the ultrasonic sensors:

Table 7: Ultrasonic Flow-Meter Test Variables

Input Variables	**Date**	06/12/12
	Pump Speed	1485 rpm
	Condensate Temperature	47 °C
	Test Start Time	12 : 40
	Test Stop Time	13 : 40
	Time Increments	1 minute
	Pipe Outside Diameter	711.2 mm
	Pipe Thickness	9.2 mm
Calculated Distance Between Sensors		339.7 mm

[52] Power Station Plant Performance Line Manager, Buhle Mkhwanazi organised the equipment and skilled personnel to conduct the performance test.

The test results are tabulated in Table 8:

Table 8: Unit 4 CEP Performance Test Results

Average Test Discharge Pressure	1616.8 kPa
Average Flow (Meter 1)	476.34 kg/s
Average Flow (Meter 2)	496.35 kg/s

The results of Table 8 were plotted on the pump performance curve in Figure 20:

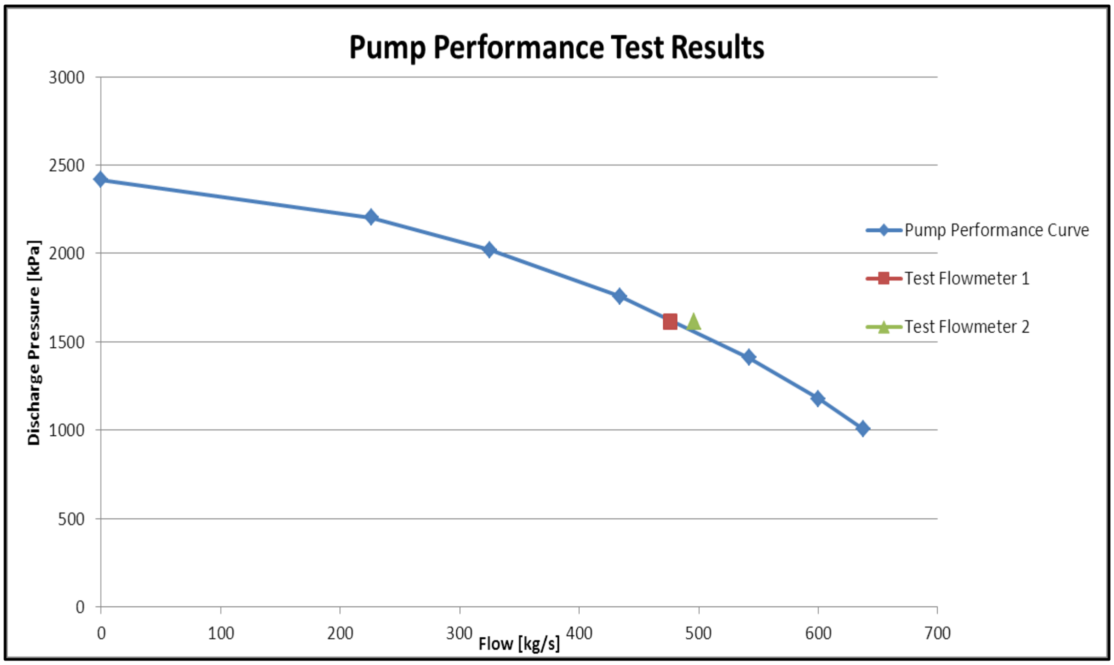

Figure 20: U4 CEP Performance Test Curve

In the curve above, flow-meter 1 has a claimed 2% measurement error; and flow-meter 2 has a claimed 7% measurement error. Taking this into consideration, both measurements are within the specific error margins; and as a result, the measurements serve as verification for each other. Taking flow-meter 1 as the more accurate flow-meter, the pump is believed to be performing according to its performance curve, as expected.

The data for the pump power consumed from the plant operating software revealed an average power consumption of 960 kW. At this power consumption, the efficiency of the pump was 82%, which also corresponds with the performance curve in APPENDIX C.

7.2 Performance Test Anomaly

From the performance test, it was observed that the mass flow through the pump was more than the specified flow at the best efficiency point (BEP) (476 kg/s instead of the 430 kg/s design flow in Figure 20). The increased flow requirement is the result of leaking valves and vessels in the system; and this forces the CEPs to pump more water to maintain the Condenser and the De-aerator level. The pump performance curve in APPENDIX C shows that an increase in the mass flow through the pump would inherently cause an increase in pump power consumption. As a result, a VSD would be required to operate at a speed higher than that which was initially calculated in section 6.3. This increase in pump speed would change the feasibility of the project, as Equation 5 shows how an increase in VSD speed influences the power consumption of the pump.

A good indicator of an increased flow requirement at a power station is the demineralised water consumption of the unit. High demineralised water consumption relates to a high volume of leakages in the system. At the power station investigated, the units were all operating at close to the same demineralised water consumption on the test day; so it was assumed that all the units were performing at similar inefficiencies.

From the performance test, it can be deduced that the pump is constantly operating at a reduced efficiency under normal operating conditions – due to this increased mass flow.

The EES calculation inputs were adjusted in APPENDIX D to take into account the increase in mass flow. The mass flow for the pump power consumed equation was changed to 476 kg/s; and according to the pump performance curve in APPENDIX B, the new head delivered is 123.5 m.

When using these values in the EES code (APPENDIX D), instead of the design values, and specifying a drop in efficiency of 1%[53] to a final efficiency of 78.5%, the new power consumed by the pump is 734.6 kW. This value was subtracted from the full load fixed speed power in Table 6; and as a consequence, the energy saving expected was reduced from 319.6 kW to 189.7 kW.

7.3 Interpretation of Performance Test Results

From the pump-performance test, an energy saving of 189.7 kW can be achieved when retrofitting the plant with VSD technology (while the pumps are still operating with increased flow). This is 20.5% of the original power consumed (section 2.3.1); while the initial saving of 319.6 kW is 34.6% of the initial power consumed as calculated in section 6.3.

Inefficiencies in the system have reduced this saving by 14.1%, which would have a significant impact on the feasibility of the project. This reduced saving supports the necessity of a performance-testing phase in determining the feasibility of a VSD retrofit. In some cases, a large enough decrease in efficiency could disqualify a VSD flow control retrofit at the specific plant.

It is, therefore, proven that a recommendation for the expected energy savings can only be made once a performance test has been conducted on the plant under investigation.

[53] This is an assumption based on how much the efficiency would drop if the flow increases in the same proportion at a fixed speed.

8 Evaluation of the Power Station Load Profile

Since the power consumed by the pump would increase with an increase in the flow requirement (see Figure 8 in section 2.4.1), it may be assumed that a lower than average plant unit load would yield a reduced average pump power consumption.

The power station investigated operates with a “two shifting” operating policy when the demand on the national grid is low. This means that some of the units in the plant are often taken off-load at night during off-peak times, and started up again during daytime to meet the morning and evening peaks. Units are not cooled down; and this allows for a quick start-up. This operating policy reduces the running costs of the plant, because no fuel is used during off-peak times.

Apart from 2-shifting, the station also runs on a load-following operating policy. The loads of individual units in service are not controlled by the unit operators, but by Eskom’s Johannesburg-based *National Control*. National Control looks at the load demand, forecasts the upcoming load, and plans all the different power station units’ load supply, accordingly. Using this grid-demand forecast, Eskom controls the load of each unit in the fleet – remotely from a central location.

Due to its two-shifting and load-following operating philosophies, the power station investigated does not run close to Maximum Continuous Rating[54] (MCR) on average; and for this reason, a detailed evaluation of the power station load profile was performed.

The data were collected from the station’s operating software, which has the capability of storing all relevant plant data to be recalled whenever this is necessary. The data for the generated capacity were recalled from 1 January 2011 until 31 December 2011, in 1-hour increments. This data were then averaged, to show what the load has been in specific hourly increments.

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[54] MCR is a term used to express 100% load or full load conditions.

Figure 21 shown below illustrates the load profile for an average day at the power station:

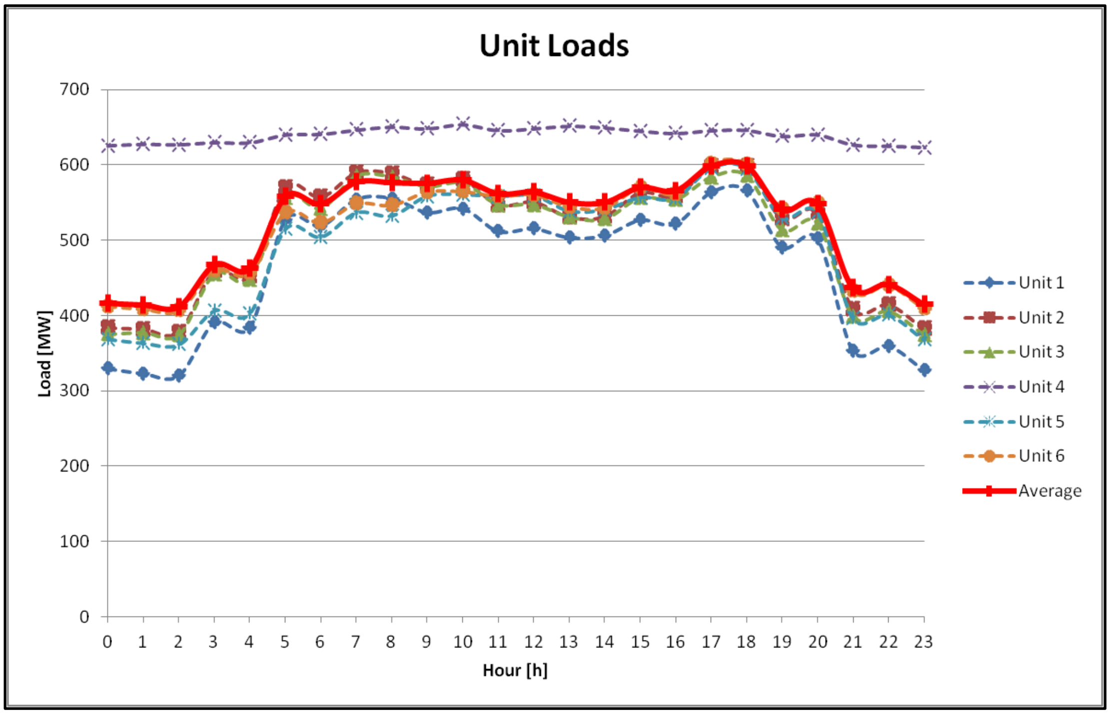

Figure 21: Power Station Load Profile

Of the 6 units at the power station under investigation, all perform at approximately the same load profile to cater for a larger demand in peak periods and a lesser demand during off-peak periods. The only exception was the 4th unit, which was not operating with a 2-shifting and load-following policy, due to maintenance and reliability issues being experienced at the time.

The solid line in Figure 21 illustrates the average load profile for the average day; and this would represent the baseline of the power station. The total average generated capacity over a whole years' test data was calculated to be 523 MW per unit, 74.2% of MCR. However, this does not take into account the times when the unit is off load for planned or unplanned maintenance.

To get a more accurate representation of the actual average load when the plant was in operation, the total average load of the plant must be divided by the time the plant is in actual operation. The first step was to calculate the percentage of time a plant unit is in actual operation; and the load data were filtered into 10% load bands.

The results are shown in the Figure 22 below:

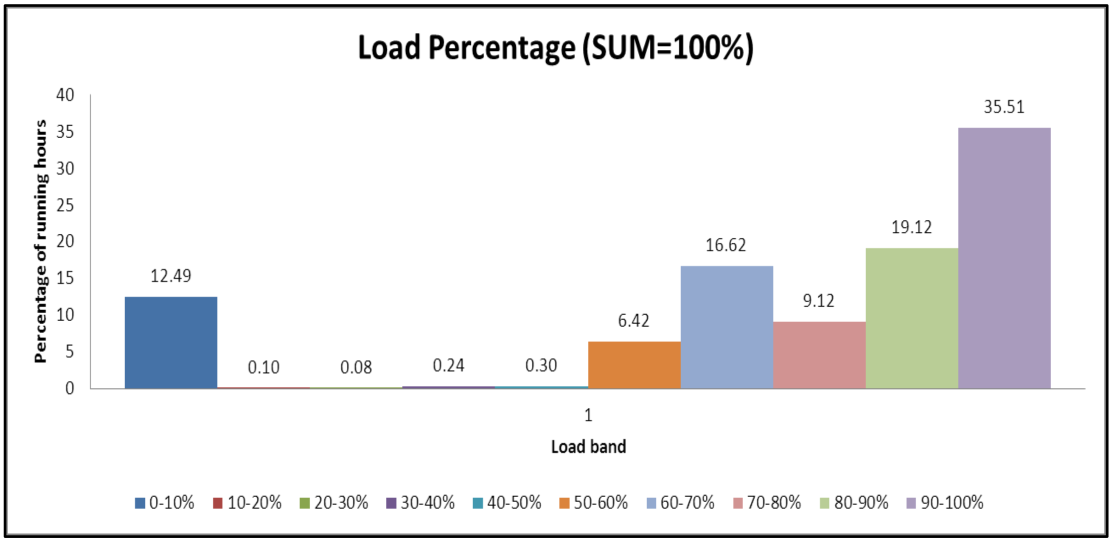

Figure 22: Load Percentage vs. Load Bands

The graph above shows that an average unit is operating at 0-10% load 12.5% of the time, 60-70% load 16.6% of the time, 90-100% load 35.5% of the time, etc.

A unit would never operate at a load below 10%; and for this reason, the 12.5% in Figure 22 was regarded as the time spent off load. The balance of time spent would reflect the Unit Capability Factor or UCF, which in this case was 87.5%. This means that the unit was in operation 87.5% of the time in 2011.

If the total average load (74.2%) is divided by the UCF (87.5%), the product (84.8%) should reflect the actual average load while a unit is in operation. Figure 23 below demonstrates how the time that units spend off-load affects the average load of the unit.

The 84.8% actual average load on the unit is a much more accurate representation of the conditions that would ultimately affect the CEP performance. This will be elaborated later on in this Dissertation.

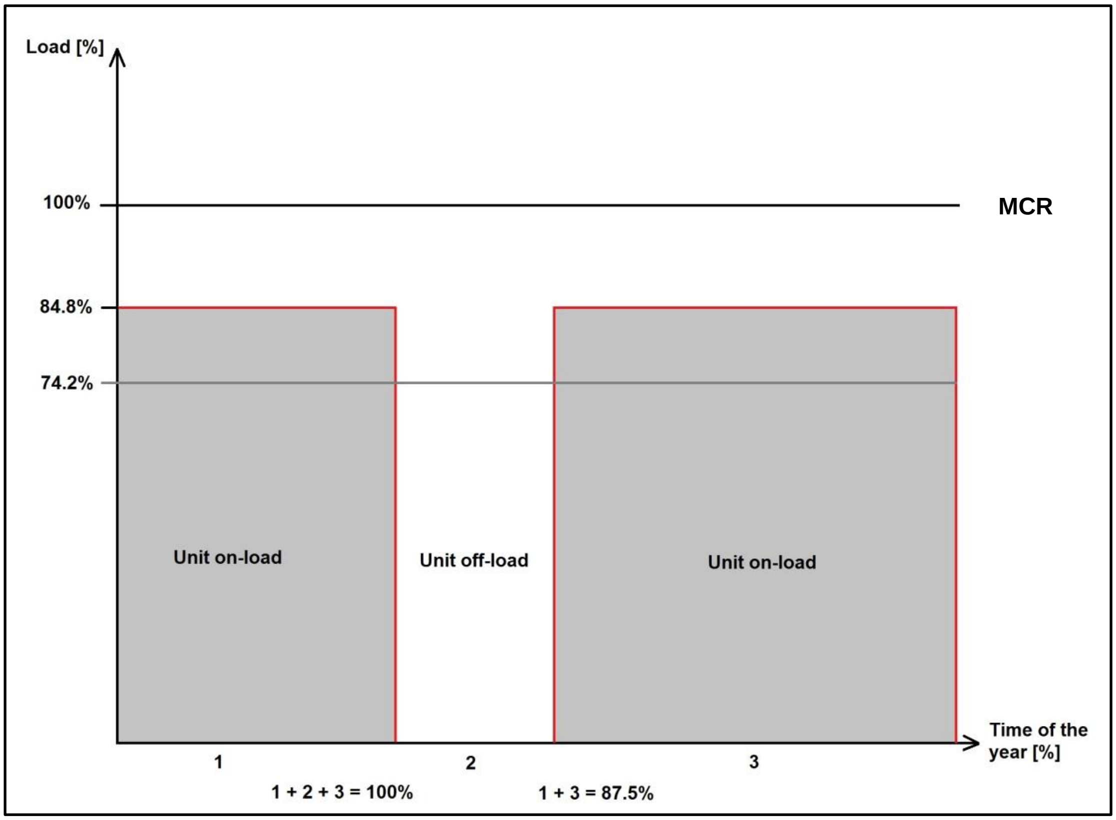

Figure 23: Average Load Demonstration

9 Calculation of the Exact Efficiency Improvement Achievable when Implementing a VSD Retrofit

9.1 Calculation of the Savings in Power Consumed

To calculate the *most-likely-to-achieve* savings, the heat-balance diagrams (APPENDIX A) and the pump performance curve (APPENDIX C) were used.
According to section 8, the average load while the unit is in operation for the power station investigated was 84.8%. At this load, the flow through the pump would be 365.7 kg/s (this is calculated using the equation for the condensate mass flow in Table 14 in APPENDIX F). From section 7.2, we know that inefficiencies in the system will increase the mass flow through the CEP by 46 kg/s, subsequently increasing the flow through the pump to 411.7 kg/s again.

This value will be used as the average flow through the pump when the unit is in operation. When using the “goal-seek” function in Excel, and the equation for the main condensate mass flow (Table 14 in APPENDIX F), the load can be determined for an average operational mass flow of 411.7 kg/s. This new *fictional* load of the unit was calculated to be 95.6%.

From the equations derived for the pump-power consumption in Table 14 of APPENDIX F, Figure 24 below shows the relationship between the power consumed by the pump when controlled by valve throttling vs. VSD control; and Figure 25 shows the savings that can be expected for different loads.

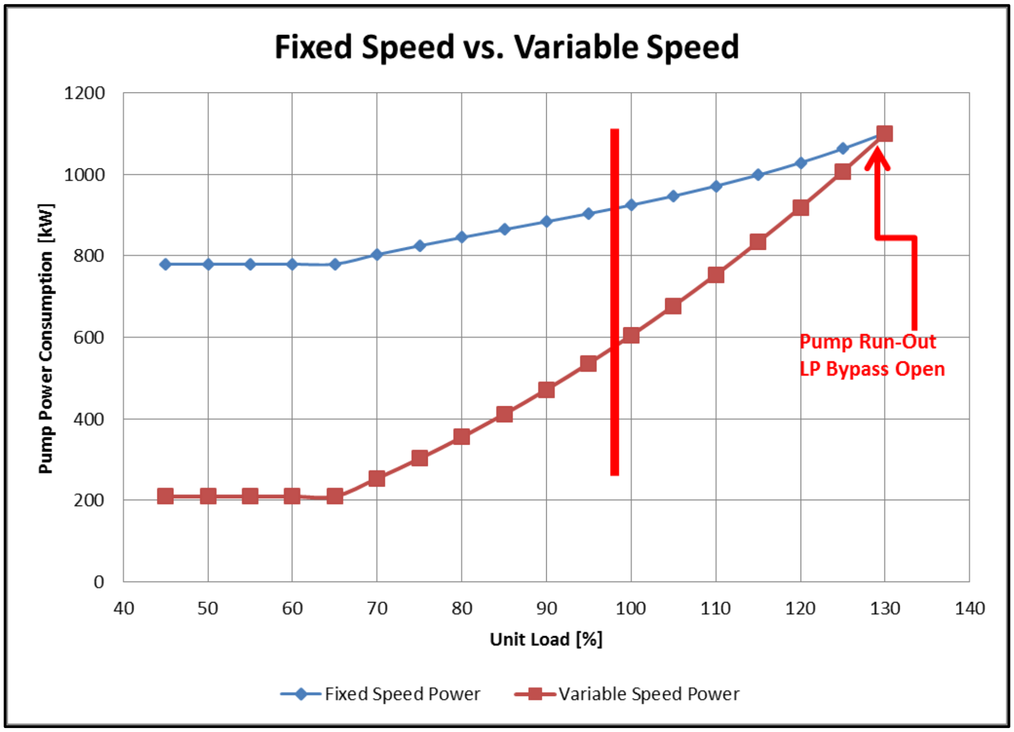

Figure 24: DOL vs. VSD Pump Power Consumption

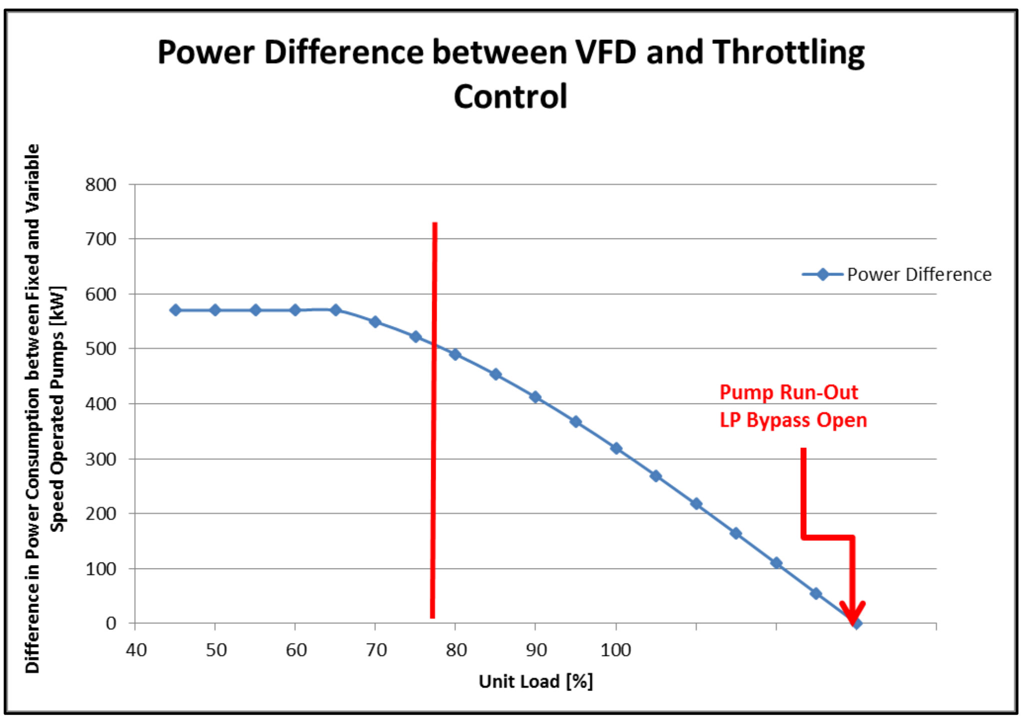

Figure 25: Power Savings Achievable for Different Loads

Using the projected savings curve in Figure 25, the saving can be calculated at the 95.6% fictional load. The results are tabulated below:

Table 9: Results for the Actual Savings Expected at Fictional Load

Unit Load	Fixed Speed Power	VSD Power	Power Savings
100 %	924.7 kW	605.0 kW	319.7 kW
84.8 %	864.0 kW	409.2 kW	454.8 kW
95.6 %	906.4 kW	544.5 kW	361.9 kW

Table 9 shows that 319.4 kW savings can be expected at full load conditions; but due to the load variability, this saving can be increased to 454.8 kW. Leakages in the power generation cycle lead to increased flow through the pumps, thereby negatively affecting the saving that could be expected. The actual saving that can be expected, based on the performance of the system, is 361.9 kW. The impact of the efficiency of the plant can be calculated by subtracting the savings calculated above from the CEP power consumption. This increased the efficiency of the plant from 44.847% to 44.865%, increasing the overall Rankine efficiency of the plant by 0.018%, which is not high, but will decrease the running costs of the plant.

9.2 Flownex Verification and Simulation Results

Flownex was used as a tool to verify all the calculations, and to create a transient model of the power station under design conditions.

In APPENDIX F, Figure 48 shows a screenshot of the Flownex input and output workbook. The model calculated an energy saving of 361.86 kW at the fictional average load of 95.6%; and the pump would then be operating at a speed of 1218.3 rpm.

The biggest advantage of the Flownex model was that it could simulate transient conditions when operating with variable speed pump flow-control. The following important observations were made, when operating the model in transient conditions:

1. The flow through the gland steam condenser (GSC) decreased from 25% of the feed water flow (as calculated in section 5.2.1) to 22%. This does not

have any significant impact on the cycle, with only a 0.3 °C decrease in GSC outlet temperature, and a 1.8 kJ/kg decrease in main feed water enthalpy. If more serious issues arise, orifice C16 between the GSC inlet and outlet (see APPENDIX E) can be removed, and the size adjusted to force 25% flow through the gland steam condenser as it was initially designed.

2. When variable speed pump flow-control was used, instead of throttling control, the condensate control valve in the model was opened 100%. With the recirculation valve still operating, as with throttling control, to protect the pump against minimum flow, the load was reduced to see if the minimum flow protection operates as required. Due to the water pressure at the inlet of this valve being lower, as with throttling control (lower pump speed = lower discharge head), the recirculation valve would not be able to open enough to protect the pump from minimum flow at lower loads. This means that the recirculation valve control logic would need to be adapted to fulfil the minimum-flow protection requirements.

No other significant changes were made to the model, when substituting conventional valve flow control for VSD flow control. It is recommended to use a detailed model to evaluate the response time of the drive, when the LP bypass open signal is given.

If the LP Bypass is signaled as open, the VSD would have to revert to full speed, as soon as possible, in order to meet the De-aerator and the LP bypass flow requirements. This can only be simulated if a final decision is made on how the decentralized control system (DCS) would control the flow though the pump. (The different control options are discussed in Chapter 8 of the Variable Speed Pumping handbook.)

9.3 LP Drain-Recovery System's Impact on Efficiency

As explained in section 2.3.2, the LP Drain-Recovery System was designed to increase plant efficiency, and taking this system out-of-service would mean:

1. An increased flow would be required through the CEPs, to make up for this flow.

2. The enthalpy of the feed water entering into the DST would be lower, ultimately decreasing the overall cycle efficiency.

The EES model can be used to determine what the impact would be on the Rankine efficiency of the cycle.

The first step was to add the LP drain-recovery flow to the flow through the CEP (mass flow 1 should be equal to mass flow 16 in APPENDIX A); and the second step is not to simulate any increase in enthalpy over the LP drain pump discharge inlet to the feed water line (enthalpy 13 should be equal to enthalpy 16 in APPENDIX A).

By implementing these two steps, the EES model calculated the Rankine efficiency of the cycle to be 44.83%, 0.02% lower than the initial 44.85% efficiency. CEP power increased by 54.4 kW; and this could have been sent out in the form of extra electricity generated.

9.3.1 Increasing the LP Drain-Recovery System's Availability

The following recommendations were made by local engineers for increasing the availability of the LP Drain-Recovery System[55]:

1. Replace the mechanical level indicators (floating indicators) with electric pressure differential transmitters (or whatever new technology is on the market). This should increase the ease of control of the Flashbox level, as the sensitivity and margins of the indicator increase.
2. Put a procedure in place, where plant personnel are required to calibrate the valves and other transmitters on a regular basis, especially every time when the unit is off load for maintenance activities.
3. Set up contacts at other pump repair centres that can repair or refurbish the LP drain pumps, when the OEM is overloaded. It has been proven countless times that a secondary repair company can do the same quality work in a much shorter timeframe. Regular inspection intervals during the repair process would guarantee the quality of the maintenance.

[55] This was discussed during a meeting with the power station condensate, LP feed heating, and cooling water system engineer, Alistair Stuart. Some of these measures had already been implemented with success.

10 Economic Evaluation

If a VSD is used to control the flow through the CEPs, an energy saving of 361.9 kW can be achieved. This equates to an energy saving of 8.7 MWh per day, and 2688 MWh per year (if multiplied by the UCF calculated in section 9).

10.1 The Cost of Generating Electricity

According to the Eskom Modification Procedure, the cost of electricity must be calculated as the "cost to produce" electricity (called the System Marginal Price, or SMP), and not by the price that the consumer pays for the electricity. The Eskom Power Exchange website[56] is updated hourly with the SMP, as well as a forecast on the load demand for the day.

The average SMP can change considerably from one month to the next, and is dependent mostly on the demand for electricity. An increase in the demand for electricity would mean that the gas turbine power stations would also have to be put into service; and these power stations generate electricity at a higher cost than the fossil-fuel-fired, nuclear, or hydro-power stations. Electricity demand is also constantly changing, due to economic developments and seasonal load changes.

The average monthly SMP was calculated for the years of 2011 and 2012 and plotted in Figure 26.

[56] This website is available through Eskom's intranet: http://powi.eskom.co.za/~powi/

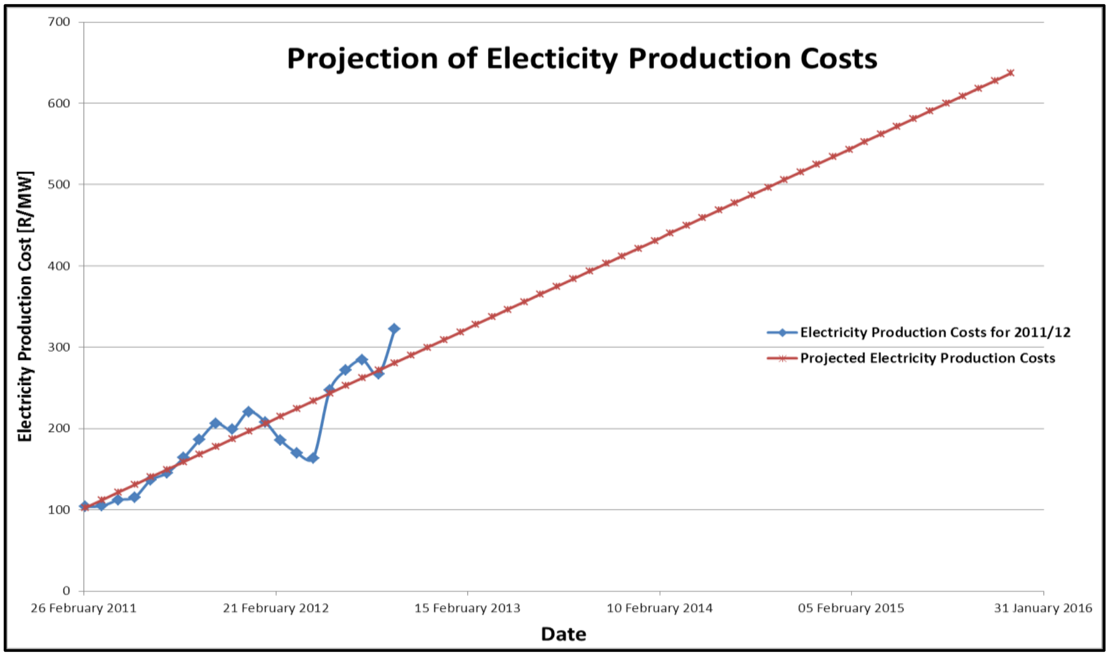

Figure 26: System Marginal Price 2011/12

From the curve, no exponential tendency can be observed (the wavelike behavior of the curve suggests seasonal changes), so a linear trend line was used to predict the SMP.

If $January\ 2011 = 1$ and $February\ 2011 = 2$, etc.

(This is referred to as the date counter.)

Then, the equation used to represent the projected cost is:

$$y = 9.3794961766x + 74.478273782$$

where y = Electricity Production Cost [R/MW]

and x = Date Counter

This equation will be used to predict the SMP from the project implementation date until the end of the life of the plant. The SMP already includes the effect of inflation.

10.2 Implementable Options

In this chapter, six different options will be compared. The main areas of investigation are:

1. Throttling control
2. Electrical VSD control
    a. One single VSD per unit driving one of the pumps
    b. One single VSD per unit with a synchronous transfer, allowing it to drive either of the pumps
    c. Two VSDs per unit, each driving their respective pumps
3. HVSC control
    a. One HVSC per unit, driving one of the pumps
    b. Two HVSCs per unit, each driving their respective pumps

The implementable options will be weighed against the BASE CASE, or option 1, which is the option where no changes are made to the current plant.

Due to the different options having different valve maintenance requirements, the valve maintenance cost must be catered for in the economic analysis.

For the base case, valve maintenance will stay as currently in place (regular replacement and regular services), which will be explained in the next section.

For any configuration with a single variable speed drive/coupling, valve maintenance will be reduced to regular maintenance intervals (which is due to valves being utilized at lower rates than previously).

For any configuration with two variable speed drives/couplings, no valve maintenance will be required any more (this is mainly due to the second drive/coupling taking the place of the valve to ensure system redundancy).

If the valves are functioning as per design requirements, no transient hydraulic shock should be encountered under normal operating conditions (especially large enough to cause the inlet to the DST to crack open).

10.2.1 Option 1 – Base Case

No changes to the current plant:

The flow control is still implemented by means of a mechanical control (control valve in Figure 27). For this option, no initial capital investment would be required, and monthly costs consist of exchanging the valve internals with new ones every three years (refurbishing the internals have proven unsuccessful in the past), and the cost of fixing the De-aerator inlet piping when it is leaking (Figure 27) – as a result of hydraulic shock in the pipeline.

Hydraulic shocks in the pipeline are caused by control valves being "sticky" (not opening and closing smoothly) after continuous use. These costs include welding costs to weld the cracks closed, and the 48 hours downtime that is required to let the unit cool down and perform the weld.

Increased operating costs, due to inefficiencies forcing more flow through the pumps (identified in section 7.2), are also included in the monthly costs.

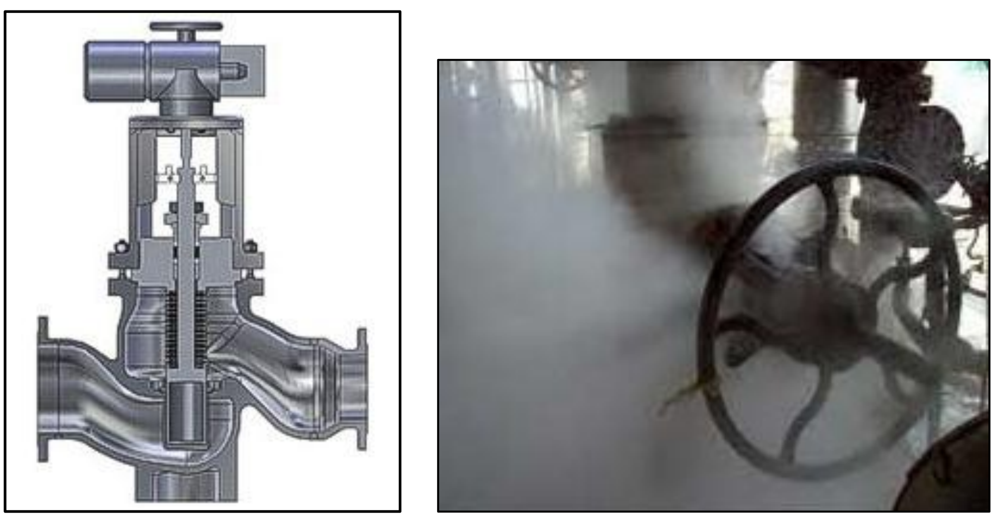

Figure 27: Plug-Type Control Valve and Steam Leak

10.2.2 Option 2 – One VSD per Unit

One single VSD per power station unit driving one of the CEPs[57]:

For this option, initial capital costs include the purchase, installation and commissioning costs of a single VSD. This single VSD would be used to drive only one of the pumps. The second pump would be used as a stand-by, and the internals of the pump would be exchanged when there is an opportunity to do so (intervals at the system engineer's discretion). Monthly costs include maintenance of the single VSD, as well as reduced maintenance on the control valve, which would still have to be used as back-up when the VSD fails (the MTBF is 100,000 hours, but the redundant system must be maintained to guarantee a steady supply of electricity by the unit).

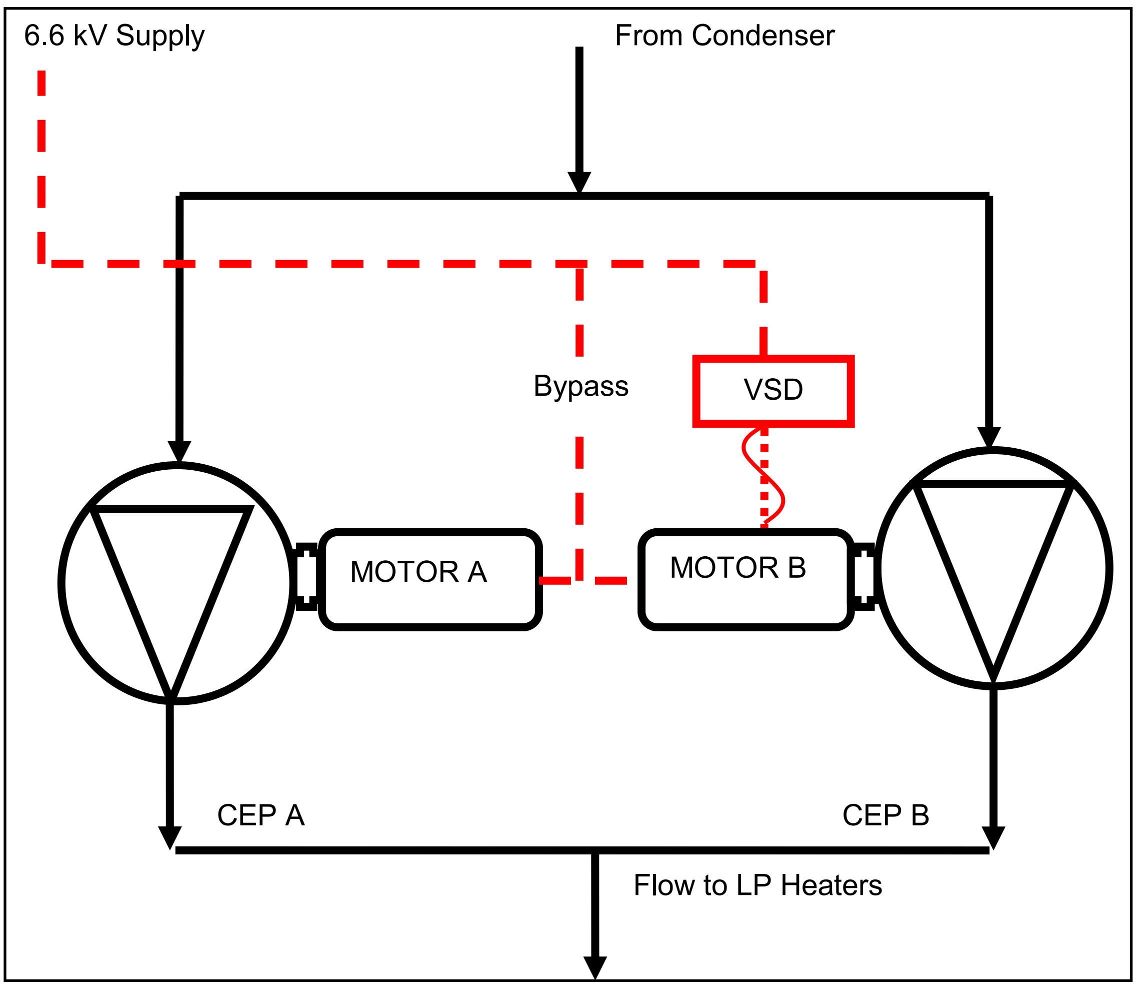

Figure 28: Retrofit Option 2 – One VSD

[57] This plant configuration was discussed with VSD suppliers as a feasible retrofit option.

10.2.3 Option 3 – One VSD per Unit with a Synchronous Transfer

One single VSD with a synchronous transfer allowing it to control either of the two pumps (but not simultaneously)[58]:

With all VSD options, the VSD can be bypassed if there is a failure. Utilizing a synchronous transfer, the VSD can control either pump A or pump B, with the second pump available as a fixed speed for back-up. This means that if the VSD controls the speed of pump A, pump B would be available as the fixed speed back-up when pump A fails. The capital costs include the acquisition, commissioning and installation of a single VSD, and that of a synchronous transfer. Monthly costs include the reduced costs of maintaining the valve, as well as the maintenance costs of the VSD and the synchronous transfer.

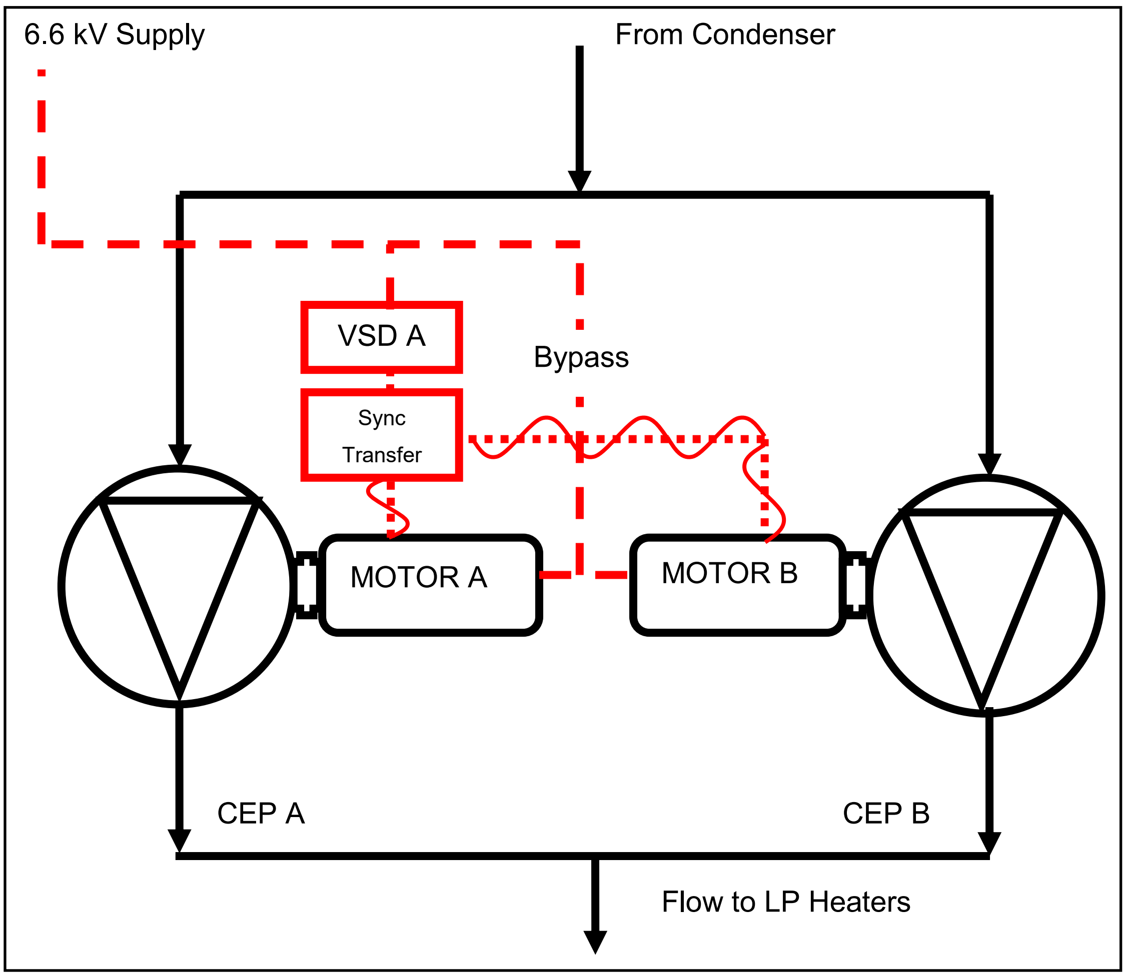

Figure 29: Retrofit Option 3 – One VSD with a Synchronous Transfer

[58] This plant configuration was discussed with VSD suppliers as a feasible retrofit option.

10.2.4 Option 4 – Two VSDs per Unit

Two VSDs controlling two separate pumps[59]:

With this option, every pump motor has its own VSD controlling the motor speed. This is especially beneficial when two pumps are run at once regularly; but this is not the case for the power station under investigation. The initial capital investment includes the acquisition, installation and commissioning of two VSDs. Monthly costs involve no maintenance on the valve, as it would no longer be required as a back-up, but this would involve mostly the maintenance of the two VSDs.

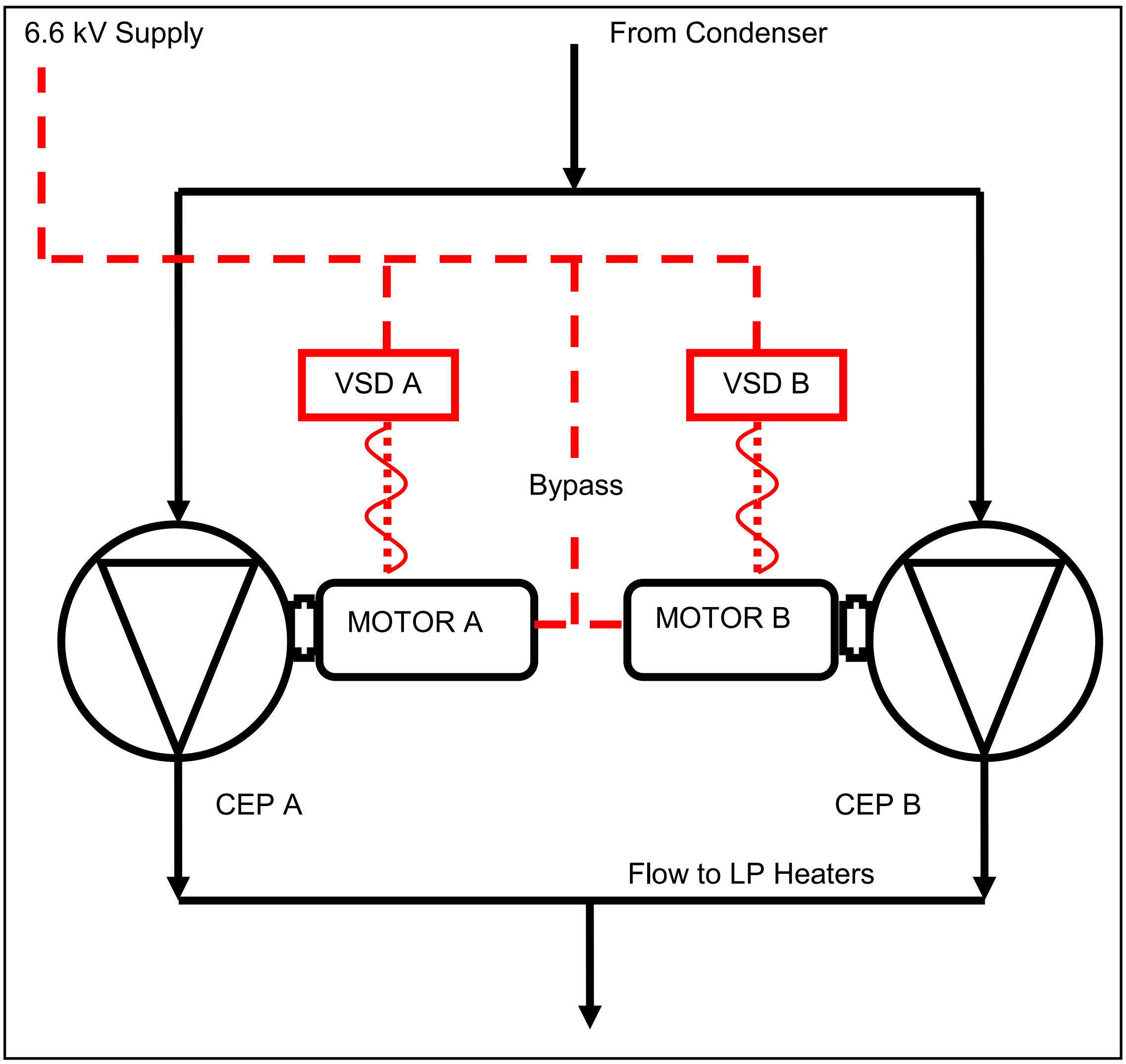

Figure 30: Retrofit Option 4 – Two VSDs

[59] This plant configuration was discussed with VSD suppliers as a feasible retrofit option.

10.2.5 Option 5 – One HVSC per Unit

One single HVSC controlling a single pump[60]:

With this option, no savings are recognized, as the motor is running at a fixed speed as before. Capital costs involve the acquisition, installation and commissioning of one HVSC with its hydraulic pack and cooling system. Monthly costs include maintaining the HVSC, as well as the reduced maintenance on the control valve.

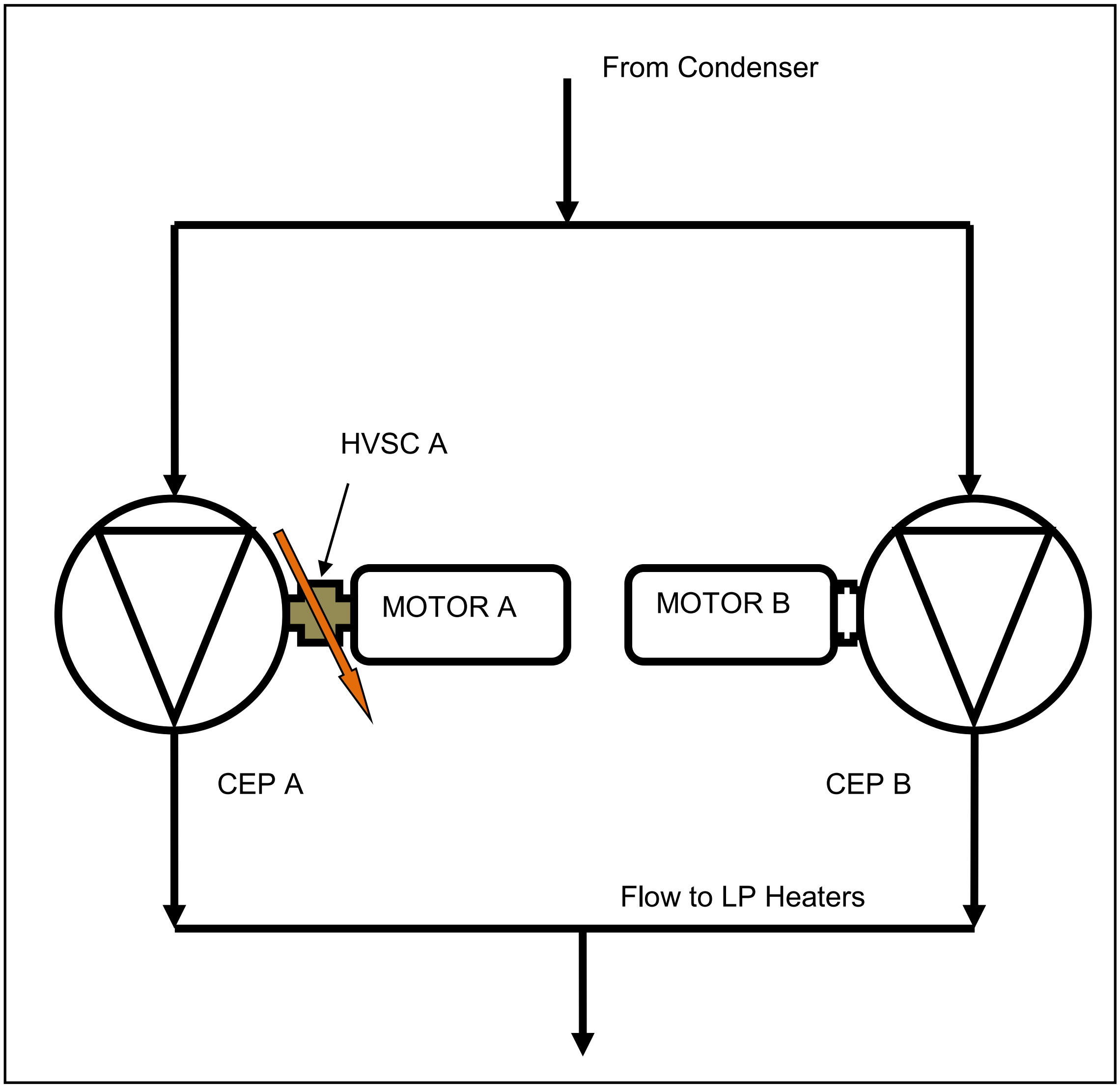

Figure 31: Retrofit Option 5 – One HVSC

[60] This plant configuration was discussed with HVSC suppliers as a feasible retrofit option.

10.2.6 Option 6 – Two HVSCs per Unit

Two HVSCs controlling the speeds of two different pumps[61]:

There is still no savings potential in this option, but the increased reliability and controllability is now possible. The capital costs would include the acquisition, installation, and commissioning costs of two HVSCs – with their hydraulic packs and cooling water networks. Maintenance on the valve is nil; and the monthly costs only include maintaining the HVSC and its hydraulic pack.

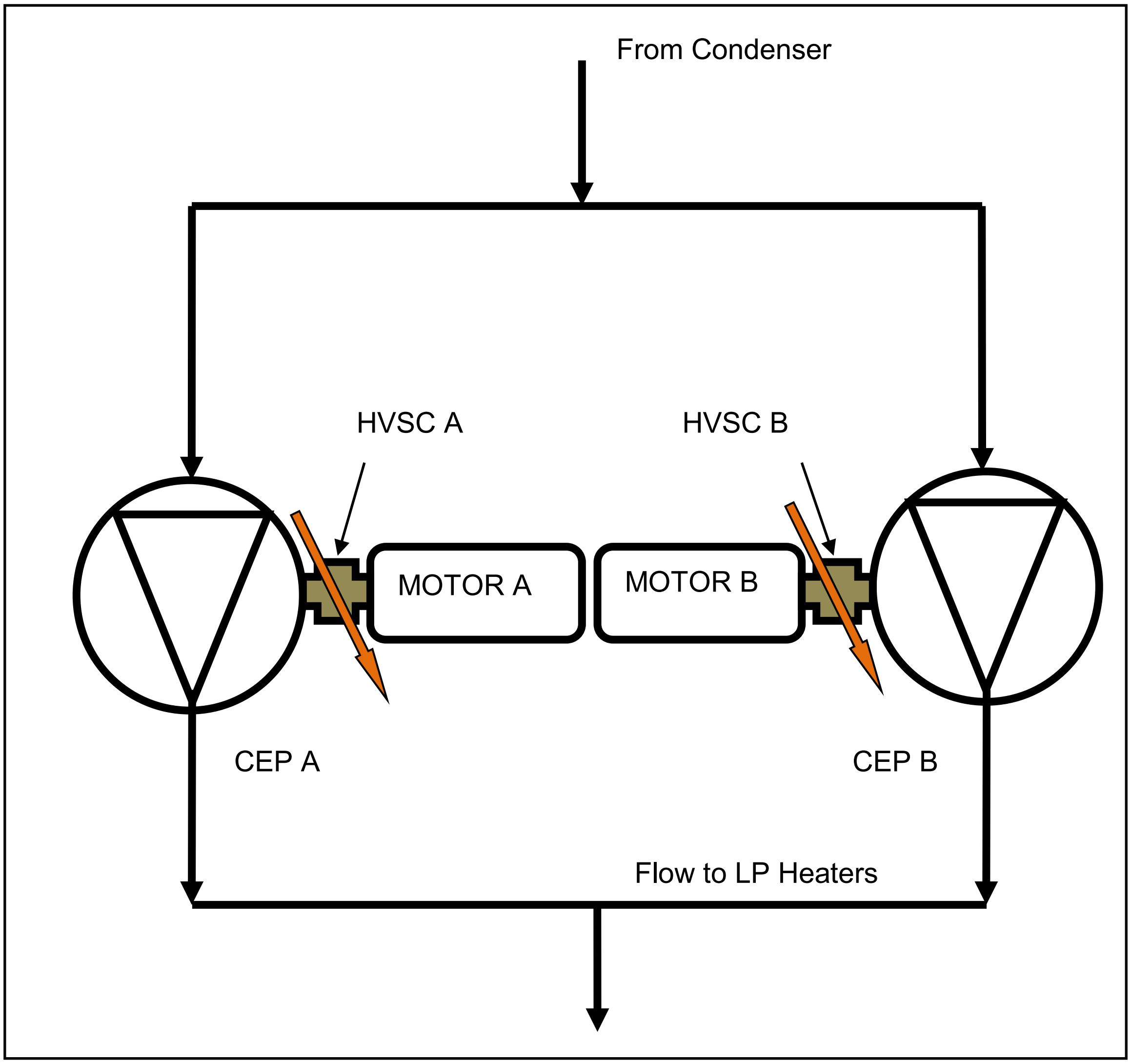

Figure 32: Retrofit Option 6 – Two HVSCs

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[61] This plant configuration was discussed with HVSC suppliers as a feasible retrofit option.

10.3 Cost Analysis

A cost analysis was performed comparing the 6 options, as discussed in section 10.2 concerning the capital investment costs and the monthly operating costs. Costs were accumulated from quotations and other information received by the various pump and variable speed drive manufacturers[62]; and these were compared with similar Eskom projects (with the help of an internal project engineer[63]); and they were found to be accurate for the purpose of this study.

The purpose of the economic evaluation was to determine the payback period for the respective retrofit options. This would reveal when each of options 2 to 6 would start to benefit financially in reduced operating costs, as a result of their individual capital investment costs, when compared with the base case, or option 1. These payback periods can be compared, in order to see which of the options would be the most economical to implement over the remaining life of plant (LOP).

The total project costs associated with the different options are displayed in Figure 33 and Figure 34. Because option 1 is the base case, the payback periods for options 2 to 6 would be where their respective cost curves intersects with the cost curve of option 1.

Payback Periods:

Option 2: *1 year and 6 months*

Option 3: *2 years and 4 months*

Option 4: *2 years and 9 months*

Option 5: *18 years and 11 months*

Option 6: *34 years and 11 months*

The base case (in Figure 33) incurs no initial capital investment costs, but plant maintenance and operating costs increase each year with inflation. All the other options have an initial capital investment (increasing from option 2 to option 6), but have reduced operating and maintenance costs, depending on the configuration of the retrofit. When compared, option 1 has a 0% initial capital investment cost, and option 6 has a 100% initial capital investment cost.

[62] These manufacturers are Eskom vendors; and the preferred pump and motor OEM suppliers.

[63] Costs between similar projects were compared in a meeting with Eskom Senior Pump Engineer, Kumar Rupnarain.

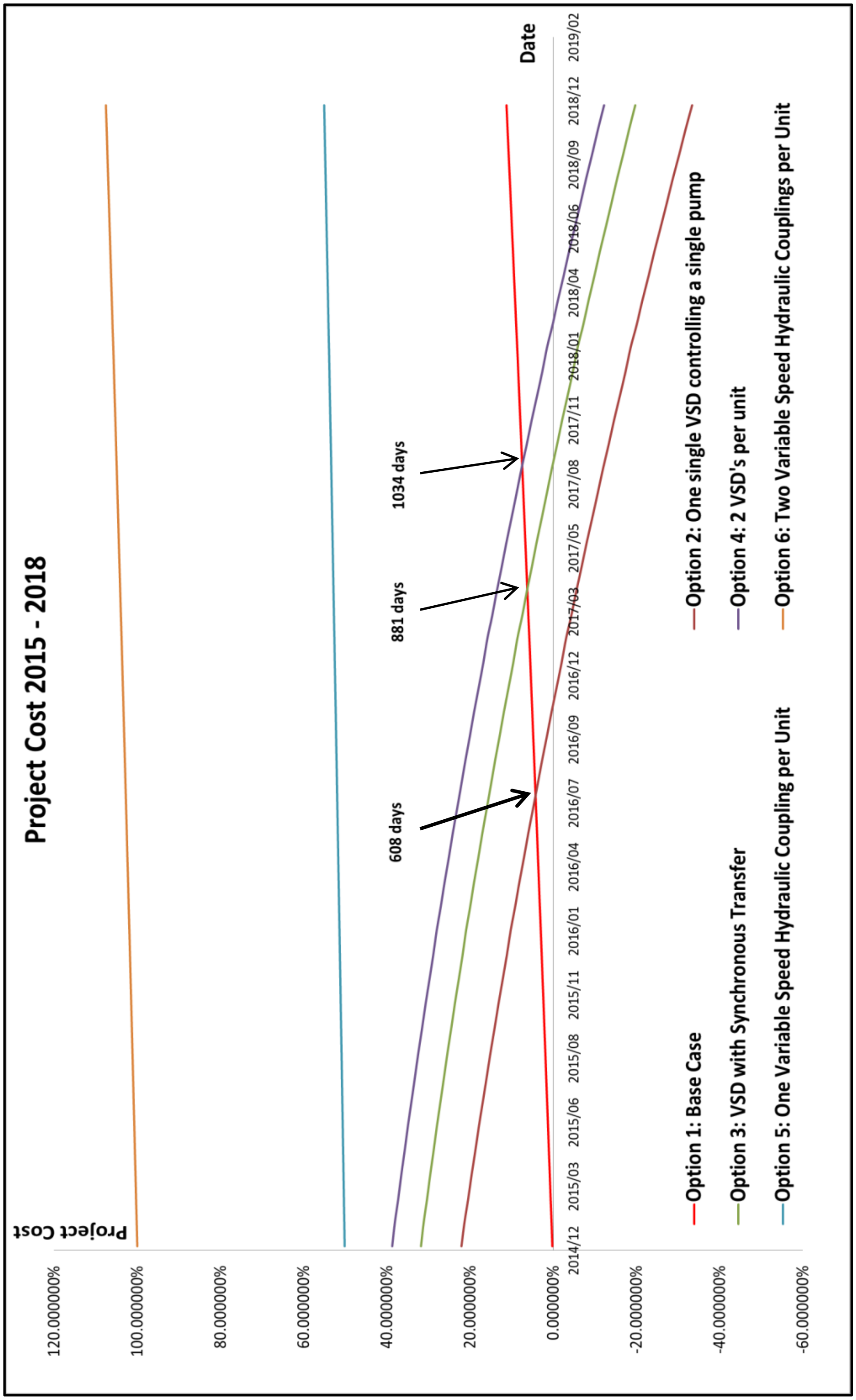

Figure 33: Total Retrofit Project Costs 2015-2018

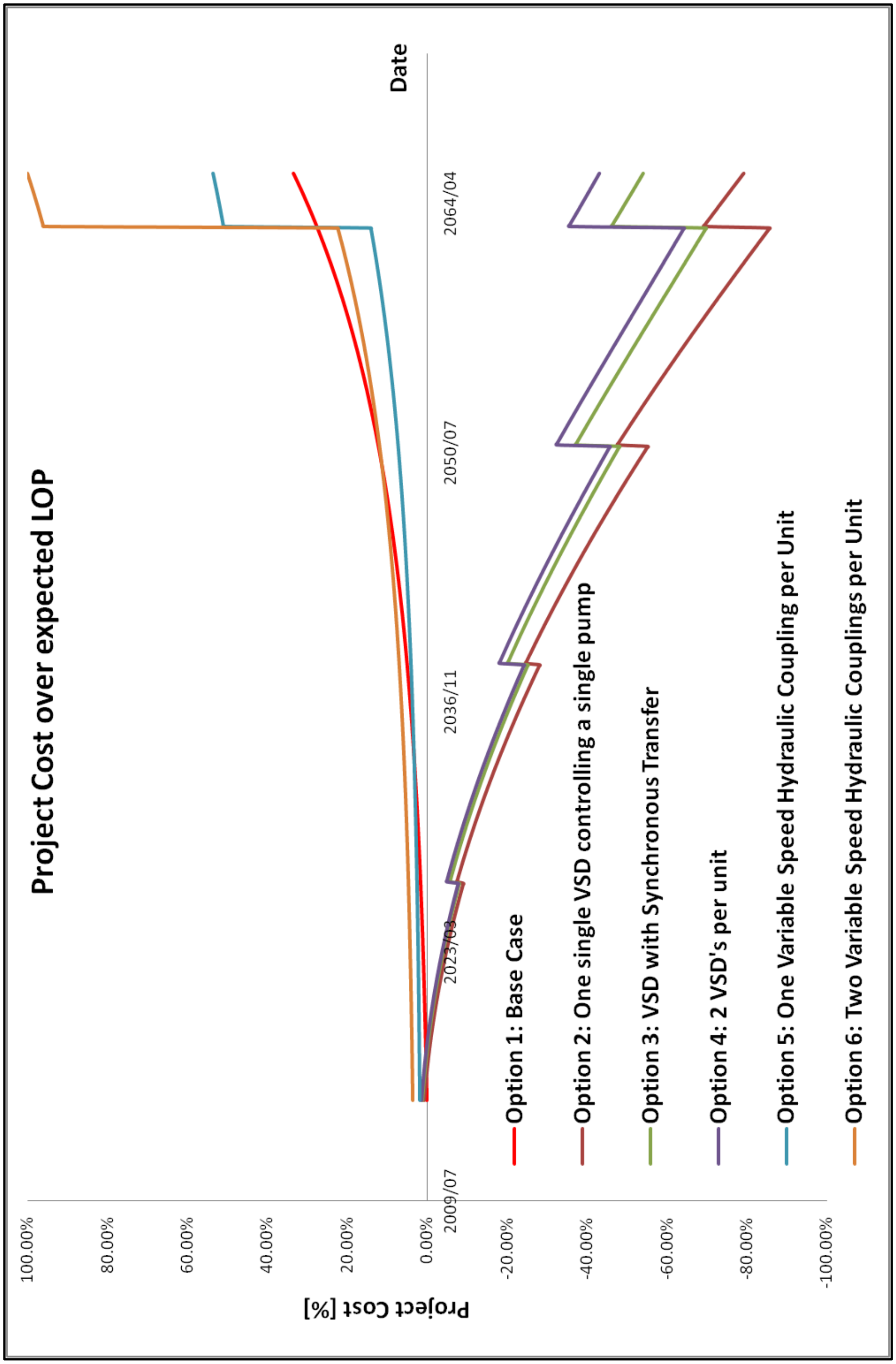

Figure 34: Total Retrofit Project Cost over LOP

From the two graphs above, it may be observed that the payback period increases from option 2 to option 6. This means that from a financial perspective, the options can be rated from best to worse, as follows: option 2, option 3, option 4, option 5, option 6, and option 1, respectively.

Also noteworthy is the significant difference in the payback period when comparing the VSD flow control with the HVSC control. This is mainly because no significant energy savings can be achieved from implementing HVSC control.

The spikes in the cost curves are the result of equipment being replaced in fixed intervals due to the end of the operational lifetime, or obsolescence. From Figure 34, it is evident that while an HVSC is in operation, a VSD would have to be replaced three times in the same life cycle. With the high energy savings expected, it is still economically viable to install a VSD.

There are many other aspects to consider, aside from the financial impact, such as reliability, accuracy of control, environmental friendliness, etc. (Europump, 2004)[64]. These options should be rated, weighed and compared with each other with the use of a generic scoring model. This will be explained in the following paragraph.

10.4 Generic Scoring Model

A generic scoring model[65] was developed to evaluate the different options, based on how well they perform when compared on a *weighted table*. On the weighted table, all the influencing factors are weighed, according to their importance and relevance. For example, capital investment and plant complexity are two different ranking criteria; and when weighed, capital investment would score a 10/10; while plant complexity would score much lower (3/10), as it is less important. All the options would receive a score for their performance when measured against specific criteria

[64] Advantages, disadvantages, and other important considerations are mentioned in Section 11 of the *Variable Speed Pumping* handbook.
[65] Developed from the Criteria Ranking and Weighting principles in Sections 4.3 and 4.4 of the Engineering Design Principles handbook (Hurst, 1999).

(Table 10), and these scores would first be multiplied by their weight – before they were compared.

The list of criteria worth considering when doing the economic evaluation was compiled in Table 10, and presented to all Eskom's power station system engineers, pump specialists, and OEM representatives at the Eskom monthly *Critical Pump Forum*. All the stakeholders at the *Critical Pump Forum* were asked to evaluate the criteria and weigh each criterion on a 1 to 10 scale with reference to their individual importance. The members attending this forum did not suggest adding or removing any of the criteria presented, and the scores, as seen in Table 10, are the averages of all their individual scores (with the outliers in the data removed).

Table 10: Economic Evaluation Criteria

Criteria	***Score [_ /10]***
Financial Justification	8.8
Energy Savings	8.8
Environmental Impact	7.1
Accuracy of Control	6.9
Complexity of Plant	6.9
Maintenance	7.6
Operating	6.6
Plant Health	7.4
Reliability	8.2
Ease of Retrofit	6.7
Redundancy	6.8

According to Section 4 of the Engineering Design Principles handbook (Hurst, 1999), a *criterion-ranking matrix* can be used to rank the criteria in the order of their relevance when compared on a one-to-one basis. The scores for each criterion are added and ranked accordingly. The following criterion-ranking matrix was developed for the study:

Table 11: Criterion-Ranking Matrix

Weight	Financial Justification	Energy Savings	Environmental Impact	Accuracy of Control	Complexity of Plant	Maintenance	Operating	Plant Health	Reliability	Ease of Retrofit	Redundancy	SUM	Percentage	Rank
Financial Justification	/	1	0	1	1	1	1	1	0	1	0	7	12.73%	4
Energy Savings	0	/	0	1	1	1	1	0	0	1	0	5	9.09%	6
Environmental Impact	1	1	/	1	1	1	1	1	1	1	0	9	16.36%	2
Accuracy of Control	0	0	0	/	1	0	0	0	0	1	0	2	3.64%	9
Complexity of Plant	0	0	0	0	/	0	0	0	0	0	0	0	0.00%	11
Maintenance	0	0	0	1	1	/	0	0	0	1	0	3	5.45%	8
Operating	0	0	0	1	1	1	/	0	0	1	0	4	7.27%	7
Plant Health	0	1	0	1	1	1	1	/	0	1	0	6	10.91%	5
Reliability	1	1	0	1	1	1	1	1	/	1	0	8	14.55%	3
Ease of Retrofit	0	0	0	0	1	0	0	0	0	/	0	1	1.82%	10
Redundancy	1	1	1	1	1	1	1	1	1	1	/	10	18.18%	1
											Total	55		

The criterion-ranking scores in the matrix in Table 11 were combined with the weight scores from the Critical Pump Forum stakeholders in Table 10, to reveal a *scrutiny factor* for each criterion. This scrutiny factor was multiplied by the scores given for each of the criteria for every option considered; and these generated the following scores (Table 12):

Table 12: Retrofit Option Weights and Scores

Criteria	Option 1	Option 2	Option 3	Option 4	Option 5	Option 6	Scrutiny Factor	Ranking
Redundancy	10	8	7	8	9	9	0.1229	1
Environmental Impact	1	10	10	9	3	3	0.1175	2
Reliability	6	9	9	9	9	9	0.1180	3
Financial Justification	4	10	9	7	4	3	0.1156	4
Plant Health	1	8	9	10	6	7	0.0979	5
Energy Savings	1	10	10	10	3	3	0.1008	6
Operating	7	8	9	9	8	9	0.0777	7
Maintenance	1	6	4	4	10	9	0.0777	8
Accuracy of Control	3	9	10	10	9	10	0.0651	9
Ease of Retrofit	10	7	6	6	3	2	0.0560	10
Complexity of Plant	9	3	2	2	3	2	0.0508	11
Weighted Score	37.00%	68.21%	66.09%	65.05%	50.25%	49.76%		

10.5 Interpretation of Results

According to the scores the respective options received in Table 12, leaving the plant in its current condition would be the least-economic solution of all the options. Installing an HVSC is a more economical prospect, with a payback period of almost 19 years (Figure 34); but careful consideration must be taken, when the HVSCs are installed. According to Figure 34, an HVSC should be installed, and in full operation for at least 19 years per coupling, in order for it to be a feasible retrofit. For example, when the remaining operational lifetime of a power station is 50 years, the HVSC would need to be replaced after 48 years (according to the regular procedure). It would, however, not be feasible to replace the HVSC after that, as the plant will be shut down in two years, 17 years before the payback period.

Replacing all the equipment before the end of the LOP would not be economical, as the replacement cost of the equipment is too high. The recommended approach (when HVSC flow control is chosen) would be to postpone the installation of an HVSC until the remaining LOP is less than – or the same – as the expected operational lifetime of the HVSC.

The scoring model concludes that retrofitting the current plant, in order to accommodate an electrical VSD flow control is a more feasible option than utilizing mechanical or hydraulic-flow control. The scoring model does not distinguish much between the different VSD layouts; but the following is evident when comparing options 2, 3 and 4:

- Option 4 received the lowest score of the three, because the higher initial capital costs would result in much higher replacement costs every 12 years (Figure 34). Also, since only one pump is in operation, and the other is on stand-by, it is not worthwhile to install an additional VSD that would have to be replaced every 12 years – due to obsolescence – even though the remaining running hours might be few.
- Option 3 received the second lowest score of the three, mainly because its capital costs are very close to the capital costs of option 4, and also because the added complexity of the plant puts it within the range of option 4.

- Option 2 received the highest score, due to cost and VSD redundancy. This option has the lowest initial capital costs (almost half the cost of option 4 and 5, and a quarter of the cost of option 6). In terms of VSD redundancy, the pump internals of the pump controlled by the VSD and its motor can be exchanged with the standby unit during the first weekend maintenance opportunity – in the event of a failure. This means that if planned correctly, the unit would never have to control the condensate flow with a CEP at a fixed speed for more than a week.
- The VSD controls of options 3 and 4 operates with a less complex redundant system than option 2, because failure of one of the pumps would allow the stand-by pump to immediately kick in at a variable speed. The higher initial capital costs of options 3 and 4 still outweigh the less complex redundancy, since plant operation would not be disrupted when the pump is controlled at a fixed speed. Only the efficiency of the system would decrease.

From the economic analysis, it may be concluded that the most economically feasible option would be to retrofit the current plant with one single VSD. The control system would have to be able to control the flow through the pump by speed variation under normal operating conditions, as well as throttling control, in case of an emergency. In emergency conditions, the mechanical control valve would control the flow through the stand-by pump, in the event of a VSD, motor, or pump failure.

With the control valve in place, as the redundant system, plant personnel would feel safer and more comfortable in implementing VSD flow control on the other pump, since they are familiar with the valve throttling control system and its capabilities.

11 Conclusions and Recommendations

The research in this dissertation was identified by focusing on two major areas of efficiency improvement and energy savings; and these areas are the CEP Flow Control Technique, and the LP Bypass Drains Recovery System Functionality. The task was to evaluate these areas in terms of their feasibility and applicability with regards to energy-efficiency improvement measures, and to recommend a conceptual layout of plant equipment that would result in the lowest cost over the operational lifetime of the equipment, while maintaining reliability, production demand and plant health.

11.1 Conclusions

11.1.1 Research Area 1 Conclusion: CEP Flow Control

After evaluating the information that was available in the during the literature survey research period, it was identified that there is a need for a systematic approach that will aid the large industrial pump user in determining the feasibility of a VSD flow-control retrofit.

Regarding the CEP flow control, a comparison between conventional valve-throttling control, HSVC control, and electrical VSD control revealed that the most energy-efficient option would be to retrofit the flow control system with a single electrical VSD. This decision was supported by the literature survey; while the feasibility and economic evaluation also took the financial perspective and other aspects into consideration when retrofitting the plant.

Retrofitting the current plant with a single VSD would lead to an annual saving of between 2688 MWh (pg. 75) and 3,370 MW (pg. 81) in auxiliary power with a payback period of only 1.5 years (Figure 33). Over the remaining operational lifetime of the plant, the financial savings would be roughly R400 million (Figure 34).

Installing a synchronous transfer, or another VSD per unit, is also feasible in terms of energy efficiency, but the final decision should lie with the local system engineer, as different power stations run on different maintenance strategies.
The decision of a single VSD is based purely on a financial approach, and with the conventional system in line for redundancy, plant personnel would also trust the system more.

11.1.2 Research Area 2 Conclusion: LP Drain-Recovery System

With all the current maintenance and operation issues identified, paragraph 9.3 calculated the impact of the system's availability on the Rankine efficiency of the cycle. If the system is not in operation, the Rankine efficiency of the cycle would decrease by 0.02%, which does not sound like much, but the extra 54.4 kW consumed by the CEPs is regarded as an electricity production loss.

Using the cost calculated in paragraph 10.1, the current financial implication is a production loss of +/- R 600 per day, or R 219, 000 per annum.

In section 9.3.1, measures were identified that would help site personnel increase the availability of the system, and as a result, increase the efficiency of the cycle, while maintaining plant health. Implementing these measures should increase the availability of the LP Drains Recovery System, and as a result; reduce the operating costs over the remaining LOP of the power station.

11.2 Recommended Evaluation Approach

Load variability on the pumps, as well as the actual pump performance test results had a noteworthy impact on the actual savings achievable (as is evident in paragraph 9). Based on these impacts, the following systematic approach (developed in section 3) is detrimental in evaluating the feasibility of a retrofit:

- Prequalify the pumping application with the tools provided in section 2.6.3.2.

- Conduct a performance test on one or more of the CEPs at the power station to get a better understanding of the current performance of the pump and determine whether the pumps are operating at their design duty points.
- Collect generated load data from the power station to construct a load output baseline, showing the load profile for a typical day, and the average load generated by the power station.
- Calculate the average mass-flow rate that would be expected though the pump for this average load generated, and determine the pump power consumption at this duty point.

Any anomalies between the design flow rate and the actual flow rate through the pumps should be picked up during the pump performance tests (section 7.1); and this would be added to the average pump flow rate. The average flow rate through the pumps can be determined by making use of the load profile and the heat-balance diagrams (to determine what flow rate corresponds with the average load generated).

Once the average CEP flow rate is known, the pump performance curve (APPENDIX C) can be used to determine the corresponding head when not throttling on the discharge (this is where the resistance curve intersects with the pump performance curve). Using the values for flow and head as well as the motor, pump and drive efficiency (available from the manufacturer), the Pump Power Consumption Equation (Equation 6) can be used to determine the exact energy savings that can be expected at this average flow rate.

The Economic Evaluation can only be conducted when the exact savings expected have been calculated. Table 5 in section 6.3 shows that calculating the energy savings expected using the wrong method will give vastly different results than that which was calculated using the correct method (Table 9 in section 9.1). The calculated savings expected will have a detrimental effect on the feasibility of the project.

12 References

ABB Drives Technical Guide Book. (2013). Retrieved 09 11, 2013, from ABB: http://www05.abb.com

Al-Bahadly, I. (2007). Energy Saving with Variable Speed Drives in Industry Applications. In W. International (Ed.), *Conference on Circuits, Systems, Signal and Telecomminucations* (pp. 53-58). Gold Coast, Australia: WSEAS Int.

Barnes, M. (2003). *Practical Speed Drives and Power Electronics.* (V. Mehra, Ed.) Oxford: Elsevier.

Bekink, P. (2012, December 6). Amersfoort - Tuthukani Plant Performance CEO.

BPMA Website. (n.d.). *Use Variable Speed Drives*. Retrieved 09 11, 2013, from BPMA (Better Pumping Practices): http://www.bpma.org.uk/page.asp?node=122&sec=Use_variable_speed_drives

Carrier Corporation. (2005, 10). *Carrier.* Retrieved 09 11, 2013, from Operation and Application of Variable Speed Technology: www.carrier.com

Cruz, B. D. (2009). Pump Characteristics and ISO Efficiency Curves. In L. S. Ltd (Ed.), *PUMPS: Maintenance, Design, and Reliability Conference 2009 – IDC Technologies*, (pp. 1-20).

Eskom website. (n.d.). *Eskom company Information*. (Eskom) Retrieved June 11, 2012, from Eskom Company Information: http://www.eskom.co.za/c/40/company-information/

Europump. (2004). *Variable Speed Pumping : A guide to successful applications.* (E. a. Institute, Ed.) Elsevier.

Europump, Hydraulic Institute. (2000). *Pump Life Cycle Costs.* Oxford: Elsevier.

Ferreira, F. J. (June 2011). Ecoanalysis of Variable Speed Drives for Flow Regulation in Pumping Systems. *IEEE Transactions on Industrial Electronics , 58* (6), 2117 - 2125.

Hector, J. (2013, June 24). Johannesburg - Eskom Energy Efficiency Project Leader.

Hurst, K. (1999). *Engineering Design Principles.* Elsevier Science and Technology.

J.Tsou. (1998). *Heat Rate Improvement Reference Manual.* EPRI, Palo Alto.

Johannes Feuchter. (2012, 07 08). Voith SVNL Training Manual. Johannesburg, Gauteng, South Africa: VOITH.

Lindsay, B. (2013, January 28). Johannesburg - Actom Motors Commercial Manager.

M. Pugh. (2000). *Condensate Pump Application and Maintenance Guide.* EPRI, Palo Alto.

Mkhwanazi, B. (2012, December 6). Amersfoort - Majuba Power Station Plant Performance Line Manager.

Moncrief, W. (2001). *Guide to the Industrial Application of Motors and Variable Speed Drives.* EPRI, Palo Alto, CA.

P. Ruestman. (1999). *EPRI Heat Rate Improvement Reference Training Manual.* EPRI, Palo Alto.

Pemberton, M. (January 2005). Variable Speed Pumping: Myths and Legends. *World Pumps , 2005* (460), 22 - 24.

Roy-Aikins, P. J. (2012, February 19). Johannesburg - Eskom Chief Engineer.

Rupnarain, K. (2013, February 12). Johannesburg - Eskom Senior Technologist Engineer.

Sulzer Pumps. (1998). *Centrifugal Pumps Handbook.* Oxford: Elsevier.

Thulani S. Gcabashe. (2005). Eskom Energy Efficiency Pledge. Johannesburg.

van der Westhuizen and Cattaert, W. a. (2009). Power Station Pump Selection: Part 1. *World Pumps* , 16-19.

Westhuizen, W. v. (2013, March 06). Johannesburg - Eskom Corporate Pump Specialist.

13 Appendices

APPENDIX A[66]

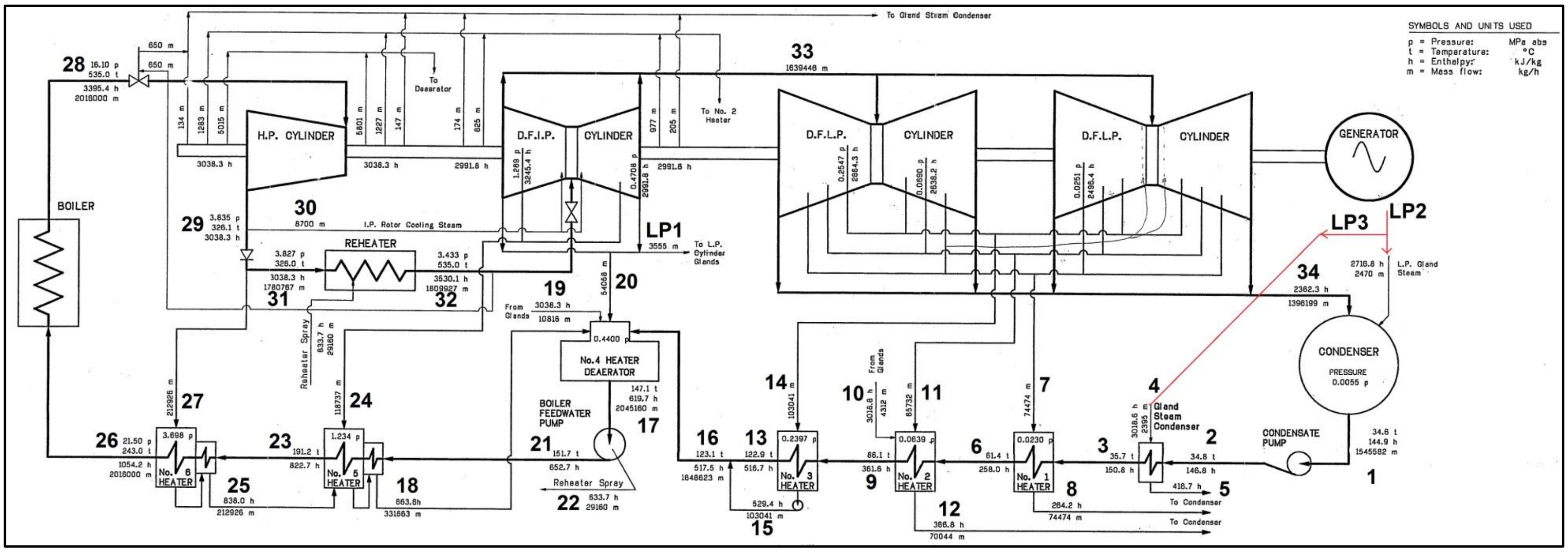

Figure 35: 100% Load Heat Balance Diagram for the Power Station Investigated with Reference Numbers

[66] The information in this appendix is confidential, and available from the author of this dissertation by special request only.

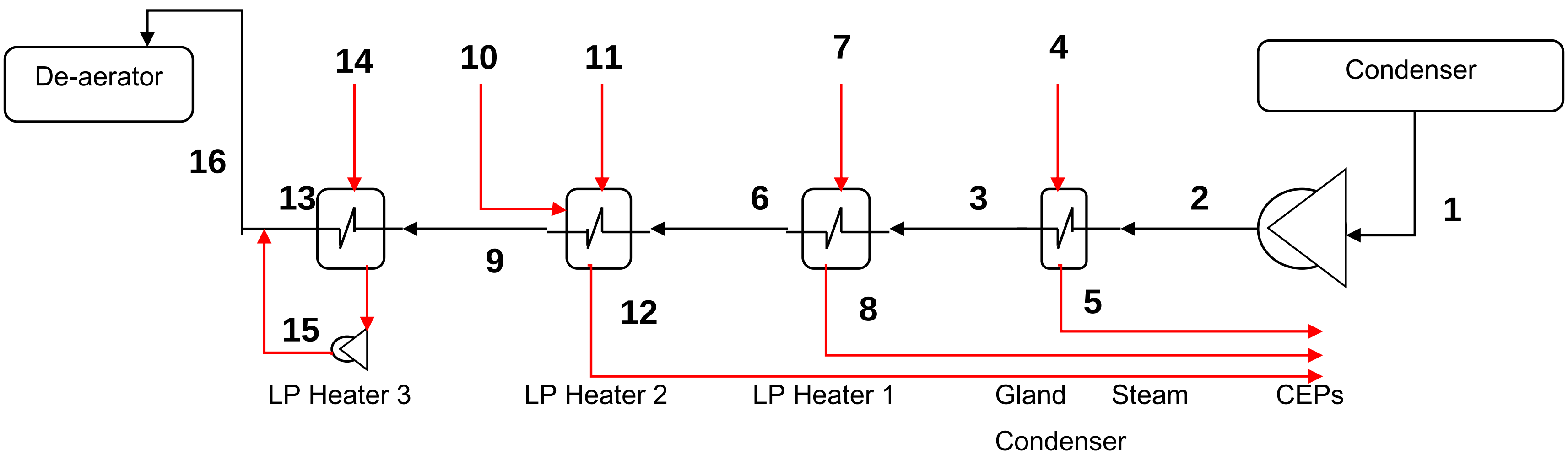

Figure 36: Condensate & LP Feed-Heating Heat-Balance Diagram

Table 13: Heat-Balance Diagram Inputs for Different Loads

LOAD		Condenser	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	DST
100%	P [MPa abs]	0.0055							0.0251				0.069			0.2547			0.44
	T [°C]		34.6	34.8	35.7			61.4			86.1				122.9			123.1	
	h [kJ/kg]		144.9	146.8	150.8	3018.6	416.7	258	2496.4	264.2	361.6	3018.8	2638.2	366.8	516.7	2864.3	529.4	517.5	
	m [kg/h]		1545582			2395			74474	74474		4312	65732	70044		103041	103041	1648623	
90%	P [MPa abs]	0.0055							0.0229				0.0624			0.2305			0.3973
	T [°C]		34.6	34.8	35.8			59.5			83.7				120			120.2	
	h [kJ/kg]		144.9	146.8	151.1	2977.6	416.7	250.1	2499.5	255.6	351.3	3022.9	2640.1	355.8	504.3	2866.8	515.5	505	
	m [kg/h]		1396942			2358			61854	61854		3813	57669	61482		91194	91194	1488136	
80%	P [MPa abs]	0.0055							0.0206				0.0556			0.2054			0.3529
	T [°C]		34.6	34.8	35.9			57.4			80.8				116.6			116.7	
	h [kJ/kg]		144.9	146.8	151.5	2969.4	416.7	241.2	2503.3	245.9	339.3	3032.6	2642.4	343.3	489.7	2869.8	499.8	490.3	
	m [kg/h]		1242524			2272			49557	49557		3281	49401	52682		79135	79135	1321659	
60%	P [MPa abs]	0.0055							0.0163				0.0426			0.156			0.2664
	T [°C]		34.6	34.8	36.1			52.7			74.5				108.7			108.8	
	h [kJ/kg]		144.9	146.8	152.4	2943.2	416.7	221.7	2497	225	312.9	3037.9	2629.6	315.7	456.3	2850.8	463.7	456.7	
	m [kg/h]		953735			2124			29181	29181		2232	35097	37329		57495	57495	1011232	
45%	P [MPa abs]	0.0055							0.0133				0.0333			0.1207			0.2048
	T [°C]		34.6	34.8	36.4			48.7			68.9				101.5			101.6	
	h [kJ/kg]		144.9	146.8	153.6	2919.5	416.7	205	2492.5	207.2	289.3	3042.3	2618.1	291.2	426.2	2833.3	431.6	426.5	
	m [kg/h]		745122			2025			16825	16825		1440	25390	26830		42607	42607	787729	

APPENDIX B

EES Code

>>Variables<<

Condenser Hotwell to CEP inlet

$P_1 = 5.5$ [kPa]

$T_1 = 34.6 + 273.15$ [K]

$h_1 = 144.9$ [kJ/kg]

$m_1 = \frac{1.54558 \times 10^6}{3600}$ [kg/s]

$s_1 = 0.499702$ [kJ/kg.K]

CEP out

$T_2 = 34.8 + 273.15$ [K]

$h_2 = 146.8$ [kJ\kg]

$s_2 = \mathbf{s}(\text{'Steam'}, T = T_2, h = h_2)$

GSC

$T_3 = 35.7 + 273.15$ [K]

$h_3 = 150.8$ [kJ/kg]

$s_3 = \mathbf{s}(\text{'Steam'}, T = T_3, h = h_3)$

$h_4 = 3018.8$ [kJ/kg]

$m_4 = \frac{2395}{3600}$

$T_4 = T_{29} - 2$ [K]

$s_4 = \mathbf{s}(\text{'Steam'}, T = T_4, h = h_4)$

$P_4 = \mathbf{P}(\text{'Steam'}, T = T_4, h = h_4)$

$h_5 = 416.7$ [kJ/kg]

LP 1

$P_{LP1} = 23$ [kPa]

$P_{LP1} = 23$ [kPa]

$T_6 = 61.4 + 273.15$ [K]

$h_6 = 258$ [kJ/kg]

$s_6 = \textbf{s}('Steam', T = T_6, h = h_6)$

$h_7 = 2496.4$ [kJ/kg]

$$m_7 = \frac{74474}{3600}$$

$s_7 = \textbf{s}('Steam', P = P_{LP1}, h = h_7)$

$T_7 = \textbf{T}('Steam', P = 25.1, h = h_7)$

$h_8 = 264.2$ [kJ/kg]

LP 2

$P_{LP2} = 63.9$ [kPa]

$T_9 = 86.1 + 273.15$ [K]

$h_9 = 361.6$ [kJ/kg]

$s_9 = \textbf{s}('Steam', T = T_9, h = h_9)$

$h_{10} = 3018.8$ [kJ/kg]

$$m_{10} = \frac{4312}{3600}$$

$h_{11} = 2638.2$ [kJ/kg]

$P_{11} = 69$ [kPa]

$$m_{11} = \frac{65732}{3600}$$

$s_{11} = \textbf{s}('Steam', P = P_{11}, h = h_{11})$

$h_{12} = 366.8$ [kJ/kg]

$s_{11} = \mathbf{s}\,('Steam', P = P_{11}, h = h_{11})$

$h_{12} = 366.8$ [kJ/kg]

$m_{12} = \dfrac{70044}{3600}$

LP 3

$P_{LP3} = 239.7$ [kPa]

$T_{13} = 122.9 + 273.15$ [K]

$h_{13} = 516.7$ [kJ/kg]

$s_{13} = \mathbf{s}\,('Steam', T = T_{13}, h = h_{13})$

$h_{14} = 2864.3$ [kJ/kg]

$P_{14} = 254.7$ [kPa]

$m_{14} = \dfrac{103041}{3600}$

$s_{14} = \mathbf{s}\,('Steam', P = P_{14}, h = h_{14})$

$T_{14} = \mathbf{T}\,('Steam', h = h_{14}, P = P_{14})$

$h_{15} = 529.4$ [kJ/kg]

$T_{16} = 123.1 + 273.15$ [K]

$h_{16} = 517.5$ [kJ/kg]

$m_{16} = \dfrac{1.64862 \times 10^{6}}{3600}$

$s_{16} = \mathbf{s}\,('Steam', T = T_{16}, h = h_{16})$

Deaerator

$P_{DA} = 440$ [kPa]

$T_{17} = 147.1 + 273.15$ [K]

$h_{17} = 619.7$ [kJ/kg]

$h_{17} = 619.7$ [kJ/kg]

$$m_{17} = \frac{2.04516 \times 10^{6}}{3600}$$

$s_{17} = 1.81292$ [kJ/kg.K]

$h_{18} = 663.6$ [kJ/kg]

$$m_{18} = \frac{331663}{3600}$$

$h_{19} = 3038.3$ [kJ/kg]

$$m_{19} = \frac{10816}{3600}$$

$$m_{20} = \frac{54058}{3600}$$

$P_{20} = P_{33}$

$h_{20} = h_{33}$

$s_{20} = \mathbf{s}\,(\text{'Steam'}, P = P_{33}, h = h_{33})$

BFP's

$T_{21} = 151.7 + 273.15$ [K]

$h_{21} = 652.4$ [kJ/kg]

$s_{21} = \mathbf{s}\,(\text{'Steam'}, T = T_{21}, h = h_{21})$

$h_{22} = 633.7$ [kJ/kg]

$$m_{22} = \frac{29160}{3600}$$

HP 1

$P_{HP1} = 1234$ [kPa]

$T_{23} = 191.2 + 273.15$ [K]

$T_{23} = 191.2 + 273.15$ [K]

$h_{23} = 822.7$ [kJ/kg]

$s_{23} = \mathbf{s}\,(\text{'Steam'}, T = T_{23}, h = h_{23})$

$$m_{24} = \frac{118737}{3600}$$

$h_{24} = 3245.4$ [kJ/kg]

$P_{24} = 1289$ [kPa]

$s_{24} = \mathbf{s}\,(\text{'Steam'}, P = P_{24}, h = h_{24})$

$h_{25} = 838$ [kJ/kg]

$$m_{25} = \frac{212926}{3600}$$

HP 2

$P_{HP2} = 3698$ [kPa]

$P_{26} = 21500$ [kPa]

$T_{26} = 243 + 273.15$ [K]

$h_{26} = 1054.2$ [kJ/kg]

$$m_{26} = \frac{2.016 \times 10^{6}}{3600}$$

$s_{26} = \mathbf{s}\,(\text{'Steam'}, T = T_{26}, h = h_{26})$

$m_{27} = m_{25}$

$h_{27} = h_{29}$

$P_{27} = P_{29}$

$s_{27} = \mathbf{s}\,(\text{'Steam'}, P = P_{27}, h = h_{27})$

Boiler Out

$P_{28} = 16100$ [kPa]

$P_{28} = 16100$ [kPa]

$T_{28} = 535 + 273.15$ [K]

$h_{28} = 3395.4$ [kJ/kg]

$m_{28} = m_{26}$

$s_{28} = \mathbf{s}(\text{'Steam'}, T = T_{28}, h = h_{28})$

HP Turbine

$P_{29} = 3835$ [kPa]

$T_{29} = 326.1 + 273.15$ [K]

$h_{29} = 3038.3$ [kJ/kg]

$s_{29} = \mathbf{s}(\text{'Steam'}, T = T_{29}, h = h_{29})$

$$m_{hp1} = \frac{134}{3600}$$

$$m_{hp2} = \frac{1283}{3600}$$

$$m_{hp3} = \frac{5015}{3600}$$

$$m_{hp4} = \frac{5801}{3600}$$

$$m_{hp5} = \frac{1227}{3600}$$

$$m_{hp6} = \frac{147}{3600}$$

$$m_{30} = \frac{8700}{3600}$$

$s_{30} = s_{29}$

Reheater

$P_{31} = 3827$ [kPa]

$$P_{31} = 3827 \quad [kPa]$$

$$T_{31} = 326 + 273.15 \quad [K]$$

$$h_{31} = h_{29}$$

$$m_{31} = \frac{1.78077 \times 10^{6}}{3600}$$

$$s_{31} = \mathbf{s}\,(\text{'Steam'}, T = T_{31}, h = h_{31})$$

$$P_{32} = 3433 \quad [kPa]$$

$$T_{32} = 535 + 273.15 \quad [K]$$

$$h_{32} = 3530.1 \quad [kJ/kg]$$

$$m_{32} = \frac{1.80993 \times 10^{6}}{3600}$$

$$s_{32} = \mathbf{s}\,(\text{'Steam'}, T = T_{32}, h = h_{32})$$

IP Turbine

$$P_{33} = 470.8 \quad [kPa]$$

$$m_{33} = \frac{1.63945 \times 10^{6}}{3600}$$

$$h_{33} = 2991.8 \quad [kJ/kg]$$

$$s_{33} = \mathbf{s}\,(\text{'Steam'}, P = P_{33}, h = h_{33})$$

$$T_{33} = \mathbf{T}\,(\text{'Steam'}, h = h_{33}, P = P_{33})$$

$$m_{ip1} = \frac{174}{3600}$$

$$m_{ip2} = \frac{825}{3600}$$

$$m_{ip3} = \frac{977}{3600}$$

$$m_{ip3} = \frac{977}{3600}$$

$$m_{ip4} = \frac{205}{3600}$$

$$m_{lp1} = \frac{3555 + 650}{3600}$$

$$m_{lp2} = \frac{4205}{3600}$$

$$m_{lp3} = \frac{1735}{3600}$$

LP Turbines

$h_{34} = 2362.3$ [kJ/kg]

$$m_{34} = \frac{1.3962 \times 10^{6}}{3600}$$

$s_{34} = \mathbf{s}\,(\text{'Steam'}, P = P_1, h = h_{34})$

$T_{34} = \mathbf{T}\,(\text{'Steam'}, h = h_{34}, P = P_1)$

$h_{35} = 2716.8$ [kJ/kg]

$$m_{35} = \frac{2470}{3600}$$

$s_{35} = \mathbf{s}\,(\text{'Steam'}, P = P_1, h = h_{35})$

>>Mass Balance<<

$m_{CON.b} = m_1 - m_{34} - m_{35} - m_4 - m_7 - m_{12}$ Condenser

$m_{LP2.b} = m_{12} - m_{11} - m_{10}$ LP Heater 2

$m_{LP3.b} = m_{16} - m_1 - m_{14}$ LP Heater 3

$m_{DA.b} = m_{17} - m_{16} - m_{20} - m_{19} - m_{18}$ Deaerator

$m_{BFP.b} = m_{17} - m_{26} - m_{22}$ Boiler Feed Pump

$m_{BFP.b} = m_{17} - m_{26} - m_{22}$ Boiler Feed Pump

$m_{HP1.b} = m_{18} - m_{24} - m_{25}$ HP Heater 1

$m_{HPT.b} = m_{28} - m_{31} - m_{27} - m_{30} - m_{hp1} - m_{hp2} - m_{hp3} - m_{hp4} - m_{hp5} - m_{hp6}$ HP Turbine

$m_{RH.b} = m_{32} - m_{31} - m_{22}$ Reheater

$m_{IPT.b} = m_{32} + m_{30} - m_{33} - m_{24} - m_{20} - m_{ip1} - m_{ip2} - m_{ip3} - m_{ip4} - m_{lp1}$ IP Turbine

$m_{LPT.b} = m_{33} + m_{lp1} - m_{34} - m_{7} - m_{11} - m_{14} - m_{lp2}$ LP Turbine

$m_{GSHPDA.b} = m_{19} - m_{hp3} - m_{hp4}$ Gland Steam to Deaerator

$m_{GSHPLP2.b} = m_{10} - m_{hp2} - m_{hp5} - m_{ip2} - m_{ip3}$ Gland Steam to HPH 2

$m_{GSHPGSC.b} = m_{4} - m_{hp1} - m_{hp6} - m_{ip1} - m_{ip4} - m_{lp3}$ Gland Steam to Gland Steam Condenser

>>Work Calculations<<

$W_{CEP} = m_1 \cdot (h_2 - h_1)$

$W_{BFP} = (m_{28} + m_{22}) \cdot (h_{22} - h_{17}) + m_{28} \cdot (h_{21} - h_{22})$ Spray water to reheater is tapped off between stages

$W_{HPT} = m_{28} \cdot (h_{28} - h_{29})$

$W_{IPT} = m_{32} \cdot h_{32} + m_{30} \cdot h_{29} - (m_{32} + m_{30}) \cdot h_{24} + (m_{32} + m_{30} - m_{24}) \cdot (h_{24} - h_{33})$ IP rotor cooling steam is added and bled steam to HP heater 1 is removed

$W_{LPT} = m_{33} \cdot (h_{33} - h_{14}) + (m_{33} - m_{14}) \cdot (h_{14} - h_{11}) + (m_{33} - m_{14} - m_{11}) \cdot (h_{11} - h_7) + (m_{33} - m_{14} - m_{11} - m_7) \cdot (h_7 - h_{34})$

Bled steam to LP heater 1, 2 and 3 removed

$W_{turbine} = W_{HPT} + W_{IPT} + W_{LPT}$

$W_{net} = W_{turbine} - W_{CEP} - W_{BFP}$

>>Heat Calculations<<

$Q_{boiler} = m_{28} \cdot (h_{28} - h_{26})$

$Q_{reheat} = m_{32} \cdot h_{32} - m_{22} \cdot h_{22} - m_{31} \cdot h_{31}$

$Q_{net} = Q_{boiler} + Q_{reheat}$

>>Efficiency<<

$$\eta_{Rankine} = \frac{W_{net}}{Q_{net}} \cdot 100$$

EES Results

Unit Settings: [kJ]/[K]/[kPa]/[kg]/[degrees]

$\eta_{Rankine}$ = **44.847883** [%]	h_1 = 144.9 [kJ/kg]	h_{10} = 3019 [kJ/kg]	h_{11} = 2638 [kJ/kg]	h_{12} = 366.8 [kJ/kg]	h_{13} = 516.7 [kJ/kg]
h_{14} = 2864 [kJ/kg]	h_{15} = 529.4 [kJ/kg]	h_{16} = 517.5 [kJ/kg]	h_{17} = 619.7 [kJ/kg]	h_{18} = 663.6 [kJ/kg]	h_{19} = 3038 [kJ/kg]
h_2 = 146.8 [kJ\kg]	h_{20} = 2992 [kJ/kg]	h_{21} = 652.4 [kJ/kg]	h_{22} = 633.7 [kJ/kg]	h_{23} = 822.7 [kJ/kg]	h_{24} = 3245 [kJ/kg]
h_{25} = 838 [kJ/kg]	h_{26} = 1054 [kJ/kg]	h_{27} = 3038 [kJ/kg]	h_{28} = 3395 [kJ/kg]	h_{29} = 3038 [kJ/kg]	h_3 = 150.8 [kJ/kg]
h_{31} = 3038 [kJ/kg]	h_{32} = 3530 [kJ/kg]	h_{33} = 2992 [kJ/kg]	h_{34} = 2362 [kJ/kg]	h_{35} = 2717 [kJ/kg]	h_4 = 3019 [kJ/kg]
h_5 = 416.7 [kJ/kg]	h_6 = 258 [kJ/kg]	h_7 = 2496 [kJ/kg]	h_8 = 264.2 [kJ/kg]	h_9 = 361.6 [kJ/kg]	m_1 = 429.3 [kg/s]
m_{10} = 1.198 [kg/s]	m_{11} = 18.26 [kg/s]	m_{12} = 19.46 [kg/s]	m_{14} = 28.62 [kg/s]	m_{16} = 458 [kg/s]	m_{17} = 568.1 [kg/s]
m_{18} = 92.13 [kg/s]	m_{19} = 3.004 [kg/s]	m_{20} = 15.02 [kg/s]	m_{22} = 8.1 [kg/s]	m_{24} = 32.98 [kg/s]	m_{25} = 59.15 [kg/s]
m_{26} = 560 [kg/s]	m_{27} = 59.15 [kg/s]	m_{28} = 560 [kg/s]	m_{30} = 2.417 [kg/s]	m_{31} = 494.7 [kg/s]	m_{32} = 502.8 [kg/s]
m_{33} = 455.4 [kg/s]	m_{34} = 387.8 [kg/s]	m_{35} = 0.6861 [kg/s]	m_4 = 0.6653 [kg/s]	m_7 = 20.69 [kg/s]	$m_{BFP.b}$ = -0 [kg/s]
$m_{CON.b}$ = 0.0 [kg/s]	$m_{DA.b}$ = -0.000 [kg/s]	$m_{GSHPDA.b}$ = 0.000 [kg/s]	$m_{GSHPGSC.b}$ = -0.000 [kg/s]	$m_{GSHPLP2.b}$ = 0 [kg/s]	m_{hp1} = 0.03722 [kg/s]
$m_{HP1.b}$ = 0.000 [kg/s]	m_{hp2} = 0.3564 [kg/s]	m_{hp3} = 1.393 [kg/s]	m_{hp4} = 1.611 [kg/s]	m_{hp5} = 0.3408 [kg/s]	m_{hp6} = 0.04083 [kg/s]
$m_{HPT.b}$ = -0.000 [kg/s]	m_{ip1} = 0.04833 [kg/s]	m_{ip2} = 0.2292 [kg/s]	m_{ip3} = 0.2714 [kg/s]	m_{ip4} = 0.05694 [kg/s]	$m_{IPT.b}$ = -0.000 [kg/s]
m_{lp1} = 1.168 [kg/s]	m_{lp2} = 1.168 [kg/s]	$m_{LP2.b}$ = 0.000 [kg/s]	m_{lp3} = 0.4819 [kg/s]	$m_{LP3.b}$ = -0.000 [kg/s]	$m_{LPT.b}$ = 0 [kg/s]
$m_{RH.b}$ = 0.0 [kg/s]	P_1 = 5.5 [kPa]	P_{11} = 69 [kPa]	P_{14} = 254.7 [kPa]	P_{20} = 470.8 [kPa]	P_{24} = 1289 [kPa]
P_{26} = 21500 [kPa]	P_{27} = 3835 [kPa]	P_{28} = 16100 [kPa]	P_{29} = 3835 [kPa]	P_{31} = 3827 [kPa]	P_{32} = 3433 [kPa]
P_{33} = 470.8 [kPa]	P_4 = 4230 [kPa]	P_{DA} = 440 [kPa]	P_{HP1} = 1234 [kPa]	P_{HP2} = 3698 [kPa]	P_{LP1} = 23 [kPa]
P_{LP2} = 63.9 [kPa]	P_{LP3} = 239.7 [kPa]	Q_{boiler} = 1.311E+06 [kJ/s]	Q_{net} = 1.578E+06 [kW]	Q_{reheat} = 266733 [kJ/s]	s_1 = 0.4997 [kJ/kg.K]
s_{11} = 7.427 [kJ/kg-K]	s_{13} = 1.56 [kJ/kg-K]	s_{14} = 7.384 [kJ/kg-K]	s_{16} = 1.562 [kJ/kg-K]	s_{17} = 1.813 [kJ/kg.K]	s_2 = 0.5056 [[kJ/kg.K]]
s_{20} = 7.357 [kJ/kg-K]	s_{21} = 1.889 [kJ/kg-K]	s_{23} = 2.268 [kJ/kg-K]	s_{24} = 7.322 [kJ/kg-K]	s_{26} = 2.734 [kJ/kg-K]	s_{27} = 6.511 [kJ/kg-K]
s_{28} = 6.426 [kJ/kg-K]	s_{29} = 6.524 [kJ/kg-K]	s_3 = 0.5186 [[kJ/kg.K]]	s_{30} = 6.524 [kJ/kg-K]	s_{31} = 6.525 [kJ/kg-K]	s_{32} = 7.253 [kJ/kg-K]
s_{33} = 7.357 [kJ/kg-K]	s_{34} = 7.705 [kJ/kg-K]	s_{35} = 8.801 [kJ/kg-K]	s_4 = 6.438 [kJ/kg-K]	s_6 = 0.8517 [kJ/kg-K]	s_7 = 7.509 [kJ/kg-K]
s_9 = 1.15 [kJ/kg-K]	T_1 = 307.8 [K]	T_{13} = 396.1 [K]	T_{14} = 471.7 [K]	T_{16} = 396.3 [K]	T_{17} = 420.3 [K]
T_2 = 308 [K]	T_{21} = 424.9 [K]	T_{23} = 464.4 [K]	T_{26} = 516.2 [K]	T_{28} = 808.2 [K]	T_{29} = 599.3 [K]
T_3 = 308.9 [K]	T_{31} = 599.2 [K]	T_{32} = 808.2 [K]	T_{33} = 538 [K]	T_{34} = 307.7 [K]	T_4 = 597.3 [K]
T_6 = 334.6 [K]	T_7 = 338.2 [K]	T_9 = 359.3 [K]	W_{BFP} = 18425 [kW]	W_{CEP} = 815.7 [kW]	W_{HPT} = 199976 [kW]
W_{IPT} = 262382 [kW]	W_{LPT} = 264495 [kW]	W_{net} = 707612 [kW]	$W_{turbine}$ = 726853 [kW]		

Calculation time = .0 sec

Figure 37: EES Results

APPENDIX C[67]

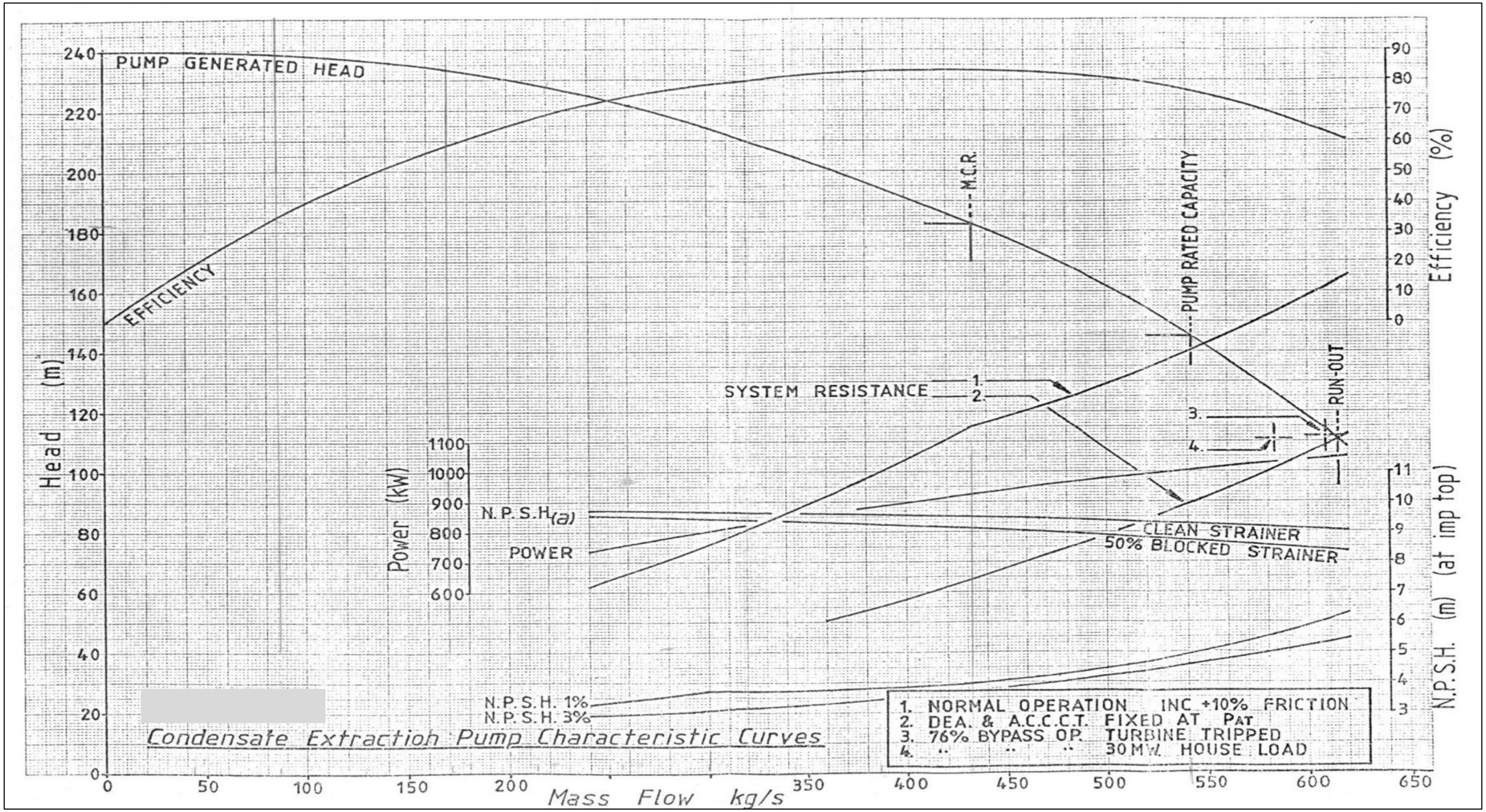

Figure 38: CEP Pump Curve

[67] The information in this appendix is confidential, and only available from the author of this dissertation by special request.

APPENDIX D

EES Calculation of Power Saving

According to the Pump affinity laws the following three equations are true:

N = Pump Shaft Speed [rpm]

Q = Flow [kg/s]

H = Head / Pressure [m]

P = Power [kW]

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

$$\frac{H_1}{H_2} = \left[\frac{N_1}{N_2}\right]^2$$

$$\frac{P_1}{P_2} = \left[\frac{N_1}{N_2}\right]^3$$

$$\dot{m}_1 = Q_1 \cdot \rho$$

$$\dot{m}_2 = Q_2 \cdot \rho$$

$\rho = 985$ [kg/m^3]

$$P_{H1} = \frac{\dot{m}_1 \cdot g \cdot H_1}{1000}$$ kg/s = m^3/s * kg/m^3

$$\eta_1 = \frac{P_{H1}}{P_1} \cdot 100$$

$$P_{H2} = \frac{\dot{m}_2 \cdot g \cdot H_2}{1000}$$

$$\eta_2 = \frac{P_{H2}}{P_2} \cdot 100$$

>>>Condition 1 - Rated

$N_1 = 1485$ [rpm]

$\dot{m}_1 = 542$ [kg/s]

$H_1 = 146$ [m]

$P_1 = 1004.25$ [kW]

$g = 9.81$ [m/s]

>>>Condition 2 - VSD

$\dot{m}_2 = 429.3$ [kg/s]

>>>Work at condition 2

$\dot{m}_{actual} = 429.3$ [kg/s]

$H_{actual} = 115$ [m]

$$P_{actual} = \frac{\dot{m}_{actual} \cdot g \cdot \frac{H_{actual}}{1000}}{\eta_{actual}}$$

$\eta_{actual} = 0.78$

EES Results for Affinity Law Calculations

Unit Settings: [kJ]/[C]/[kPa]/[kg]/[degrees]

$\eta_1 = 77.3$ [%]	$\eta_2 = 77.3$ [%]	$\eta_{actual} = 0.78$ [%]	$g = 9.81$ [m/s]
$H_1 = 146$ [m]	$H_2 = 91.6$ [m]	$H_{actual} = 115$ [m]	$\dot{m}_1 = 542$ [kg/s]
$\dot{m}_2 = 429.3$ [kg/s]	$\dot{m}_{actual} = 429.3$ [kg/s]	$N_1 = 1485$ [rpm]	$N_2 = 1176$ [rpm]
$P_1 = 1004$ [kW]	$P_2 = 499$ [kW]	$P_{actual} = 620.9$ [kW]	$P_{H1} = 776.3$ [kW]
$P_{H2} = 385.8$ [kW]	$Q_1 = 0.5503$ [m^3/s]	$Q_2 = 0.4358$ [m^3/s]	$\rho = 985$ [kg/m^3]

Calculation time = .0 sec

EES Calculation of Power Saving with Increased Flow

>>>Condition 2 - VSD

$$\dot{m}_2 = 429.3 \quad [kg/s]$$

>>>Work at condition 2

$$\dot{m}_{actual} = 476 \quad [kg/s]$$

$$H_{actual} = 123.5 \quad [m]$$

$$P_{actual} = \frac{\dot{m}_{actual} \cdot g \cdot \frac{H_{actual}}{1000}}{\eta_{actual}}$$

$$\eta_{actual} = 0.785$$

EES Results at Increased Flow

Unit Settings: [kJ]/[C]/[kPa]/[kg]/[degrees]

η_1 = 77.3 [%]	η_2 = 77.3 [%]	η_{actual} = 0.785 [%]	g = 9.81 [m/s]
H_1 = 146 [m]	H_2 = 91.6 [m]	H_{actual} = 123.5 [m]	$\dot{m}_1$ = 542 [kg/s]
$\dot{m}_2$ = 429.3 [kg/s]	$\dot{m}_{actual}$ = 476 [kg/s]	N_1 = 1485 [rpm]	N_2 = 1176 [rpm]
P_1 = 1004 [kW]	P_2 = 499 [kW]	P_{actual} = 734.6 [kW]	P_{H1} = 776.3 [kW]
P_{H2} = 385.8 [kW]	Q_1 = 0.5503 [m^3/s]	Q_2 = 0.4358 [m^3/s]	ρ = 985 [kg/m^3]

Calculation time = .0 sec

APPENDIX E[68]

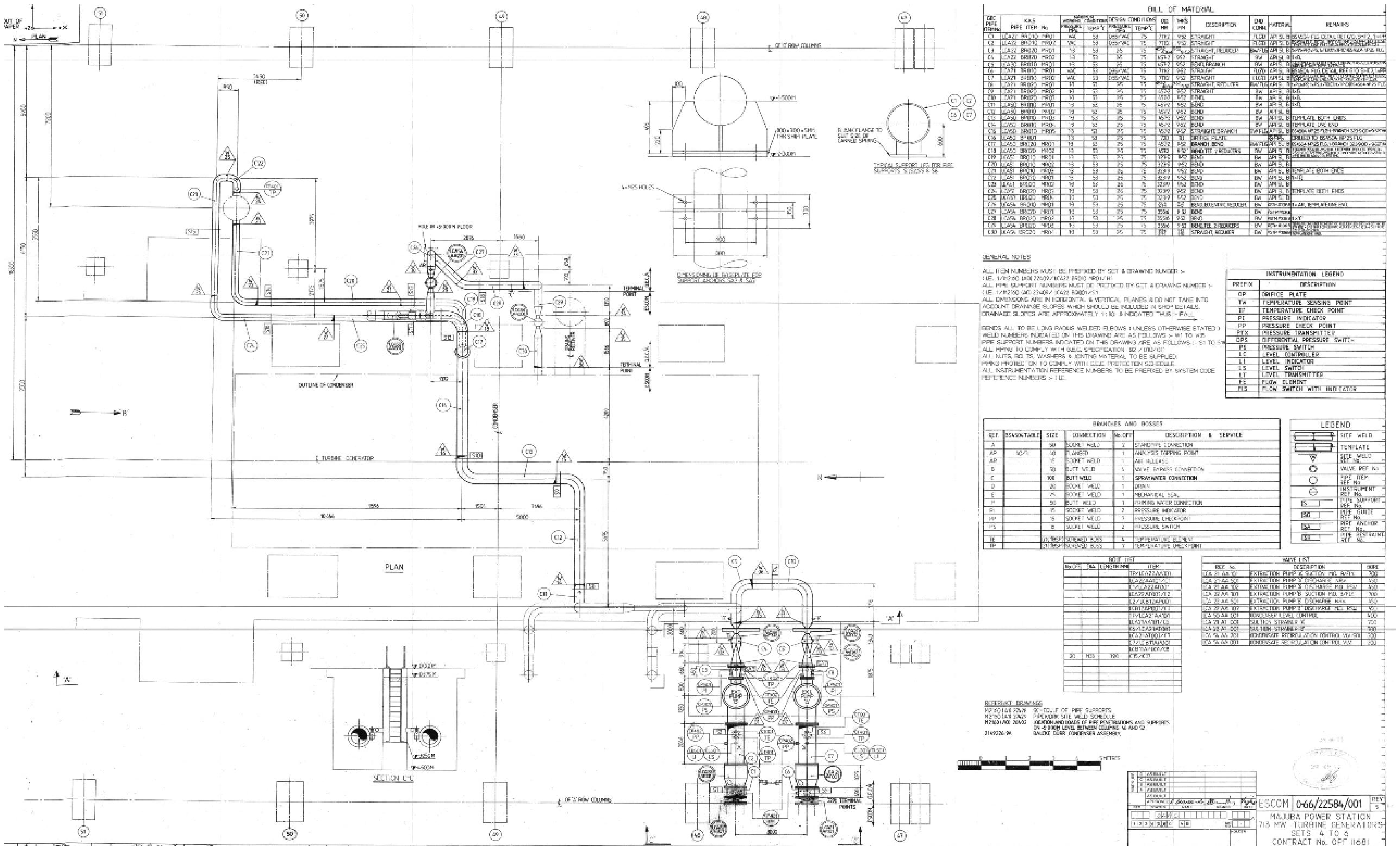

Figure 39: Condensate System Detailed Plant Arrangement Drawing A

[68] The information in this appendix is confidential, and only available from the author of this dissertation by special request.

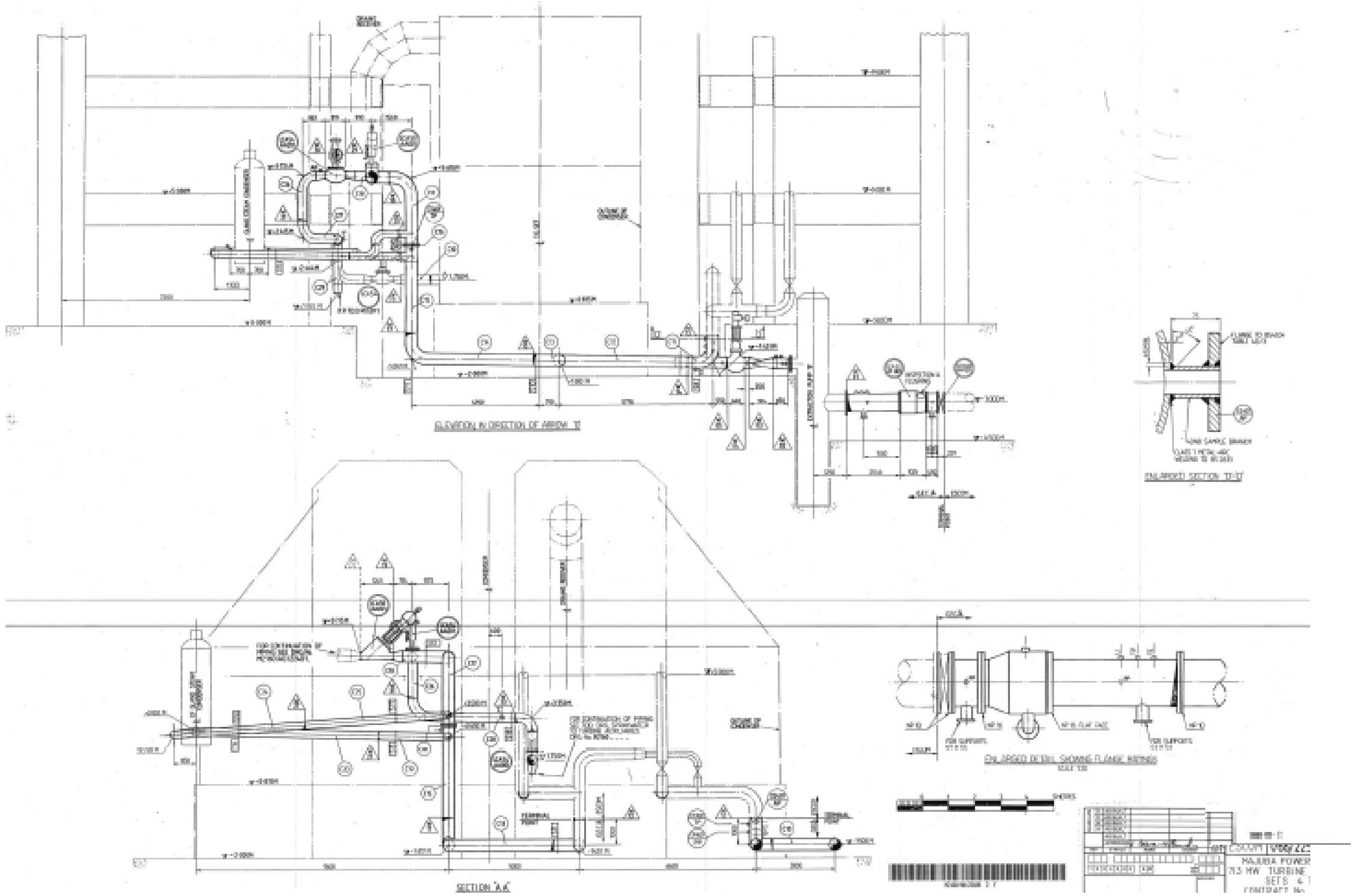

Figure 40: Condensate System Detailed Plant Arrangement Drawing B

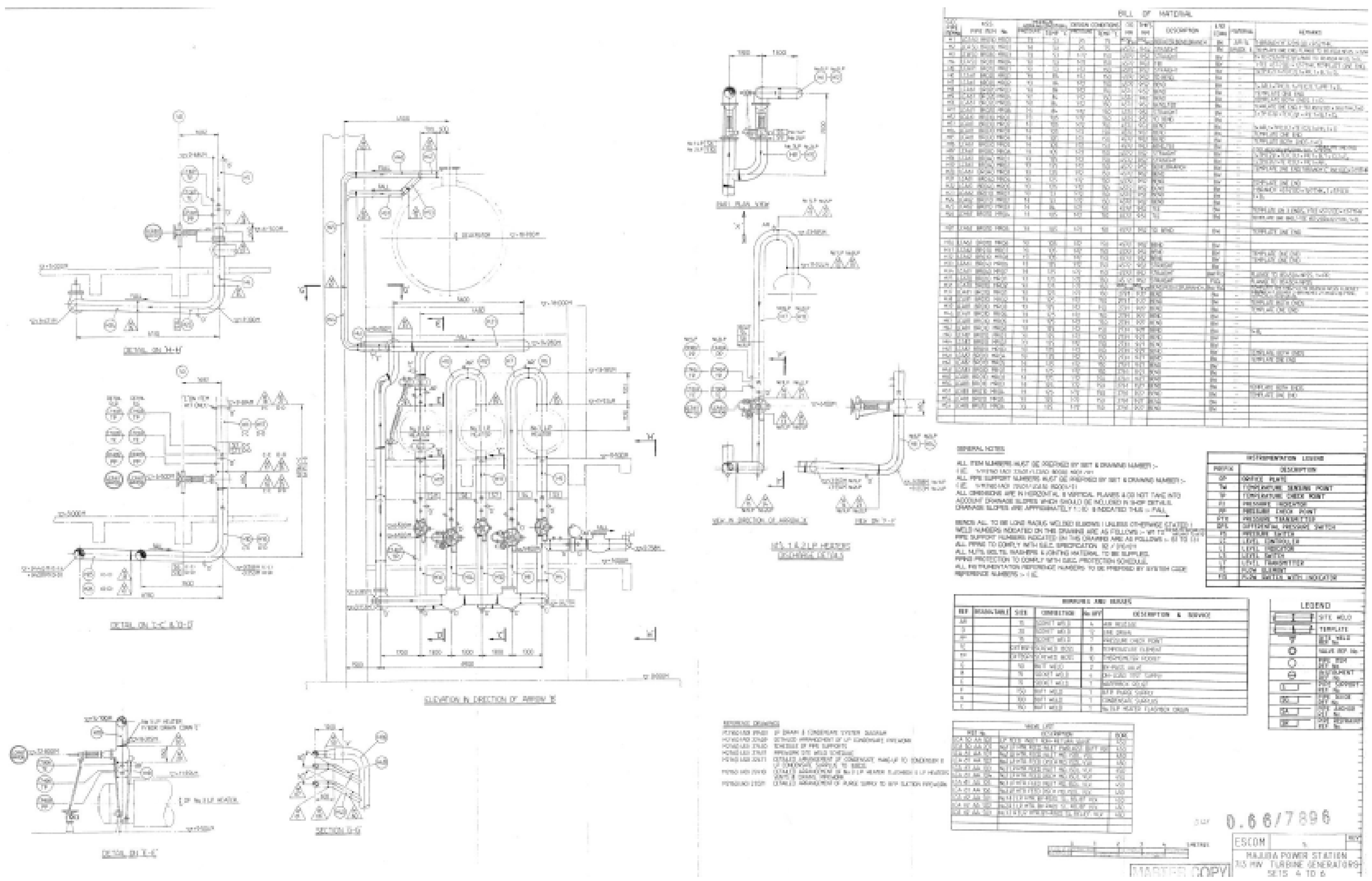

Figure 41: LP Feed-Heating System Detailed Plant Arrangement Drawing A

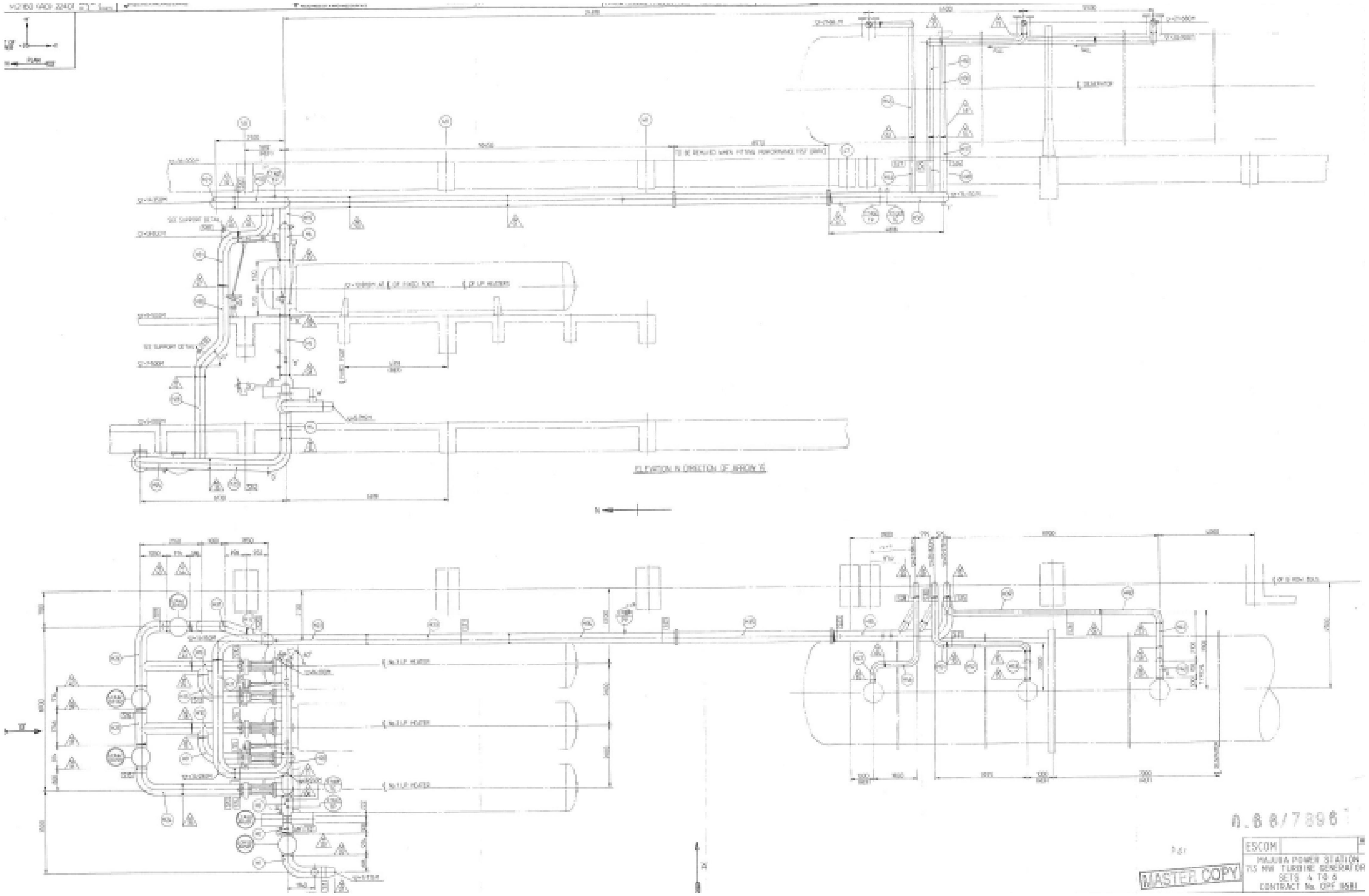

Figure 42: LP Feed-Heating System Detailed Plant Arrangement Drawing

APPENDIX F

F.1. Flownex Modeling in Steady State

The model was split into two sections, as was explained in section 5.2.1.; the LP Feed-Heating System and the Condensate System.

Figure 43 shows the interface of the LP Feed Water System (all plant in between the condensate control valve suction and where the stork sprayers spray into the De-aerator):

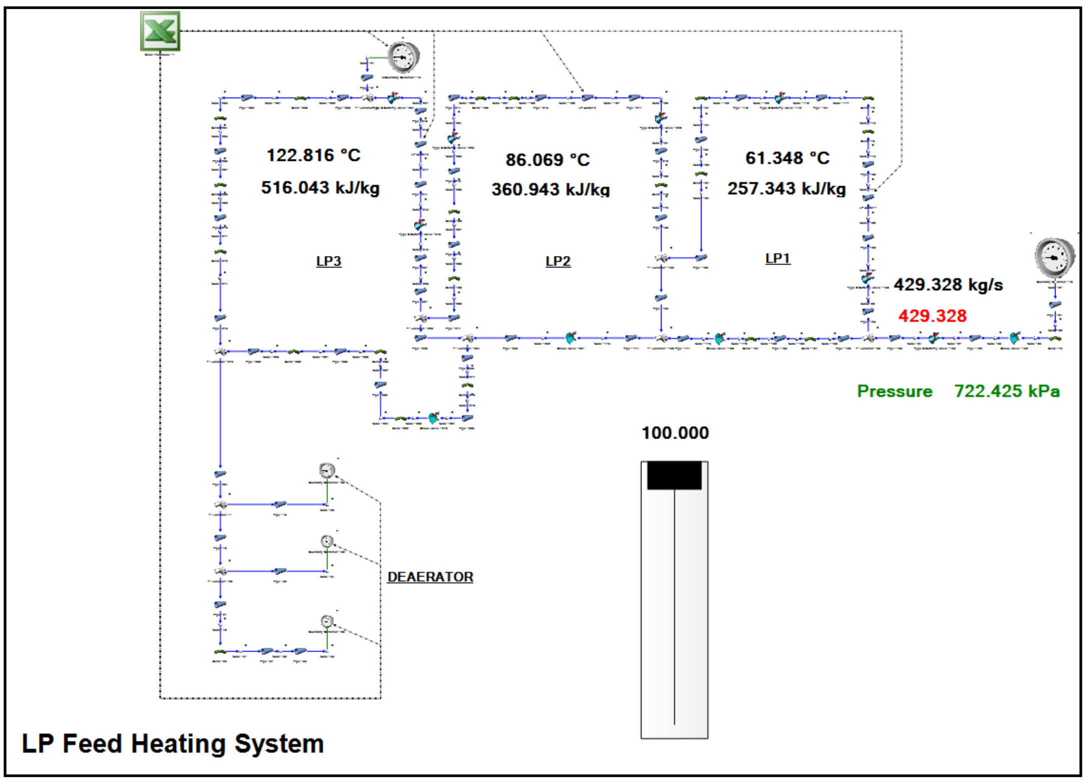

Figure 43: Flownex LP Feed-Heating System Interface

Figure 44 shows the interface of the Condensate System (all plant in between the Condenser outlet and the condensate control valve discharge):

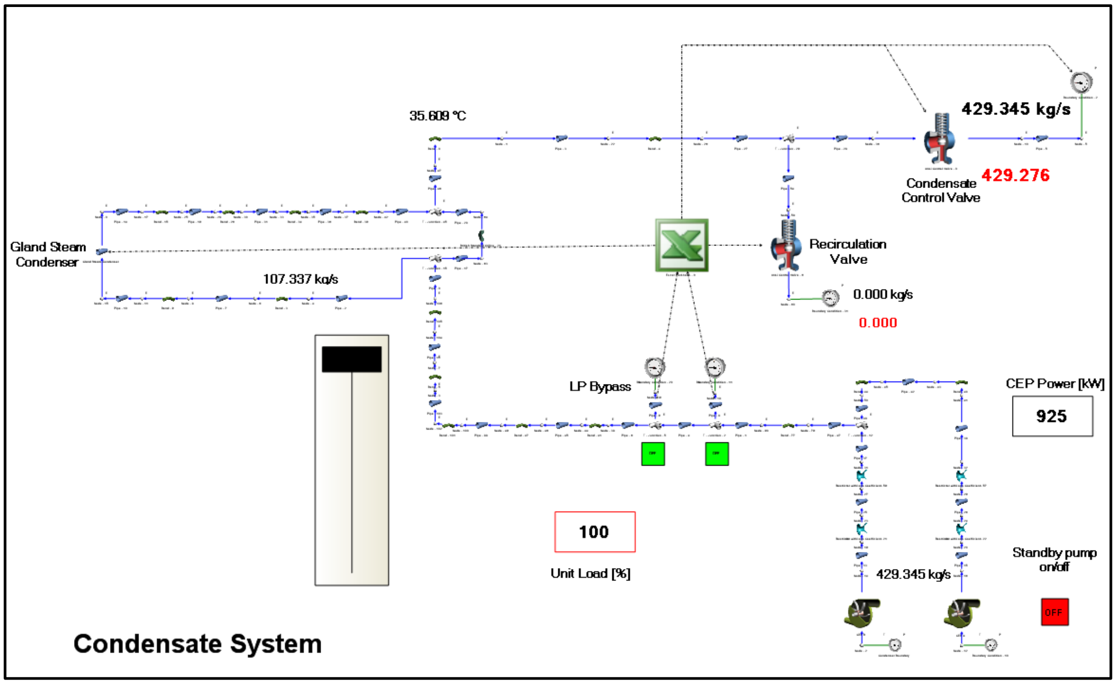

Figure 44: Flownex Condensate System Interface

The physical plant parameters (from APPENDIX E) and the thermodynamic set points (from APPENDIX A) were included in the model; and this allowed the model to be solved at steady state (full load conditions).

F.2. Demonstration of Equation Derivation for Simulating the Transient Model

To allow the model to simulate transient conditions, all the thermodynamic set points had to be changed to equations with one single variable. The single variable chosen to dictate all these equations was the unit load.

The following example explains how the predictive equations were derived:

Third order equation: $y = ax^3 + bx^2 + cx + d$

Where,

$y = thermodynamic\ setpoint(like\ the\ de - aerator\ pressure)$

and $x = unit\ load\ [MW]$

In this example, $\boldsymbol{a}, \boldsymbol{b}, \boldsymbol{c}$ and $\boldsymbol{d}$ will be dependent on the characteristic curve predicting the De-aerator pressure. The pressure in the De-aerator was plotted against the load from the data collected from the heat-balance diagrams (APPENDIX A), and yielded the following curve:

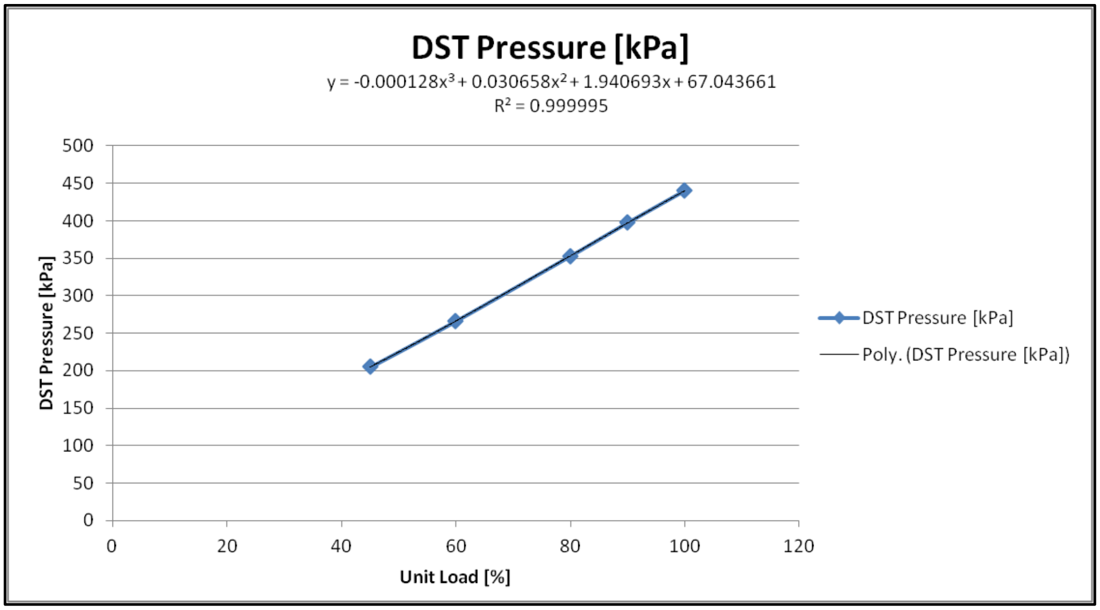

Figure 45: De-aerator Pressure vs Load Curve

A polynomial trend line was fitted to the curve, as a regression line, to statistically predict the pressure of the DST at any load. The R^2 value featured below is the coefficient of determination (COD); and it illustrates how well the regression line fits the actual data curve. Increasing the order of the polynomial function usually increases the COD; and for this reason, the order of the polynomial function had to be increased to the third order – to get to a COD close to 1.

The equation [$y = -0.000128x^3 + 0.030658x^2 + 1.940693x + 67.043661$] could thus be used to determine the pressure in the De-aerator (y) at different loads (x).

F. 3 Development of the Transient Model in Flownex

The equations for all the set point inputs into the Flownex model are listed in Table 14:

Table 14: Equations Derived for Flownex Transient Modelling

	Thermodynamic Set Points	**Polynomial Trend Line Equation**	**Coefficient of Determination (R^2)**
1	**De-aerator Pressure**	$y = -1.28e\text{-}04x^3 + 3.0658e\text{-}02x^2 + 1.940693x + 6.7043661e\text{+}01$	0.999995
2	**Stork Sprayer Pressure Drop**	$y = -3.4482184e\text{-}08x^3 – 8.5619029046e\text{-}05x^2 + 1.18004719998439e\text{-}01x + 27.027774565585700$	0.999854
3	**LP Drain-Recovery System Mass Flow**	$y = -8.3265e\text{-}07x^4 + 2.3871e\text{-}04x^3 – 2.4196e\text{-}02x^2 + 1.3208x – 1.6943e\text{+}01$	1.000000
4	**LP Heater 1 Heat Load**	$y = -1.6333658961e\text{-}03x^4 + 4.6162288112e\text{-}01x^3 - 4.4334540430e\text{+}01x^2 + 2.2915371534e\text{+}03x - 3.8070605211e\text{+}04$	1.000000
5	**LP Heater 2 Heat Load**	$y = -1.6366164876e\text{-}03x^4 + 4.6799590388e\text{-}01x^3 - 4.7665617211e\text{+}01x^2 + 2.5229847315e\text{+}03x - 3.5498140822e\text{+}04$	1.000000
6	**LP Heater 3 Heat Load**	$y = -1.3769767035e\text{-}03x^4 + 3.8253142327e\text{-}01x^3 - 3.7431488545e\text{+}01x^2 + 2.2026830862e\text{+}03x - 2.4198351595e\text{+}04$	0.999999
7	**Gland-Steam Condenser Heat Load**	$y = 9.2338811394e\text{-}05x^4 - 2.8267641093e\text{-}02x^3 + 3.1471540914e\text{+}00x^2 - 1.4458500390e\text{+}02x + 3.7380327633e\text{+}03$	1.000000
8	**Main Condensate Mass Flow**	$y = -9.9105973760e\text{-}06x^4 + 2.8378466711e\text{-}03x^3 - 2.9159750900e\text{-}01x^2 + 1.6709315186e\text{+}01x - 1.7241502888e\text{+}02$	1.000000
9	**Control Valve Discharge Pressure**	$y = -1.2309e\text{-}04x^3 + 3.9043e\text{-}02x^2 + 1.7042e\text{+}00x + 1.9899e\text{+}02$	1.000000
10	**Condensate Control Valve Mass Flow**	$y = -1.537e\text{-}02x^6 + 1.9513e\text{-}01x^5 - 1.1829e\text{-}01x^4 - 6.54515x^3 + 2.584604e\text{+}01x^2 + 3.666990e\text{+}01x - 3.354e\text{-}02$	0.999610
11	**Recirculation Control Valve Mass Flow**	$y = 1.917e\text{-}02x^6 – 5.0798e\text{-}01x^5 + 4.60272x^4 – 1.586780e\text{+}01x^3 + 1.643196e\text{+}01x^2 – 5.823904e\text{+}01x + 2.7549030e\text{+}02$	0.999720
12	**Control Valve Signal**	$y = -1.977e\text{-}11x^6 + 7.32786e\text{-}09x^5 - 1.01492970e\text{-}06x^4 + 6.128721967e\text{-}05x^3 - 1.38289474423e\text{-}03x^2 + 6.576670629804e\text{-}02x + 4.8127402407060e\text{-}01$	0.999999
13	**Condensate Control Valve Opening with LP Bypass Closed**	$y = -1.69228e\text{-}10x^6 + 7.4154061e\text{-}07x^5 - 1.3308011851e\text{-}05x^4 + 1.250684471320e\text{-}03x^3 -$	0.999700

		$6.4876645896800e-02x^2 + 1.766419407950830x – 1.9463876667379600e+01$	
14	**Recirculation Control Valve Opening with LP Bypass Closed**	$y = 2.2330e-11x^6 - 7.208977e-09x^5 + 8.19421092e-07x^4 - 3.4709142511e-05x^3 + 8.4288193585e-05x^2 + 1.2400596501443e-02x + 4.43770470176989e-01$	0.999893
15	**Condensate Control Valve Opening with LP Bypass Open**	$y = -7.222e-12x^6 + 2.791803e-09x^5 - 4.08399561e-07x^4 + 2.7404987969e-05x^3 - 7.52758711531e-04x^2 + 6.303440407819e-03x + 3.13735713486782e-01$	0.999788
16	**Recirculation Control Valve Opening with LP Bypass Open**	$y = 2.5693e-11x^6 - 8.494964e-09x^5 + 1.016106547e-06x^4 - 5.0233990334e-05x^3 + .78386896536e-04x^2 - 6.118686682098e-03x + 7.07780824426872e-01$	0.999892
17	**Pump Power Consumption at Fixed Speed**	$y = 7.563884e-04x^3 - 1.961044861e-01x^2 + 2.08160859017e+01x + 4.77458572062e+01$	0.999866
18	**Pump Power at Variable Speed**	$y = -3e-02x^2 + 1.49e+01x + 7e+01$	1.000000

In the same order, as in

Table 14, the following procedure was followed to develop the transient model:

1. Derivation of the De-aerator Pressure Equation.
2. Derivation of the Stork-Sprayer Pressure Drop equation (this had to be subtracted from the De-aerator Pressure equation to get to the actual pressure at the inlet of the De-aerator.
3. Derivation of the LP Drain-Recovery System mass flow equation.
4. Derivation of LP Heater 1 Heat Load.
5. Derivation of LP Heater 2 Heat Load.
6. Derivation of LP Heater 3 Heat Load.
7. Derivation of the Gland Steam Condenser Heat Load.
8. Derivation of the main condensate mass flow (before the LP drain-recovery flow was introduced).
9. The pressure drop was now available over the LP Feed Heating System. The Designer function in Flownex was used to determine the Condensate Control Valve discharge pressure for every load – from 45% to 100% load in 5% increments. These values were plotted on a curve, and an equation was derived to predict the pressure for any given load.

10. Derivation of the Condensate Control Valve mass flow equation with a control signal as the variable.
11. Derivation of the Recirculation Control Valve mass flow equation with a control signal, as the variable.
12. Derivation of the control-signal equation for any given load.
13. The pressure drop over the Condensate System is now available; and the Designer function in Flownex was used with the discharge pressure of the Condensate Control Valve to determine the Condensate Control Valve Opening for loads from 45% to 100%, in 5% increments. These data were plotted on a curve; and an equation was derived to predict the Condensate Control Valve Opening – for any given load.
14. The same was done to derive an equation for the Recirculation Control Valve Opening – for any given load.
15. Exactly the same simulation was performed, as in bullet 13; but this time, the LP Bypass Spray Water Valves were opened, giving the equation for the Condensate Control Valve Opening for any given load during emergency conditions.
16. Exactly the same simulation was performed, as in bullet 14; but this time, the LP Bypass Spray Water Valves were opened, giving the equation for the Recirculation Control Valve Opening for any given load during emergency conditions.
17. An equation was derived for the Pump Power Consumption at fixed speed (utilizing valve-throttling control) for any given load.
18. An equation was derived for the Pump Power Consumption at variable speeds (utilizing electrical VSD flow control) for any given load.

Table 15 shows the results for the DST stork sprayer simulation in Flownex performed under bullet 2 above. In Figure 43, it can be observed that there are three different inlet lines from the Feed Water System into the DST. This pressure drop for each of the individual lines was deducted from the DST pressure (bullet 1 above) to reveal the actual DST inlet pressure for each of these lines.

Table 15: DST Stork Sprayers Pressure Drop Results

Predicted Pressure Drops					Difference			New Predicted Mass Flows		
Load	Line 1	Line 2	Line 3	Deaerator Pressure	Line 1	Line 2	Line 3	Line 1	Line 2	Line 3
100	43.57123	43.00555	42.17701	440.0840	483.6552	483.0896	482.2610	483.6552	483.0896	482.261
95	42.89895	42.35059	41.54735	418.6893	461.5883	461.0399	460.2367	461.5883	461.0399	460.2367
90	42.17851	41.64964	40.87621	397.0091	439.1876	438.6588	437.8853	439.1876	438.6588	437.8853
85	41.43282	40.9249	40.18394	375.1391	416.5719	416.0640	415.3230	416.5719	416.064	415.323
80	40.67944	40.19335	39.4861	353.1749	393.8543	393.3682	392.6610	393.8543	393.3682	392.661
75	39.93053	39.46675	38.79347	331.2123	371.1428	370.6790	370.0058	371.1428	370.679	370.0058
70	39.19287	38.75166	38.11208	309.3470	348.5398	348.0986	347.4590	348.5398	348.0986	347.459
65	38.46789	38.04943	37.44314	287.6747	326.1425	325.7241	325.1178	326.1425	325.7241	325.1178
60	37.75163	37.35619	36.78312	266.2910	304.0427	303.6472	303.0742	304.0427	303.6472	303.0742
55	37.03473	36.66284	36.12368	245.2918	282.3266	281.9547	281.4155	282.3266	281.9547	281.4155
50	36.30249	35.95511	35.45172	224.7728	261.0752	260.7279	260.2245	261.0752	260.7279	260.2245
45	35.5348	35.21348	34.74936	204.8295	240.3643	240.0430	239.5788	240.3643	240.043	239.5788

Table 16: Unit Load vs Condensate and Recirculation Valve Control Signal

Load	Control Signal
100	6.532491522
95	6.29701836
90	6.051844688
85	5.796265624
80	5.528963267
75	5.248176062
70	4.951988598
65	4.638951854
60	4.309132621
55	3.965076452
50	3.61126738
45	3.251343043

Table 17 tabulates the power consumed by the fixed-speed pump, the power consumed by the VSD pump, and the difference between these values for every load:

Table 17: Power Consumption Saving for Different Unit Loads

Load		**Power Consumed**		**Power Difference**
		Fixed Speed	***VSD***	
130		1101.457	1100	1.456524137
125		1062.945	1007.5	55.44509336
120		1028.811	919	109.8107208
115		998.4861	834.5	163.9861151
110		971.404	754	217.403985
105		946.997	677.5	269.4970392
100		924.698	605	319.6979864
95		903.9395	536.5	367.4395353
90		884.1544	472	412.1543945
85		864.7753	411.5	453.2752729
80		845.2349	355	490.2348791
75		824.9659	302.5	522.4659218
70		803.4011	254	549.4011096
65		780	209.5	570.5
60		780	209.5	570.5
55		780	209.5	570.5
50		780	209.5	570.5
45		780	209.5	570.5

F.4 CEP Variable Speed Control Interface:

After the EXCEL inputs were included in the model design, and the heat balance diagrams confirmed that the model operates at design conditions in transient situations (For example: the red design mass flow and the black model mass flow values in Figure 43 and Figure 44 should be very close to each other), one of the CEPs in the model in Figure 44 was exchanged for a CEP with electrical VSD control. The new plant interface is shown in Figure 46:

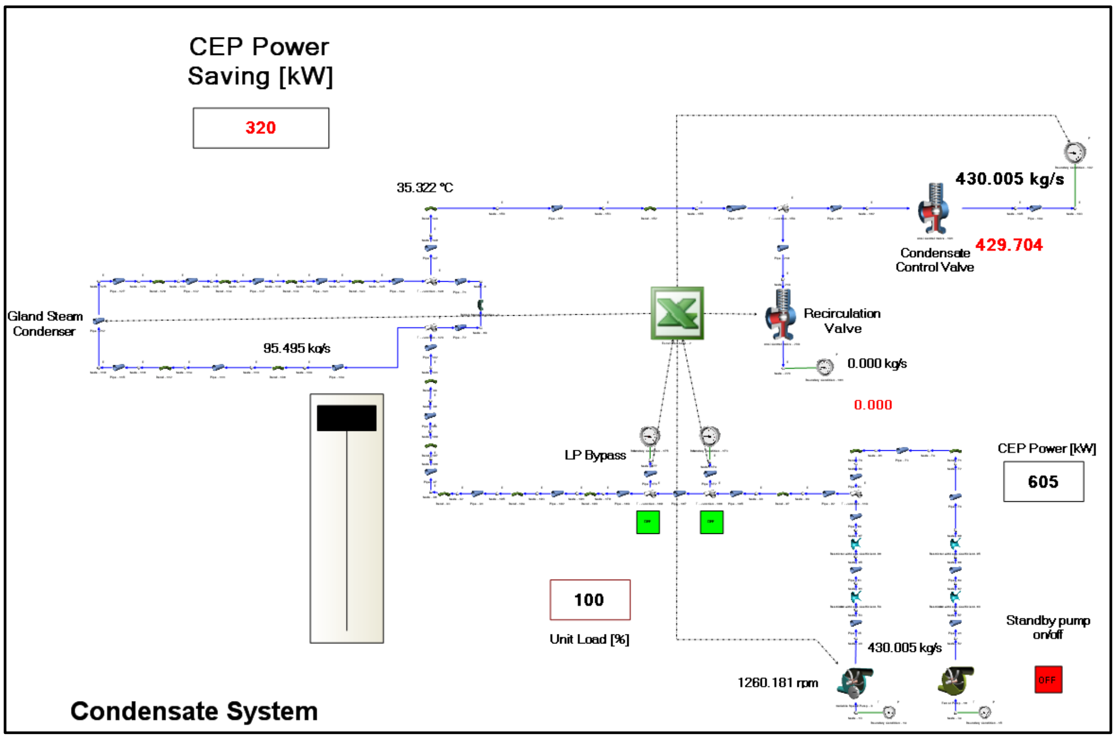

Figure 46: Flownex Condensate System Interface with VSD Flow Control

The energy saving expected at full load is displayed in the notice box at the top left corner of the interface; and it shows the 320 kW achievable at full loads.

Even when operating the pumps at a lower average load, the energy savings achievable correspond with those in Table 9 in section 9.

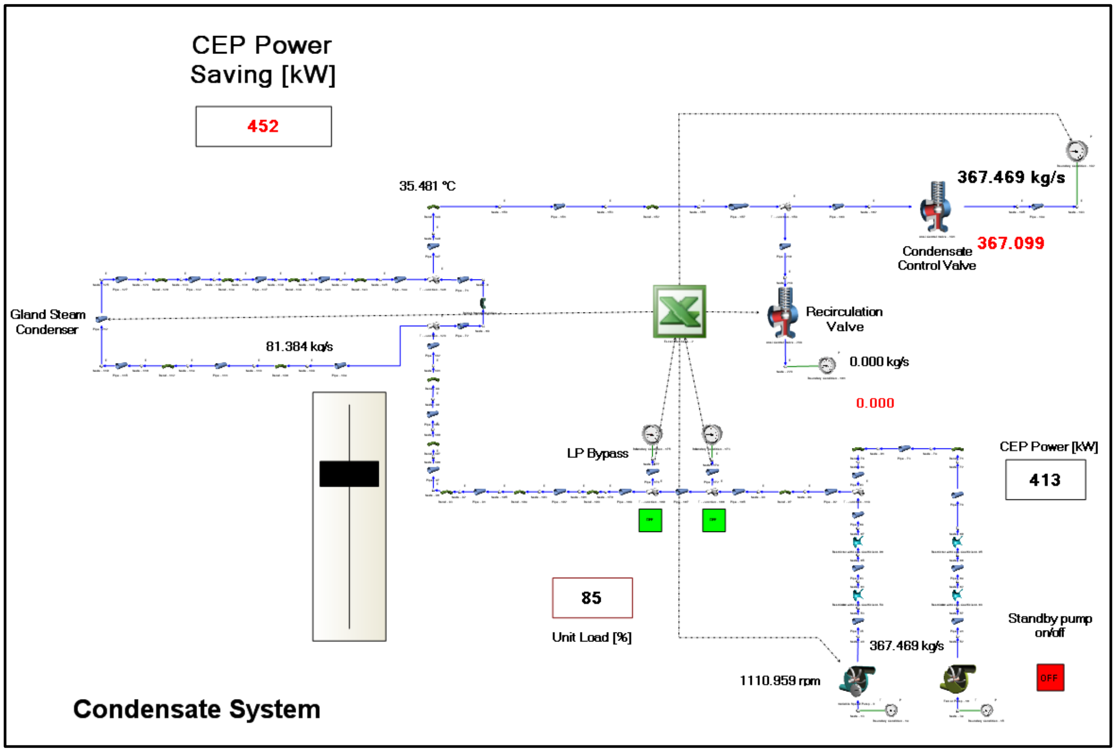

Figure 47: Flownex Condensate System Plant Interface with VSD Control (85% load)

For the fictional load of 95.6% displayed in Table 9, the calculated energy savings would be **361.9 kW**, which corresponds to the Flownex input EXCEL spread sheet value in Figure 48:

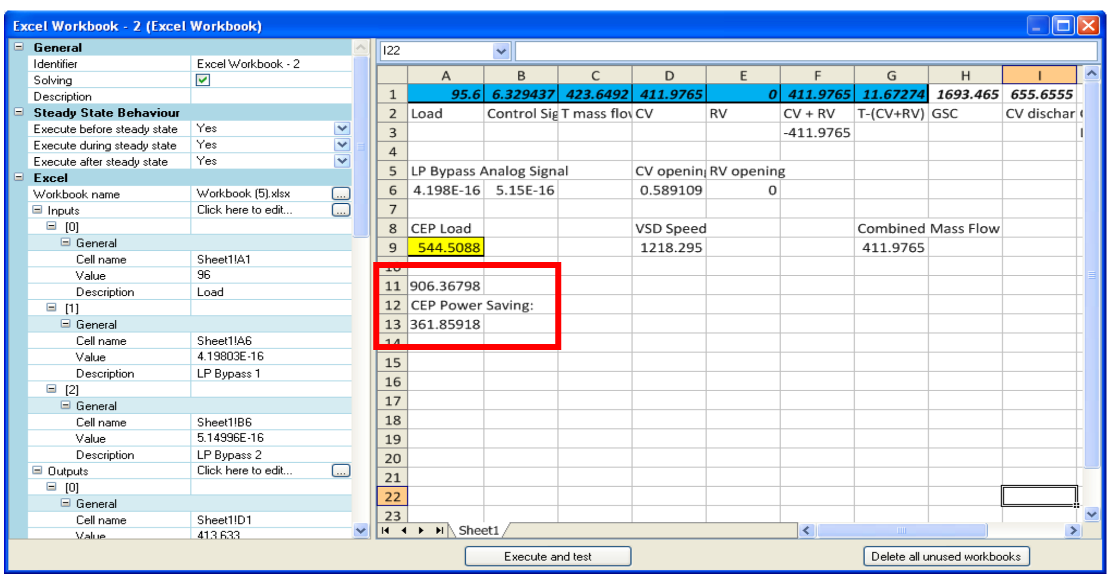

Figure 48: Flownex EXCEL Workbook