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# Optimum Time to Refurbish a Boiler Feed Pump at Eskom's Power Stations

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MRTSTE 007

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requirements of the degree of Master of Engineering

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## DECLARATION

I, Steven Peter Mortlock (Identity Number: 851121 5037 084) hereby declare that the work contained in this dissertation is my own work and has not been submitted before to any other university to obtain a degree. Some of the information contained in this dissertation has been gained from various journal articles, text books, manuals, etc., and has been referenced accordingly. The research in this dissertation was supported and funded by Eskom as part of the Eskom Power Plant Engineering Institute initiative.

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*(Signature of candidate)*

\_\_\_\_\_ day of \_\_\_\_\_ 20\_\_\_\_

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## ABSTRACT

A key focus area in power generation today is to improve equipment and plant efficiency, with this study specifically aimed to determine the optimum time to refurbish a boiler feed pump by analysing the internal wear of the annular seals.

The boiler feed pump (BFP) installed in a power station is either driven by a steam turbine or an electric motor and the BFP is the highest auxiliary power consumer within the power station. In the 13 Eskom generating fossil fired power stations the installed BFPs consume approximately 1000 MWs of auxiliary power continuously.

The focus of the study is to understand how the BFPs efficiency changes with time as a result of increased internal clearances and the effect this has on the auxiliary power consumption.

An in-depth analysis of the efficiency degradation has been studied to understand the contributing factors to these changes and specifically the change in leakage rate and friction in the annular seals of the impellers and axial thrust balancing devices.

Since the commissioning of the power stations and BFPs, data has been recorded during the refurbishment periods of the increases in clearances of the annular seals over a period of time. This information was then used to determine how the BFP wear rate of the internal components would increase over the operating life of the BFP.

The leakage around the impellers and the axial thrust balance devices as well as the friction has been modelled to predict a BFPs performance as a result of the change in internal clearances. The predicted model results of the leakage have been validated by using Sulzer Pumps design tables which, together with the OEM pump designers, the optimum time to refurbish a boiler feed pump can be predicted or determined.

On-site performance testing was done by contractors to determine the operating parameters and actual performance of the BFPs. The BFPs were tested with certain limitations and found not to be within the accuracy required for the efficiency change as a result of internal wear.

A one percentage point efficiency improvement of the BFPs installed across the 13 Eskom Generating Power Stations would result in an annual saving of 79 million kWh of electrical power which would be enough to supply 158,000 homes in Africa with electricity annually.

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**NOMENCLATURE**

A	Surface roughness
a	area
AVT(R)	All Volatile Treatment (reducing)
API	American Petroleum Institute
BEP	best efficiency point
BC	boundary conditions
BFP	boiler feed pump
BHN	Brinell Hardness Number
$c_{ax}$	axial velocity in annular seal
CFD	computer-aided fluid design
CSS	Customer Support Services
$d$	diameter
$d_2$	impeller outside diameter
$d_{EK}$	diameter of the leakage in axial thrust balance device
$d_{sp}$	diameter at which annular seal leakage occurs
EPPEI	Eskom Power Plant Engineering Institute
EFP	electric feed pump
EPRI	Electric Power Research Institute
FAC	flow accelerated corrosion
$f_L$	influence of leakage flow on disk friction
$f_{R,La}$	influence of impeller roughness
$f_q$	number of impeller eyes
g	gravity
h	pressure difference
H	head generated
$H_p$	static pressure rise in the impeller
$H_{sp}$	pressure difference across annular seal (flow is radially inwards)
$H_{s3}$	pressure difference across annular seal (flow is radially outwards)
$H_{EK}$	pressure difference across axial thrust balance device
$H_{tot}$	total head generated by the pump

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ISO	International Organization for Standardization
$k$	rotation of fluid in the impeller sidewall gap
K	experimental factor
L	length
$L_{sp}$	length of annular seal
m	empirical constant
n	speed
$n_q$	specific speed
OEM	original equipment manufacturer
OT	oxygenated treatment
$P$	power at coupling
$p$	pressure
$P_u$	useful power
$P_{RR}$	disk friction power
$P_{S3}$	inter-stage seal power
$P_L$	leakage power loss
$P_{er}$	friction power loss
$P_{vh}$	hydraulic power loss
$P_{rec}$	recirculation power loss
$P_m$	mechanical power loss
$p_2$	static pressure at the outlet of the impeller
$P_{u,opt}$	useful power at BEP
$P_i$	inner power
PS	power station
$Q$	volumetric flow rate
$Q_{sp}$	leakage flow rate through a seal at impeller inlet
$Q_{S3}$	leakage flow rate through inter-stage seal
$Q_E$	leakage flow rate through axial thrust balancing device
$Q_h$	leakage flow rate through auxiliaries
NPSH	net positive suction head
NDE	non-drive end

$Q_{opt}$	volumetric flow rate at BEP
Re	Reynolds Number
$Re_u$	Reynolds Number in circumferential component
$R_G$	degree of reaction
r	radius
$r_{sp}$	radius of annular seal leakage
s	gap width
SFP	steam feed pump
TOO	take-off open
TOC	take-off closed
UCLF	Unplanned Capacity Load Factor
$u$	circumferential velocity
$u_2$	circumferential velocity at impeller blade trailing edge
$u_{sp}$	circumferential velocity at annular seal
v	velocity
$Z_{la}$	number of impeller blades
$Z_{ST}$	number of stages
<b><u>Greek Symbols</u></b>	
$\varepsilon$	surface roughness
$\zeta_{EA}, \zeta_K$	loss coefficients
$\gamma_{sp}$	impeller discharge coefficient at annular seal
$\Psi_{opt}$	head coefficient at BEP
$\varphi_{sp}$	flow coefficient of impeller sidewall gap
$\rho$	density
$\eta_v$	volumetric efficiency
$\eta_h$	hydraulic efficiency
$\eta$	efficiency
$\mu$	dynamic viscosity
$\lambda$	Friction coefficient

## 1. BACKGROUND

With the rapidly growing demand for electricity in South Africa, Eskom have embarked on a capital expansion programme to increase the power generation capacity from 44,000 MW to 75,000 MW by 2025. In order to ensure the success of this capital investment the need for specialised skills has been identified and the Eskom Power Plant Engineering Institute has been established to meet some of these requirements.

It has been identified that the current engineering courses in South African universities do not fully meet the specific subject skills required for the power plant industry. Together with the shortage of qualified engineers with the necessary skills, this has created an opportunity for Eskom and South African universities to collaborate and overcome these challenges together.

Eskom is the biggest producer of electricity in South Africa. A power station however uses auxiliary power to drive numerous equipment in the power station to produce electricity. The combined auxiliary power required for all of Eskom's power stations to drive equipment makes Eskom the biggest consumer of electricity as a single entity in South Africa.

At the University of Cape Town an energy efficiency centre of excellence has been established where the focus of projects look at how efficiently various systems in the power stations are operating.

Therefore the current project is formulated keeping in mind the efficiency improvement in generating electricity. This project focuses on the economic lifecycle of the BFPs considering the effect that the internal wear has on the annular seal leakage in a BFP.

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## 2. INTRODUCTION

In a power station the biggest consumer of auxiliary power is a boiler feed pump, which consumes between 2.5 % and 4 % of the total generated power of the power station.

The auxiliary power of the boiler feed pumps has a direct influence on the power stations heat rate and the overall cycle performance and efficiency. When a BFP starts to wear the internal losses will increase and the BFP will absorb more power to supply the required flow and pressure to the boiler.

At the 13 Eskom fossil-fired generating power stations there are 237 installed BFPs, of which 101 are continuously running, consuming approximately 1000 MWs of auxiliary power constantly. The majority of the BFPs installed in South African power stations are designed and manufactured by Sulzer Pumps, with whom Eskom have a Partnering Agreement.

Records of the internal wear of BFPs have been recorded over many years by the various site managers and pump inspectors, which show a trend in the wear behaviour at each of the power stations. The power stations which have been analysed in this study are Lethabo, Kendal, Matla, Matimba and Kriel, due to the availability of data, records of internal clearances and information relating to the design of the BFP.

The changes in clearances between rotating and stationary parts were modelled through various methodologies and the change on the BFP efficiency was predicted. The work published by Dr Johann Gülich of Sulzer Pumps Switzerland (Gülich, 2008) was a major source for the prediction of fluid flow around an impeller and the leakage through the annular seals of a centrifugal pump.

The period spent at the Sulzer Pumps head offices with the design engineers in Winterthur, Switzerland was focused on validating the models created, using the in-house empirical Sulzer Design Tables as well as CFD software modelling tools.

The collaboration formed between Eskom and Sulzer Pumps (Winterthur) was the first of its kind for Eskom and a pump manufacturer, with the intention to understand the needs of the client and the areas that require technological improvements.

To validate the leakage through the rotating and stationary parts and the effect this has on the efficiency of the BFP, on-site performance testing of some of the BFP systems was done at the power stations to achieve unity between the models and the operating system.

The effect of the annular seal leakage on efficiency was predicted and the life cycle cost management of each BFP was determined. Each power station was calculated individually in order to determine the optimum refurbishment intervals.

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### 3. LITERATURE REVIEW

#### 3.1 BOILER FEED PUMPS (BFP)

A BFP is a multi-stage centrifugal pump that generates the highest pressure in a power station and supplies demineralised water to the boiler in a coal fired power station.<sup>1</sup>

During the late 1970s and the early 1980s low availability and reliability as a result of BFP failures contributed to high unplanned capacity load factor (UCLF) losses in power stations worldwide. During this period EPRI initiated a research and development programme in various aspects of BFP design and operation over a period of 10 years to improve the reliability and availability of BFPs. This initiative was extremely successful with huge improvements in BFP availability and reliability today (EPRI, 1994).

The rotordynamics of the BFP was one of the focal points of the research and development which included many studies conducted on the behaviour of the annular seals and seal coefficients (Florjancic, 1990).

Results of the studies improved the reliability and reduced the costs associated with unplanned load losses at a power station. The design of the BFPs improved (Energy Research and Consultants Corporation, 1978) and with increased redundancy and superior designs this increased the availability of the power stations considerably.

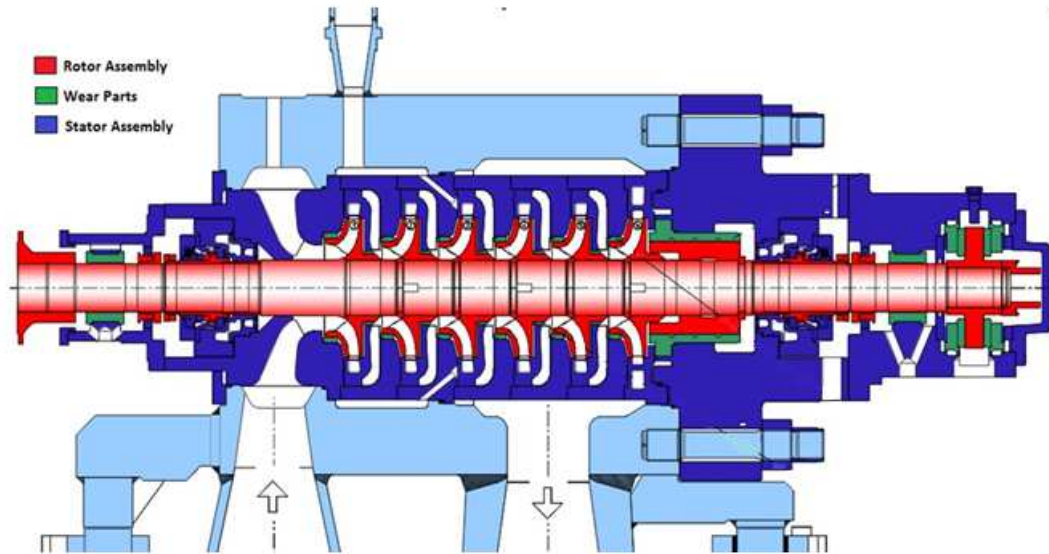
BFPs have become more reliable and have exceeded the OEMs recommended running hours in many instances (System Engineers, 2013)<sup>2</sup>. With the current focus on energy efficiency as a result of the shortfall of electricity supply in South Africa the operating life cycle cost is becoming a focal point and area for research and development (Caldo, February 2008).

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<sup>1</sup> Specific to BFPs that are being studied in this dissertation

<sup>2</sup> At Matimba Power Station an Electric Feed Pump (EFP) recently ran for 124,000 running hours with many more currently having exceeded 80,000 running hours

Figure 1 below highlights the rotating and stationary components and specifically the internal wear parts on a BFP which are a focus for this study.



**Figure 1: Sectional view of a typical BFP showing wear areas (Sulzer Pumps)**

### 3.2 PUMP EFFICIENCY

To determine the useful power ( $P_u$ ) transferred to the liquid the disk friction ( $P_{RR}$ ), losses dissipated in inter-stage seal ( $P_{S3}$ ), leakage losses ( $P_L$ ), friction losses ( $P_{er}$ ), hydraulic losses ( $P_{vh}$ ), recirculation losses ( $P_{Rec}$ ), mechanical losses ( $P_m$ ) and noise need to be considered as depicted in figure 2.

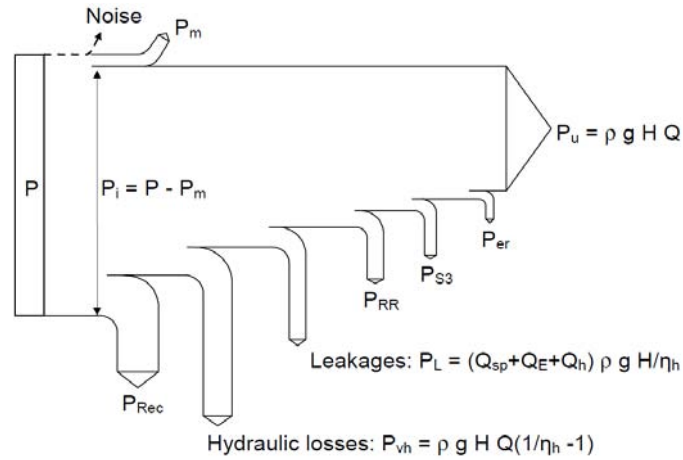


Figure 2: Power balance of a pump (Gülich, 2008)<sup>3</sup>

### 3.3 SECONDARY LOSSES

Disk friction losses and leakage losses are considered as separate phenomena that can be measured in special test circuits or even inside a pump. The close interaction between the main flow at the impeller outlet and the flow in the impeller sidewall gaps<sup>4</sup>, leakage through annular seals and the disk friction losses are difficult to capture experimentally or numerically (Gülich, 2008)<sup>5</sup>.

#### 3.3.1 Volumetric Efficiency

The volumetric flow through the impeller is greater than the measured flow of the pump by the amount of leakage flow and the ratio of the measured flow  $Q$  to the impeller flow (Stepanoff, 1957).  $Q$ ,  $Q_{sp}$ ,  $Q_E$  and  $Q_h$  are used to calculate the volumetric efficiency using Equation 1 (Gülich, 2008)<sup>6</sup>.

$$\frac{Q}{(Q + Q_{sp} + Q_E + Q_h)} = \eta_v \quad (1)$$

Usually the volumetric efficiency only takes into account the leakage between the impeller and the casing at the impeller eye for multistage pumps and single-stage

<sup>3</sup> Section 3.5 (p 85)

<sup>4</sup> See Figure 3 (p 7)

<sup>5</sup> Section 3.6 (p 85)

<sup>6</sup> Table 3.5 (p 135)

pumps. The inter-stage leakage in multistage pumps and leakage through the balancing device need to be treated separately (Stepanhoff, 1957).

The leakage flow has the stage pressure of the impeller when it passes through the impeller sidewall gap.

### 3.3.2 Disk Friction Losses

With a disk rotating in a fluid, shear stresses corresponding to the local friction coefficient occur on its surface which depends on the Reynolds Number and the surface roughness.

The disk friction power increases with the third power of the speed and the fifth power of the diameter, assuming geometrical similarity and the ratio of the outside diameter of the rotating component and the length are constant. (Gülich, 2008)<sup>7</sup>

Numerous tests have confirmed that the change in axial casing clearance has only a very small effect on the disk friction.

With the Reynolds Number increasing, the friction coefficient drops, while an increase in roughness of the rotating or stationary components increases the friction power which is evident in Figure 6 and 7.

According to the measurements in (Geis, 1985) the disk friction of a machined disk with surface roughness  $\epsilon(\max) = 120 \mu\text{m}$  was practically identical to that of a polished disk with  $\epsilon = 0$ .

All the rotating surfaces need to be considered when determining the disk friction losses in a pump. In radial pump impellers the losses due to the friction on the shrouds by far exceed the contribution of all other components (Gülich, 2008)<sup>8</sup>.

Equation 2 is used to calculate the disk friction power loss (Gülich, 2008)<sup>9</sup> for radial impellers with Reynolds Number greater than 100,000. It is shown that the leakage through the annular seals on the power loss is influenced by the factor,  $f_L$  (influence of leakage flow on disk friction) which is negligible with the changes in annular seal clearances.

$$\frac{P_{RR}}{P_{u,opt}} = \frac{770 f_{R,L} a f_L}{n_q^2 \psi_{opt}^{2.5} Re^{0.2} f_q} \quad (2)$$

Through almost the entire range of different specific speed ( $n_q$ ) pumps, leakage loss is approximately equal to one half of the disk friction loss (Stepanhoff, 1957).

For pumps of the same specific speed but of different size and speed, both the pump output and power consumed by the disk friction are proportional to  $n^3 d^5$  and their ratios remain the same. This consideration disregards the effect of the

<sup>7</sup> Section 3.6.1 (p 85)

<sup>8</sup> Section 3.6.1 (p 89)

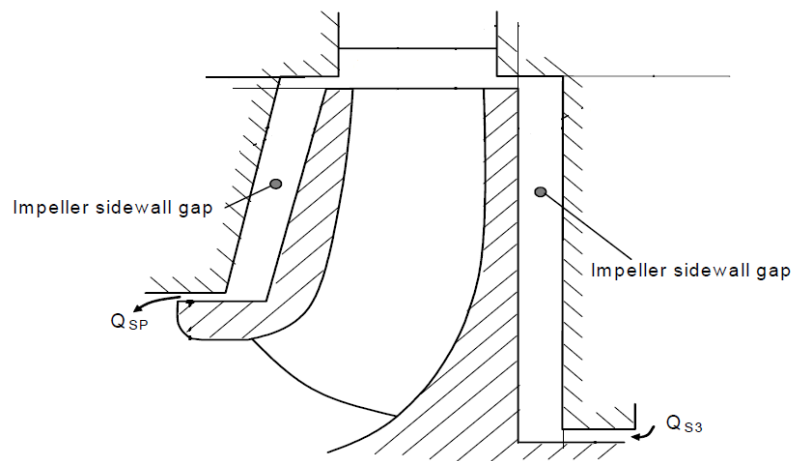
<sup>9</sup> Table 3.5 (p 135)

Reynolds Number on the value of the coefficient  $K$  and the hydraulic efficiency of the pump, as it affects the pumps output (Stepanhoff, 1957)<sup>10</sup>.

### 3.3.3 Leakage Losses through Annular Seals

A close-running clearance between the impeller and the casing limits the impeller leakage. This leakage through the annular seals from the discharge of the impeller to the inlet reduces the pump efficiency. The entire mechanical energy transferred by the impeller to the leakage flow is throttled in the seal and converted into heat. In the same way, through the axial thrust balance device the entire discharge pressure of the pump is throttled back to the suction pressure. (Gulich, 2008)<sup>11</sup>

In figure 3 the leakage through the annular seals can be seen, on the front shroud as  $Q_{SP}$  and the rear shroud  $Q_{S3}$



**Figure 3: Annular seal leakage loss around an impeller (Gulich, 2008)<sup>12</sup>**

Leakage loss is a loss of capacity through the running clearances, between the rotating element and stationary casing parts (Stepanhoff, 1957).

Leakage can take place in one or several of the following places, depending on the type of pump:

1. Between the casing and the impeller at the impeller eye
2. Between two adjacent stages in multi-stage pumps
3. Through the stuffing box
4. Through axial thrust balancing devices

<sup>10</sup> Chapter 10 (p 190)

<sup>11</sup> Section 3.6.2 (p 90)

<sup>12</sup> Table 0.2 (page XXXIII)

5. Through bleed-off bushings when used to reduce the pressure in the stuffing box
6. Past the vanes in open impeller pumps
7. At any bleed-off used for bearing and stuffing box cooling

Figure 4 details the minimum allowable diametric clearance between rotating and stationary components.

Diameter of rotating member at clearance [mm]	Minimum diametric clearance [mm]
50	0.25
50 to 64,99	0.28
65 to 79,99	0.3
80 to 89,99	0.33
90 to 99,99	0.35
100 to 114,99	0.38
115 to 124,99	0.4
125 to 149,99	0.43
150 to 174,99	0.45
175 to 199,99	0.48
200 to 224,99	0.5
225 to 249,99	0.53
250 to 274,99	0.55
275 to 299,99	0.58
300 to 324,99	0.6
325 to 349,99	0.63
350 to 374,99	0.65

**Figure 4: Minimum running clearances (API 610, 2004)**

The seal clearance is of importance since the leakage flow increases approximately to the power of one and a half times the clearance. Seemingly minor deviations from the assumed dimensions have a noticeable effect on the efficiency at low specific speeds. In the case of a pump with a  $n_q = 15$ , for example, a radial clearance change by 0.03mm from 0.25mm to 0.28mm, This means a reduction of efficiency of one percentage point. (Gülich, 2008)<sup>13</sup>

The annular seal consists of a casing ring (wear ring) and a rotating inner cylinder (impeller), with the clearance being small compared to the diameter at which the seal is located. Due to the pressure difference across the seal an axial flow is generated which is what accounts for the leakage losses.

Due to the rotation of the impeller or the axial thrust balancing device, a circumferential flow is superimposed on the axial flow. As a result a Reynolds Number for both the axial and circumferential flow needs to be considered.

Annular seals are designed in a variety of shapes as shown in Figure 5, depending on the considerations of the design, with the main objective of

<sup>13</sup> Section 3.6.2 (p 97)

preventing contact between the rotor and stator. Shaft bending as a result of the rotor weight and radial thrust, possible thermal deformations of the rotor and casing, and the tendency of the materials seizing need to be considered when designing the shape of the seal.

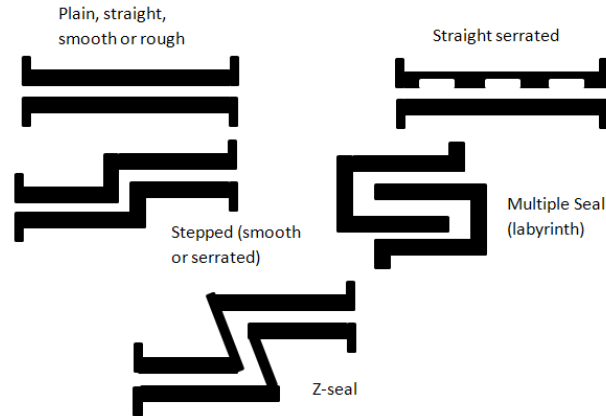


Figure 5: Types of annular seals (Gülich, 2008)

### 3.3.3.1 Turbulent Flow in Annular Seals

Experimentally determined friction coefficients are used for turbulent flow when calculating the annular seal leakages. Figure 6 below shows measurements taken on a straight, plain and smooth annular seal.

Figure 6 express that the circumferential speed and velocity through the annular seal influences the friction coefficient.

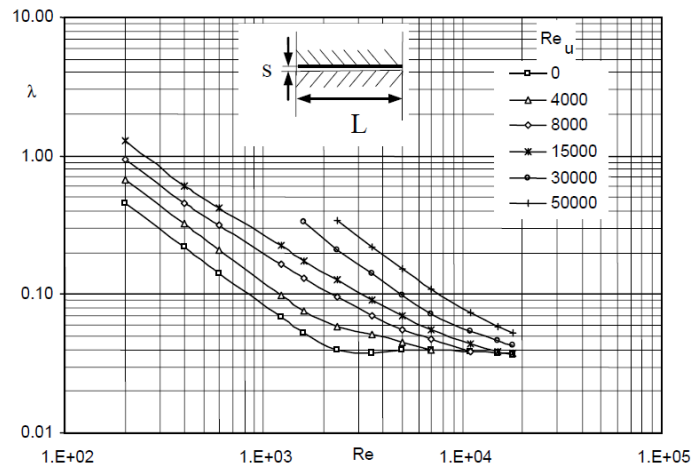
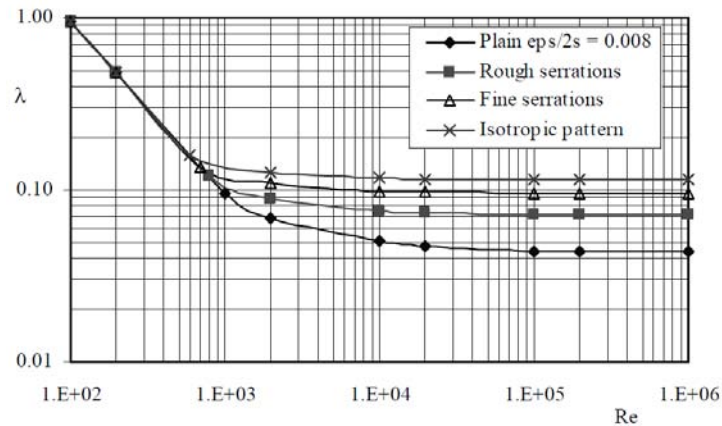


Figure 6: Friction coefficients of smooth annular seals, experimental test data,  $\epsilon/s = 0.01$ ;  $L/s = 413$ ,  $s = 0.315\text{mm}$  (Gülich, 2008)<sup>14</sup>

<sup>14</sup> Section 3.6.2 (p 91)

Depending on whether the seals are smooth, coarse, have fine serrations or have isotropic patterns at high Reynolds Numbers, the friction coefficients asymptote to different values as can be seen in Figure 7.



**Figure 7: Friction coefficients of annular seals for  $Re_u/Re = 2$  (Gülich, 2008)<sup>15</sup>**

The roughness always needs to be considered when calculating the leakage losses, provided the flow is turbulent as demonstrated in Figure 7.

### 3.3.3.2 Inlet and Outlet Losses

The inlet and outlet losses are considered as they determine the loss coefficients in the annular seals. Rotation and Reynolds Numbers both influence the loss coefficients and this makes it difficult to predict the behaviour of the fluid at this point.

A chamfer at the seal inlet can also have a major influence on the inlet coefficient.

### 3.3.3.3 Friction Coefficient in Seals

The clearance has practically no effect on the friction coefficient, provided the relative roughness  $\frac{\epsilon}{s}$  is used in the calculation. For  $Re > 10,000$  the roughness must always be considered for leakage flows since the actual ratio  $\frac{\epsilon}{s}$  is relatively large (Gülich, 2008)<sup>16</sup>.

For seals without serrations, the greatest possible roughness should be aimed for to minimise the leakage flow. There are practical limits to this as the manufacturing tolerances required for the seal diameters cannot be achieved with a very high roughness.

<sup>15</sup> Section 3.6.2 (p 93)

<sup>16</sup> Section 3.6.2 (p 97)

### 3.3.3.4 Effect of Rotation on Leakage – Rotation Factor

The greater the fluid rotation (i.e. the greater  $k$ ), the greater the pressure drop in the sidewall gap. This results in a smaller pressure difference over the seal and resulting leakage flow as described in the Equation 3.

$$\Delta P = \frac{1}{2} \rho * k^2 * u_2^2 * (1 - d_{sp}^2) \quad (3)$$

Since  $k$  increases with growing leakage, a radially inward leakage depends on itself and partly limits itself (Gülich, 2008)<sup>17</sup>.

The rotation factor can be estimated (Gülich, 2008) according to Equation 4, which is a function of the Reynolds Number (circumferential component) and seal geometry.

For  $Q_{sp}$  radially inwards  $k = 0.9 * y_{sp}^{0.087}$

For  $Q_{sp}$  radially outwards  $k = 0.24 * y_{sp}^{-0.096}$

$$y_{sp} = Re_{u_2}^{0.3} \frac{s * d_{sp}}{d_2^2} * \sqrt{\frac{s}{L_{sp}}} \quad (4)$$

### 3.3.4 Mechanical Losses

The mechanical losses are a result of the radial bearings, the axial bearing and the shaft seals and can sometimes include auxiliary equipment driven by the pump shaft. The mechanical efficiency of larger pumps is around 99.5% or more, and as a result, during the operation of a BFP under normal conditions the mechanical losses can be considered to be constant over the BFP life (Gülich, 2008)<sup>18</sup>.

<sup>17</sup> Section 9.1 (p 499)

<sup>18</sup> Section 3.6.6 (p 101)

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### 3.4 WEAR RING

The purpose of wear rings is to provide a renewable surface between the stationary boundary of the annular seal between the front and back hubs of the impellers. In high speed boiler feed pumps a wear ring is not mounted on the impellers, therefore the hubs themselves are the rotating boundary of the annulus. The radial clearance of this annulus is the difference between the impeller hub radius and the wear ring radius.

The wear ring surface is going to wear and during a refurbishment it is replaced with new parts in almost every case (Inspectors, 2012). Normally the bore of the replacement wear ring is undersize on the bore so that the wear surfaces on the impeller hubs can be machined accordingly and the new wear ring machined to establish the desired clearance.

The wearing surface of the wear ring and the impeller hubs usually have configurations such as both flat or one flat and the other one grooved. Older BFPs designed 30 years ago had flat surfaces, followed by a period when it was thought that grooved rings would yield less leakage and be more efficient, and be more wear and gall-resistant and less likely to seize. Some grooves were even machined in a spiral to reduce the amount of leakage through the annulus. Now that the rotordynamic characteristics of BFPs are able to be modelled relatively easily with present computer codes, the flat rings and hubs are becoming more popular because of their high dampening coefficients needed for stable rotordynamics (EPRI, 1994)<sup>19</sup>.

Grooved rings are usually either V-grooved, like a standard screw thread, or grooved with a flat land between the rotating and stationary surfaces. In many cases, the grooving was shaped as a labyrinth tooth to reduce the leakage rate.

The wear is always on the inner surface of the wear ring, where it has come into contact with the hub of the impeller. The wear can be any of the following three types or a combination of all three (EPRI, 1994)<sup>20</sup>.

1. Scoring of the wear ring and impeller hub surface by foreign particles entering the annulus. This usually occurs during initial plant start-up or after an outage when the boiler is brought up from a cold shutdown.
2. Friction rubbing of the wear ring and/or the impeller hub.
3. Galling of the wear ring and/or impeller hub.

When the clearances in the annular seals have worn to one and a half to two times the original clearance the wear rings are considered “worn” and will need to be replaced.<sup>21</sup>

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<sup>19</sup> Section 2.10 (p 2-139)

<sup>20</sup> Section 2.10 (p 2-140)

<sup>21</sup> Recommendation in OEM Manuals

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### 3.4.1 Materials used in Wear Rings

In order to ensure reliable operation of the BFPs due to the close running and high speeds the correct materials need to be specified with the correct heat treatment processes.

BFP wear rings are almost always a hardened 13% chrome nickel stainless steel material to provide corrosion and wear resistance. The wear ring is usually harder than the impeller hub by at least 50 BHN. To prevent galling and seizure, latest experiences show that 100 BHN is necessary (EPRI, 1994)<sup>22</sup>.

A hardness difference of 100 BHN is recommended when materials of BHN < 450 are used (Gülich, 2008)<sup>23</sup>.

Typical material used for wear rings in Eskom boiler feed pumps are steel grades DIN 1.4059 or DIN 1.4057<sup>24</sup> which are heat-treated to between 350 and 380 BHN<sup>25</sup> to ensure the recommended hardness differences.

### 3.5 HYDRAULIC PERFORMANCE TESTING

A performance test for the centrifugal pump is conducted according to ISO 9906, which is the international standard to test its operating parameters to either class 1, 2 or 3. The classes vary due to tolerance factors, fluctuations and uncertainties of measurement.

Performance testing can either be carried out on a test bed, under controlled conditions or on-site where the pump is installed.

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<sup>22</sup> Section 2.10 (p 2-139)

<sup>23</sup> Section 14.4.4 (p 831)

<sup>24</sup> Information obtained from Sulzer Pumps inspection reports

<sup>25</sup> Hardness values obtained from heat treatment certificates

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## 4. ESKOM BOILER FEED PUMPS

The BFPs are either driven by a steam turbine or an electric motor. Typically the “older” power stations have a 100% SFP<sup>26</sup> which operates continuously to maintain the unit load and two, 50% EFPs<sup>27</sup> which are on standby. The “newer” power stations typically have three 50% EFPs which operate two EFPs continuously, and a third EFP which is on standby and will be used in the case of the unavailability of one of the other EFPs.

### 4.1 PARTNERING AGREEMENT BETWEEN SULZER AND ESKOM

During the late 1970s, failures in boiler feed pump systems resulted in significant unavailability losses to the utility industry worldwide. Eskom also suffered from these losses and as a result of these effects on plant performance, Eskom Generation and Sulzer Pumps entered into a partnering agreement in 1993 with the goal to jointly improve performance (Cadle, Catteart, & van der Westhuizen, 2008).

Since the start of this service alliance, this partnering agreement has improved the pumps included in the contract to availability above 99.9%.

The partnering agreement with Sulzer Pumps have increased the number of pumps to 850 pumps of which there are 58 SFPs and 176 EFPs<sup>28</sup>

### 4.2 ELECTRICITY PRICING

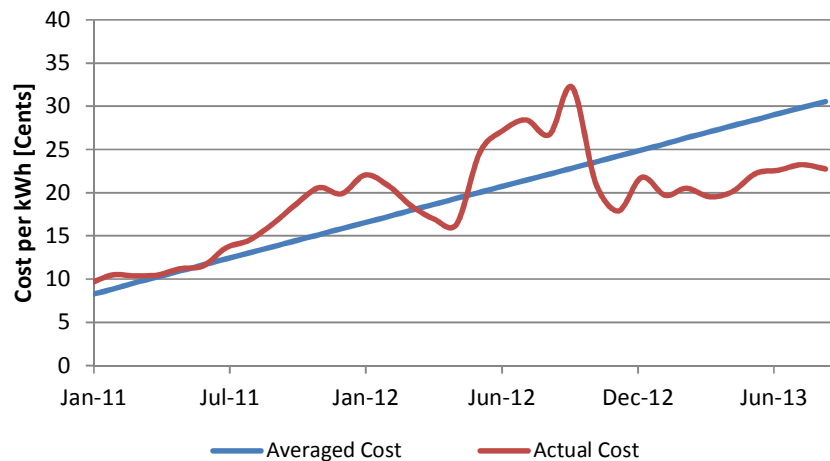
Eskom uses an average generating cost for electricity across all the power stations in the Eskom fleet; these values are detailed in Figure 8.

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<sup>26</sup> Refers to 100 % of flow required by the boiler

<sup>27</sup> Refers to 50 % of flow required by the boiler

<sup>28</sup> Data from the Eskom/Sulzer partnering agreement schedule of installed equipment



**Figure 8: Eskom generation cost to produce electricity (Eskom Power Price, 2013)**

The excess energy used as a result of inefficiencies at a power station is difficult to quantify, but with the shortage of current electricity supply in South Africa, using a generation cost is a somewhat conservative approach. The costs of running the peaking power stations, such as the open cycle gas turbines, are in excess of 250 cents per kWh which would in certain periods increase the costs as a result of inefficiencies in power stations astronomically from the averaged generation cost of 29 cents per kWh.

When Eskom have an excess capacity of power, then the savings are in the costs to produce the power. When there is a shortage of supply the cost would be in the savings in peaking power stations used.

#### 4.3 DATA CAPTURE

All the power stations have records of BFP refurbishments that were done by various pump repairers which were located and populated with the actual pump running hours at the time the pumps were taken out of service. The measurements on the annular seals and the balancing device were the basis for the analysis of the BFPs internal wear rates.

When a BFP is taken out of its barrel casing at the site and sent for repairs, the dimensions of the BFP are measured to determine the repair work needed. The inspector (from the pump repairers) inspects all the components and makes an assessment of what can be re-used or replaced and the repair method to be used.

The axial thrust balance device recorded clearances on some of the sites were limited as a result of records not documented correctly.

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The power stations that have been analysed had the majority of the information recorded as a result of the site managers keeping well-documented records of running hours and changes in clearances.

#### **4.4 BFP REFURBISHMENT COSTING**

To determine the refurbishment costs of a BFP, previous refurbishments were analysed by Sulzer Pumps Customer Support Services and brought to present day value.

What is generally considered in a BFP refurbishment (Wolfgang Miller, 2013) are the following:

- Site installation
- Gaskets
- Partnering reconditioned pull-out
- Workshop repair pull-out
- Partnering reconditioned NDE cover
- Workshop repair NDE cover
- Free issue spares
- Shoulder rings to impellers
- Ring to thrust collar
- Tension ring to thrust collar
- Locking washer to thrust collar
- Joint ring
- Ring to balancing piston
- Lock nut to balancing piston
- Tension ring
- Drive End shaft sleeve
- Spacer ring to shaft sleeve
- NDE shaft sleeve
- Balancing piston
- Lock bush to balancing disc
- Drive end locking washer to shaft nut
- Throttling bush
- Wear rings and consumables

#### 4.5 WATER CHEMISTRY

Eskom power stations use demineralised water in the steam/water cycle; since 1988 they have used oxygenated treatment (OT). A guideline for the OT chemistry is given in Table 1.

None of the power stations analysed in this study operate with AVT(R) or AVT(O).

**Table 1: Economiser Inlet Chemistry Guideline Value for OT Chemistry (Electric Power Research Institute, 2005)<sup>29</sup>**

pH at 25°C	8.0-8.5* 9.0-9.4**
Ammonia (ppm)	0.02-0.07* 0.28-1.1**
Specific Conductivity (µS/cm) at 25°C	0.25 – 1* 2.7 – 6.7**
Cation conductivity (µS/cm) at 25°C	< 0.15
Iron (ppb)	< 2
Oxygen (ppb)	30-150* 30-50**

\* - once-through boilers

\*\* - drum boilers

The content of the medium being pumped is important when analysing the effect it will have on the internal wear of a BFP.

For this study, the effect that the demineralised water has on the BFPs annular seals should be evident in the wear of the annular seals.

From inspection reports that are done during a refurbishment minimal evidence of abrasive wear is present due to the feed water pumped.

<sup>29</sup> Table 2-4 (p 2-7)

## 5. ANNULAR SEAL LEAKAGE LOSS – MODELLING

Considering the annular seals in a typical SFP with the pressure difference across the seal using Bernoulli's equation, this will be used to determine how the change in area affects the leakage rate.

Considering a typical 4-stage SFP design with the following parameters:

Total Head: 2000m

Head per stage: 500m (4 stages)

Front Shroud Diameter of Impeller at Seal: 250mm

Rear Shroud Diameter of Impeller at Seal: 200mm

Diametric Clearance: 0.6mm (new) and 1mm (worn)

Front Shroud - Inner Diameter of Wear Ring: 250.6mm (new) and 251(worn)

Rear Shroud - Inner Diameter of Wear Ring: 200.6mm (new) and 201(worn)

$$a = \frac{\pi d_{outer}^2}{4} - \frac{\pi d_{inner}^2}{4} \quad (5)$$

Area (new) = 235.9 mm<sup>2</sup>

Area (worn) = 393.5 mm<sup>2</sup>

Assuming that the pressure drop across the annular seal on the front shroud of the impellers would be 0.75 (degree of reaction) of head per stage and 0.25 of head per stage on the rear shroud. The pressure drop across the balancing device would be the total head generated by the SFP.

It will be assumed that the inlet pressure and the outlet pressure are made up completely of static components and the pressure drop is dissipated through the seal.

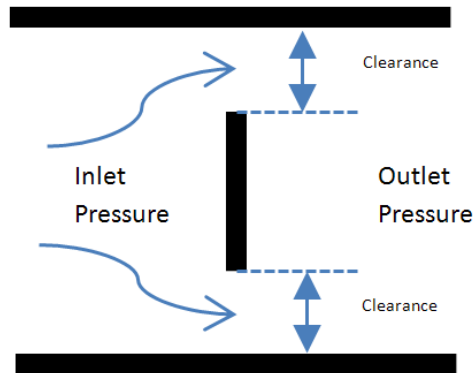
According to Bernoulli's equation for ideal flow the velocity through the annular seal as seen in Figure 9 would be:

$$v = \sqrt{2gh} \quad (6)$$

To obtain the flow rate through the annular seal Equation 7 below would be used:

$$Q = v x a \quad (7)$$

Using Equation 6 it assumes that the discharge coefficient, seal surface, length of seal, pre-rotation of fluid, rotating components and friction coefficients will be constant and the change in clearance would be the only influencing factor.



**Figure 9: Pressure drop across a plate**

The clearance from new to worn diametric clearance is assumed to happen linearly in 40,000 running hours (approximately 5 years) of operation at which point it would be refurbished.

The generation cost of electricity for Eskom is averaged at R0.29 per kWh and over its design life it would consume an additional 15.2 million kWh (based on 5 years for 1 SFP) as a result of the change in internal clearances.

**Table 2: Life cycle cost breakdown for the increased clearance of the annular seals for a typical SFP**

Cost Breakdown	
Total Cost for Balancing Device	R 3,200,084
Total Cost for Impellers	R 1,199,389
<b>Total Cost</b>	<b>R 4,399,473</b>

This seems to be a very simplistic way of looking at the influence the wear has on the flow rate, but with all the parameters remaining constant in theory it is the change from new too worn which is of concern and not the actual starting point.

The flow field inside whirling and non-whirling seals can be dramatically different (Gerald L. Morrison, 2000) which will be shown how to incorporate it in section 5.1 (p 20) with refined techniques.

## 5.1 MODELLING THEORIES

Bulk-flow theory is based on an analogy between turbulent flow as a result of a pressure gradient and flow as a result of a sliding surface (Hirs, 1970).

It is shown that the wall shear stress depends on two dimensionless groups, friction factor and Reynolds Number. The actual rate of flow in a lubricated film with respect to each surface can be treated as two superimposed components (Hirs, 1970).

Equation 8 (Y. Hsu, 2002) represents the friction factor on the left and the Reynolds Number on the right of the equation.

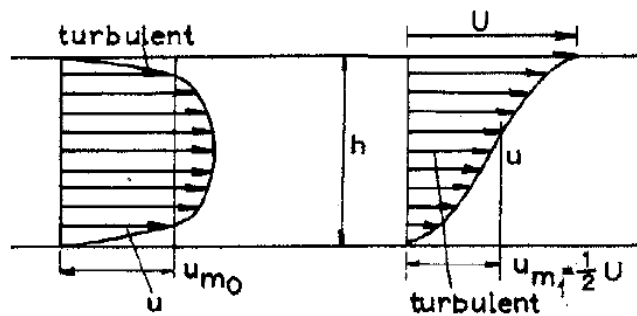
$$\frac{\tau}{\frac{1}{2}\rho u_m^2} = n \left( \frac{\rho u_m h}{\mu} \right)^m \quad (8)$$

Equation 9 below shows that the mean flow velocity is relative to the cube root of the film thickness.

$$u_m \sim \sqrt[2+m]{\frac{1}{h}} \quad (9)$$

This illustrates that with increased clearance between stationary and rotating parts the velocity through the seal will increase.

Figure 10 on the left shows the turbulent flow between two surfaces under the influence of a pressure gradient and the turbulent flow on the right due to sliding forces.



**Figure 10: Turbulent flow between two surfaces as a result of pressure gradient and sliding surfaces (Hirs, 1970)**

In the turbulent flow region the amount of flow does not depend on the shaft speed, but the friction is affected in the laminar region and will be found to increase as the speed increases. According to (Schneckenberg, 1932) the maximum eccentricity may increase the amount of leakage to 2.5 times the amount for a concentric annular channel. In the turbulent region the amount of flow increases due to the eccentricity of the annular seal.

### 5.1.1 Centrifugal Pumps – Dr Johann Gülich

The hydraulic design of the Sulzer Pumps BFPs hydraulics were based principally on research done over many years by Sulzer Pump with certain equations published in (Gülich, 2008). These equations together with the empirical data from Sulzer Pumps were used to determine the annular seal leakage flow rate in the various power stations.

The pressure difference acting over the seal is estimated accurately enough by using a degree of reaction ( $R_G$ ) of 0.75 for specific speeds ( $n_q$ ) lower than 40. A rotation factor ( $k$ ), which is the angular velocity of the water, is applied to the drop in pressure between the impeller outlet and the seal as a result of the rotation of the fluid.

Using the equations below the leakage through the annular seals can be calculated.

The static pressure rise created by the impeller

$$H_p = R_G H \quad (10)$$

Pressure difference across seal if leakage flow is radially inwards

$$\Delta H_{sp} = H_p - k^2 \frac{u_2^2}{2g} \left\{ 1 - \frac{d_{sp}^2}{d_2^2} \right\} \quad (11)$$

Pressure difference across seal if leakage flow is radially outwards

$$\Delta H_{s3} = H - H_p + k^2 \frac{u_2^2}{2g} \left\{ 1 - \frac{d_{sp}^2}{d_2^2} \right\} \quad (12)$$

Pressure difference across axial thrust balance device

$$\Delta H_{EK} = (z_{ST} - 1)H + H_p + k^2 \frac{u_2^2}{2g} \left\{ 1 - \frac{d_{EK}^2}{d_2^2} \right\} \quad (13)$$

Axial velocity for each annular seal can be calculated for each of the following  $\Delta H_{sp}, \Delta H_{s3}, \Delta H_{EK}$  by substituting in to  $\Delta H$  below,  $i$  = number of chambers

$$c_{ax} = \sqrt{\frac{2g\Delta H}{\zeta_{EA} + \lambda \frac{L_{sp}}{2s} + \sum_i \left( \frac{d_{sp}}{d_{si}} \right)^2 \left( \frac{s}{s_i} \right)^2 \left\{ \zeta_K + \lambda_i \frac{L_i}{2s_i} \right\}}} \quad (14)$$

## Loss coefficients

 $\zeta_{EA} = 1 \text{ to } 1.1 \text{ inlet + outlet losses}$ 
 $\zeta_K = 1 \text{ to } 1.3 \text{ loss per chamber}$ 

Influence of rotation using a friction coefficient for turbulent flow

$$\lambda = \frac{0.31}{\left\{ \log\left(A + \frac{6.5}{Re}\right) \right\}^2} * \left\{ 1 + 0.19 \left( \frac{Re_u}{Re} \right)^2 \right\}^{0.375} \quad (15)$$

## Reynolds Numbers

$$Re = \frac{2 s c_{ax}}{\nu} \quad u_{sp} = \frac{\pi d_{sp} n}{60} \quad Re_u = \frac{2 s u_{sp}}{\nu} \quad (16)$$

## Leakage flow rate

$$Q_{sp} = \pi d_{sp} s c_{ax} \quad (17)$$

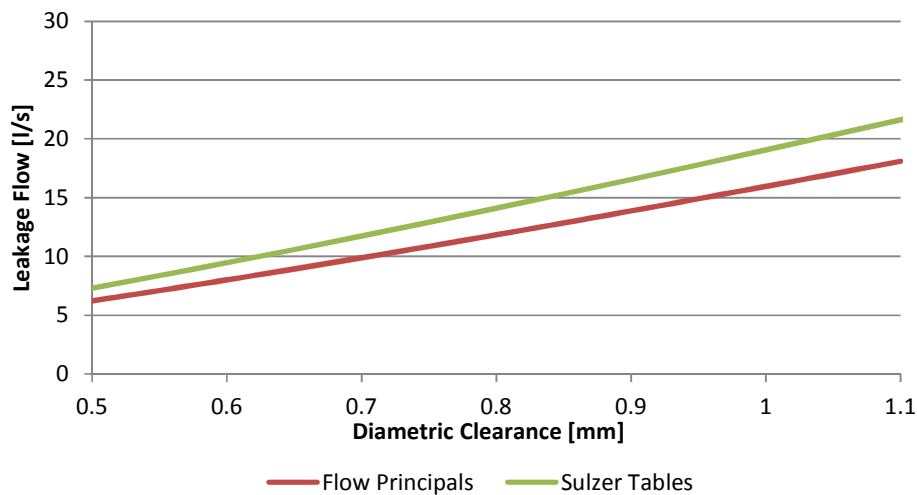
### 5.1.2 Sulzer Tables

The design tables used by Sulzer Pumps (Switzerland) when determining the hydraulics of a pump are proprietary information with some of the calculations published in the Centrifugal Pumps book (Gülich, 2008). The empirical factors from test data, which were developed experimentally, are used to interpret values more accurately.

For the equations used in section 5.1.1 (p 21) to determine the annular seal leakage losses for inline pumps the deviation in predicted leakage losses is a result of the value A, and the constants 0.19 and 0.31 in Equation 15.

The loss factors depend on the geometry of the annular seal and the seal surface roughness. The values used in Equation 15 are a generic empirical equation not considering the specific annular seal design.

Figures 11 to 15 show the comparison between the Centrifugal Pumps book (Gülich, 2008) and the Sulzer Pumps tables for Lethabo Power Station SFP where specific values for the constants of the specific pump have been included.



**Figure 11: Lethabo SFP suction impeller – Front shroud comparison**

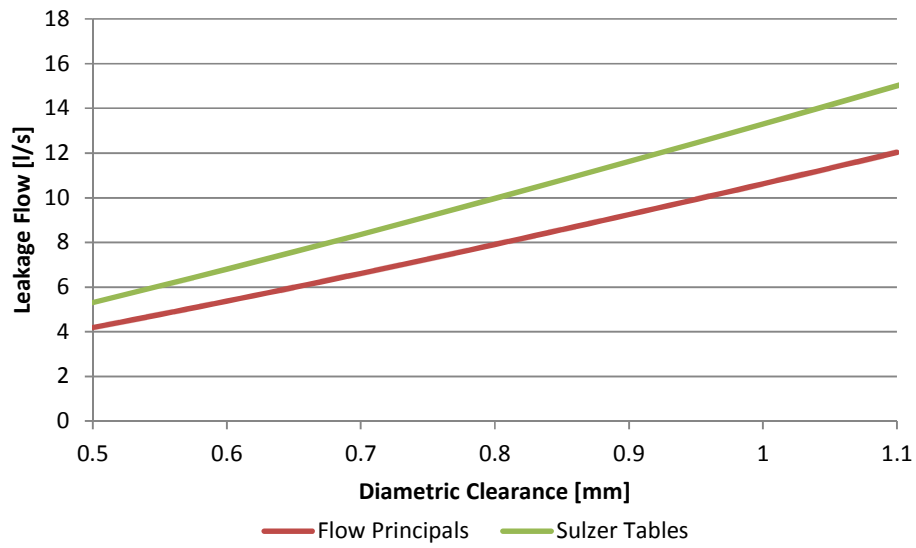


Figure 12: Lethabo SFP suction impeller – Rear shroud comparison

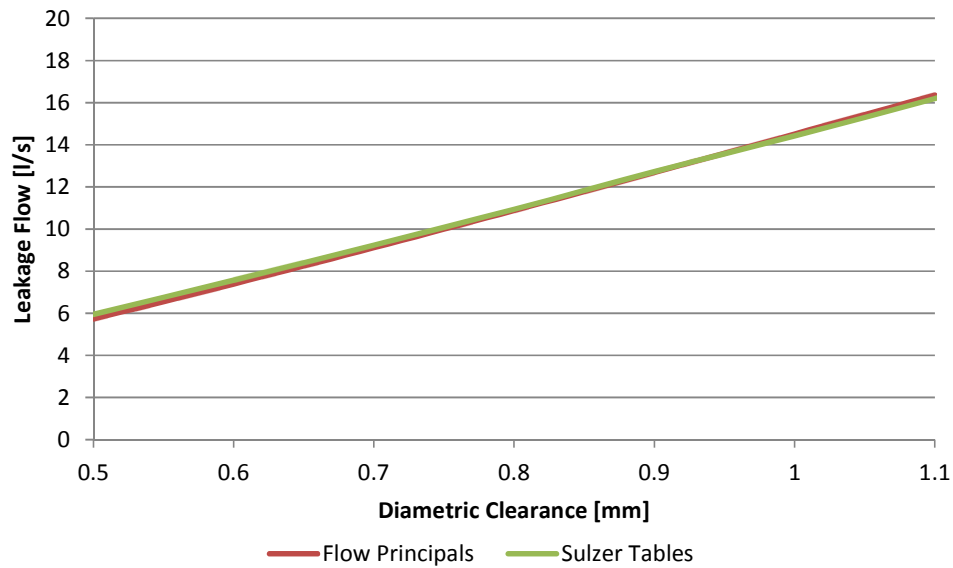
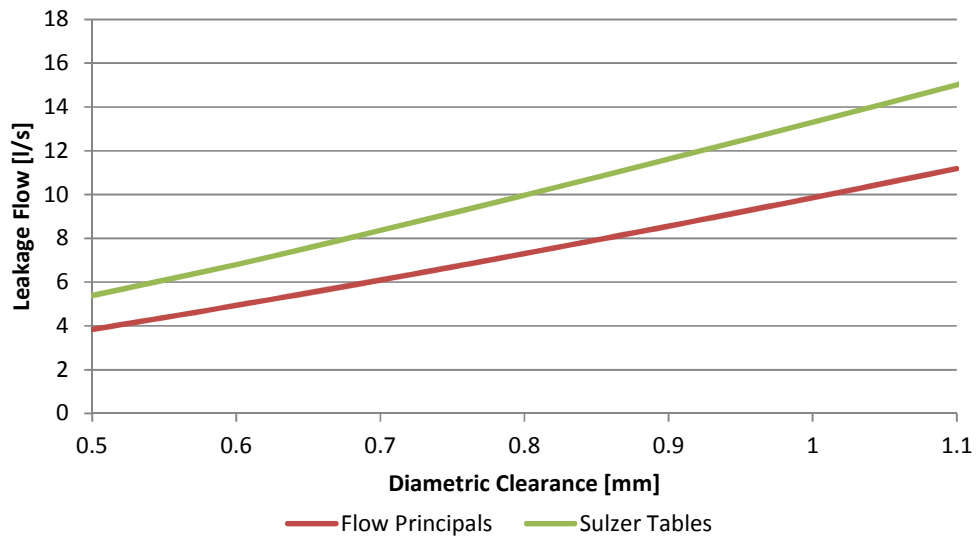
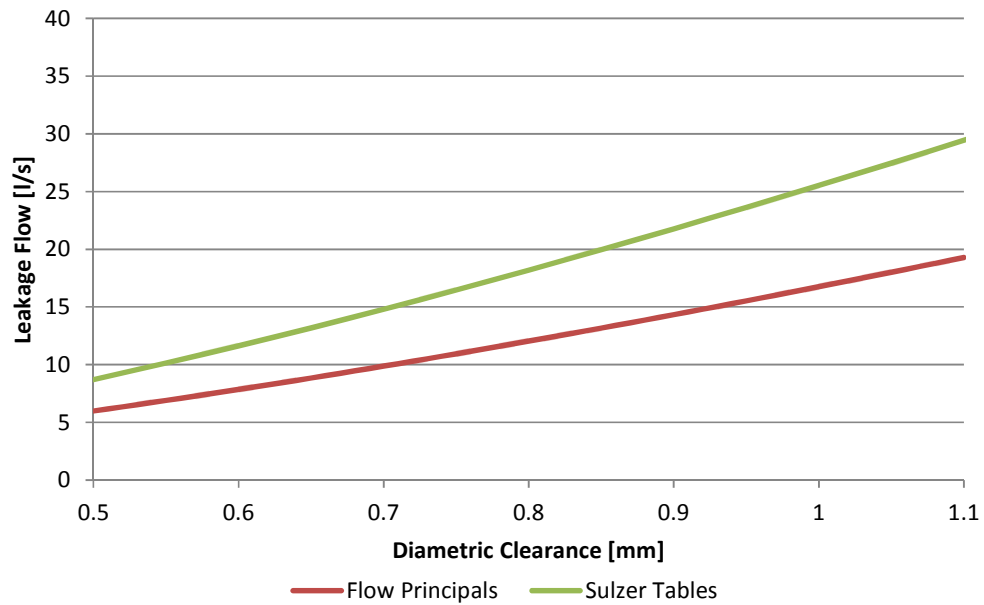


Figure 13: Lethabo SFP 2<sup>nd</sup> to last stage impeller – Front shroud comparison



**Figure 14: Lethabo SFP 2<sup>nd</sup> to last stage Impeller – Rear Shroud Comparison**

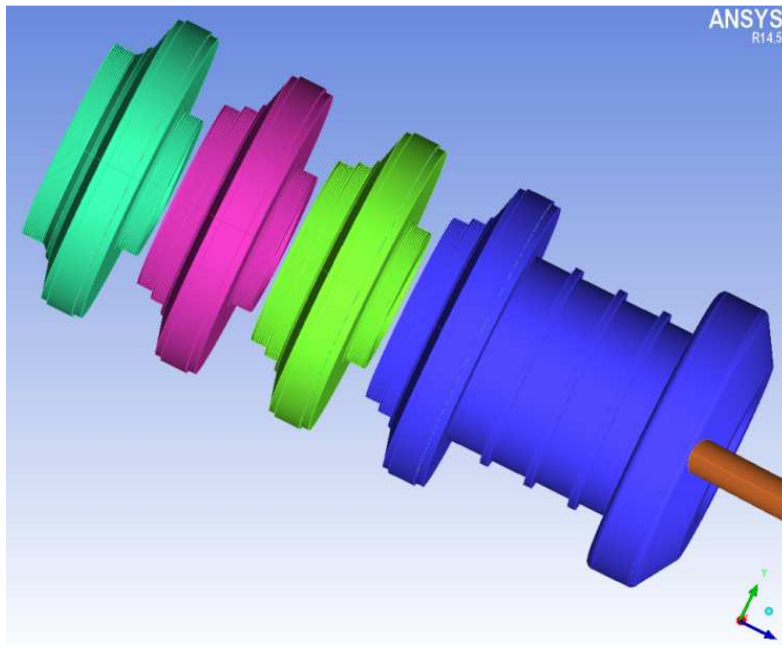


**Figure 15: Lethabo SFP axial thrust balance device comparison**

### 5.1.3 ANSYS Simulation for Annular Seal Leakage

To determine the accuracy of the annular seal leakage prediction using the laws of fluid flow in (Gülich, 2008) ANSYS 14.5 will be used<sup>30</sup>. The geometry was created for the impeller sidewall gaps and axial thrust balance device for Lethabo SFP.

Figure 16 depicts the geometry created<sup>31</sup> for the 4 stages and the axial thrust balance device.



**Figure 16: Lethabo SFP geometry used for modelling leakage in annular seals**

The geometry was created using the SFP design drawings, which were then meshed with ICEM CFD 14.5<sup>32</sup>.

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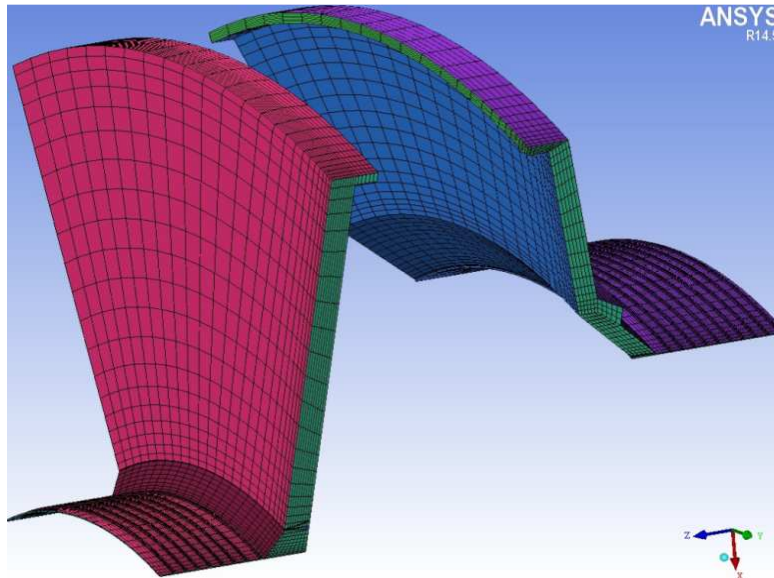
<sup>30</sup> Sulzer Pumps (Switzerland) had a collaboration with Eskom to validate the results of the research

<sup>31</sup> Geometry was created by Sulzer Pumps Intern using the Lethabo SFP design drawings

<sup>32</sup> Meshing was done together with Dr Arnaldo Rodrigues (PD Hydraulics Manager – Sulzer Pumps, Switzerland)

Figure 17 depicts a picture of the mesh created on a 45-degree section of the suction impeller for the Lethabo SFP. Various methods could have been used to create the mesh, but it was determined that the best method would be by using a 45 degree section.

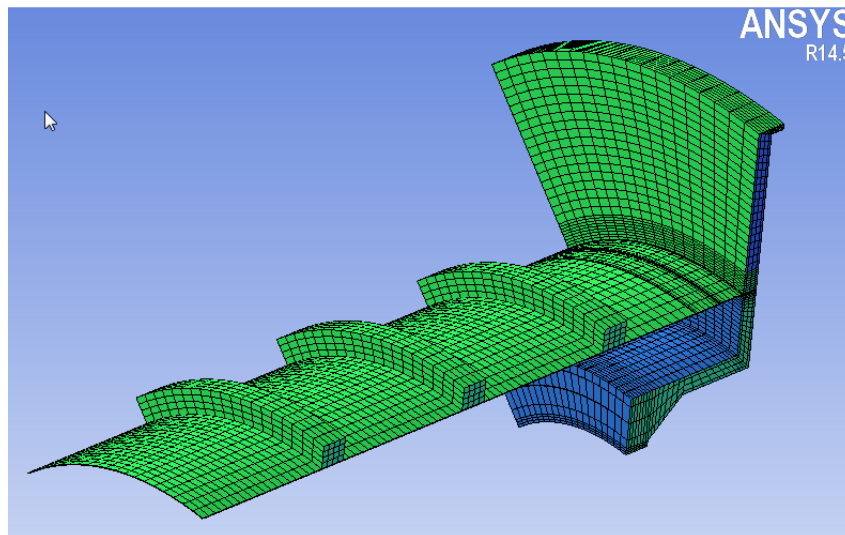
The 45 degree section would then be rotated when the mesh was finalised to give the complete impeller sidewall gaps between the rotating and stationary components.



**Figure 17: Lethabo SFP mesh on suction impeller (45 degree section)**

Figure 18 shows the mesh created on a 45 degree section of the Lethabo SFP axial thrust balance device.

The 45 degree which is the gap between the balance piston and throttling bush would be rotated once the meshing was successful.



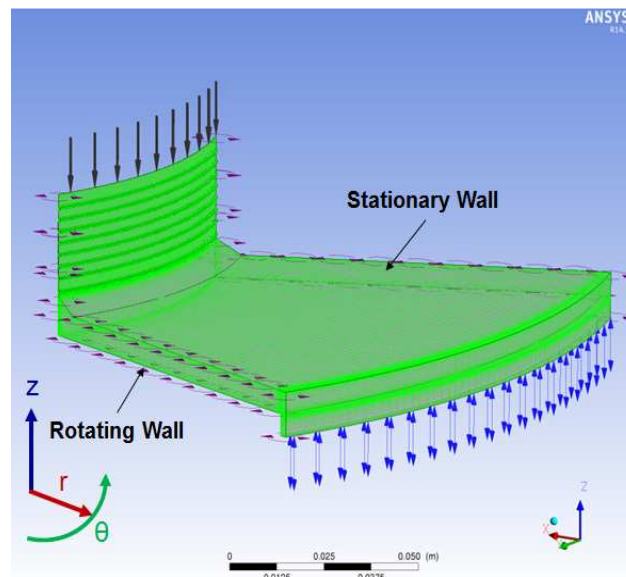
**Figure 18: Lethabo SFP mesh on axial thrust balance device annular seal (45 degree section)**

With the mesh complete and rotated to form the complete geometry it was run with fluid dynamics CFX 14.5 to determine the fluid flow with changing clearances in the annular seals using the setup described in Table 3.

**Table 3: ANSYS CFX Setup<sup>33</sup>**

<b>Simulation type</b>	RANS, Single-phase
<b>Advection schemes</b>	<b>Momentum:</b> High resolution <b>Turbulence:</b> High resolution
<b>Turbulence model</b>	k- $\omega$ SST
<b>Boundary conditions</b>	<b>Walls:</b> non-slip wall conditions, automatic near wall treatment, wall angular velocity defined for rotating wall <b>Run 1:</b> Coarse mesh with mass flow BC for the inflow and an average static pressure BC for the outflow. <b>Run 2:</b> Coarse mesh with total pressure BC for the inflow, and opening BC for the outflow. <b>Run 3:</b> Final run: Refined mesh with total pressure BC for the inflow, and opening BC for the outflow.
<b>Convergence Criteria</b>	RMS Residuals in the range of $10^{-5}$ , Imbalances < 1%, Convergence of mass flow rate
<b>Fluid</b>	Water @ 180°C, $\rho = 904.9 \text{ kg/m}^3$
<b>Machine Data</b>	$n = 5143 \text{ rpm}$

The boundary conditions are shown in Figure 19 using the setup described in Table 3



**Figure 19: CFX setup for suction impeller rear shroud**

<sup>33</sup> CFX setup configured by Laurent Chatagny (Sulzer Pumps, Switzerland)

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The radial clearances between rotating and stationary components were determined to be 0.25mm, 0.5mm and 0.7mm.

For each case, 3 simulations would be run using CFX to determine the leakage flow rate through the annular seals which could then be compared to the Sulzer Pumps design tables.

After some effort from Sulzer Pumps, Switzerland engineers the outcome was that a reliable solution could not be obtained since the investigations showed that the inlet and outlet were not sufficiently constrained by using pressure boundary conditions. The recommendation was to therefore carry out a coupled calculation.

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## 5.2 PUMP EFFICIENCY

From the losses described below (power to seals and bearings, disk friction, hydraulic and mechanical losses) it has been analysed that the power consumed by these losses would not influence the efficiency of the pump over its operating life.

In the case where they have had substantial damage the component/system would either be repaired or replaced and brought back to design specifications. The changes are therefore negligible as described for each secondary loss as a contribution to efficiency changes. In order to influence any of these losses the design of the component would need to be considered and done by the OEM (G. Ludwig, 2003).

### 5.2.1 Power to Seals and Bearings

The power to seals and bearing should remain fairly constant throughout the life of the pump as the bearings and seals are inspected<sup>34</sup> at different intervals confirming that they are operating according to design. In the case that a pump experiences problems on the bearings or seals they will be replaced and thus be brought back to design specification.

### 5.2.2 Disk Friction Losses

The disk friction losses can be assumed to be constant as the changes due to internal wear has been modelled<sup>35</sup> and the effects found to be minimal.

The Lethabo SFP axial thrust balance device was modelled to understand the effect the internal wear will have on the power consumed. What was found was that the power consumed from friction decreases with increased clearance by a negligible amount, being a change of 0.32 kW at worn clearances.

### 5.2.3 Hydraulic Losses

The hydraulic losses represent a major part of the losses in a BFP, which are created by a relative motion between the pump and the fluid. These forces are as a result of inertia and viscosity.

As a result of the leakage losses not affecting the friction forces it is assumed that they will be constant over the life of the BFP.

### 5.2.4 Mechanical Losses

The radial bearings, axial bearings and shaft seals contribute to the mechanical losses for large pumps which has a mechanical efficiency of approximately 99.5%.

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<sup>34</sup> Inspections are conducted by the power station maintenance and engineering department

<sup>35</sup> Modelled according to (Gülich, 2008)Table 3.6 (p 136)

A change in the mechanical losses will be negligible to the overall pump efficiency and can be assumed to be constant.

### 5.3 EFFICIENCY CHANGE FOR EACH POWER STATION

To determine the change of a BFPs performance it was assumed that the only factor affecting the efficiency was the wear in annular seals, and that the other contributory factors which are detailed in section 5.2 would remain constant.

Equation 18 (Gulich, 2008)<sup>36</sup> was used to determine the change in the BFP efficiency ( $\eta$ ) as the volumetric efficiency ( $\eta_v$ ) would change with internal wear.

$$\eta = \frac{P_u}{P} = \frac{\rho g H_{tot} Q}{P} = \frac{\eta_v \eta_h \rho g H_{tot} Q}{\rho g H_{tot} Q + \eta_v \eta_h (\sum_{st} P_{RR} + \sum P_{S3} + P_m + P_{er} + P_{Rec})} \quad (18)$$

Using the recorded data to predict the change in clearance for each power station the increased power consumption as a result of this change was modelled to show the change it would have on efficiency.

The new clearances for the annular seals as per design drawing were assumed as the power consumption at zero running hours or “new”. Using the trendlines from the wear data for each impeller and axial balance thrust balance device the increase in power consumption was taken for each interval.

The power consumed for each BFPs impeller and axial thrust balance device is detailed in the appendices in the section power consumed through annular seals.<sup>37</sup>

The BFPs were modelled according to their design points<sup>38</sup> and the additional power consumed once internal wear starts is detailed in the efficiency change for each power station.

<sup>36</sup> Table 3.5 (p 135)

<sup>37</sup> Lethabo (p 113), Kendal (p 119), Matla (p 130), Kriel (p 140) and Matimba (p 148)

<sup>38</sup> Lethabo (p 109), Kendal (p 116), Matla (p 123), Kriel (p 132) and Matimbe (p 146)

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## 6. ESKOM POWER STATIONS

### 6.1 LETHABO POWER STATION



#### 6.1.1 Overview

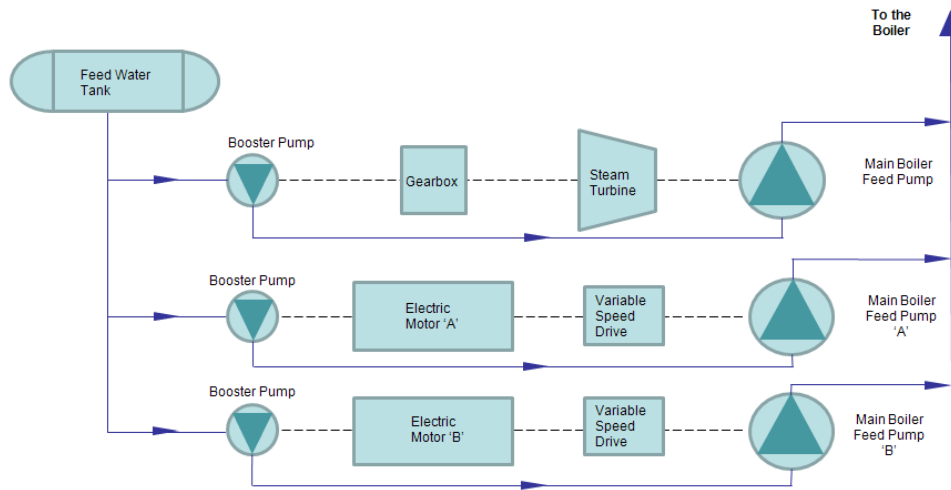
Lethabo Power Station is based in the Free State province, consisting of six 618 MW units with a total installed capacity of 3708 MW

The commissioning of the boiler feed pumps (BFPs) at Lethabo Power Station started in 1986 with the last unit being commissioned in December 1990 (Eskom Holdings Limited, c2014).

#### 6.1.2 Boiler Feed Water Pumps System Description

The feed water passes from the feed water tank via a suction strainer to the booster pump; with the purpose of the booster pump being to meet the necessary NPSH requirements of the main boiler feed pumps due to their high running speeds. The feed water is then pumped by the booster pump via an intermediate strainer before entering the main pump which is designed to meet the flow and pressure requirements of the boiler.

The unit configuration consists of two 50% electric-driven boiler feed pumps (EFPs) and one 100% steam-driven boiler feed pump (SFP). The unit can operate with either the 100% SFP or the two 50% EFPs to meet the full load of the unit. Lethabo uses the EFPs primarily for start-up, shut-down or when operating at lower loads and in the event that the SFP is not available. The SFP is used during continuous operation to maintain the load at each unit in the power station.



**Figure 20: Overview of the boiler feed pump plant at Lethabo Power Station**

**Table 4: Lethabo SFP pump type and designation**

OEM	Sulzer Pumps
Type	HPT Pk 350 – 405 - 4s/33

The steam-driven main boiler feed pump is directly coupled to and speed-regulated with the auxiliary turbine speed. The speed-regulated auxiliary turbine also drives a reduction gearbox (ratio 4:1) which drives the booster pump at a quarter of the speed of the main pump.

**Table 5: Lethabo EFP pump type and designation**

OEM	Sulzer Pumps
Type	HPT Pk 250 – 305 – 5s/33

There are two electric-driven boiler feed pump trains. The main electric-driven boiler feed pump is driven by electric motors via a variable-speed turbo gear, with the booster pump directly coupled to the motor at its running speed.

### 6.1.3 Power Station Operating Conditions

#### 6.1.3.1 Operation of Boiler Feed Pumps

The operation of the power station during commissioning and at a time when the country had excess capacity, the BFPs would have seen more starts and stops. Over the period from 1995 to 1997 the water chemistry was converted from reduced all volatile treatment to oxygenated treatment to mitigate flow accelerated corrosion and was the first drum boiler to be converted (Dooley, March 2005). With the increased demand on the supply of electricity more recently, Lethabo operating units have run more continuously at higher loads for the last 10 – 15 years.

#### 6.1.3.2 Boiler Feed Pump Refurbishment

Since the commissioning of the power station the EFPs installed in unit 5 and 6 are still original, with unit 1 to 4 having been refurbished between 1992 and 1996. Having conducted no refurbishments or replacements of the EFPs since 1996 they still have low running hours in the range between 12,000 and 20,000 running hours as a result of the high availability of the SFPs.

The availability of the SFPs has been good and therefore the EFPs have not been needed to maintain the load on the units for extended periods. Sulzer Pumps are in the process of refurbishing one EFP to assess the condition with the intention of refurbishing all installed EFPs. As the EFPs have not been refurbished, no records are available of clearances and therefore the wear rate effect on EFP efficiency cannot be considered. Table 6 indicates how the averages have increased from the previous operating regime to the last refurbishments that were done.

**Table 6: Lethabo SFP and EFP running hours<sup>39</sup>**

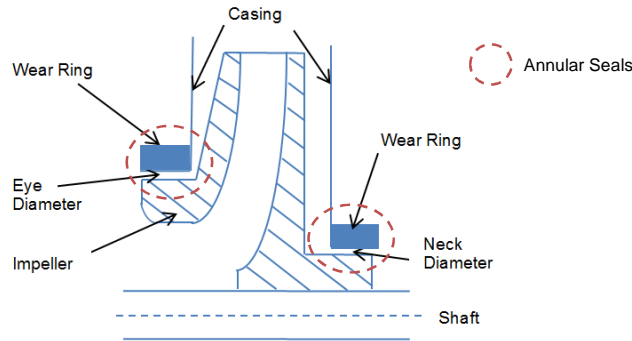
	Average Running Hours	Average Installed Hours	Average Running Time
Overall Refurbished SFPs	47,925	65,199	73.5%
Last Refurbishment of each Units SFPs	74,859	90,189	83%
Currently Installed EFPs	11,817	176,076	6.7%

Table 6 shows that the average running period for the more recent SFPs has increased as a result of the increasing power demands.

<sup>39</sup> Records from Wolfgang Miller, Sulzer Pumps site manager

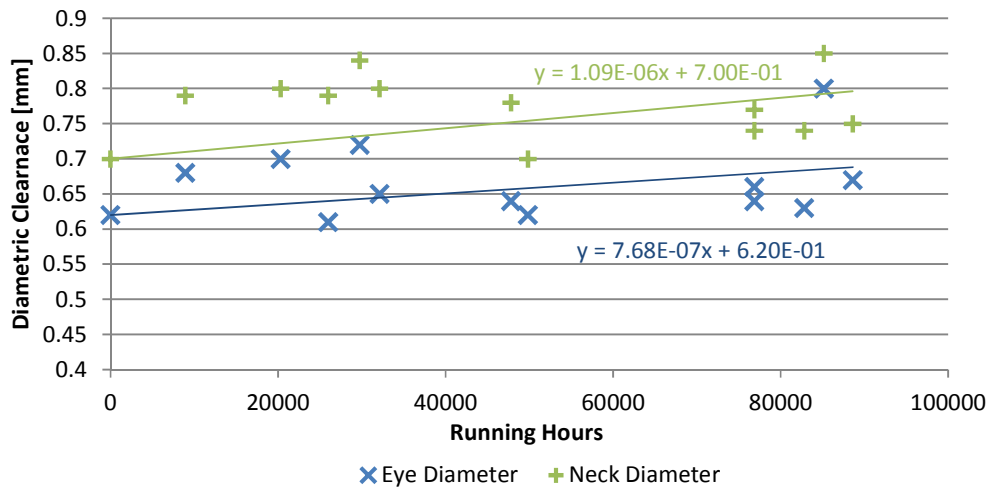
SFP refurbishments were done by Sulzer Pumps, OEM of the pumps, between 1991 and 1996 as well as from 2001 to 2012; these refurbishments are detailed in Figure 22 and for the 2<sup>nd</sup> to 4<sup>th</sup> stage impeller they are detailed in Appendix A.2 (p 110).

The suction impeller has a straight annular seal as shown in Figure 21 with a manufactured eye diameter of 279.45 mm h7. The wear ring is fitted on the first casing element and the impeller rotating inside the wear ring. The neck diameter is a straight annular seal of diameter 199.3 mm h7. The wear ring will be machined to suit the impeller once the impellers are machined to tolerance.



**Figure 21: Lethabo SFP suction impeller sectional view**

Figure 22 details the suction impeller diametric clearance at various running hours which were measured during refurbishments of the SFPs.



**Figure 22: Lethabo suction impeller wear data<sup>40</sup>**

The refurbishments that were conducted between zero and 40,000 running hours were all refurbishments that would have seen AVT(R) water chemistry, with high levels of iron transport (Dooley, March 2005) and FAC. This would have been a

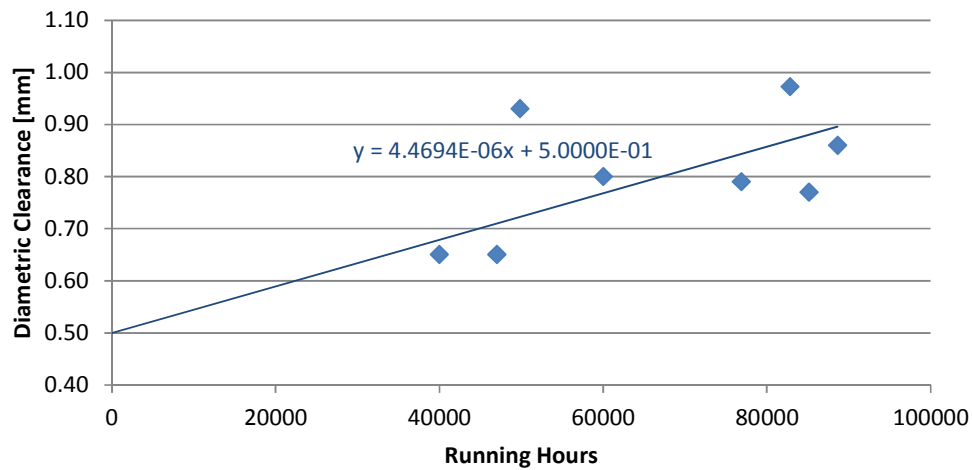
<sup>40</sup> Sulzer Pumps refurbishment records from inspection reports

contributory factor to the higher wear experienced. With the refurbishment between 40,000 and 90,000 hours only being exposed to OT this has resulted in a linear trend and more realistic wear rate for the Lethabo for operating conditions in the future.

The graphs in Appendix A.2 (p 110) for the second to last stage impellers illustrate a similar behaviour as in Figure 22 for recorded clearances at zero to 40,000 running hours this can be seen.

The commissioning of the power station during this period could have also been a contributory factor, but the latest refurbishments give a more realistic wear on the annular seals and are therefore considered as a linear trend.

The recorded clearance data for the axial thrust balance device shown in Figure 23 indicates the internal wear behaviour when clearances were recorded creating a linear trend.



**Figure 23: Lethabo axial thrust balance device wear data<sup>41</sup>**

The predicted wear of each impeller and the balancing device is detailed in Table 7, which was taken from the data obtained from the refurbished pumps. The trend was plotted as linear from the indications of the plotted values and the y-intercept was obtained from assembly reports which were averaged to obtain the “new clearance”.

Table 7 details the linear wear trendlines from the graphs using the equation:

$$y = mx + c$$

<sup>41</sup> Sulzer Pumps refurbishment records from inspection reports

**Table 7: Lethabo linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	7.68E-07	0.62	n/a	n/a	8.90E-07	0.7
2nd Stage Impeller	9.10E-07	0.62	9.10E-07	0.62	1.09E-06	0.7
3rd Stage Impeller	1.02E-06	0.62	1.02E-06	0.62	1.54E-06	0.7
Last Stage Impeller	9.33E-07	0.62	1.11E-06	0.62	n/a	n/a
	<b>m</b>			<b>c</b>		
Balancing Device	4.53E-06			0.50		

During a SFP refurbishment, depending on the condition of the impeller and the amount of scoring or pick up in the wear ring areas, the impeller will for most cases be skimmed to a smaller diameter. For cases of excessive wear it will be machined to a smaller diameter and then be welded up and then re-machined to the correct diameter.

For most cases it will be skimmed to a smaller diameter as a result of minimal wear. The wear ring is then fitted in the casing element for each stage and is left unbored. Once it is fitted in the casing element it is machined according to the new diameter of the impeller to ensure the correct clearance is maintained between the stationary and rotating parts.

The minimum clearances allowed for the impellers on the eye diameters are 0.58mm and on the neck diameter is 0.48mm (API 610, 2004).

#### 6.1.4 On-site Performance Testing

On-site testing was previously carried out by the Technology Services International (TSI), Eskom Performance and Testing department between 2000 and 2002 and more recently by a contractor Tuthukani Plant Performance Investigations in 2012<sup>42</sup>.

The performance tests in Table 8 were carried out with details of the tests in Appendix A.4, (p 114).

<sup>42</sup> All on-site performance testing was done by Mr Peter Bekink initially working for Eskom and then as a consultant

**Table 8: Lethabo on-site performance tests detail**

Test Number	Date	Unit	SFP Running Hours
1	June 2001	1	36,777
2	September 2002	6	42,528
3	July 2012	4	30,393
4	July 2012	6	36,322

The performances of the SFPs were tested at steady state conditions at generating loads between 613 MW and 630 MW.

From the on-site performance testing there is no consistency between the tests conducted and the link with the running hours of the SFPs.

With accessibility limited to measuring the various parameters at specific locations, certain assumptions are taken into account which limits the accuracy of the tests.

As a result of the tolerance on the measuring equipment under specified conditions and the conditions at Lethabo being somewhat limiting. Results are given which do not reflect the accuracy required of quantifying annular seal leakage of the SFP when tested.

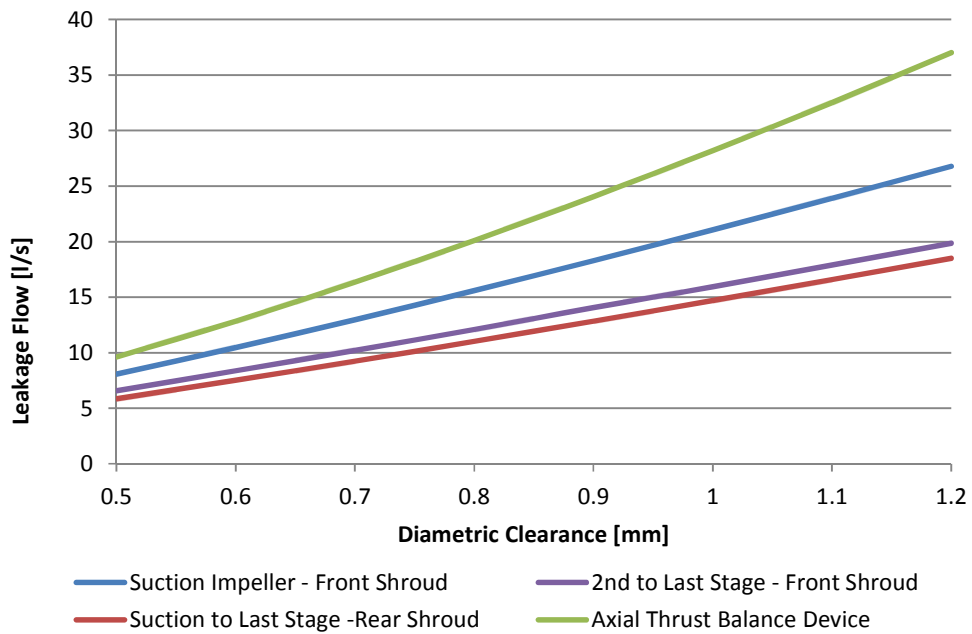
The tests do give an indication that the SFP is operating within 1% to 2% accuracy in efficiency which is enough to determine if there are major complications in the SFP and whether the SFP is operating at parameters which are not within design. The inaccuracy in the on-site performance testing can be seen in Appendix A.4 (p114).

The testing is, however, not accurate enough to determine the effect of internal wear on the performance of the pumps.

### 6.1.5 Annular Seal Leakage Modelling

The SFP has been modelled according to the equations in Section 5.1.1 (p 21) by the fluid-flow principles (Gülich, 2008) and additionally using the correct constants for the equations using Sulzer Pumps, Switzerland empirical data.

Figure 24 details the change in leakage flow through the annular seals as the clearance in the annular seal changes.



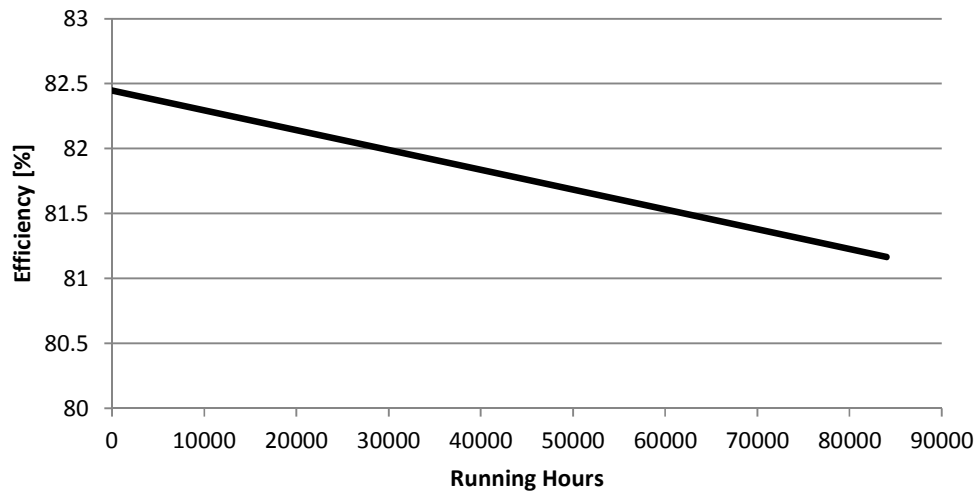
**Figure 24: Lethabo SFP annular seal leakage modelling results**

### 6.1.6 Efficiency Change

Due to the operation of the BFP system at Lethabo Power Station the SFP would only be considered from an efficiency point as the EFPs have low running hours compared to the power station running hours. The EFPs will need to be considered from a reliability aspect as their reliability is essential to ensuring a unit is operational when the SFPs are not available and the redundancy is essential.

The change in performance of the SFP was assumed that the only notable change to the efficiency would be the increase in leakage through internal wear of the annular seals.

The internal wear according to Table 7 on the annular seals was trended and the change in leakage flow was determined as shown in Figure 24. Together this was used to define the change in efficiency as shown in Figure 25.



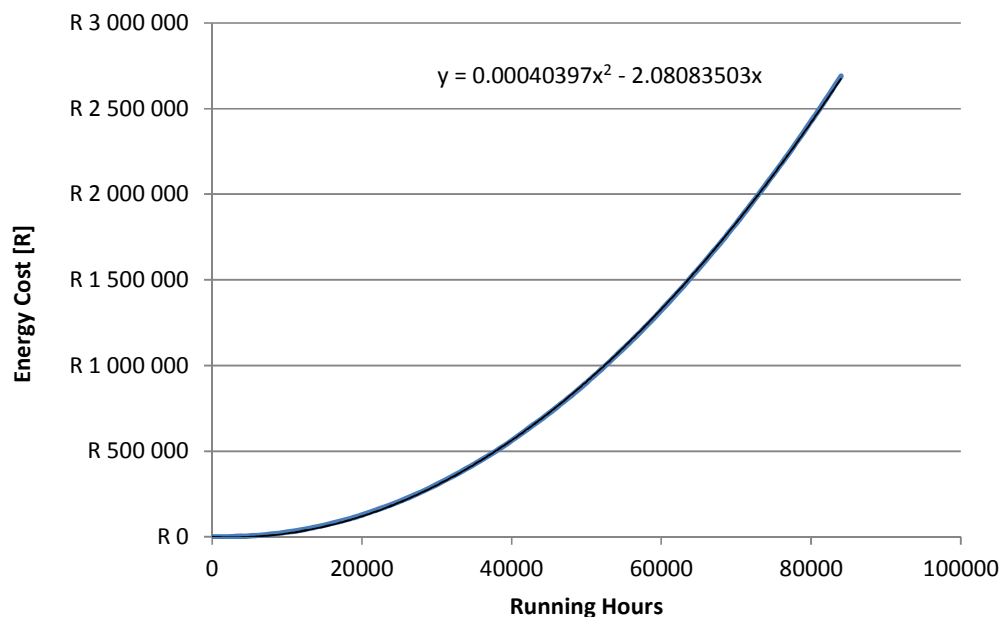
**Figure 25: Lethabo SFP – Change in efficiency over operating life**

### 6.1.7 Refurbishment Intervals

In order to determine the refurbishment intervals of the SFP considering the impact that the wear rate has on the power consumption of the BFP, the changes in leakage rates through the annular seals have been interpreted into a cost using the Eskom generation cost.

It is assumed that the SFP will be taken out of service when a strategic general overhaul is done on the unit and the refurbishment will not result in a load loss on the unit (i.e. it will be part of a planned outage in the power station unit). The costs for the exchange of a SFP on-site have also been considered.

Figure 26 predicts the energy cost as a result of the predicted internal wear from Table 7 (p 37) and combining the leakage flow rate from Figure 24 (p 39)



**Figure 26: Lethabo additional energy cost as a result of internal wear**

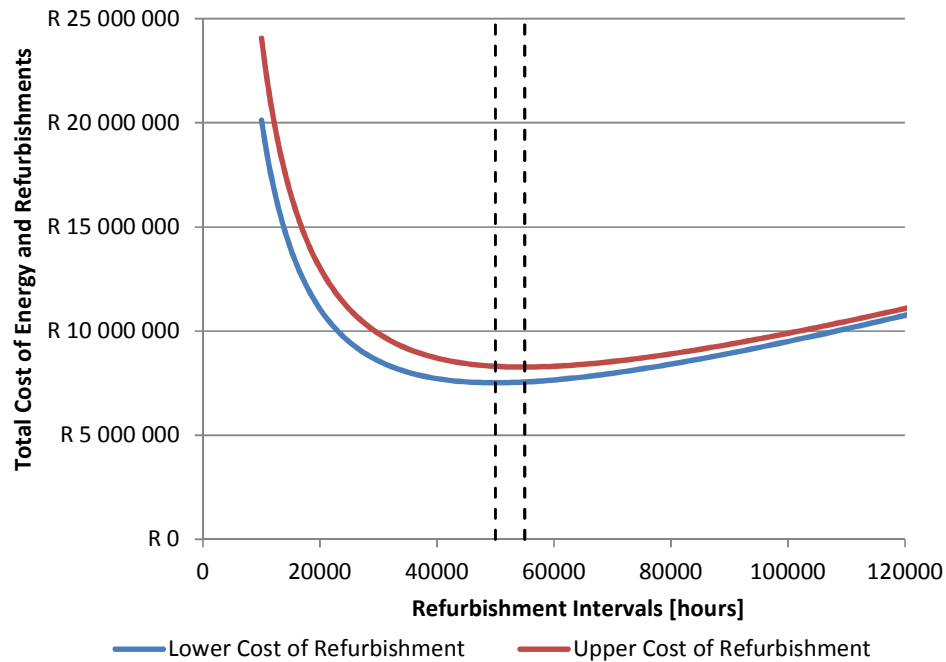
Assuming there is no major damage on the SFP the refurbishment cost will range between R1 million and R1.2 million.<sup>43</sup>

The refurbishment cost is dependent on the number of parts that either need to be replaced or require work to bring them back to design specifications.

The SFPs at Lethabo have been operated in a manner that has not seen extensive damage to the SFP internals and have been refurbished timeously at the current recommended running hours.

<sup>43</sup> Values averaged by Sulzer Pump CSS department from previous refurbishments

Figure 27 takes into consideration the additional energy required as a result of the internal wear of the SFP and the refurbishment costs (R 1 – 1,2 million) over the life of the plant. The design life is considered as 25 years with an availability of 90%.



**Figure 27: Optimum time to refurbish a Lethabo SFP**

It can be seen that the optimum time to refurbish the SFPs at Lethabo would be between 50,000 and 55,000 running hours depending on the refurbishment cost. With the current recommended running hours at Lethabo being 80,000 running hours, reducing the intervals between refurbishments should reduce the cost of refurbishment.

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## 6.2 KENDAL POWER STATION



### 6.2.1 Overview

Construction of Kendal began in July 1982, having six units with a generation capacity of 686 MW each and a total installed capacity of 4116 MW with the first unit being commissioned in 1988 (Eskom Holdings Limited, c2014).

It has an indirect dry-cooling system, which results in a much smaller usage of water.

### 6.2.2 Boiler Feed Water Pumps System Description

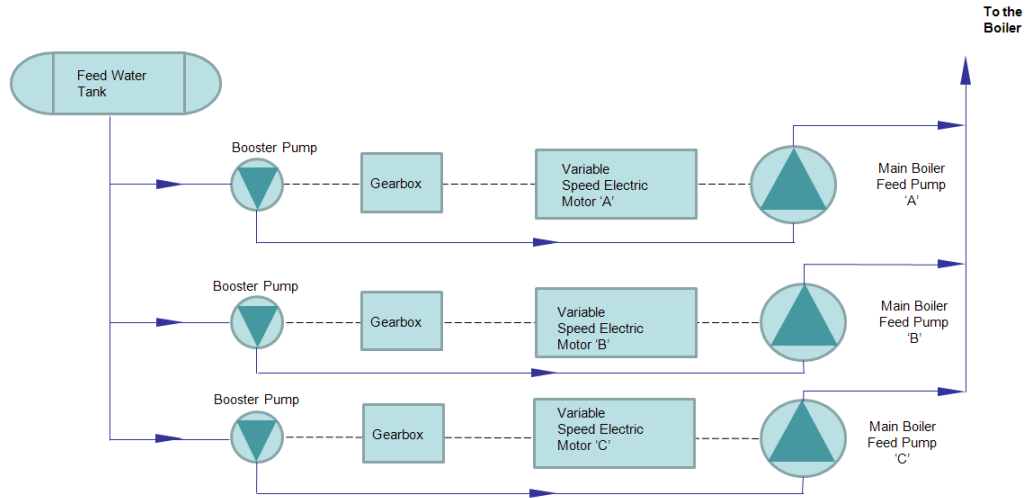
The feed water passes from the feed water tank to the booster pump, whose duty is to meet the necessary NPSH requirements of the main electric boiler feed pumps.

A connection pipe and intermediate strainer take the feed water from the booster pump into the main pump where the EFP is required to meet the boiler system requirements. Each unit at Kendal is equipped with three 50%-duty EFPs, with two EFPs running in parallel that will meet the full load requirements of each unit. Reduced load can be achieved with one EFP in service.

The EFPs are driven by variable speed electric motors which are directly coupled to the main boiler feed pump. The booster pump is driven by the variable speed electric motor through a speed reduction gearbox (ratio 1:4).

The BFP system is controlled fully or semi-automatically from a central control room.

Figure 28 below provides an overview of the Kendal boiler feed pump plant.



**Figure 28: Overview of boiler feed pump plant at Kendal Power Station**

The Sulzer Pumps electric-driven main boiler feed pump is a horizontal, multi-stage barrel type centrifugal pump, with a pull-out type cartridge design. The suction and discharge branches are welded onto the pipelines in the power station and as a result of the barrel type EFP; the welds do not have to be cut out to remove the cartridge.

**Table 9: Kendal EFP pump type and designation**

OEM	Sulzer Pumps
Type	HPT Pok 300-350

## 6.2.3 Power Station Operating Conditions

### 6.2.3.1 Operation of Electric Feed Pumps

The operation and maintenance philosophy of the EFPs is to attempt to run all three EFPs for a similar amount of time throughout each year. Typically they will try to have the three EFPs running with 20,000 running hours intervals between them (for example EFP A at 20,000 running hours, EFP B at 40,000 running hours and EFP C at 60,000 running hours) which will ensure that they do not all reach the recommended refurbishment running hours at the same time.

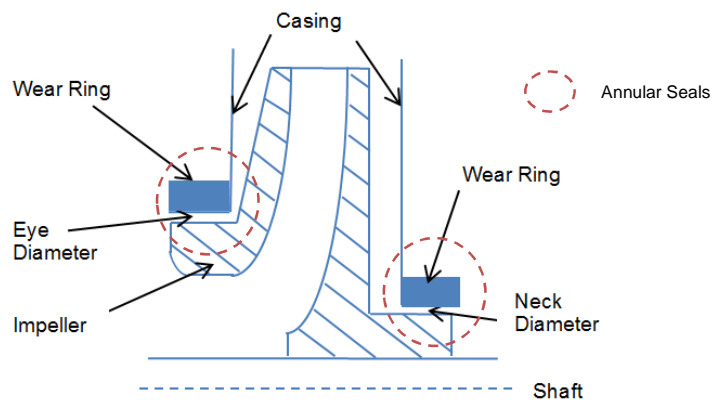
### 6.2.3.2 Electric Feed Pump Refurbishment

The average running hours after which the EFPs are refurbished is 64 692 running hours, all of which were done by the OEM (Sulzer Pumps).

Numerous refurbishments were done on the EFPs between 1999 and 2003 from where the analysis of refurbishments was taken, with running hours of between 50,000 and 75,000 running hours between refurbishments.

From the data collected, as seen in Figure 30, it was realised that the EFPs are refurbished timeously and consequently have not seen excessive damage as a result of extending the running hours.

The suction impeller has a straight annular seal as seen in Figure 29, with a manufacturing eye diameter of 244.53 mm h8. The wear ring is fitted on the first casing element and the impeller rotates inside the wear ring. The neck diameter is a straight annular seal of diameter 174.39 mm h8.



**Figure 29: Kendal EFP suction impeller sectional view**

Details of the 2<sup>nd</sup> to last stage impellers wear data can be found in Appendix B.2 (p 116).

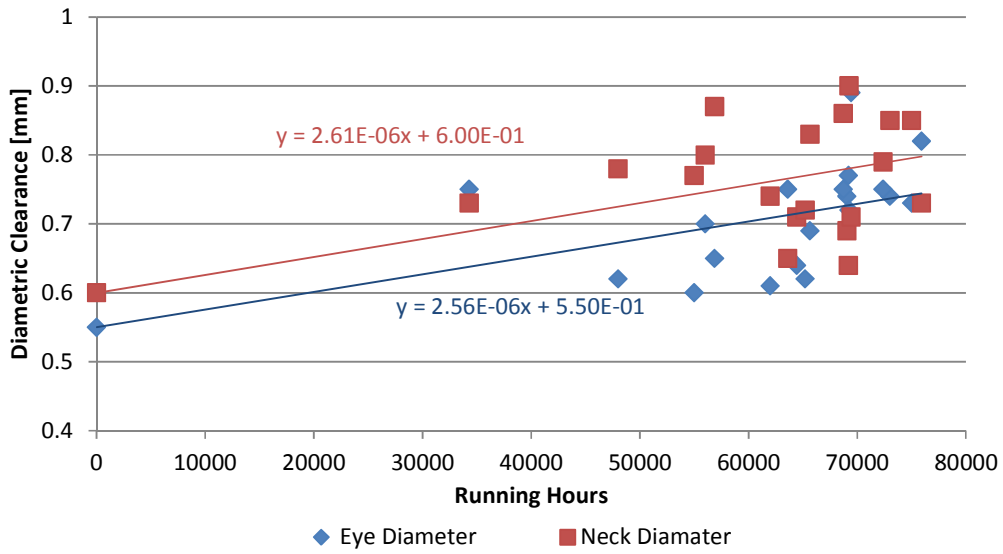


Figure 30: Kendal suction impeller wear data<sup>44</sup>

With all the refurbishments done at higher running hours it is assumed that the trend would be linear throughout, starting at a new clearance which was averaged from assembly reports after an EFP refurbishment was done. The tolerance in the scatter is somewhat substantial, but the linear behaviour depicted can be justified as a result of the new clearance being an averaged value. A “worn” clearance as detailed by the OEM would be 0.9mm diametric clearance and this is not yet reached before a refurbishment is done.

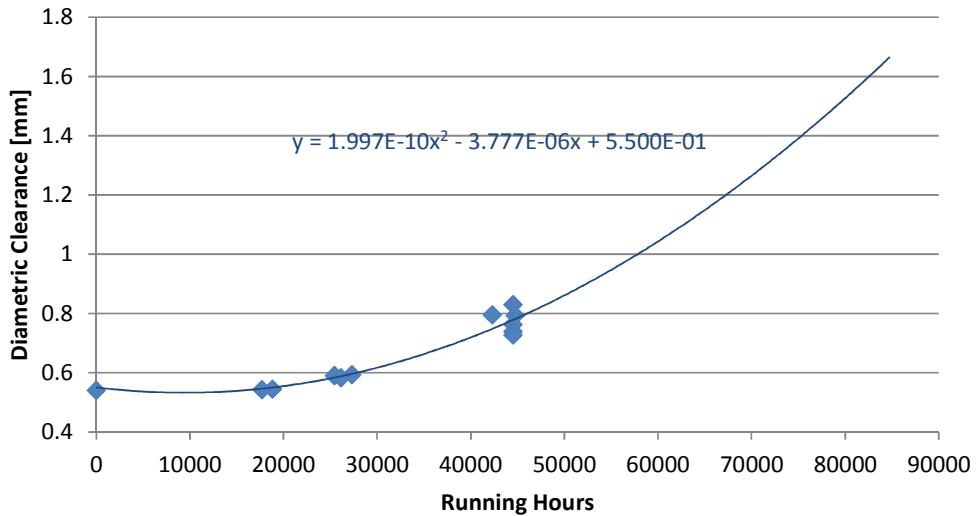


Figure 31: Kendal axial thrust balance device wear data<sup>45</sup>

<sup>44</sup> Sulzer Pumps refurbishment records from inspection reports

The NDE cover can be dismantled on site if the NDE end cover is leaking or in order to inspect the condition of the axial thrust balance device. As a result records are maintained by the site manager and are detailed in Figure 31 which are at more distributed running hours.

Table 10 details the linear wear trendlines from the impeller graphs using the equation:

$$y = mx + c$$

Table 10 details the parabolic wear trendline to the second order from the balancing device graph using the equation:

$$y = dx^2 + ex + f$$

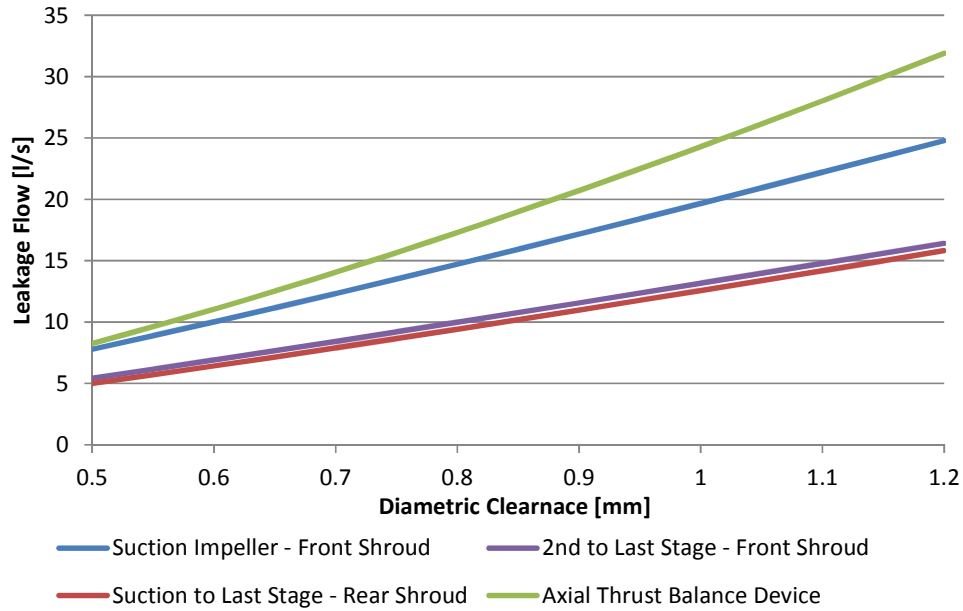
**Table 10: Kendal linear and exponential wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	2.56E-06	0.55	n/a	n/a	2.61E-06	0.6
2nd Stage Impeller	2.29E-06	0.55	2.00E-06	0.55	2.40E-06	0.6
3rd Stage Impeller	2.49E-06	0.55	2.19E-06	0.55	2.72E-06	0.6
Last Stage Impeller	2.66E-06	0.55	2.38E-06	0.55	n/a	n/a
	d		e		f	
Balancing Device	2.00E-10		3.78E-06		0.55	

<sup>45</sup> Records obtained from Sulzer Site Manager

### 6.2.4 Annular Seal Leakage Modelling

The EFP was modelled according to the fluid flow principles according to (Gülich, 2008) using Sulzer Pumps, Switzerland empirical data.



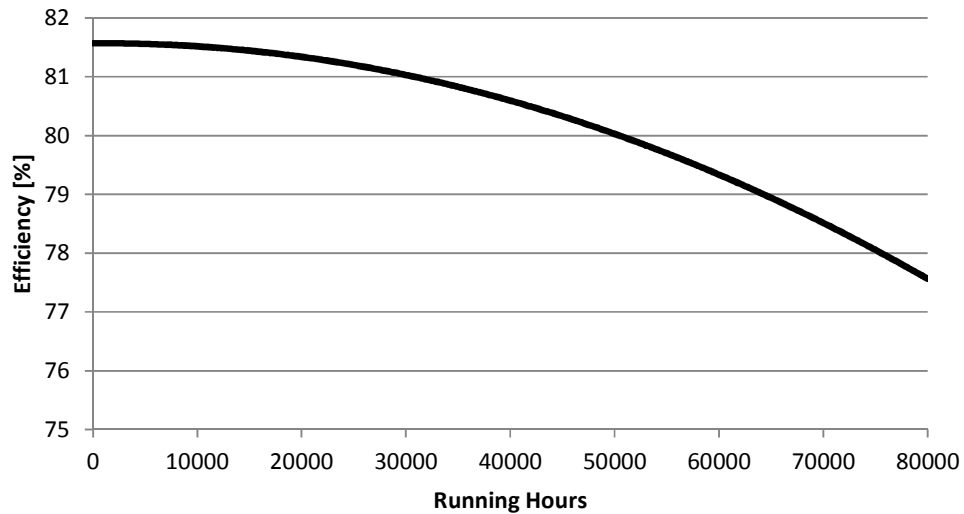
**Figure 32: Kendal EFP annular seal leakage modelling results**

The suction impeller has a straight annular seal and the 2<sup>nd</sup> to last stage impeller has a stepped seal. In Figure 32 it is seen that the effect of the seal configuration, for this particular EFP design, significantly reduces the amount of leakage through the annular seal.

### 6.2.5 Efficiency Change

The internal wear in the EFP on the annular seal has an effect on the leakage and as a result reduces the efficiency of the EFP as it wears, which increases the power requirements of the BFP system.

The effect of internal wear on efficiency is clearly shown in Figure 33

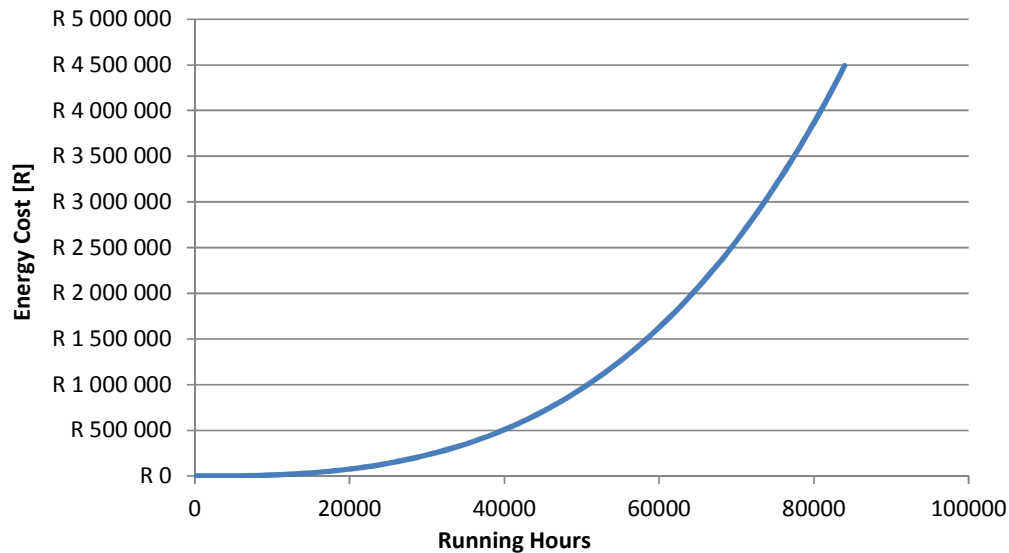


**Figure 33: Kendal EFP – Change in efficiency over operating life**

The change in efficiency is not linear as a result of the axial thrust balance device which has an exponential wear rate.

### 6.2.6 Refurbishment Intervals

The energy cost given in Figure 34 is as a result of the internal wear, therefore when the EFP is new it is assumed that it will not use any additional power but as it starts to wear the additional power is the energy cost shown in Figure 34.



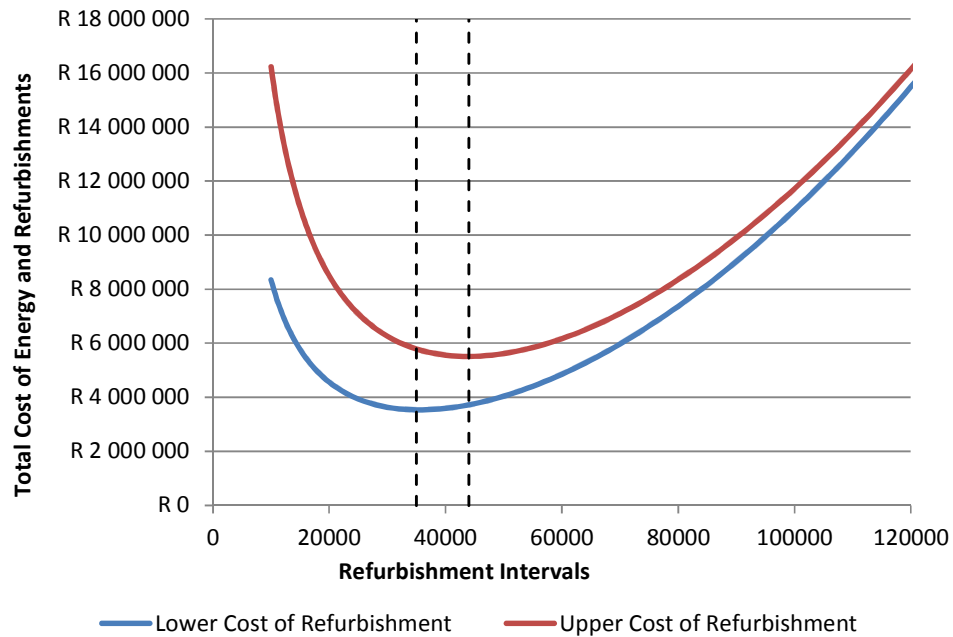
**Figure 34: Kendal additional energy cost as a result of internal wear**

The Kendal EFPs are 50%-duty feed pumps and to quantify the cost for each unit in the power station, the combination of two 50% EFPs is required. This will allow one to compare it to a power station that operates with a 100% SFP.

The refurbishment costs at present value ranges between R600,000 and R1.2 million, depending on the condition of the EFP. Factors that influence the refurbishment cost are the number of previous repairs done and the amount of internal wear. The lower refurbishment costs are primarily as a result of EFPs that have seen minimal wear and only the necessary wear parts are replaced.

The balance water return line flow and pressure are monitored on site and this should be monitored to estimate the deterioration in the EFP. The wear in the axial thrust balance device is directly related to the flow and pressure and as the clearance increases the change can be monitored.

The refurbishment costs for Kendal EFPs have a significant difference between its lower and upper refurbishment cost (R 600,000 - R1.2 million). The wear has an exponential behaviour at higher running hours and this is linked to the wear of the components in the EFP which affects the refurbishment costs.



**Figure 35: Optimum time to refurbish a Kendal EFP**

Now considering the energy consumed as result of internal wear and the cost of refurbishments with an upper and lower cost as shown in Figure 35.

Figure 35 shows that the optimum time for refurbishment would be between 35,000 and 44,000 running hours depending on the cost of refurbishment. Considering the history of refurbishments, the cost would be reduced by refurbishing more frequently.

## 6.3 MATLA POWER STATION



### 6.3.1 Overview

Matla Power Station was a milestone for power generation in South Africa when it was built during the 1980s. Construction began in 1974 and by July 1983 the power station could produce 3600 MW, with a unique concrete constructed boiler house. The power station has six units that make up the power station, each of which has a capacity of 600 MW (Eskom Holdings Limited, c2014).

### 6.3.2 System Description

The feed water pumps supply the feed water from the deaerator storage tank through the HP heaters to the boiler.

The system consists of a 100% duty pump, driven by a steam turbine and two 50% duty pumps driven by electric motors and a variable speed fluid drive coupling. The general arrangement can be seen in the schematic layout shown in Figure 36. Either the SFP or the two EFPs running in parallel can maintain the load requirements of each unit in the power station.

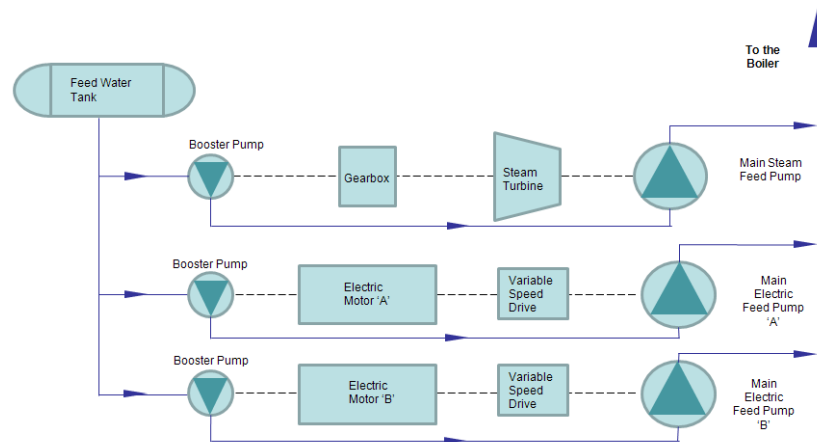


Figure 36: Overview of boiler feed pump plant at Matla Power Station

**Table 11: Matla SFP pump type and designation**

OEM	Sulzer Pumps
Type	HPT Pok 40

The main steam feed pump is directly coupled and speed-regulated by the turbine, at the same time driving the booster pump through a speed reduction gearbox.

**Table 12: Matla EFP pump type and designation**

OEM	Sulzer Pumps
Type	HPT Pok 32

The electric feed pump has a variable-speed fluid drive coupling connected to the main pump and the fixed-speed electric motor can vary its duty point depending on the requirements of the system. The booster pump is directly coupled to the other side of the fixed-speed electric motor.

### 6.3.3 Power Station Operating Conditions

#### 6.3.3.1 Operation of Boiler Feed Pumps

The SFP is used to maintain the load on the unit according to design and with the high availability of the SFP has resulted in it being in service for most of the time. As detailed in the previous section the EFPs are used mainly for start-up and shut-down and as a result of the high availability of the SFPs, the EFPs have not been needed to maintain load on any of the units for extended periods.

There was a period between 1993 and 2008 where the EFPs were not refurbished and their installed hours increased to an average of 15,000 running hours, with the majority of these EFPs still not refurbished.

Having the EFPs installed over a 20-year period has resulted in each EFP experiencing a high number of starts and stops. If the SFP is not available and the EFPs are used to maintain the load they will run at low efficiencies due to the increased annular seal clearances as seen in section 6.3.3.4 (p 58).

#### 6.3.3.2 Boiler Feed Pump Refurbishment

The BFP refurbishments analysed for the SFPs and EFPs were done for the period between 1989 and 2009. Three of the EFPs had been damaged to an extent that no records of wear were documented with one of the SFPs having oval annular seals which made it impossible to determine the amount of wear.

Refurbishments have been carried out by the OEM (Sulzer Pumps) since 1980 with the details of the wear on the impellers and balance device detailed in the following sections with additional data in Appendix C.2 (p 124).

**Table 13: SFP and EFP running hours<sup>46</sup>**

	Average Running Hours	Average Installed Hours	Average Running Time
SFPs	37,901	44,753	84.7%
EFPs	2884	42,391	6.8%

#### 6.3.3.3 SFP Wear History

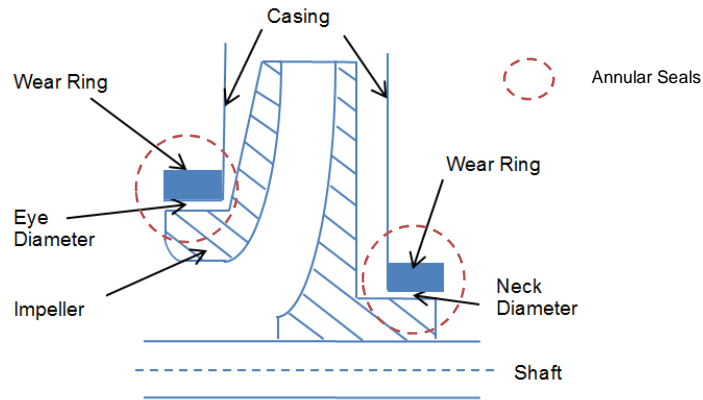
The SFPs have been refurbished between 5 and 7 times on each unit since January 1981, which means that the SFPs are refurbished on average every 5 – 6 years.

The target for refurbishment is every 6 years for the SFPs, but from the history of the refurbishments the running hours on the currently installed SFPs are

<sup>46</sup> Records from Sulzer Pumps Site Manager

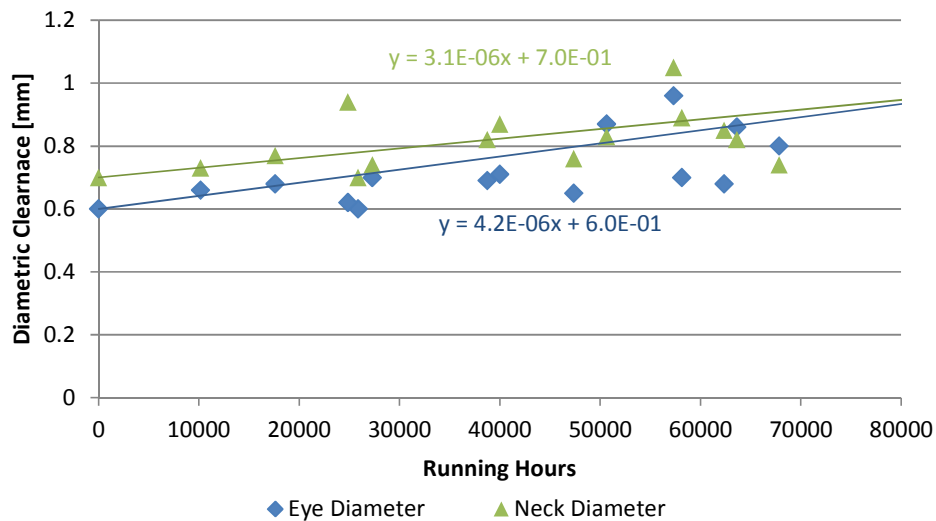
increasing and the opportunity for refurbishment is less as a result of a constrained Eskom grid and increased energy demand.

The suction impeller has a straight annular seal as shown in Figure 37, with a manufacturing eye diameter of 279.45 mm h7. The wear ring is fitted on the first casing element with the impeller rotating inside the wear ring. The neck diameter is a straight annular seal of diameter 199.3 mm h7.



**Figure 37: Matla SFP suction impeller sectional view**

Figure 38 details the suction impeller diametric clearances at various running hours when various refurbishments were done on the Matla SFPs.

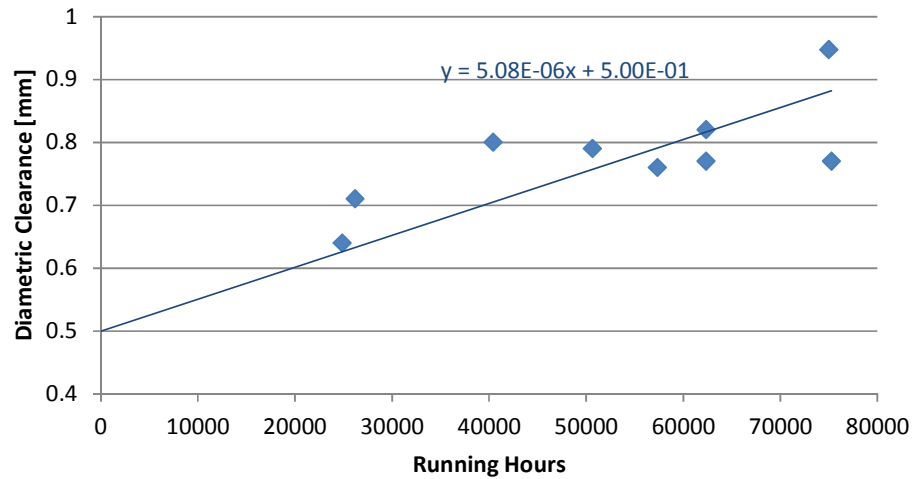


**Figure 38: Matla SFP suction impeller wear data<sup>47</sup>**

Recorded data of the 2<sup>nd</sup> to last stage impeller wear data is detailed in Appendix C.2 (p 124).

<sup>47</sup> Records from Sulzer Pumps inspection reports

Figure 39 details recorded data of the axial thrust balance device that were recorded at various intervals on the SFPs.



**Figure 39: Matla SFP axial thrust balance device wear data<sup>48</sup>**

The predicted behaviour of each impeller and the balancing device is detailed in Table 14 which was drafted from the data obtained from inspection reports during the refurbishment. The trend was plotted as linear from the indications of the plotted values and the y-intercept was obtained from assembly reports which were averaged to obtain the “new clearance”.

Table 14 details the linear wear trendlines from the graphs using the equation:

$$y = mx + c$$

**Table 14: Matla SFP linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	4.2E-06	0.6	n/a	n/a	3.1E-06	0.7
2nd Stage Impeller	3.9E-06	0.6	2.9E-06	0.6	3.6E-06	0.7
3rd Stage Impeller	3.6E-06	0.6	2.8E-06	0.6	3.7E-06	0.7
Last Stage Impeller	4.0E-06	0.6	3.1E-06	0.6	n/a	n/a
	<b>m</b>			<b>c</b>		
Balancing Device	5.08E-06			0.5		

<sup>48</sup> Recorded data by Sulzer Pumps Site Manager

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During a SFP refurbishment, depending on the condition of the impeller and the amount of scoring or pick up that the impeller has, it can be welded up and then machined to the correct diameter. In most cases it will be machined down to a smaller diameter as a result of wear on the annular seal face of the impeller.

The unbored wear ring is then fitted in the casing element for each stage and machined to specified tolerance of the annular seal according to the annular seal face of the impeller.

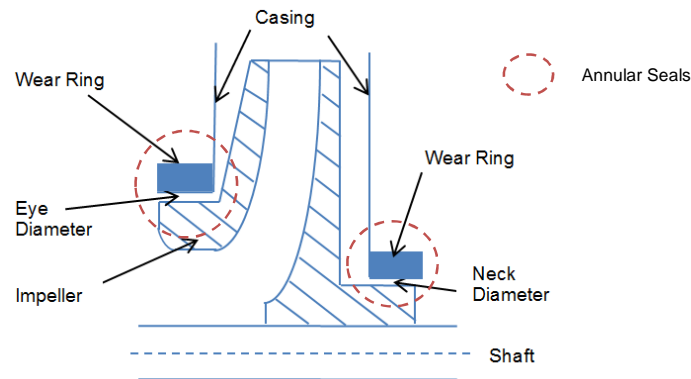
The minimum clearances allowed for the impellers on the eye diameters are 0.58mm and on the neck diameter it is 0.48mm (API 610, 2004).

### 6.3.3.4 EFP Wear History

The EFPs have been refurbished between three and five times with the exception of unit 5A which has had seven refurbishments.

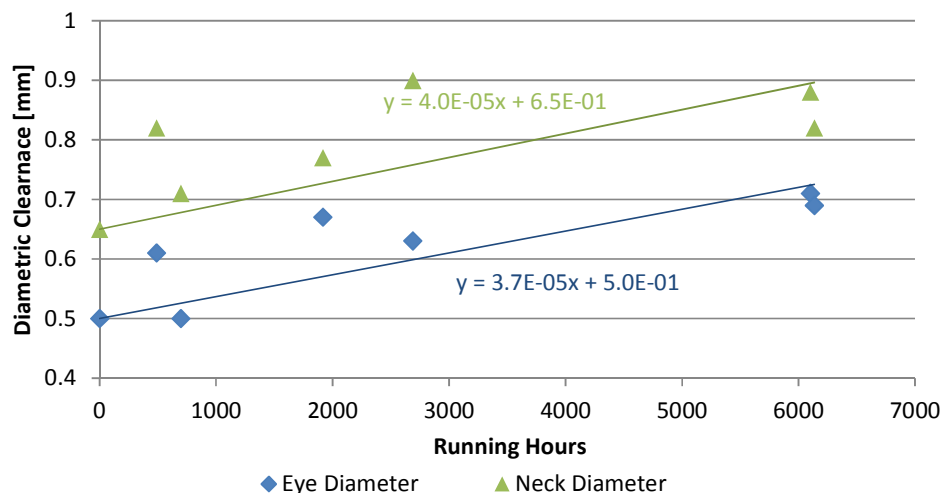
The average period of refurbishment is every eight years, which is in line with the target years of exchanges, but currently the EFPs installed have nearly all surpassed their refurbishment period.

The suction impeller has a straight annular seal as shown in Figure 40; manufacturing diameter of 214.5 mm h7, with the wear ring fitted on the first casing element and the impeller rotating inside the wear ring. The neck ring has a straight annular seal with a manufacturing diameter of 149.35 mm h7.



**Figure 40: Matla EFP Suction Impeller Sectional View**

Figure 41 gives details of the diametric clearances measured when a refurbishment was done at the specific running hours of the EFP.

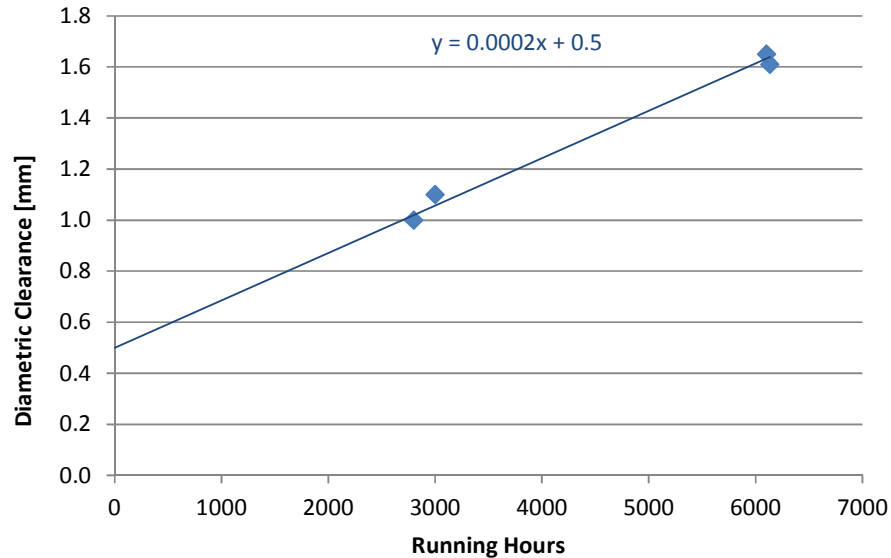


**Figure 41: Matla EFP suction impeller wear data<sup>49</sup>**

<sup>49</sup> Records from Sulzer Pumps inspection reports

Recorded data of the EFP 2<sup>nd</sup> to last stage impeller information is illustrated in Appendix C.3 (p 126).

Figure 42 details recorded data of the axial thrust balance device that were recorded at various intervals on the EFPs.



**Figure 42: Matla EFP axial thrust balance device wear data**

Only four records were available for the Matla EFP, which however had consistency of clearance relative to running hours as shown in Figure 42.

Table 15 details the linear wear trendlines from the graphs using the equation:

$$y = mx + c$$

**Table 15: Matla EFP linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	3.7E-05	0.5	n/a	n/a	4.0E-05	0.65
2nd Stage Impeller	4.3E-05	0.5	3.1E-05	0.5	3.6E-05	0.65
3rd Stage Impeller	5.1E-05	0.5	4.5E-05	0.5	3.8E-05	0.65
4th Stage Impeller	5.5E-05	0.5	4.7E-05	0.5	4.5E-05	0.65
Last Stage Impeller	4.9E-05	0.5	4.6E-05	0.5	n/a	n/a
	<b>m</b>			<b>c</b>		
Balancing Device	0.0002			0.5		

During an EFP refurbishment, depending on the condition of the impeller and the amount of scoring or pick up on the impeller, it can be welded up and then

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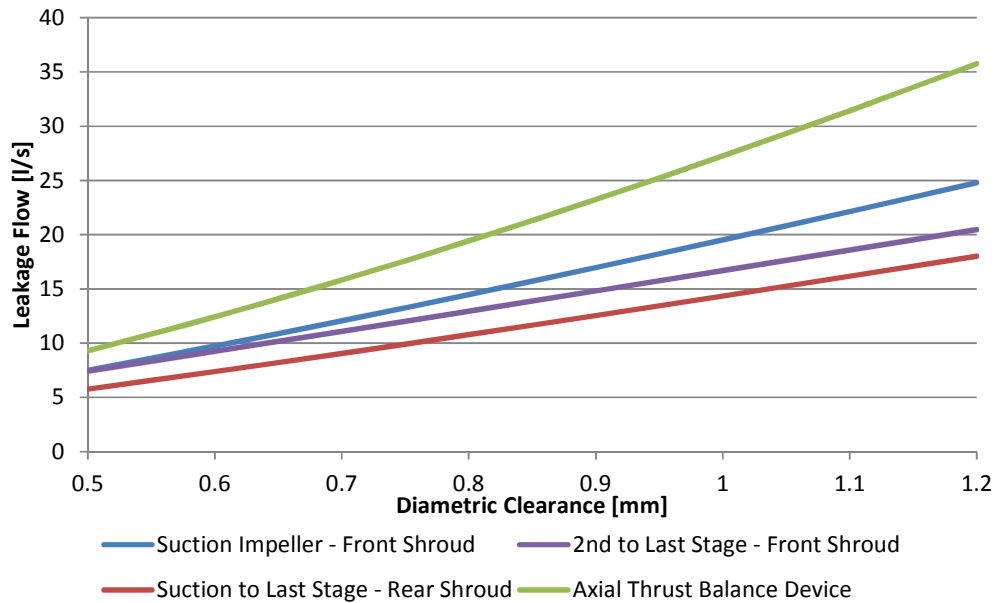
machined to the correct diameter, but in most cases it will be machined down to a smaller diameter as a result of wear on the annular seal face of the impeller.

The unbored wear ring is then fitted in the casing element for each stage and machined to specified tolerance of the annular seal according to the annular seal face of the impeller.

The minimum clearances allowed for the impellers on the eye diameters are 0.5mm and on the neck diameter it is 0.43mm (API 610, 2004).

### 6.3.4 Annular Seal Leakage Modelling

The SFP was modelled according to the fluid flow principles according to (Gülich, 2008) using Sulzer Pumps, Switzerland empirical data.



**Figure 43: Matla SFP annular seal leakage modelling results**

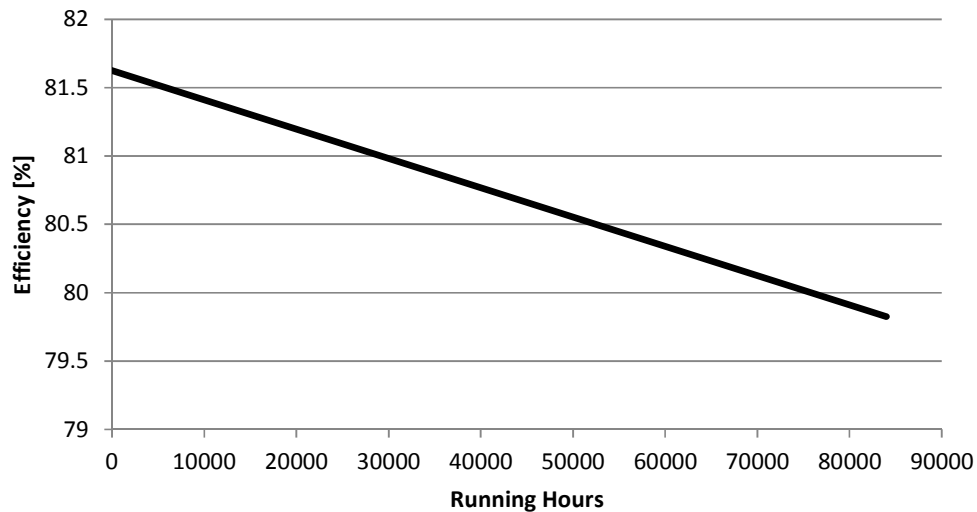
The suction impeller has a straight annular seal and the 2<sup>nd</sup> to last stage impeller has a stepped seal. In Figure 43 above it is seen that the effect of the seal configuration for this particular design reduces the amount of leakage through the annular seal.

### 6.3.5 Efficiency Change

Due to the operation of the BFP system at Matla Power Station the SFP would only be considered from an efficiency point as the EFPs have low running hours compared to the power station running hours. The EFPs will need to be considered from a reliability aspect as this is essential to ensuring a unit is operational when the SFPs are not available and the redundancy is then needed.

To understand the change in performance of the SFP it was assumed that the only notable change to the efficiency would be the increase in leakage through internal wear of the annular seals.

The wear on the annular seals was trended as shown in Figure 38 (p55) and the change over time was modelled to describe the change in efficiency as shown in Figure 44.

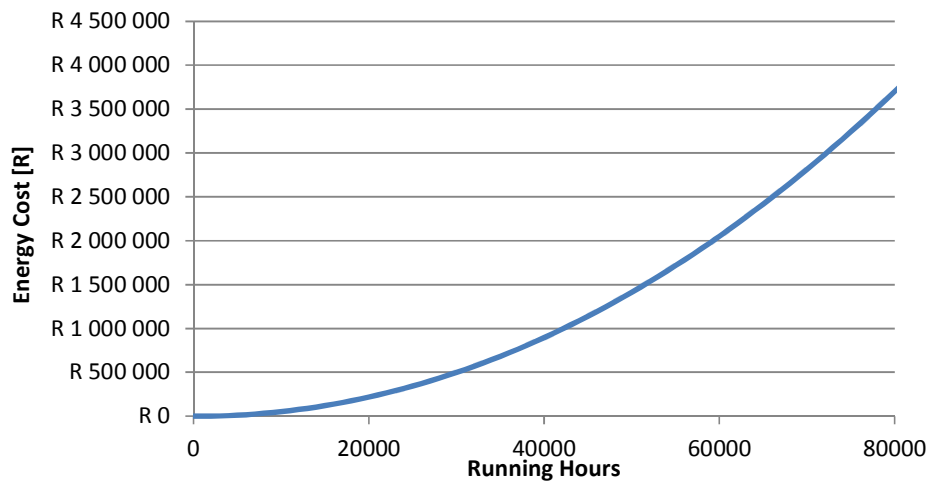


**Figure 44: Matla SFP – Change in efficiency over operating life**

### 6.3.6 Refurbishment Intervals

The majority of the SFPs at Matla have been refurbished timeously and not been run to excessive operating hours that result in unnecessary<sup>50</sup> internal damage. There was however one SFP that had been run to 73,828 running hours which experienced severe damage and required a shaft replacement and weld repairs on the impellers.

The accumulated energy cost in Figure 45 is as a result of the internal wear which causes the clearance in the annular seals to increase and use additional power.

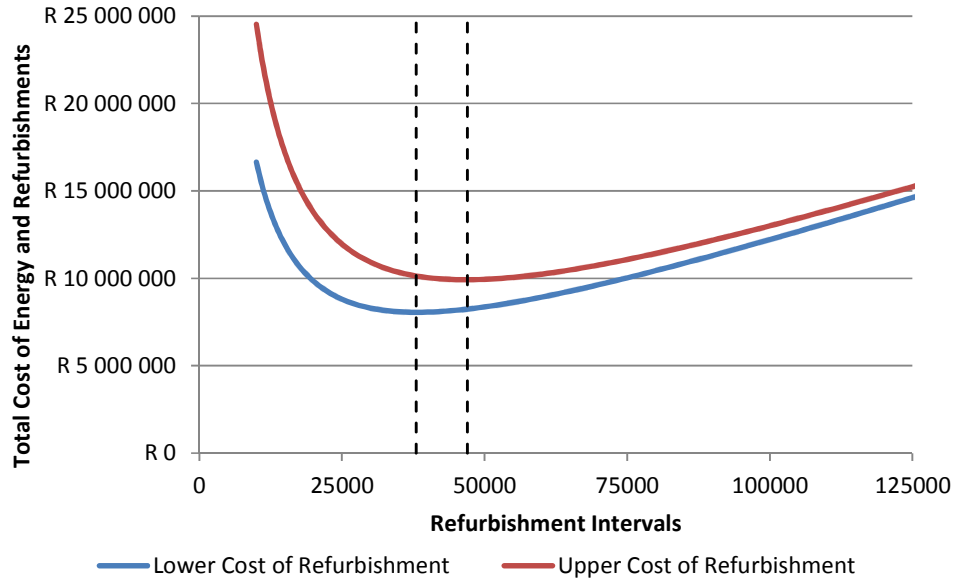


**Figure 45: Matla SFP additional energy cost as a result of internal wear**

The cost for refurbishment without excessive damage to any parts has been between R800,000 and R1.2 million. The refurbishment costs are used to for the lower and upper cost of refurbishment calculation in Figure 46.

<sup>50</sup> Matla inspection reports done by Sulzer Pumps

The graph in Figure 46 gives the total energy cost as a result of internal wear together with the number of refurbishments over the life of the power station in order to find the minimum total cost of energy and refurbishments.



**Figure 46: Optimum time to refurbish a Matla SFP**

The optimum time for refurbishment for the Matla SFPs is between 38,000 and 47,000 running hours depending on the refurbishment cost.

## 6.4 KRIEL POWER STATION



### 6.4.1 Overview

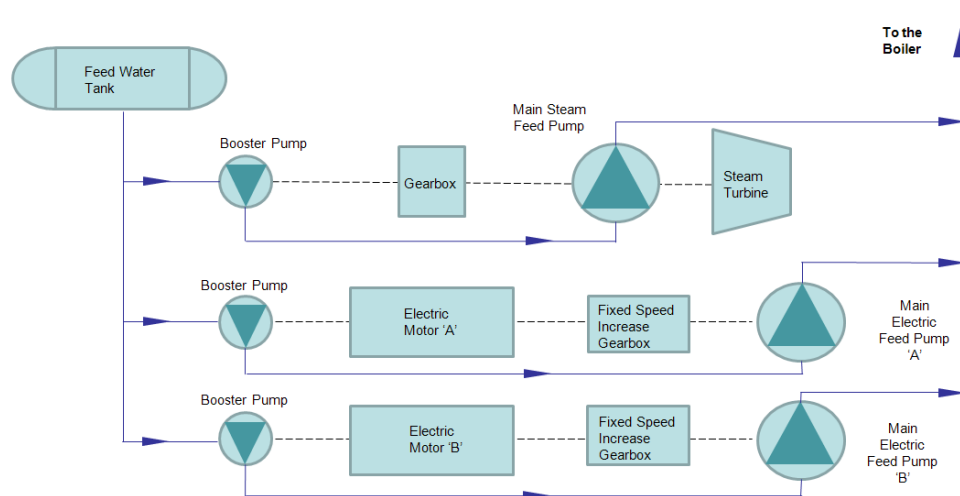
When the 3000 MW Kriel power station was commissioned in 1979 it was the largest coal-fired power station in the Southern Hemisphere (Eskom Holdings Limited, c2014).

Each unit has a 500 MW capacity with a wet direct cooling system which is evident by a steam cloud above the cooling towers.

### 6.4.2 System Description

The feed water pumps supply the feed water from the feed water tank through the high pressure heaters to the boiler.

The system consists of a 100% duty pump, driven by a steam turbine and 2 x 50% duty pumps driven by electric motors. The general arrangement can be seen in the schematic layout in Figure 47. Either the SFP or the two EFPs running in parallel can maintain the load requirements of each boiler.



**Figure 47: Overview of boiler feed pump plant at Kriel Power Station**

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The SFP is directly coupled and speed regulated by the BFPT with a speed reduction gearbox onto the booster pump.

**Table 16: Kriel SFP pump type**

OEM	Sulzer Pumps
Type	HPT Pk 30–40, 5 stages

The EFP has a fixed speed electric motor with a speed increase gearbox attached to the main pump with no variable speed control. The system requirements of the boiler are regulated via the feed-regulating control valves downstream of the EFPs.

**Table 17: Kriel EFP pump type**

OEM	Sulzer Pumps
Type	HPT Pk 32-35, 5 stages

### 6.4.3 Power Station Operating Conditions

#### 6.4.3.1 Operation of Boiler Feed Pumps

After being fully commissioned Kriel operated principally sound and as a result it had high availability of the plant and for a number of years operated effectively with minimal UCLF. The SFPs were refurbished at the recommended intervals and the availability of the SFPs was high as seen in the inspection reports of the SFPs. With the high availability of the plant long-term maintenance was somewhat neglected and as a result in the long run it has seen a lot of the SFPs become unavailable and out of service for extended periods in its later operating years.

The SFP being unavailable is not necessarily as a result of the pump specifically but in most cases as a result of the turbine or auxiliary equipment associated with it.

The unavailability of the SFPs has resulted in the EFP being required to maintain the load on some of the units on the plant for extended periods.

#### 6.4.3.2 Boiler Feed Pump Refurbishment

The refurbishments that were analysed for the SFPs at Kriel were done between 1999 and 2009 and the refurbishments on the EFPs were done between 2004 and 2009.

All the refurbishments had standard wear components replaced except for one SFP which had seized and a more extensive repair was done. This allowed for wear measurements to be accurately taken, which is evident in the wear data in Figure 49.

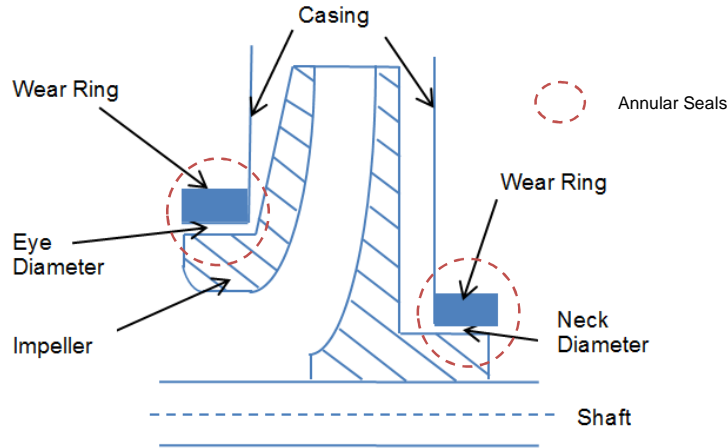
**Table 18: Kriel SFP and EFP running hours<sup>51</sup>**

	Average Running Hours	Average Installed Hours	Average Running Time
Overall Refurbished SFPs	60,181	87,622	68.65%
Overall Refurbished EFPs	5217	40047	13.03%

#### 6.4.3.3 SFP Wear History

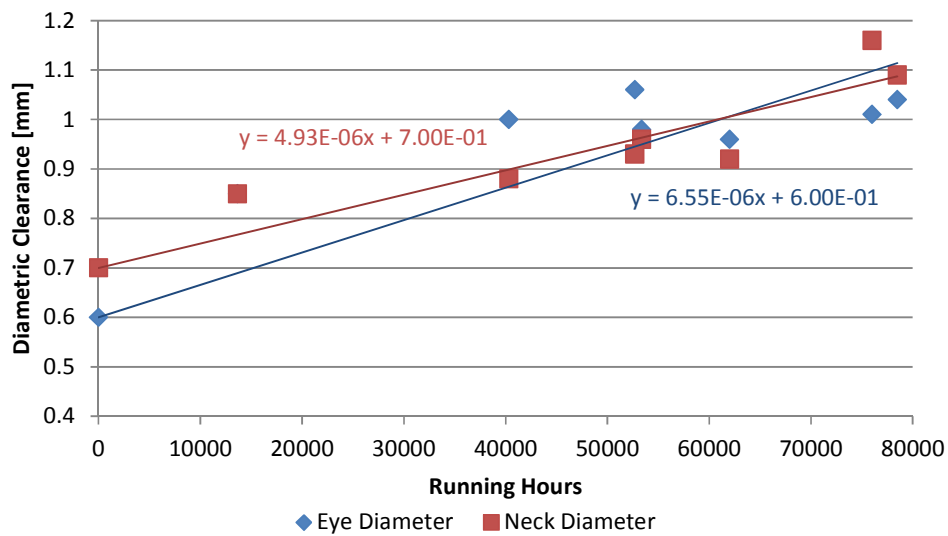
The suction impeller has a straight annular seal as seen in Figure 48; with a manufacturing diameter of 279.45 mm h7, with the wear ring fitted on the first casing element and the impeller rotating inside the wear ring. The wear ring will be machined to suit the impeller once the impellers are machined to tolerance. The neck diameter is a straight annular seal of diameter 199.3 mm h7.

<sup>51</sup> Records from Sulzer Pump Site Manager



**Figure 48: Kriel SFP suction impeller sectional view**

The internal clearances obtained for the Kriel SFP suction impeller are detailed in Figure 49.



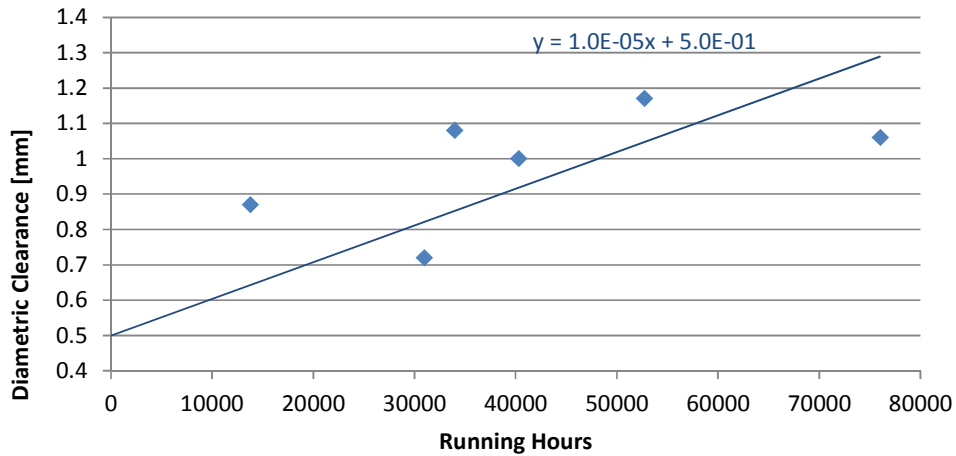
**Figure 49: Kriel SFP suction impeller wear data<sup>52</sup>**

Details of the 2<sup>nd</sup> to last stage impellers wear data can be found in Appendix D.2 (p 133).

The axial thrust balance device clearance measurements were a combination of records compiled at the power station by the site manager as well as records maintained during refurbishments by the OEM, which are detailed in Figure 50.

<sup>52</sup> Recorded data from Sulzer Pump inspection reports

Measurements of the SFP at various intervals were done by the site manager and these records are detailed in Figure 50.



**Figure 50: Kriel SFP axial thrust balance device wear data**

During a SFP refurbishment, depending on the condition of the impeller and the amount of scoring or pick-up it has in the wear ring areas, it can be welded up and then machined to the correct diameter, but in most cases it will be skimmed to a smaller diameter as a result of wear on the annular seal face of the impeller.

The wear ring is then fitted in the casing element for each stage unbored, which is machined according to the new specified tolerance of the impeller to ensure the correct clearance of the annular seal.

Table 19 details the linear wear trendlines from the Figures 49, 50, 99, 100 and 101 using the equation:

$$y = mx + c$$

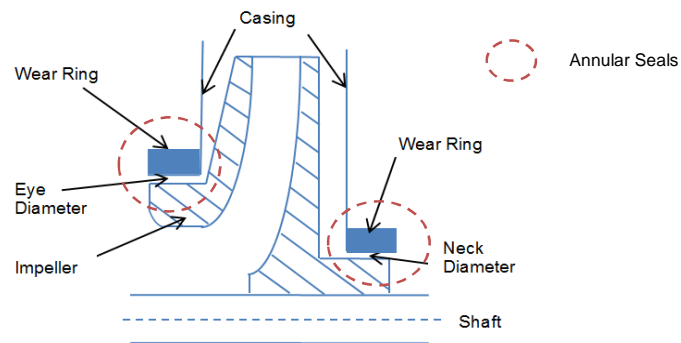
**Table 19: Kriel SFP linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	6.55E-06	0.6	n/a	n/a	4.93E-06	0.7
2 <sup>nd</sup> Stage Impeller	5.38E-06	0.6	4.29E-06	0.6	4.31E-06	0.7
3 <sup>rd</sup> Stage Impeller	5.23E-06	0.6	4.57E-06	0.6	4.51E-06	0.7
4 <sup>th</sup> Stage Impeller	5.60E-06	0.6	4.71E-06	0.6	5.05E-06	0.7
Last Stage Impeller	4.22E-06	0.6	3.94E-06	0.6	n/a	n/a
	<b>m</b>			<b>c</b>		
Balancing Device	1.0385E-05			0.5		

#### 6.4.3.4 EFP Wear History

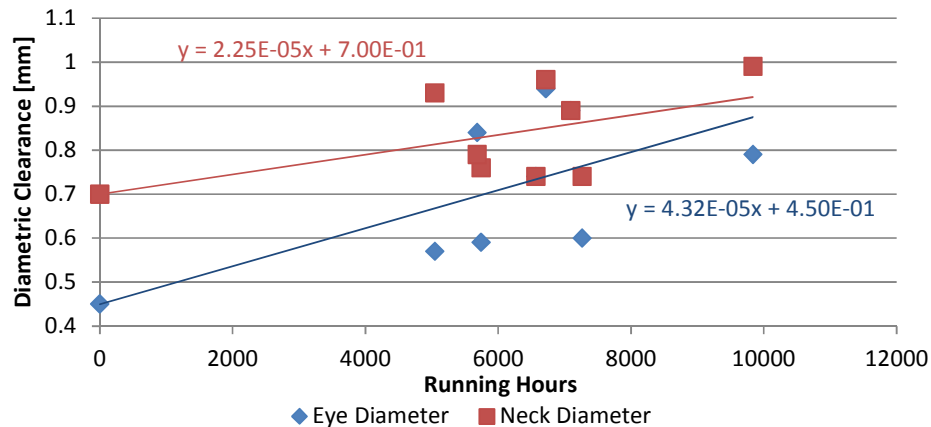
The EFPs at Kriel experienced numerous premature failures<sup>53</sup> and as a result have had a high number of refurbishments. Even with the high number of refurbishments this has not increased the reliability of the EFPs. All the units have experienced between four and nine refurbished cartridges with the exception of unit 1B which has had 12 refurbishments. The target is to have the EFPs installed for eight years before exchange, but this is rarely achieved.

The suction impeller has a straight annular seal as shown in Figure 51; with a manufacturing diameter of 214.5 mm h7. The wear ring is fitted on the first casing element with the impeller rotating inside the wear ring. The wear ring will be machined to suit the impeller once the impellers are machined to tolerance. The neck diameter is a straight annular seal which has a diameter of 149.35 mm h7.



**Figure 51: Kriel EFP suction impeller sectional view**

Figure 52 below details the diametric clearance of the suction impeller at the time of the refurbishment and is plotted according to the number of running hours.



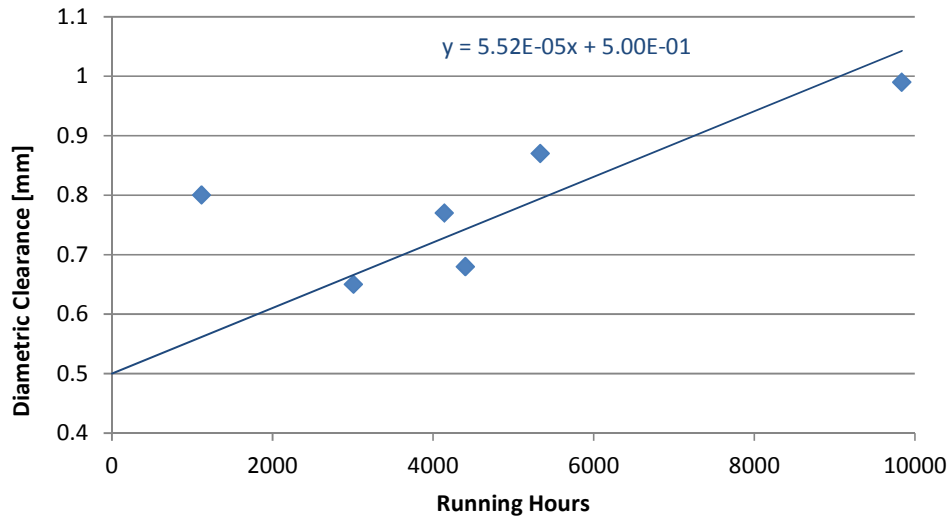
**Figure 52: Kriel EFP suction impeller wear data<sup>54</sup>**

<sup>53</sup> Details of premature failures were found in Sulzer Pumps inspection reports

<sup>54</sup> Recorded data from Sulzer Pumps inspection data

Recorded data of the 2<sup>nd</sup> to last stage impeller information is illustrated in Appendix D.3 (p 136). It is evident that the internal wear rate on the EFPs is high for such low running hours.

Figure 53 details recorded data at various intervals for the axial thrust balance device.



**Figure 53: Kriel EFP axial thrust balance device wear data<sup>55</sup>**

Table 20 specifies the linear wear trendlines from the Figure 52, 53, 103, 104, 105 and 106 using the equation below.

$$y = mx + c$$

**Table 20: Kriel EFP linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	c
Suction Impeller	4.32E-05	0.45	n/a	n/a	2.25E-05	0.7
2 <sup>nd</sup> Stage Impeller	4.98E-05	0.45	4.37E-05	0.45	2.96E-05	0.7
3 <sup>rd</sup> Stage Impeller	5.86E-05	0.45	5.03E-05	0.45	4.08E-05	0.7
4 <sup>th</sup> Stage Impeller	6.45E-05	0.45	5.37E-05	0.45	4.95E-05	0.7
Last Stage Impeller	6.92E-05	0.45	6.78E-05	0.45	n/a	n/a
	<b>m</b>				<b>c</b>	
Balancing Device	5.52E-05				0.5	

<sup>55</sup> Records from Sulzer Pumps Site Manager

#### 6.4.4 On-site Performance Testing

On-site testing was done on the SFPs at Kriel by an external contractor in 2012 as a result of uncertainties on the SFP system on unit 2 and 3.

Testing was done on the SFPs by determining the hydraulic power of the SFP by measuring the flow supplied by the SFP and the manometric head developed by the main and booster pumps. The attenuator spray water was open and the pressure and flow rate was also measured. The input power was determined using the thermodynamic testing method across the pump which is calculated assuming the conservation of energy.

The testing was carried out at steady state conditions at lower loads as the unit cannot reach full load due to deficiencies on other plant components. The test details are given in Table 21.

**Table 21: Kriel on-site performance test details**

Test Number	Date	Unit	SFP Running Hours	Efficiency
1	2012	2	21,979	83.6%
2	2012	3	14,207	78.6%

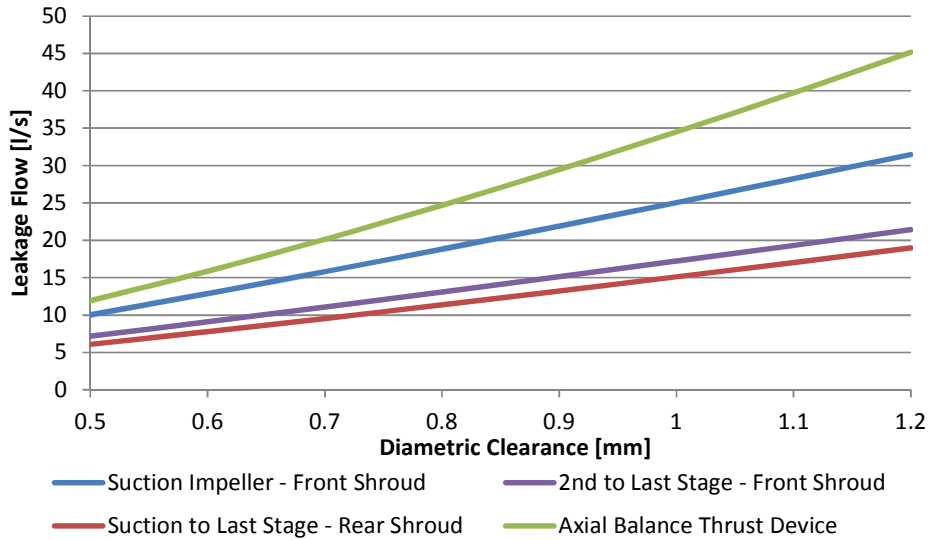
The testing was carried out on units 2 and 3 as it was suspected that there were deficiencies on the BFPT and resulted in an opportunity to carry out an efficiency test on the SFP. The data for the on-site performance testing that was done is included in Section D.5 (p143)

It is difficult to determine the shaft input power to the SFP using the thermodynamic testing method across the turbine or across the SFP. When comparing these two values there is no consistency in results, therefore the accuracy of this method is not sufficient for the testing required.

The suction temperature of the main pump is assumed from the inlet temperature of the booster pump and considering the uncertainty of the equipment used and the assumption that the temperature is consistent across the booster pump; this makes the test insufficient for the accuracy required.

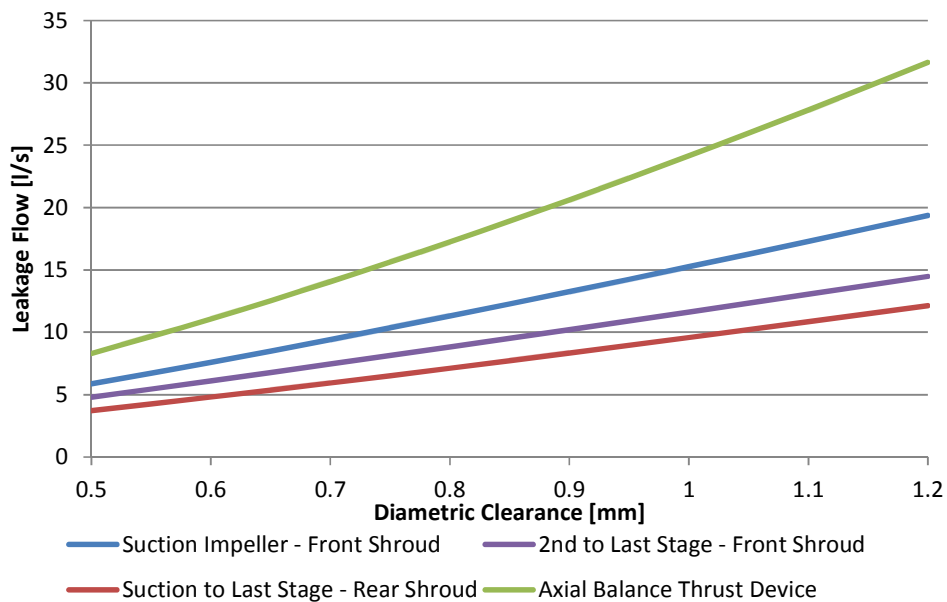
### 6.4.5 Annular Seal Leakage Modelling

The SFP was modelled according to the fluid flow principles detailed in Section 5.1.1 (p21) according to (Gülich, 2008) using Sulzer Pumps, Switzerland empirical data with corrected constants for the Kriel SFP.



**Figure 54: Kriel SFP annular seal leakage modelling results**

The SFP and EFP have a straight annular seal on the suction impeller and a stepped annular seal on the 2<sup>nd</sup> to last stage impeller, the effect of the stepped seal limiting the flow can be seen in Figure 54 and Figure 55.



**Figure 55: Kriel EFP annular seal leakage modelling results**

## 6.4.6 Efficiency Change

### 6.4.6.1 SFP Efficiency

To understand the change in performance of the SFP it was assumed that the only notable change to the efficiency would be the increase in leakage through internal wear of the annular seals.

The wear on the annular seals was trended as shown in Figures 49, 50, 99, 100 and 101 and the change in leakage rate as shown in Figure 54 over time was modelled to describe the change in efficiency. This effect on efficiency is shown in Figure 56 below.

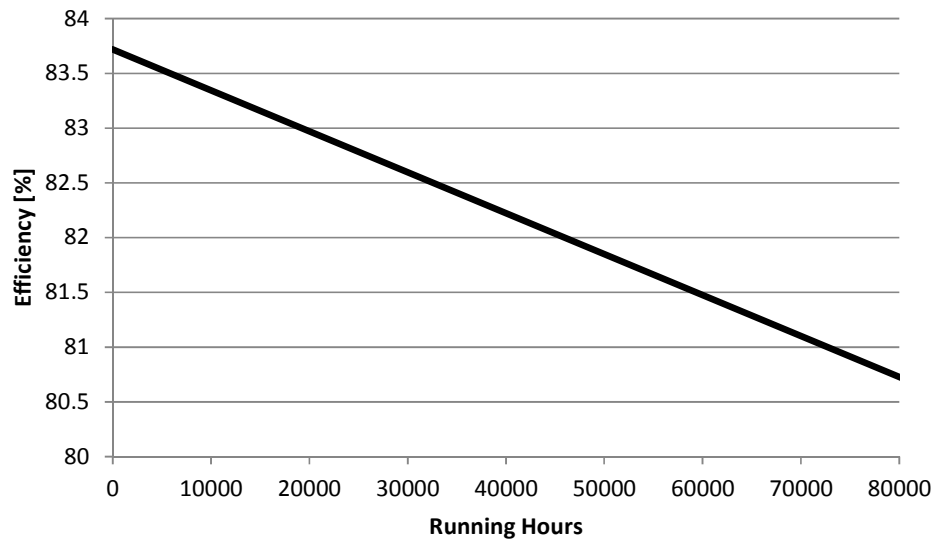


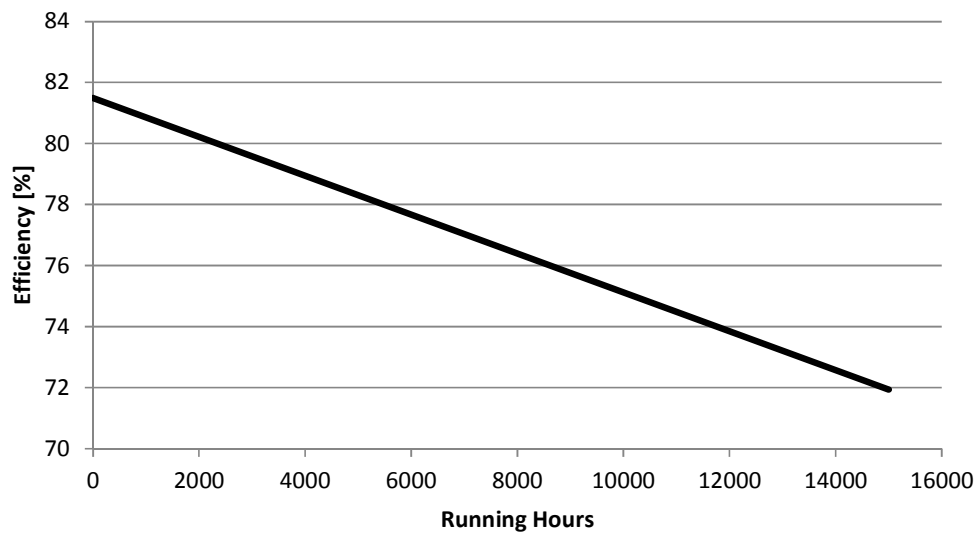
Figure 56: Kriel SFP – Change in efficiency over operating life

### 6.4.6.2 EFP Efficiency

The EFPs efficiency has been considered as a result of the high unavailability of the SFP at Kriel overall.

In addition to the low efficiencies as a result of the numerous starts and stops on the installed EFPs, the main pump does not have variable speed and causes additional losses over the feed-regulating valves, which contributes to the inefficiencies of the system.

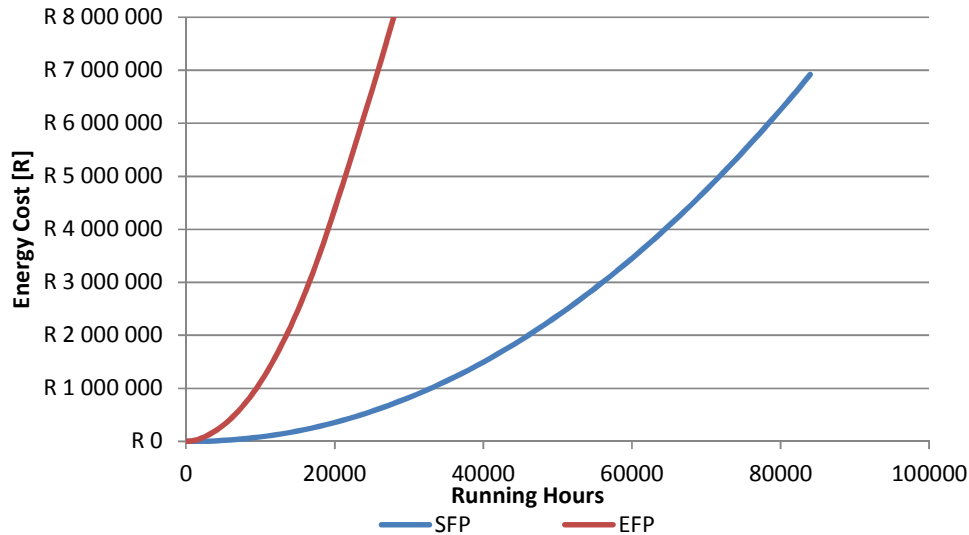
The drop in efficiency as a result of internal wear is seen in Figure 57 below of the internal wear on the annular seals as shown in Figures 52, 53, 103, 104, 105 and 106 and the change in leakage rate as shown in Figure 55 over time was modelled.



**Figure 57: Kriel EFP – Change in efficiency over operating life**

### 6.4.7 Refurbishment Intervals

The graph in Figure 58 below has been detailed with both the SFP and EFP, considering the running costs over the feed pump life cycle.



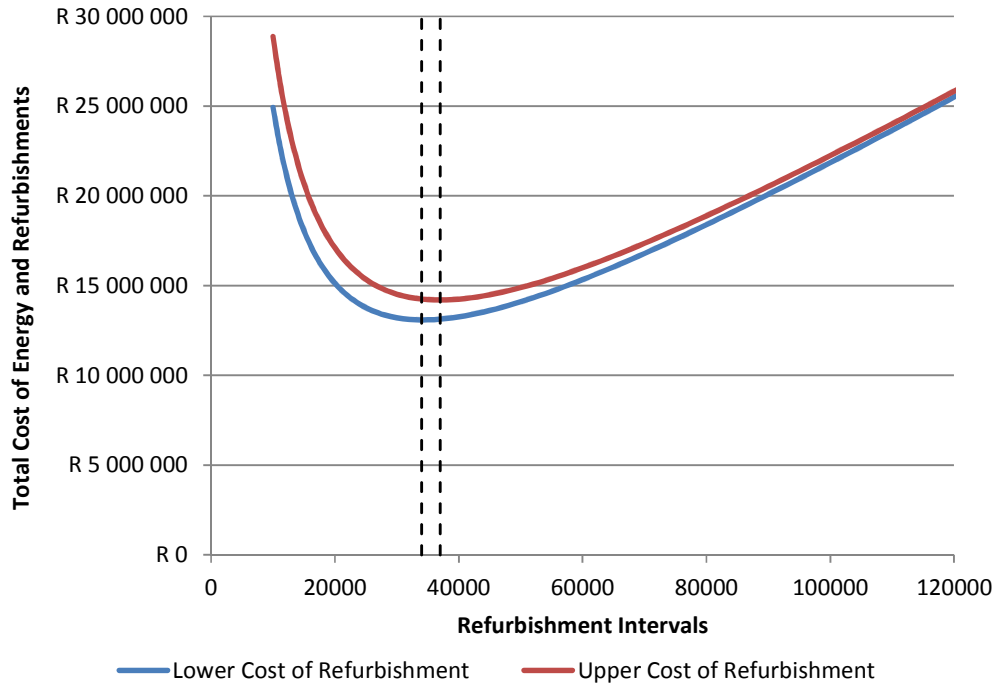
**Figure 58: Kriel feed pumps additional energy cost as a result of internal wear**

The SFPs have been refurbished timeously and until recently had a high availability. This decrease in efficiency is not directly related to the pump itself but is a result of other equipment on the system. With the exception of one SFP in 2009 the repair work required has not been excessive, with the cost of refurbishment ranging from R 1.2 million to R 1.4 million.

The EFPs have been operated for periods exceeding the OEM recommendations and as a result experienced high internal wear, causing a steep drop-off in efficiency and high costs for refurbishment in the range between R2 million to R2.5 million.

The high internal wear of the EFP is evident in Figure 58 as seen with the increase in energy costs compared to the SFP which has less internal wear.

The cost of refurbishments on the Kriel SFPs has been fairly constant, but with high internal wear and low volumetric efficiencies this has resulted in an increased total cost as can be seen in Figure 59.



**Figure 59: Optimum time to refurbish a Kriel SFP**

The optimum time for refurbishment of the Kriel SFP is between 34,000 and 37,000 running hours depending on the refurbishment costs.

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## 6.5 MATIMBA POWER STATION



### 6.5.1 Overview

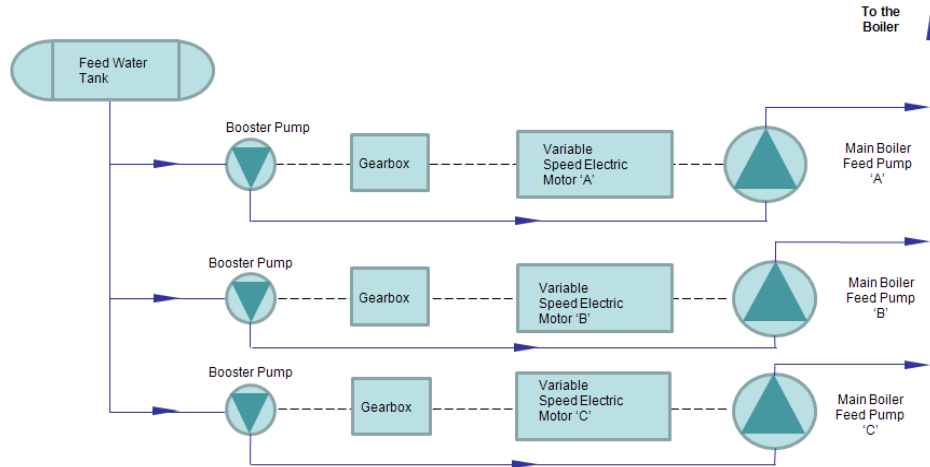
Matimba is currently the largest direct dry-cooling power station in the world, with a generating capacity of 6 x 665 MW units giving a total installed capacity of 3990 MW. It was designed as a base load operating power station and since commissioning started has had a high availability and operated predominantly at a constant full load (Eskom Holdings Limited, c2014).

### 6.5.2 Boiler Feed Water Pumps System Description

Each unit at Matimba is equipped with three 50%-duty EFPs, where two EFPs running in parallel will meet the full load requirements of each unit. Reduced load can be achieved with one EFP in service.

The feed water passes from the feed water tank to the booster pump, whose duty is to meet the necessary NPSH requirements of the electric feed pumps. A connection pipe and intermediate strainer take the feed water from the booster pump into the main pump where the EFP is required to meet the boiler system requirements.

An overview of the BFP system is shown in Figure 60 below.



**Figure 60: Overview of the boiler feed pump plant at Matimba Power Station**

The EFPs are driven by variable speed electric motors with which they are directly coupled. The booster pump is driven by the variable speed electric motor through a speed reduction gearbox.

**Table 22: Matimba EFP pump description**

OEM	Weir Pumps
Type	FK 3C-44

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### 6.5.3 Power Station Operating Conditions

#### 6.5.3.1 Operation of Boiler Feed Pumps

The EFPs at Matimba are a robust pumps that are able to survive vapour-lock and thermal shock conditions and have proven their design over two decades of operation<sup>56</sup>.

Matimba is a base load power station and with the EFPs being operated primarily near BEP, this increases the life of the EFP. The EFP is exposed to fewer starts and stops and does not operate at low flows which create instabilities as a result of mismatched angles on the impeller vanes.

Matimba does not fall under the Sulzer partnering contract and as a result has Rotek to assist with cartridge exchange at the power station. The refurbishments of the main pumps are done on an open-enquiry basis.

An efficiency study<sup>57</sup> was conducted previously on the EFPs and it was concluded that the refurbishment time of 40,000 running hours was recommended as described by the OEM. For numerous reasons the EFPs were run for extended periods past the recommended value, but still achieved high availability. More recently the EFP on unit 5A accumulated 124,000 running hours since the commissioning of the unit before being refurbished.

#### 6.5.3.2 Boiler Feed Pump Refurbishment

The average running hours for all the refurbishments on the EFPs is 49 234 running hours, which is close to the refurbishment interval as recommended by the OEM of 40,000 running hours.

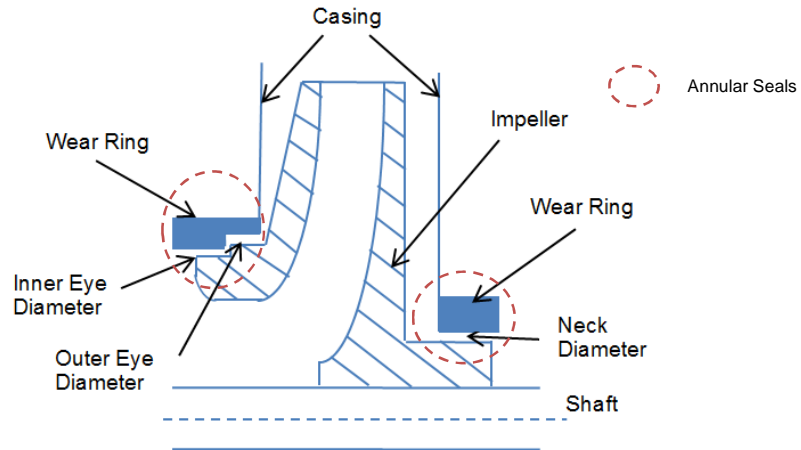
The data collected for the EFPs wear were prepared between 1997 and 2013 which are detailed in the graph in Figure 61 as well as the graphs in Appendix E.2 (p 146).

All three stages in the EFP have a stepped annular seal as shown in the sectional view in Figure 61.

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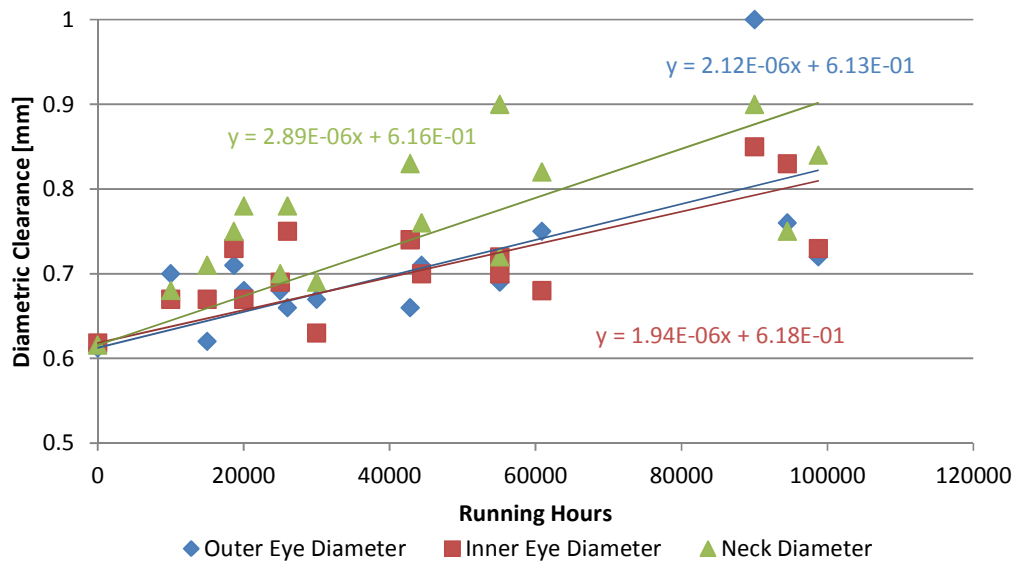
<sup>56</sup> As described in OEM Manual which was supplied during commissioning

<sup>57</sup> Matimba Report on Works Power Consumption (Matimba PS 4078384) Compiled by Jan Terblanche 13 August 1998

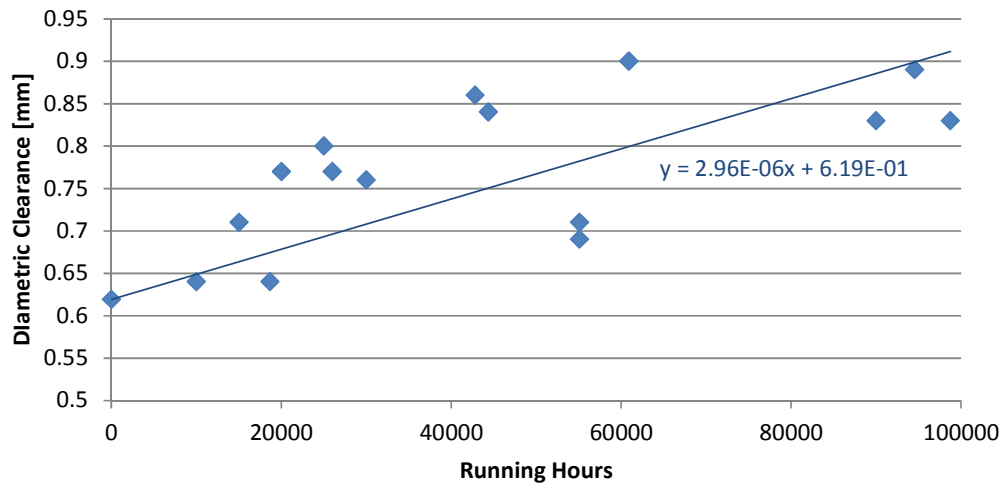


**Figure 61: Matimba EFP suction – last stage impeller sectional view**

Refurbishments were carried out by a number of different repairers over the 15 year period, with limited or sometimes no representation in South Africa for the OEM. Details in the graphs in Figures 62 and 63 and Appendix E.2 (p147) are from all the repairers.



**Figure 62: Matimba suction impeller change in clearance**



**Figure 63: Matimba axial thrust balance device wear data**

Table 23 below details the linear wear trendlines from Figures 62 and 63 and Figures in Appendix E.2 (p147) using the equation:

$$y = mx + c$$

**Table 23: Matimba linear wear trendline data for impellers and balancing device**

Component	Outer Eye Diameter		Inner Eye Diameter		Neck Diameter	
	m	c	m	c	m	C
Suction Impeller	2.10E-06	0.61	1.94E-06	0.62	2.89E-06	0.62
2 <sup>nd</sup> Stage Impeller	1.82E-06	0.62	2.17E-06	0.61	2.68E-06	0.61
Last Stage Impeller	1.79E-06	0.62	1.69E-06	0.64	n/a	n/a
	<b>m</b>			<b>c</b>		
Balancing Device	2.30E-06			0.62		

### 6.5.4 On-site Performance Testing

On-site performance testing of the EFPs has been carried out at Matimba by the Performance and Testing Department since 1994 on numerous EFPs.

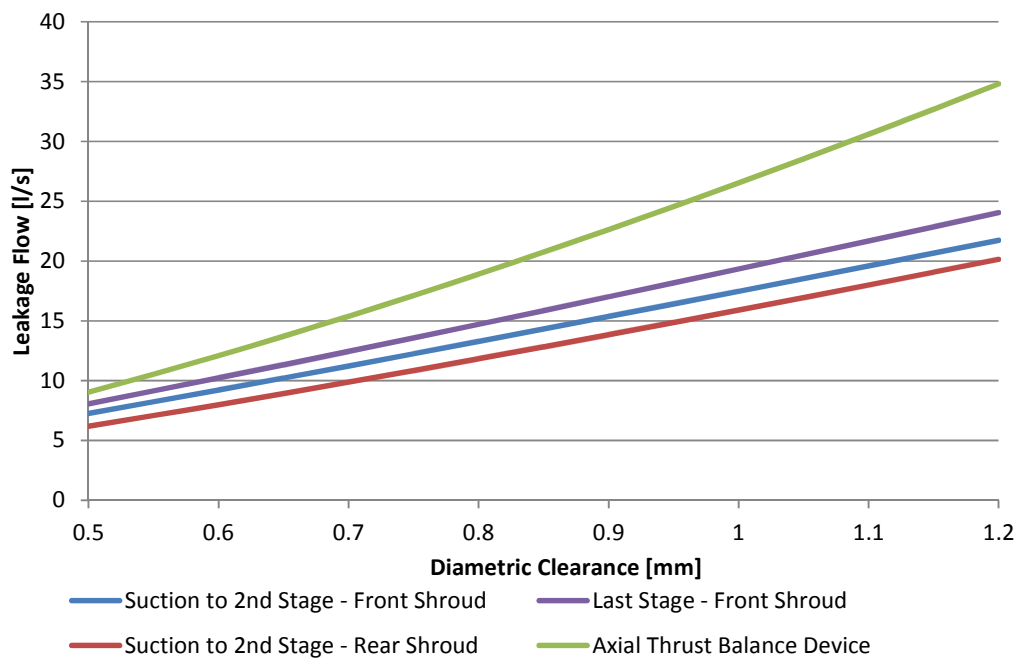
Calibrated equipment was used to measure the temperature, pressure and power input, which is captured on a data logger and plotted on the software program “EFP Efficiency” which was created by the system engineer on site.

The testing data has been thoroughly analysed and plotted to compare it to the theoretical curve at the time of the test, but there is no consistency and correlation from the tests and therefore the tolerance in recorded values is somewhat scattered.

### 6.5.5 Annular Seal Leakage

The EFP was modelled according to the fluid flow principles according to (Gülich, 2008) using Sulzer Pumps, Switzerland empirical data.

The leakage through the annular seals for the EFP have the same designed seals, with the last stage having a reduced diameter which results in a slightly higher leakage as seen in Figure 64 below.



**Figure 64: Matimba EFP annular seal leakage modelling results**

**Table 24: Input power and losses to Matimba EFP<sup>58</sup>**

<b>Total Input Power per Pump</b>	<b>8383 kW</b>
Power to Seals and Bearings	129 kW
Hydraulic Friction	674 kW
Balance Drum/Throttle Bush Friction	321 kW
Balance Drum Leakage	250 kW
Impeller Annular Seal Leakage	270 kW
<b>Total Losses</b>	<b>1644 kW</b>

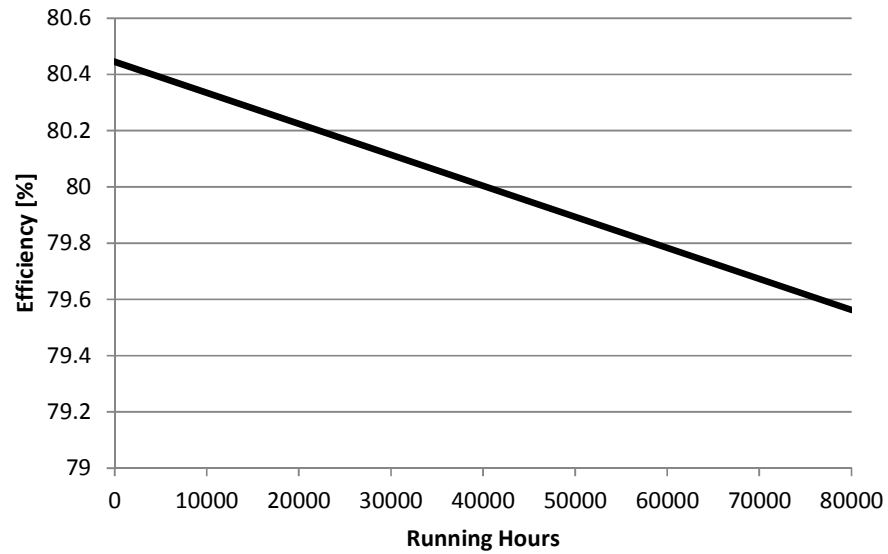
Table 24 details the power estimated by the annular seal leakage is 520 kW (250kW + 270kW) and the estimated power from the leakage modelling<sup>59</sup> when the clearances are new is 443 kW, which after 80,000 running hours of operation increases to 539 kW.

<sup>58</sup> Values requested by Eskom Corporate Specialist in 2009 which were supplied by the OEM (Weir Pumps)

<sup>59</sup> Change in annular seal leakage as a result of the internal wear as seen in figure 64

### 6.5.6 Efficiency Change

The Matimba EFP has a low efficiency of 80.4% when new and the effect of the limited wear internally shows how the efficiency drops over 80,000 running hours of operation, decreasing by 1.7%.



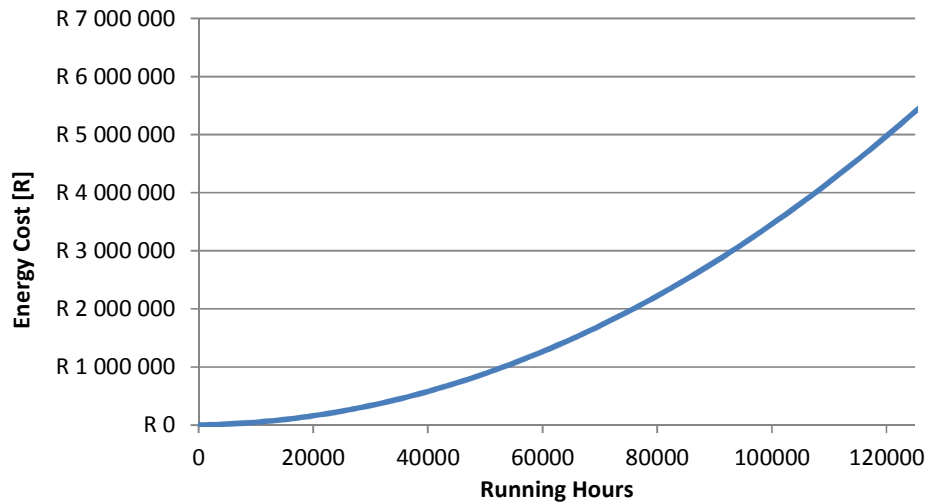
**Figure 65: Matimba EFP – Change in efficiency over operating life**

Even with a somewhat small decrease in efficiency the effect of additional power consumed as a result of the internal wear is still considerable as can be seen in the following section.

### 6.5.7 Refurbishment Intervals

With the short, stiff robust design of the EFP it is seen how the internal wear is limited and as a result, the additional energy required is somewhat limited in comparison to the other BFPs.

Figure 66 demonstrates how internal wear increases the energy cost of the EFPs.

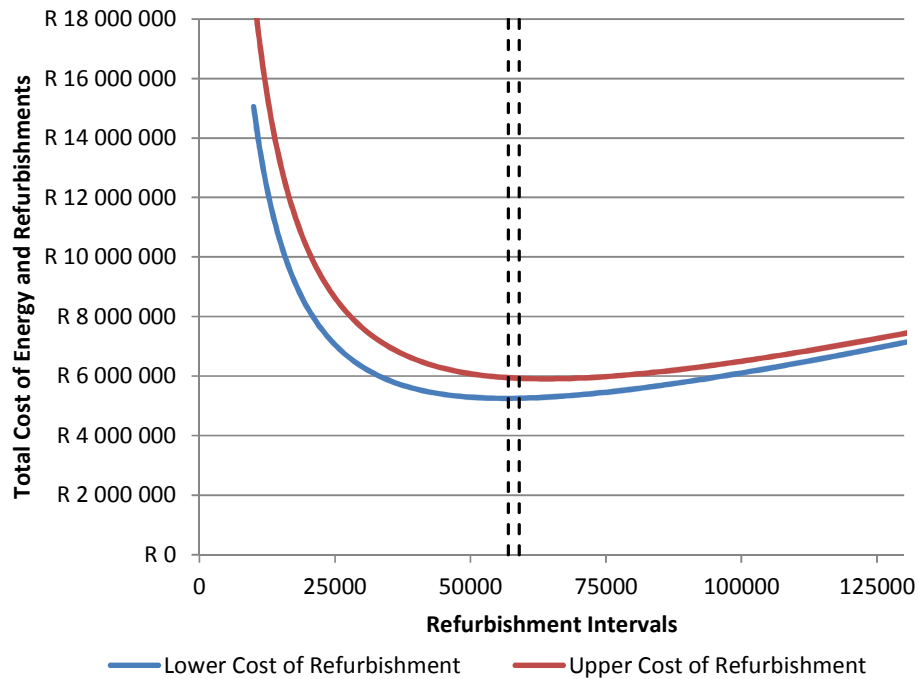


**Figure 66: Matimba EFP additional energy cost as a result of internal wear**

Typically the refurbishment costs have ranged between R 1.1 million and R 1.4 million, depending on the condition of the EFP when the refurbishment is done.

Without regular refurbishments the running hours have been extended beyond the recommended refurbishment intervals, mainly as a result of not having any permanent contract with a repairer for the refurbishment of the EFPs.

Figure 67 considers the energy and refurbishment costs for each interval over the design life of Matimba.



**Figure 67: Optimum time to refurbish a Matimba EFP**

The optimum time to refurbish an EFP at Matimba is between 57,000 and 59,000 running hours, depending on the cost of refurbishment.

## 7. COMPARISON OF BOILER FEED PUMPS

When considering the BFP systems that have been defined in the previous sections it must be remembered that reaching full load on a unit in a power station requires either 1 SFP or 2 EFPs.

In the power stations that have one SFP for continuous operation and two redundant EFPs, only the SFP has been considered for its efficiency as it is used primarily for continuous operation. The power stations that have three EFPs have been considered with two EFPs used to maintain continuous full-load operation.

### 7.1 VOLUMETRIC EFFICIENCY

The volumetric efficiency is based on the amount of internal leakage flow in the BFP compared to the amount of discharge flow from the BFP.

The flow around the inter-stage seal is not included in the calculation for the volumetric efficiency as it does not form part of the main hydraulic.

Equation 19 is used to calculate the volumetric efficiency.

$$\frac{Q}{(Q + Q_{sp} + Q_E + Q_h)} = \eta_v \quad (19)$$

Figure 68 shows the change in volumetric efficiency ( $\eta_v$ ) from when the BFP is new to when it has done 80,000 running hours of operation. The change in clearance from new to worn clearances is captured from the records of internal wear trended for each power station.

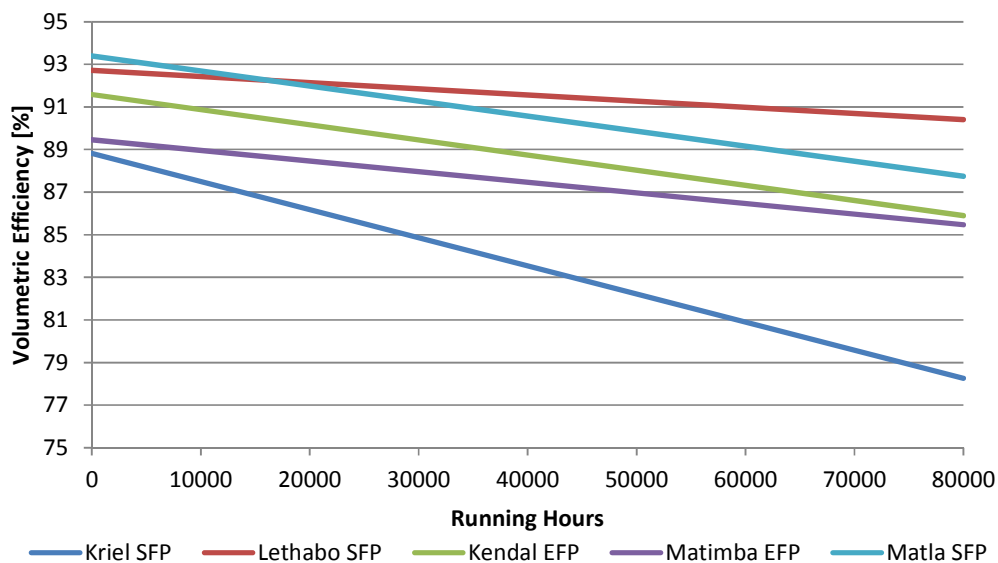
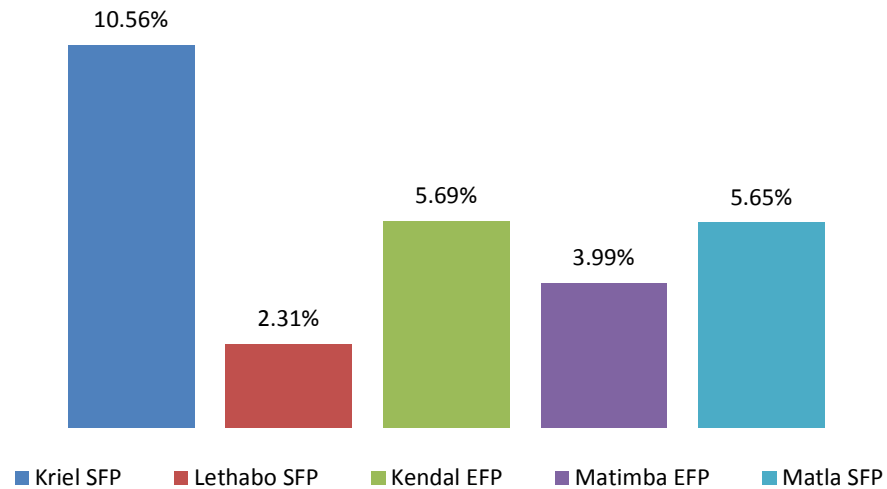


Figure 68: Volumetric efficiency comparison for power stations

The change in volumetric efficiency is shown in Figure 69 as a percentage from new to when the internal clearances have reached 80,000 running hours. The Lethabo SFP displays the least amount of change in volumetric efficiency and starting at a high efficiency over its life cycle it operates the most effectively over its operating life.



**Figure 69: Power station volumetric efficiency change as a percentage from new clearances to 80,000 running hours**

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## 7.2 THE ANNULAR SEAL LEAKAGE MODELLING

The flow rate around the impellers has been modelled, according to the equations in Section 5.1.1 (p21), for the BFPs at each power station and the results have been detailed in the graphs below for both the front and rear shroud of the impeller to compare the behaviour of the BFPs for each of the power stations.

The factors that influence the leakage flow rate are:

- Pressure across the stage
- Degree of reaction
- Impeller outside diameter
- Geometry of the seal
- Impeller speed
- Rotation factor
- Seal surface
- Loss coefficients.

All these have an influence on the amount of leakage through the seal, where some have a greater influence than others.

All the BFPs (Sulzer design) except Matimba have a straight annular seal on the suction impeller, with a larger inlet diameter to accommodate NPSH requirements. As a result the annular seal is at a larger diameter which has an impact on the flow rate.

The seal roughness can have a significant effect on the amount of leakage through the seal, but there are practical limitations.<sup>60</sup>

The Matimba 1<sup>st</sup> and 2<sup>nd</sup> stage impellers have been included in the suction impeller graphs as they have the same configuration, with the 3<sup>rd</sup> stage impeller included in the 2<sup>nd</sup> to last stage impeller graphs.

Figures 70 to 74 show the differences of each power station for the leakage flow rate through the annular seals. The factors that influence the leakage flow in each design are evident in these graphs with varying diametric clearances.

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<sup>60</sup> Effect of seal roughness can be seen in section 7.3 (p 98).

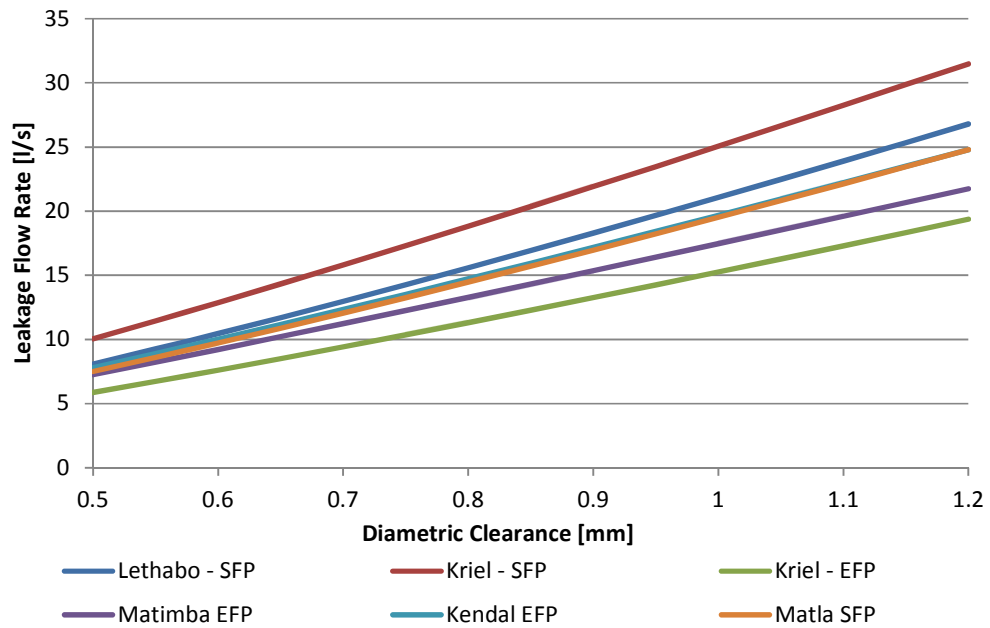


Figure 70: All power stations leakage around suction impeller – front shroud

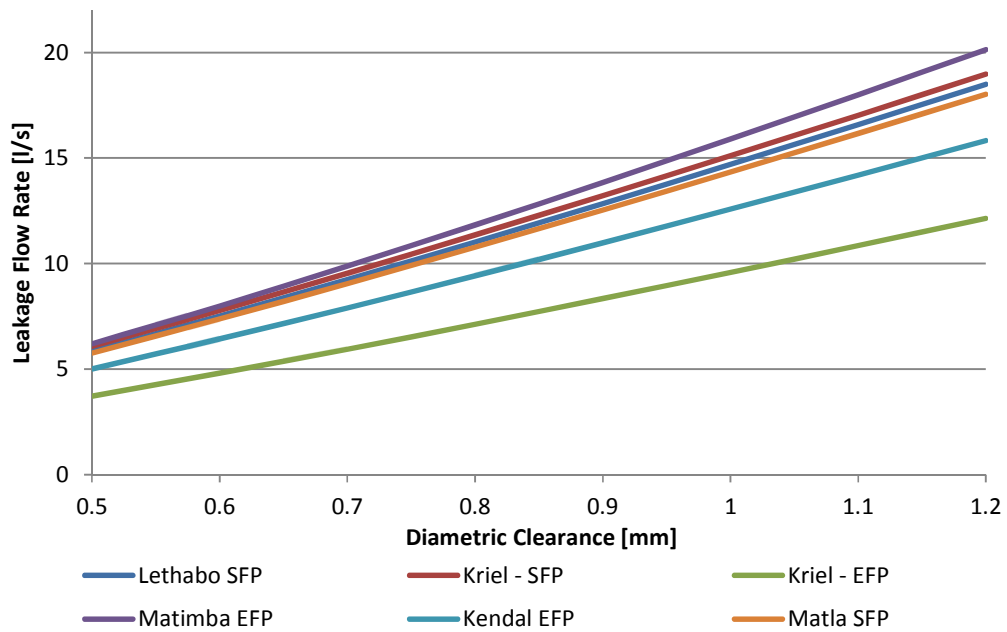
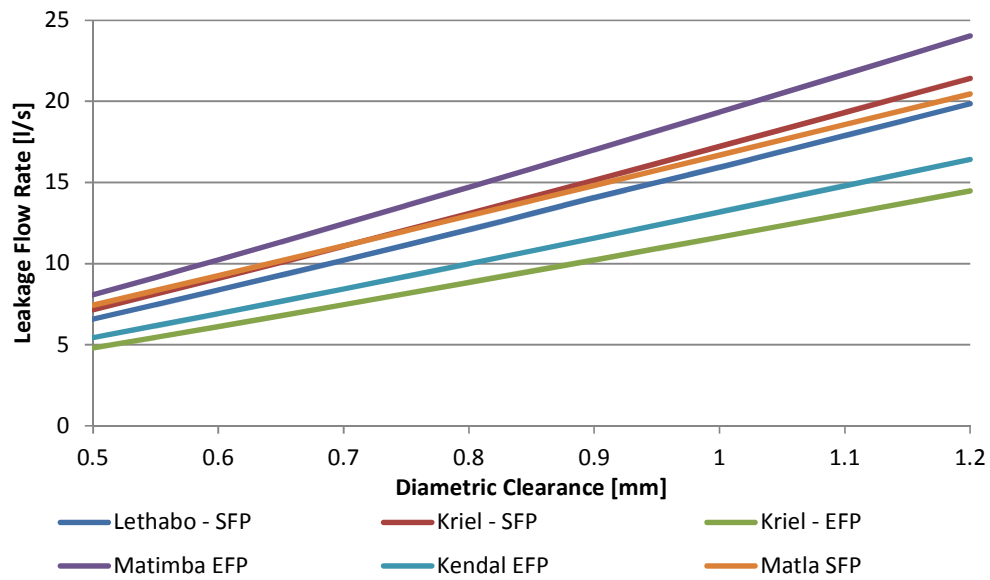
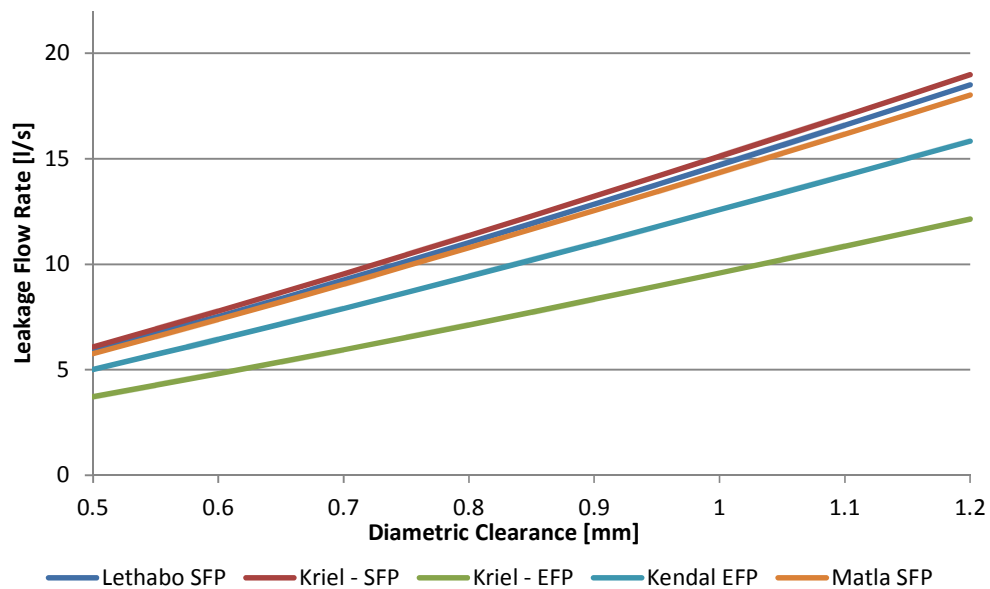


Figure 71: All power stations leakage around suction impeller – rear shroud

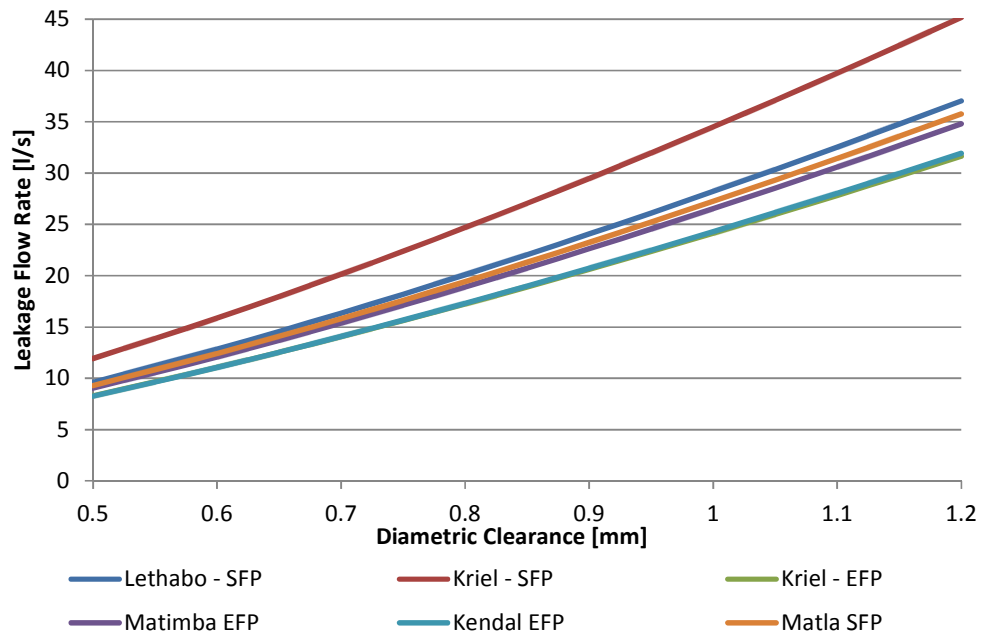


**Figure 72: All power stations leakage around 2<sup>nd</sup> – last stage impeller – front shroud**

The leakage for the Matimba last stage impeller rear shroud has not been included as the flow around the rear shroud of the last stage goes through the axial thrust balance device.



**Figure 73: All power stations leakage around 2<sup>nd</sup> – last stage impeller – rear shroud**



**Figure 74: All power stations leakage around axial thrust balance device**

The fluid force moments become destabilising and the swirl in the rear shroud clearance increases with increasing leakage flow (Yoshiki Yoshida, 1997). Considering this instability in rotor dynamics at higher clearances, this can possibly be a contributing factor to severe damage in BFPs when they have experienced high wear.

With BFPs having been run for periods that exceed OEM specifications this could be a direct result of the catastrophic failures seen at the power stations.

### 7.3 ANNULAR SEAL ROUGHNESS

The seal roughness has an important impact on the amount of leakage flow and thus impacts on the efficiency of the BFP (D. Rame Godwa, 2001).

To show the influence, the Lethabo SFP has been modelled and the seal surface has been altered to see the effect it has on the leakage rate.

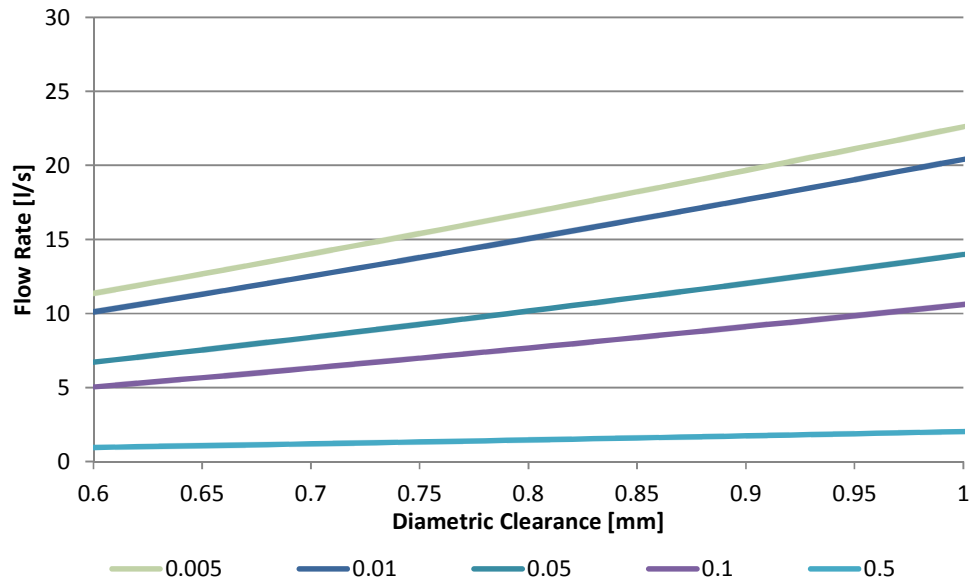
$$A = \frac{\varepsilon}{s} \quad (20)$$

A new diametric clearance was considered as 0.6mm and a worn diametric clearance was taken as 1mm as tabulated below.

**Table 25: Seal surface value (A) with varying roughness classes (Gülich, 2008)**

Roughness Class	Equivalent Sand Roughness $\varepsilon$ [ $\mu\text{m}$ ]	New Radial Clearance ( $s = 0.3\text{mm}$ ) [A]	Worn Radial Clearance ( $s = 0.5\text{mm}$ ) [A]
N5	1	0.00333	0.002
N6	2	0.00667	0.004
N7	4	0.01333	0.008
N8	8	0.02667	0.016
N9	16	0.05333	0.032
N10	32	0.10667	0.064
N11	64	0.21333	0.128
N12	128	0.42667	0.256
N13	256	0.85333	0.512

Figure 75 details the effect of altering the seal surface roughness of the Lethabo SFP, bearing in mind this is a theoretical approach and in practice these values are not achievable.



**Figure 75: Effect of annular seal surface on leakage showing various seal roughness's (A)**

Typically the wear rings and impellers after machining have a surface finish of N7 which would result in a seal roughness coefficient (A) of between 0.0133 when new and 0.008 when worn.

The seal surface roughness cannot have a value of 0.5 as practically it cannot be achieved without influencing the integrity of the damping in the annular seals and ensuring the galling effect of the two surfaces.

Von Pragenau found that the Couette factor, i.e. the tangential velocity, must be reduced in order to reach higher stability limits and better damping. The Couette factor can be reduced with higher surface roughness at the stator than at the rotor. In such a configuration, friction forces at the stator become larger and the mean tangential velocity in the annular seal is reduced (Florjancic, 1990)<sup>61</sup>.

It is evident that by having a stepped annular seal in the 2<sup>nd</sup> – last stage impeller, which is the case in most of the designs, the leakage rate can be reduced.

What is most critical for the seal configuration is the effect it will have on the rotordynamics as it is operating at high speeds.

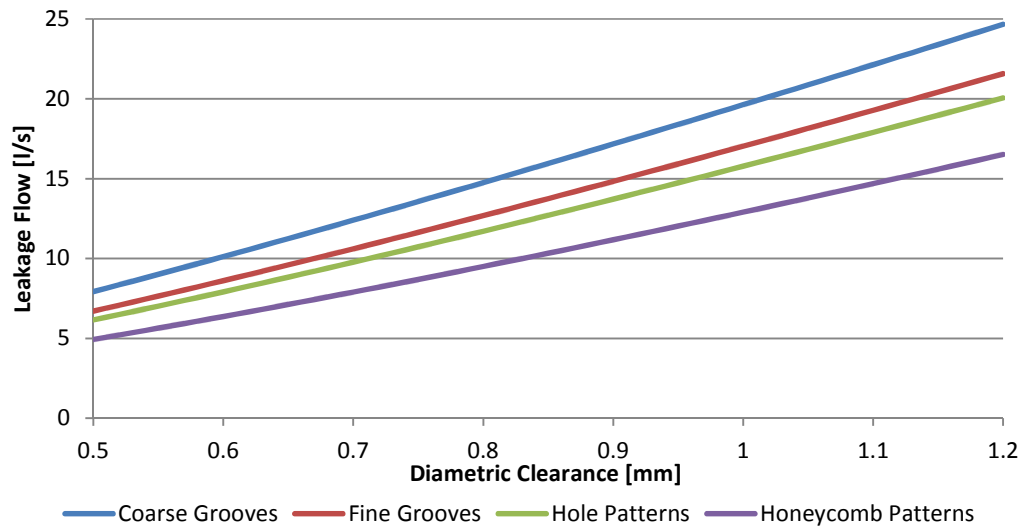
<sup>61</sup> Page 16 of Florjancic dissertation

## 7.4 ANNULAR SEAL GEOMETRY

The geometry of the annular seal surface has an effect on the friction coefficient by either having coarse grooves, fine grooves, hole patterns or a honeycomb pattern. This can be done as shown in the previous section by altering the seal roughness, but can also be influenced by altering the grooves and pattern of the seal surface.

The graph in Figure 76 shows how the leakage flow is reduced by altering the geometry.

The type of annular seal<sup>62</sup> used effects the leakage flow as seen for each power station with a straight annular seal and stepped annular seal.



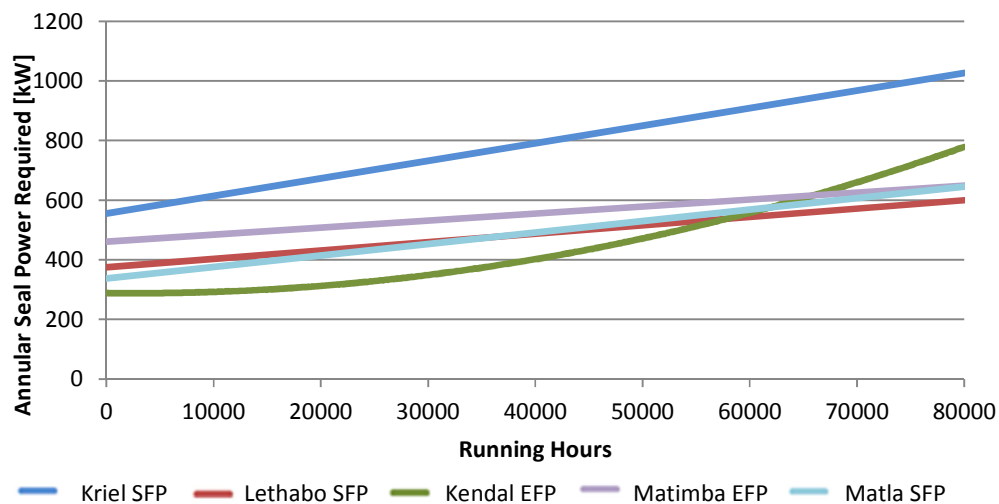
**Figure 76: Lethabo SFP front shroud influence of annular seal surface geometry**

<sup>62</sup> See Figure 5 (p 9)

## 7.5 POWER CONSUMED AS A RESULT OF LEAKAGE THROUGH ANNULAR SEALS

The leakage flow through the annular seals requires power for the demineralised water to be circulated within the BFP and as a result uses additional power which contributes to the inefficiencies within the BFP.

Figure 77 shows how the power required increases as the BFP wears internally, thus requiring more power as a result of the changes in internal leakage. The different gradients and shapes of the various BFPs are a result of different wear behaviours as well as the different dimensional configuration within the BFP which will affect the behaviour of the internal leakage.

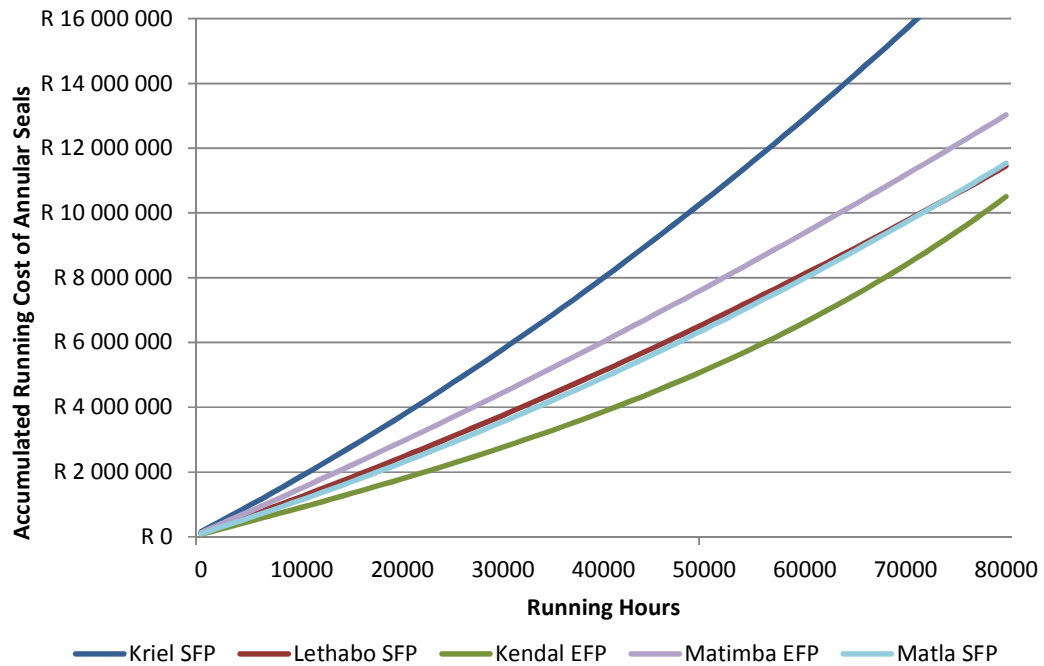


**Figure 77: Change in power consumption as a result of internal wear**

Matimba has the flattest gradient as a result of the least internal wear but power consumption is high when it starts running relative to the other BFPs due to the design of the advanced class BFP design. Kendal has an exponential behaviour as a result of the balancing device which showed such behaviour, but with the lowest consumption of power when the EFP is new. Kriel has a steep gradient as a result of the high internal wear rate that is experienced. This is possibly caused by the design being a 5-stage SFP, more flexible shaft and less robust.

Matimba, Lethabo and Matla experience very similar behaviour, but Matla and Lethabo are SFPs and Matimba is an EFP.

As a BFP starts to wear internally the clearances between the rotating and stationary parts increase, resulting in higher internal flow leakage which requires more power, as seen in the graph in Figure 77. This increased power consumption has a generation cost which is accumulated for each hour run in Figure 78.

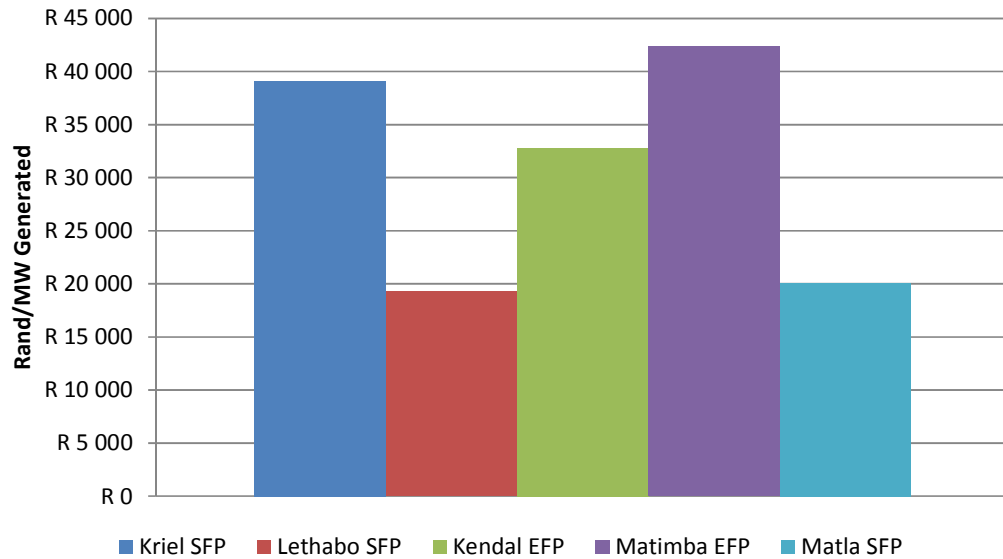


**Figure 78: Accumulated additional energy costs as a result of annular seal leakage with internal wear**

The comparison of each power stations accumulated additional energy costs as a result of internal wear in the BFPs has been made for five different power stations, of which three power stations are running one SFP and two power stations are running two EFPs to maintain full load on each unit. In order to compare the power consumption of one power station relative to another the amount of energy that is being generated per BFP system needs to be considered.

It requires two EFPs to produce the maximum load in each unit at the power station, which means the two EFPs need to be compared to one SFP. Then comparing the auxiliary power used by the BFPs in comparison with the output of that unit in the power station to have a true comparison.

Figure 79 below represents the energy cost relative to the output power produced by the unit. This gives a cost of the power required for the annular seals relative to the amount of power generated at the power station.



**Figure 79: Cost of annular seal leakage relative to the power generated**

From the graph above it can be seen that the BFPs that are the least expensive to run per unit of electricity generated are Lethabo and Matla, which are similarly designed SFPs and have been operated under similar conditions.

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## 8. CONCLUSIONS

The clearances between rotating and stationary components of a BFP are designed as small as possible to limit the leakage through the annular seals as the flow rate through the seal is directly proportional to the power consumed.

It is evident from the data collected on recorded clearances that the data would have been more accurate having had available the exact clearances for each specific BFP when they were assembled. This resulted in assumptions and averages for “new” clearances.

Over the operating life of the BFP abrasive wear occurs internally as a result of deflections on the rotor, starting and stopping of the BFPs, and thermal gradients created in the BFPs. The wear increases the clearances between rotating and stationary parts, which increases the flow through the seals.

Starting and stopping of the BFPs is considered to be the most influential of the wear in the annular seals, which is measured with a comparison between EFPs and SFPs on the same power station<sup>63</sup>. The design of the feed pumps shaft rotordynamics influences the amount of wear on each start-up, and depending on whether it is a hot, warm or cold start-up also has an influence.

The demineralised water flows through the annular seals with a high velocity and the evidence of abrasion on the annular seal surface is not seen as a major contributory factor to the increase in clearances.

As the flow rate through the annular seals increases, as a result of the internal wear, the efficiency of the BFP decreases and the power consumed increases. The cost of additional power consumed is valued at the generation cost Eskom uses to produce the power required to drive the BFPs. With the current shortage of power supply in South Africa, the relative energy cost can be increased significantly during the peak periods as the peaking power stations would be required to supply less power as a result of increased capacity. The cost is therefore conservative and would increase the energy cost, and ultimately reduce the refurbishment intervals.

The on-site testing that was carried out on the BFPs at Lethabo, Kriel and Matimba power stations showed that the uncertainties in the testing method cannot be tested to the accuracy required. An on-site test will give evidence of major issues internally in the BFP, but will not be able to predict the changes as a result of internal wear.

Prediction of the leakage flow through the annular seals using Equation 17 has some uncertainty, but the validation by Sulzer Pumps (Winterthur) has shown that the calculated values are accurate when looking at the change from new to worn.

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<sup>63</sup> This is evident in the wear data for Kriel and Matla comparing SFPs and EFPs

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Each power station in this study has been analysed individually, which has produced refurbishment intervals specific to that power station which can be used to determine the lifecycle cost of every power station.

By optimising the refurbishment intervals there is a benefit of the life cycle cost of the BFPs at the power station and the additional power that can be generated continuously will add to the capacity of the Eskom generating fleet.

It is evident that the BFPs with stiff, robust shafts<sup>64</sup> which limit the deflection, reduce the wear on the annular seal over the life of the BFPs.

Implementing the recommended refurbishment intervals on the BFPs will increase the reliability and availability of the BFPs.

Refurbishment intervals can be given to each BFP at the various power stations depending on the cost of refurbishment and cost of consumed energy. The effect of the internal wear is predicted through the change in annular seal leakage flow.

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<sup>64</sup> Matimba EFP design

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## 9. RECOMMENDATIONS

Understanding the lifecycle management of BFPs has not been an area of substantial concern in the history of power generation. After numerous circumstances of UCLF previously the major focus was on ensuring the availability of the power stations. Research and development was done on annular seals with the area of concentration on the influence of the annular seals on the rotordynamics of the pumps.

In order to effectively monitor the deterioration, numerous factors need to be recorded over many years of operation and during refurbishment intervals.

The on-site testing that was conducted demonstrated that the on-site performance test could be accomplished with certain limitations. In order to assess the deterioration of the wear components in this study, on-site testing would not be able to be done accurately enough to assess the change in BFP efficiency. On-site testing should only be conducted when major concerns are raised regarding the performance of a specific BFP.

It was found that the errors in measurements recorded would be a constant error if recorded with a starting point. The change in recorded values of specific parameters would be essential to the online monitoring of the BFPs.

Areas that need to be monitored are:

- Balance water return line pressure, flow and temperature
- Speed of BFPs from installation to cartridge exchange
- Traceability of the exchange of barrel cartridges
- Records of annular seal clearances after refurbishment
- Records of annular seal clearance during inspection
- Starts/Stops of the BFP over the operating life
- Running and installed hours of the BFP.

The collaboration that was built between Eskom and Sulzer Pumps head office through the time<sup>65</sup> in Winterthur, Switzerland is essential to the advancements required in BFP technologies.

An essential part for Eskom on all plant is to give feedback to the OEM to ensure that the developments are in line with the difficulties experienced at the power stations. A communication channel is needed through OEMs in South Africa to give feedback to the design authority in order to assess each topic of interest when it is raised.

The implementation of the project to install a test rig at a power station for BFP coating development is necessary before Eskom will install any coatings on

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<sup>65</sup> Sulzer Pumps visit to Winterthur, Switzerland in November 2013

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annular seals. The surface treatment, if successful under the operating conditions, will reduce the wear rate of the annular seals and prolong the time between refurbishment intervals.

Further analysis is necessary on materials and seal geometries used in BFP wear rings and impellers. Ensuring an isotropic hardness in the wear rings is important and the quality controls to ensure this need to be effective.

The minimum hardness difference of 100 HBN is needed between the wear ring and the impeller, unless materials that meet all the necessary requirements of 450 BHN are introduced.

With power generation expected to grow considerably in the next two decades, fossil fuels will still be the most important source of energy specifically in South Africa. BFPs are critical to the power generation process and to manage the correct refurbishment intervals through lifecycle management, Eskom need to ensure the refurbishment intervals for each power station are implemented in the maintenance programs.

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## 10. BIBLIOGRAPHY

- Eskom Power Price*. (2013). Retrieved December 10, 2013, from Official POWI website: <http://powi.eskom.co.za/~powi/>
- API 610. (2004). *Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries*. Washington, D.C: API Publishing Services.
- Cadle, J., Catteart, T., & van der Westhuizen, W. (2008). Partners in Success. *Service Agreement for Boiler Feed Pumps*, 4.
- Caldo, F. (February 2008). *Eskom's Power Crisis: Reasons, impact & possible solutions*. Centurion: Solidarity.
- D. Rame Godwa, B. P. (2001). *High-Pressure Annular Seal Leakage and Rotordynamics with Application to Turbomachinery*. Chennai: Taylor and Francis.
- Dooley, E. -R. (March 2005). *Cycle Chemistry Guidelines for Fossil Plants: Oxygenated Treatment*. Willow Way, Concord, CA: Electric Power Research Institute.
- Electric Power Research Institute. (2005). *Cycle Chemistry Guidelines for Fossil Plants: Oxygenated Treatment (1004925)*. Palo Alto, CA: EPRI.
- Energy Research and Consultants Corporation. (1978). *Survey of Feed Pump Outages*. California: EPRI.
- EPRI. (1994). *Boiler Feed Pump Operation and Maintenance Guidelines*. Columbia, Maryland: GPS TECHNOLOGIES, INC.
- Eskom Holdings Limited. (c2014). *Eskom - South African Power Utility*. Retrieved October 4, 2012, from Power Stations and Pumped Storage Schemes: [http://www.eskom.co.za/Whatweredoing/ElectricityGeneration/PowerStations/Pages/Power\\_Stations\\_And\\_Pumped\\_Storage\\_Schemes.aspx](http://www.eskom.co.za/Whatweredoing/ElectricityGeneration/PowerStations/Pages/Power_Stations_And_Pumped_Storage_Schemes.aspx)

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- Florjancic, S. (1990). *Annular Seal of High Energy Centrifugal Pumps: A New Theory and Full Scale Measurement of Rotordynamic Coefficients and Hydraulic Friction Factors*. Zurich: Swiss Federal Institute of Technology Zurich.
- G. Ludwig, S. M. (2003). *Design Factors Affecting Pump Efficiency*. Berlin - Heidelberg: Springer-Verlag.
- Geis, H. (1985). *Experimentelle Untersuchungen der Radseitenverluste von Hochdruck*. TH Darmstadt.
- Gerald L. Morrison, M. C. (2000). *Comparison of the Flow Inside Whirling and Non-Whirling Smooth Annular Seals at 50% Eccentricity*. Texas: Taylor and Francis.
- Gülich, D. J. (2008). *Centrifugal Pumps*. Verlag Berlin Heidelberg: Springer.
- Hirs, G. (1970). *Fundamentals of a Bulk-Flow Theory for Turbulent Lubricant Films*.
- Inspectors, S. P. (2012, October 16). BFP Inspections. (S. Mortlock, Interviewer)
- Schneckenberg, E. (1932). *Der Durchfluss durch Drosselspalte*. Z. Ver. deut, Ing. ASME pg 65.
- Stepanhoff, A. J. (1957). *Centrifugal and Axial Flow Pumps, 2nd Edition*. John Wiley and Sons, Inc.
- Sulzer Pumps. (n.d.). *Sulzer Pumps*. Retrieved November 14, 2013, from Sulzer Products and Services: <http://www.sulzer.com/en/Products-and-Services/Pumps-and-Systems/Barrel-Casing-Pumps/HPT>
- System Engineers. (2013). *Eskom Power Stations BFP Running Hour Records*.
- Wolfgang Miller. (2013). *BFP Refurbishment according to Partnering Contract*.
- Y. Hsu, C. B. (2002). *Fluid Flow Equations for Rotordynamic Flows in Seals and Leakage Paths*. Pasadena, CA: ASME.

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Yoshiki Yoshida, Y. T. (1997). *Measurements of LFow in Backshroud/Casing*

*Clearance of Precessing Centrifugal Impeller*. India: OPA (Overseas Publishers Association).

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## APPENDICES

## APPENDIX A - LETAHBO POWER STATION

### A.1. OPERATING POINTS

**Table 26: Lethabo SFP operating data**

Parameter	Most Economical Load	Design Point	Units
Feed Water Tank Pressure	0.652	0.655	MPa
Feed Water Tank Temperature	162.1	162.3	°C
Feed Water Tank Density	904.9	904.9	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	504.52	562.25	kg/s
Pressure at Inlet to Booster Pump	0.823	0.813	MPa
Manometric Head – Booster Pump	0.9226	1.002	MPa
Pressure at Outlet to Booster Pump	1.746	1.815	MPa
Pressure at Inlet Connection of Main Pump	1.715	1.780	MPa
Mass Flow at Extraction Connection	9.39	27	kg/s
Pressure at Extraction Connection	20.995	11.648	MPa
Mass Flow at Delivery Connection of Main Pump	495.135	535.25	kg/s
Manometric Head – Main Pump	20.19	21.026	MPa
Pressure at Delivery Connection of Main Pump	20.995	22.803	MPa
Speed of Main Pump (Take-Off Open)	5120	5395	rpm
Speed of Main Pump (Take-Off Closed)	5064	5352	rpm
Power Required at Pump Shaft (Take-Off Open)	12908	15530	kW
Power Required at Pump Shaft (Take-Off Closed)	12775	15093	kW
Main Pump Efficiency (On-site)	*84.1	82.4	%
Main Pump Specific Speed	33	33	

\*Karassik

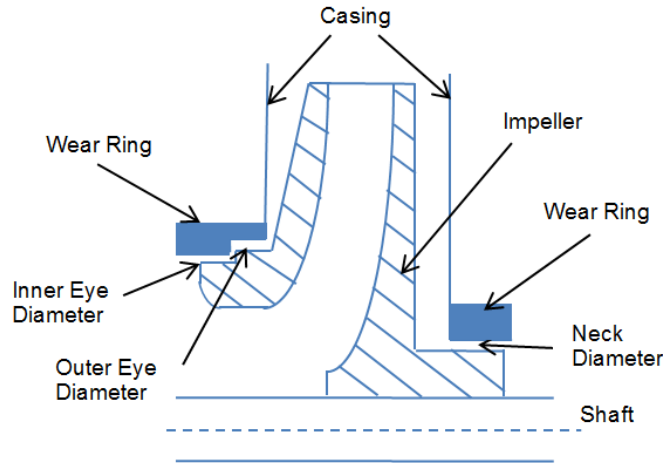
**Table 27: Lethabo EFP operating data**

Parameter	Most Economical Load	Design Point	Units
Feed Water Tank Pressure	0.652	0.655	MPa
Feed Water Tank Temperature	162.1	162.3	°C
Feed Water Tank Density	904.9	904.9	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	252.26	281.13	kg/s
Pressure at Inlet to Booster Pump	0.81	0.7927	MPa
Manometric Head – Booster Pump	1.209	1.180	MPa
Pressure at Outlet to Booster Pump	2.02	1.973	MPa
Pressure at Inlet Connection of Main Pump	1.983	1.938	MPa
Mass Flow at Extraction Connection	5.43	13.5	kg/s
Pressure at Extraction Connection	9.30	9.445	MPa
Mass Flow at Delivery Connection of Main Pump	247.57	267.63	kg/s
Manometric Head – Main Pump	19.003	20.857	MPa
Pressure at Delivery Connection of Main Pump	20.995	22.803	MPa
Speed of Main Pump (TOO)	5940	6282	rpm
Speed of Main Pump (TOC)	5901	6240	rpm
Power Required at Pump Shaft (TOO)	6455	7790	kW
Power Required at Pump Shaft (TOC)	6407	7636	kW
Main Pump Efficiency (On-site)	*83.1	81	%
Main Pump Specific Speed			

\*Karassik

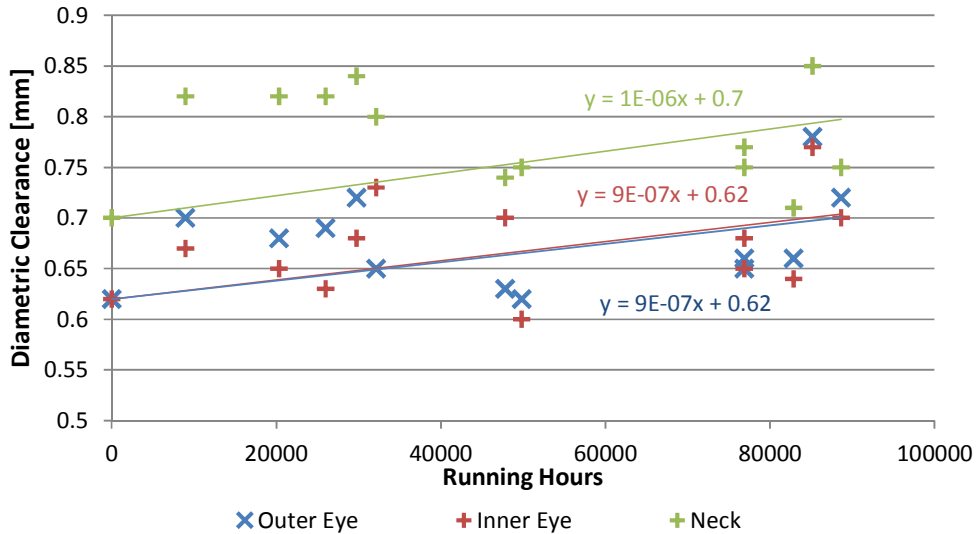
**A.2. 2<sup>ND</sup> – LAST STAGE IMPELLER – WEAR DATA**

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as shown in Figure 80, with diameters of 284.45 mm h7 and 264.45 mm h7 on the eye diameters and 199.3 mm h7 on the neck diameters when manufactured.



**Figure 80: Lethabo SFP 2<sup>nd</sup> – Last Stage Sectional View**

Figure 81 to 83 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.



**Figure 81: Lethabo 2<sup>nd</sup> stage impeller wear data**

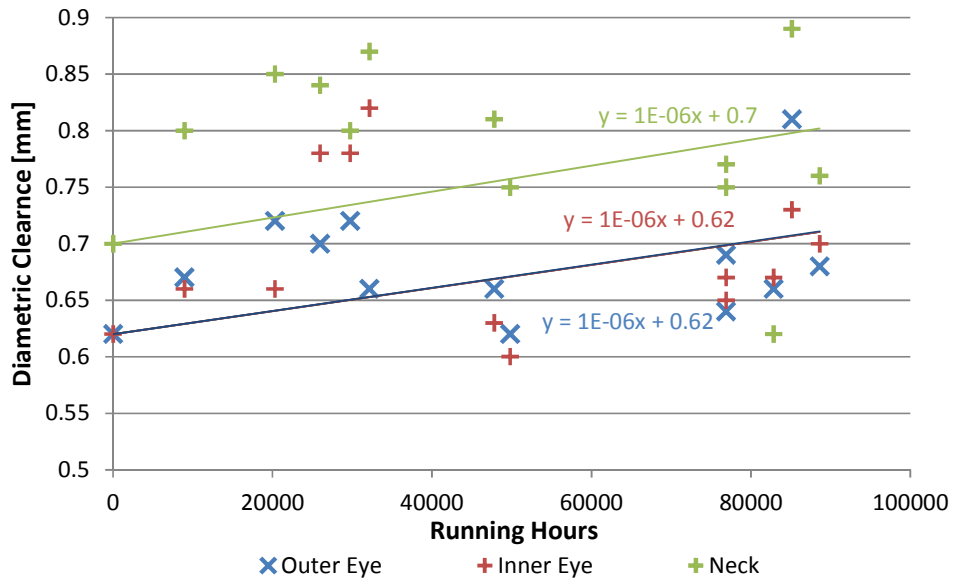


Figure 82: Lethabo 3<sup>rd</sup> stage impeller wear data

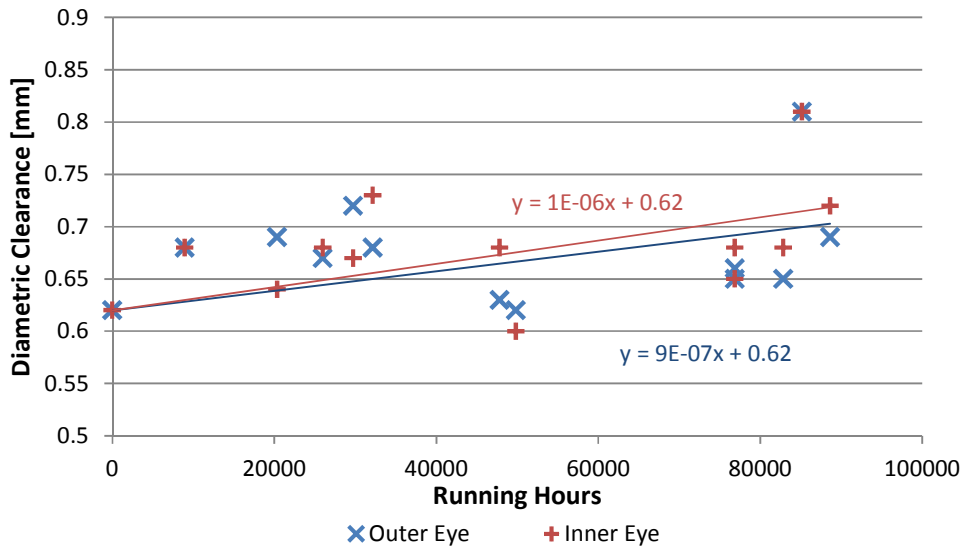


Figure 83: Lethabo last stage impeller wear data

### A.3. POWER CONSUMED THROUGH ANNULAR SEALS

Table 28: Lethabo SFP – Power consumed through annular seals

Power Consumed at a Specific Clearance [kW]									
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		Last Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	
0.5	40.74	5.24	37.71	5.24	37.71	5.24	37.71	0	139.05
0.51	41.89	5.38	39.33	5.38	39.33	5.38	39.33	0	143.21
0.52	43.06	5.53	40.93	5.53	40.93	5.53	40.93	0	147.41
0.53	44.23	5.68	42.53	5.68	42.53	5.68	42.53	0	151.64
0.54	45.40	5.83	44.12	5.83	44.12	5.83	44.12	0	155.91
0.55	46.59	5.98	45.70	5.98	45.70	5.98	45.70	0	160.22
0.56	47.78	6.13	47.29	6.13	47.29	6.13	47.29	0	164.57
0.57	48.98	6.29	48.87	6.29	48.87	6.29	48.87	0	168.96
0.58	50.19	6.44	50.44	6.44	50.44	6.44	50.44	0	173.38
0.59	51.41	6.60	52.01	6.60	52.01	6.60	52.01	0	177.84
0.6	52.63	6.75	53.58	6.75	53.58	6.75	53.58	0	182.33
0.61	53.86	6.91	55.15	6.91	55.15	6.91	55.15	0	186.87
0.62	55.09	7.07	56.72	7.07	56.72	7.07	56.72	0	191.43
0.63	56.34	7.23	58.29	7.23	58.29	7.23	58.29	0	196.03
0.64	57.59	7.38	59.85	7.38	59.85	7.38	59.85	0	200.67
0.65	58.84	7.54	61.42	7.54	61.42	7.54	61.42	0	205.34
0.66	60.11	7.71	62.98	7.71	62.98	7.71	62.98	0	210.04
0.67	61.38	7.87	64.54	7.87	64.54	7.87	64.54	0	214.78
0.68	62.65	8.03	66.11	8.03	66.11	8.03	66.11	0	219.55
0.69	63.93	8.19	67.67	8.19	67.67	8.19	67.67	0	224.36
0.7	65.22	8.36	69.24	8.36	69.24	8.36	69.24	0	229.20
0.71	66.51	8.52	70.80	8.52	70.80	8.52	70.80	0	234.07
0.72	67.81	8.69	72.37	8.69	72.37	8.69	72.37	0	238.97
0.73	69.12	8.85	73.93	8.85	73.93	8.85	73.93	0	243.91
0.74	70.43	9.02	75.50	9.02	75.50	9.02	75.50	0	248.87
0.75	71.74	9.19	77.07	9.19	77.07	9.19	77.07	0	253.87
0.76	73.06	9.36	78.63	9.36	78.63	9.36	78.63	0	258.90
0.77	74.39	9.52	80.20	9.52	80.20	9.52	80.20	0	263.96
0.78	75.72	9.69	81.78	9.69	81.78	9.69	81.78	0	269.05
0.79	77.06	9.86	83.35	9.86	83.35	9.86	83.35	0	274.18
0.8	78.40	10.03	84.92	10.03	84.92	10.03	84.92	0	279.33
0.81	79.75	10.21	86.49	10.21	86.49	10.21	86.49	0	284.51
0.82	81.11	10.38	88.07	10.38	88.07	10.38	88.07	0	289.73
0.83	82.46	10.55	89.65	10.55	89.65	10.55	89.65	0	294.97

0.84	83.83	10.72	91.23	10.72	91.23	10.72	91.23	0	300.24
0.85	85.19	10.90	92.81	10.90	92.81	10.90	92.81	0	305.54
0.86	86.57	11.07	94.39	11.07	94.39	11.07	94.39	0	310.88
0.87	87.94	11.25	95.97	11.25	95.97	11.25	95.97	0	316.24
0.88	89.33	11.42	97.55	11.42	97.55	11.42	97.55	0	321.62
0.89	90.71	11.60	99.14	11.60	99.14	11.60	99.14	0	327.04
0.9	92.10	11.78	100.73	11.78	100.73	11.78	100.73	0	332.49
0.91	93.50	11.95	102.31	11.95	102.31	11.95	102.31	0	337.96
0.92	94.90	12.13	103.90	12.13	103.90	12.13	103.90	0	343.46
0.93	96.30	12.31	105.50	12.31	105.50	12.31	105.50	0	348.99
0.94	97.71	12.49	107.09	12.49	107.09	12.49	107.09	0	354.55
0.95	99.13	12.67	108.68	12.67	108.68	12.67	108.68	0	360.13
0.96	100.54	12.85	110.28	12.85	110.28	12.85	110.28	0	365.74
0.97	101.96	13.03	111.87	13.03	111.87	13.03	111.87	0	371.38
0.98	103.39	13.21	113.47	13.21	113.47	13.21	113.47	0	377.05
0.99	104.82	13.39	115.07	13.39	115.07	13.39	115.07	0	382.74
1	106.25	13.58	116.67	13.58	116.67	13.58	116.67	0	388.46
1.01	107.69	13.76	118.28	13.76	118.28	13.76	118.28	0	394.20
1.02	109.13	13.94	119.88	13.94	119.88	13.94	119.88	0	399.97
1.03	110.57	14.13	121.49	14.13	121.49	14.13	121.49	0	405.77
1.04	112.02	14.31	123.09	14.31	123.09	14.31	123.09	0	411.59
1.05	113.48	14.49	124.70	14.49	124.70	14.49	124.70	0	417.44
1.06	114.93	14.68	126.31	14.68	126.31	14.68	126.31	0	423.31
1.07	116.39	14.86	127.92	14.86	127.92	14.86	127.92	0	429.21
1.08	117.86	15.05	129.53	15.05	129.53	15.05	129.53	0	435.14
1.09	119.32	15.24	131.15	15.24	131.15	15.24	131.15	0	441.09
1.1	120.79	15.42	132.76	15.42	132.76	15.42	132.76	0	447.06

## A.4. DATA FROM PERFORMANCE TESTING ON-SITE

On-site Performance Testing							
Date of Test		16/07/2012	26/07/2012	17/09/2002	17/09/2002	21/05/2001	21/05/2001
Running Hours		30393	36322	42528	42528	36777	36777
		1	2	3	4	5	6
Unit Load	MW	630	613	620	620	620	620
Unit number		4	6	6	6	1	1
Pump speed	RPM	5050	4960	5000	4970	5060	5059
Booster pump inlet pressure	kPa	879.5	855.7	831.5	832.9	837.4	842.2
Booster pump discharge pressure	kPa	1786.8	1731.6	1800.4	1800.7	1822	1825
Booster Pump Manometric Head	mWg			200.6	200.7	202.8	203.2
Main pump inlet pressure	kPa	1786.8	1731.6	1770.9	1771.7	1790	1796
Main pump inlet pressure	bar			17.709	17.717	17.9	17.96
Main pump discharge pressure	kPa	21418	20790	20960	20958	21114	21117
Main pump discharge pressure	bar			209.6	209.58	211.14	211.17
Main Pump Manometric Head	mWg	2192	2115.0	2135	2134.7	2147.5	2147.5
Main + Booster Pump Manometric Head	mWg	2295	2214	2240.8	2240.4	2272	2273
Feed pump Flow Rate	l/s			539.1	541.2	558	558
Booster Pump inlet temperature	°C	162	161.15	160.69	160.75	160.89	161.23
Main discharge temperature	°C	165.63	164.69	164.18	164.24	164.41	164.86
Temperature rise	°C	3.63	3.54	3.49	3.49	3.52	3.63
Inlet feed water flow rate	kg/s	495.5	523.8	491.1	491.8	506.1	506.1
Discharge feed water flow rate	kg/s			479.9	480.2	495.3	495.3
Attemperator spray water flow	kg/s			11.2	11.6	10.8	10.8
Feed water inlet enthalpy	kJ/kg	684.31	680.1	678.6	678.86	680.1	681.59514
Feed water outlet enthalpy	kJ/kg	712.05	707.10	705.56	705.82	706.69	708.62
Feed water enthalpy rise	kJ	27.74	27.00	26.96	26.96	26.57	27.02
Feed pump output power	kW	11091	10685	10672	10681	10655	10656
Feed pump input power	kW	13988	13230	13089	13102	12990	13204
Feed pump efficiency	%	79.3	80.8	81.5	81.52	82.0	80.7

## APPENDIX B - – KENDAL POWER STATION

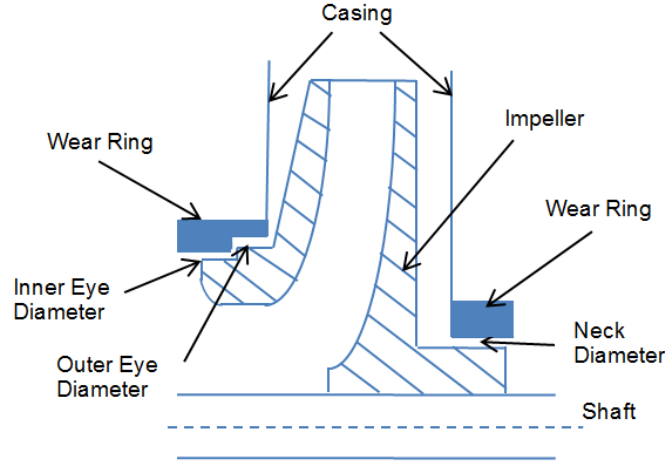
### B.1. OPERATING POINTS

**Table 29: Kendal EFP operating data**

Parameter	Most Economical Load	Design Point	Units
Feed water Tank Pressure	0.857	0.754	MPa
Feed water Tank Temperature	173.3	168	°C
Feed water Tank Density	894	899.5	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	282.5	363.4	kg/s
Pressure at Inlet to Booster Pump	0.944	0.842	MPa
Manometric Head – Booster Pump	1.34	1.49	MPa
Pressure at Outlet to Booster Pump	2.28	2.33	MPa
Pressure at Inlet Connection of Main Pump	2.27	2.31	MPa
Mass Flow at Extraction Connection	2.5	24	kg/s
Pressure at Extraction Connection	11.16	11.23	MPa
Mass Flow at Delivery Connection of Main Pump	280	339.4	kg/s
Manometric Head – Main Pump	18.39	19.35	MPa
Pressure at Delivery Connection of Main Pump	20.66	21.66	MPa
Speed of Main Pump (TOO)	5483	5861	rpm
Power Required at Pump Shaft (TOO)	7102	9319	kW
Main Pump Efficiency (On-site) – TOO	81.6	80.8	%
Main Pump Efficiency (On-site) – TOC	81.5	81.6	%
Main Pump Specific Speed			

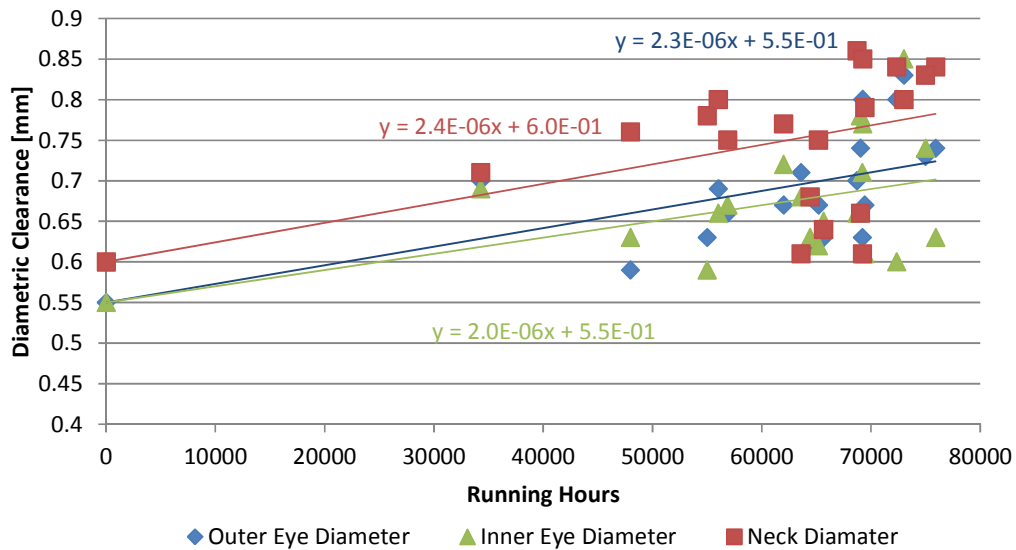
**B.2. 2<sup>ND</sup> – LAST STAGE IMPELLER WEAR DATA**

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as seen in Figure 84, with diameters of 244.53 mm h8 and 229.53 mm h8 on the eye diameters and 174.39 mm h8 on the neck diameters when manufactured



**Figure 84: Kendal EFP 2<sup>nd</sup> – last stage impeller sectional view**

Figure 85 to 87 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.



**Figure 85: Kendal 2<sup>nd</sup> stage impeller wear data**

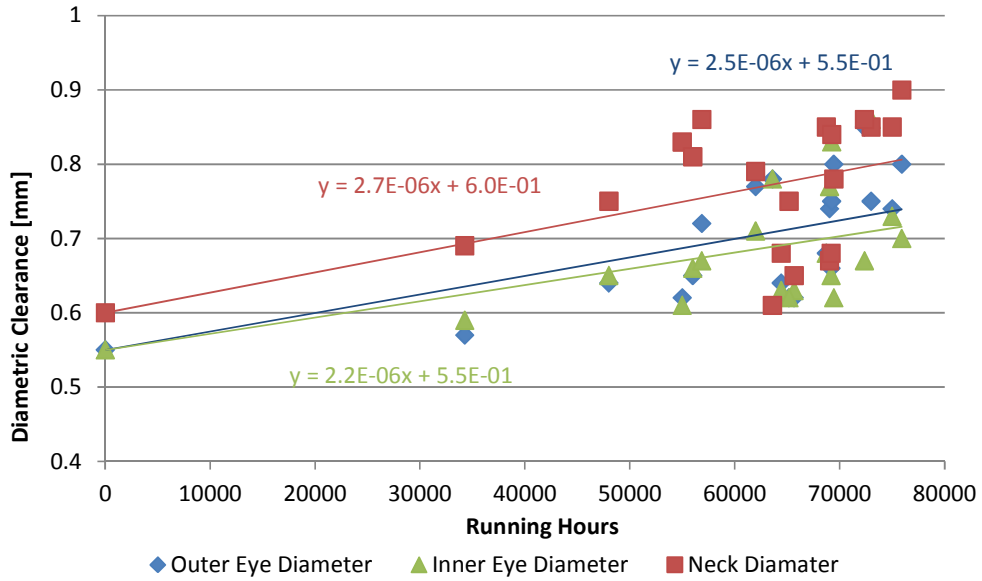


Figure 86: Kendal 3<sup>rd</sup> stage impeller wear data

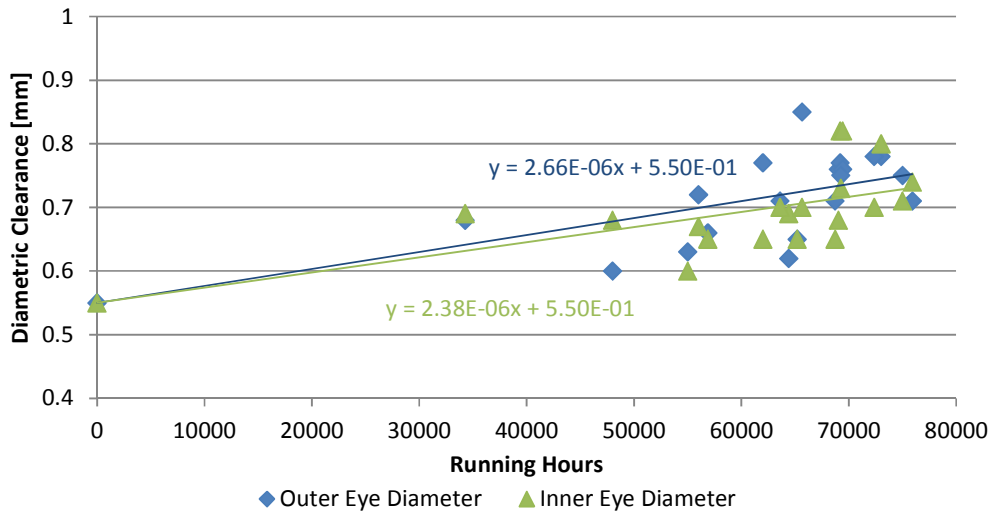


Figure 87: Kendal last stage impeller wear data

### B.3. POWER CONSUMED THROUGH ANNULAR SEALS

**Table 30: Kendal EFP – Power consumed through annular seals**

Power Consumed at a Specific Clearance [kW]									
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		Last Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	
0.5	35.7	4.5	37.5	5.3	37.5	5.3	37.5	0	111.2
0.51	36.7	4.6	38.8	5.5	38.8	5.5	38.8	0	114.6
0.52	37.7	4.7	40.1	5.6	40.1	5.6	40.1	0	117.9
0.53	38.7	4.8	41.4	5.7	41.4	5.7	41.4	0	121.3
0.54	39.7	5.0	42.7	5.8	42.7	5.8	42.7	0	124.7
0.55	40.7	5.1	44.0	6.0	44.0	6.0	44.0	0	128.2
0.56	41.7	5.2	45.3	6.1	45.3	6.1	45.3	0	131.6
0.57	42.8	5.3	46.6	6.2	46.6	6.2	46.6	0	135.1
0.58	43.8	5.5	47.8	6.4	47.8	6.4	47.8	0	138.7
0.59	44.8	5.6	49.1	6.5	49.1	6.5	49.1	0	142.2
0.6	45.9	5.7	50.4	6.6	50.4	6.6	50.4	0	145.8
0.61	47.0	5.9	51.6	6.7	51.6	6.7	51.6	0	149.5
0.62	48.0	6.0	52.9	6.9	52.9	6.9	52.9	0	153.1
0.63	49.1	6.1	54.2	7.0	54.2	7.0	54.2	0	156.8
0.64	50.2	6.3	55.4	7.1	55.4	7.1	55.4	0	160.5
0.65	51.2	6.4	56.7	7.3	56.7	7.3	56.7	0	164.2
0.66	52.3	6.5	57.9	7.4	57.9	7.4	57.9	0	168.0
0.67	53.4	6.7	59.2	7.5	59.2	7.5	59.2	0	171.8
0.68	54.5	6.8	60.5	7.7	60.5	7.7	60.5	0	175.6
0.69	55.6	6.9	61.7	7.8	61.7	7.8	61.7	0	179.5
0.7	56.7	7.1	63.0	7.9	63.0	7.9	63.0	0	183.3
0.71	57.8	7.2	64.2	8.1	64.2	8.1	64.2	0	187.2
0.72	58.9	7.4	65.5	8.2	65.5	8.2	65.5	0	191.1
0.73	60.0	7.5	66.7	8.4	66.7	8.4	66.7	0	195.1
0.74	61.1	7.6	68.0	8.5	68.0	8.5	68.0	0	199.1
0.75	62.2	7.8	69.2	8.6	69.2	8.6	69.2	0	203.1
0.76	63.4	7.9	70.5	8.8	70.5	8.8	70.5	0	207.1
0.77	64.5	8.0	71.7	8.9	71.7	8.9	71.7	0	211.1
0.78	65.6	8.2	73.0	9.0	73.0	9.0	73.0	0	215.2
0.79	66.8	8.3	74.2	9.2	74.2	9.2	74.2	0	219.3
0.8	67.9	8.5	75.5	9.3	75.5	9.3	75.5	0	223.4
0.81	69.1	8.6	76.8	9.5	76.8	9.5	76.8	0	227.6
0.82	70.2	8.8	78.0	9.6	78.0	9.6	78.0	0	231.7
0.83	71.4	8.9	79.3	9.7	79.3	9.7	79.3	0	235.9

0.84	72.5	9.0	80.5	9.9	80.5	9.9	80.5	0	240.1
0.85	73.7	9.2	81.8	10.0	81.8	10.0	81.8	0	244.4
0.86	74.8	9.3	83.0	10.2	83.0	10.2	83.0	0	248.7
0.87	76.0	9.5	84.3	10.3	84.3	10.3	84.3	0	252.9
0.88	77.2	9.6	85.5	10.5	85.5	10.5	85.5	0	257.3
0.89	78.4	9.8	86.8	10.6	86.8	10.6	86.8	0	261.6
0.9	79.5	9.9	88.0	10.7	88.0	10.7	88.0	0	265.9
0.91	80.7	10.1	89.3	10.9	89.3	10.9	89.3	0	270.3
0.92	81.9	10.2	90.6	11.0	90.6	11.0	90.6	0	274.7
0.93	83.1	10.4	91.8	11.2	91.8	11.2	91.8	0	279.1
0.94	84.3	10.5	93.1	11.3	93.1	11.3	93.1	0	283.6
0.95	85.5	10.7	94.3	11.5	94.3	11.5	94.3	0	288.0
0.96	86.7	10.8	95.6	11.6	95.6	11.6	95.6	0	292.5
0.97	87.9	11.0	96.9	11.8	96.9	11.8	96.9	0	297.0
0.98	89.1	11.1	98.1	11.9	98.1	11.9	98.1	0	301.6
0.99	90.3	11.3	99.4	12.1	99.4	12.1	99.4	0	306.1
1	91.5	11.4	100.6	12.2	100.6	12.2	100.6	0	310.7
1.01	92.7	11.6	101.9	12.4	101.9	12.4	101.9	0	315.3
1.02	93.9	11.7	103.2	12.5	103.2	12.5	103.2	0	319.9
1.03	95.1	11.9	104.4	12.7	104.4	12.7	104.4	0	324.6
1.04	96.4	12.0	105.7	12.8	105.7	12.8	105.7	0	329.2
1.05	97.6	12.2	107.0	12.9	107.0	12.9	107.0	0	333.9
1.06	98.8	12.3	108.2	13.1	108.2	13.1	108.2	0	338.6
1.07	100.0	12.5	109.5	13.2	109.5	13.2	109.5	0	343.3
1.08	101.3	12.6	110.8	13.4	110.8	13.4	110.8	0	348.0
1.09	102.5	12.8	112.0	13.6	112.0	13.6	112.0	0	352.8
1.1	103.7	12.9	113.3	13.7	113.3	13.7	113.3	0	357.6
1.11	105.0	13.1	114.6	13.9	114.6	13.9	114.6	0	362.4
1.12	106.2	13.3	115.8	14.0	115.8	14.0	115.8	0	367.2
1.13	107.4	13.4	117.1	14.2	117.1	14.2	117.1	0	372.0
1.14	108.7	13.6	118.4	14.3	118.4	14.3	118.4	0	376.9
1.15	109.9	13.7	119.6	14.5	119.6	14.5	119.6	0	381.8
1.16	111.2	13.9	120.9	14.6	120.9	14.6	120.9	0	386.7
1.17	112.4	14.0	122.2	14.8	122.2	14.8	122.2	0	391.6
1.18	113.7	14.2	123.5	14.9	123.5	14.9	123.5	0	396.5
1.19	115.0	14.3	124.7	15.1	124.7	15.1	124.7	0	401.5
1.2	116.2	14.5	126.0	15.2	126.0	15.2	126.0	0	406.4
1.21	117.5	14.7	127.3	15.4	127.3	15.4	127.3	0	411.4
1.22	118.7	14.8	128.6	15.5	128.6	15.5	128.6	0	416.4
1.23	120.0	15.0	129.8	15.7	129.8	15.7	129.8	0	421.5
1.24	121.3	15.1	131.1	15.8	131.1	15.8	131.1	0	426.5
1.25	122.5	15.3	132.4	16.0	132.4	16.0	132.4	0	431.6

1.26	123.8	15.5	133.7	16.2	133.7	16.2	133.7	0	436.7
1.27	125.1	15.6	134.9	16.3	134.9	16.3	134.9	0	441.8
1.28	126.4	15.8	136.2	16.5	136.2	16.5	136.2	0	446.9
1.29	127.6	15.9	137.5	16.6	137.5	16.6	137.5	0	452.0
1.3	128.9	16.1	138.8	16.8	138.8	16.8	138.8	0	457.2
1.31	130.2	16.3	140.1	16.9	140.1	16.9	140.1	0	462.3
1.32	131.5	16.4	141.3	17.1	141.3	17.1	141.3	0	467.5
1.33	132.8	16.6	142.6	17.2	142.6	17.2	142.6	0	472.7
1.34	134.1	16.7	143.9	17.4	143.9	17.4	143.9	0	478.0
1.35	135.3	16.9	145.2	17.6	145.2	17.6	145.2	0	483.2
1.36	136.6	17.1	146.5	17.7	146.5	17.7	146.5	0	488.5
1.37	137.9	17.2	147.8	17.9	147.8	17.9	147.8	0	493.7
1.38	139.2	17.4	149.0	18.0	149.0	18.0	149.0	0	499.0
1.39	140.5	17.5	150.3	18.2	150.3	18.2	150.3	0	504.4
1.4	141.8	17.7	151.6	18.4	151.6	18.4	151.6	0	509.7
1.41	143.1	17.9	152.9	18.5	152.9	18.5	152.9	0	515.0
1.42	144.4	18.0	154.2	18.7	154.2	18.7	154.2	0	520.4
1.43	145.7	18.2	155.5	18.8	155.5	18.8	155.5	0	525.8
1.44	147.0	18.4	156.8	19.0	156.8	19.0	156.8	0	531.2
1.45	148.3	18.5	158.0	19.2	158.0	19.2	158.0	0	536.6
1.46	149.6	18.7	159.3	19.3	159.3	19.3	159.3	0	542.0
1.47	150.9	18.8	160.6	19.5	160.6	19.5	160.6	0	547.5
1.48	152.2	19.0	161.9	19.6	161.9	19.6	161.9	0	552.9
1.49	153.6	19.2	163.2	19.8	163.2	19.8	163.2	0	558.4
1.5	154.9	19.3	164.5	20.0	164.5	20.0	164.5	0	563.9
1.51	156.2	19.5	165.8	20.1	165.8	20.1	165.8	0	569.4
1.52	157.5	19.7	167.1	20.3	167.1	20.3	167.1	0	574.9
1.53	158.8	19.8	168.4	20.4	168.4	20.4	168.4	0	580.5
1.54	160.1	20.0	169.6	20.6	169.6	20.6	169.6	0	586.0
1.55	161.5	20.2	170.9	20.8	170.9	20.8	170.9	0	591.6
1.56	162.8	20.3	172.2	20.9	172.2	20.9	172.2	0	597.2
1.57	164.1	20.5	173.5	21.1	173.5	21.1	173.5	0	602.8
1.58	165.4	20.7	174.8	21.2	174.8	21.2	174.8	0	608.4
1.59	166.8	20.8	176.1	21.4	176.1	21.4	176.1	0	614.1
1.6	168.1	21.0	177.4	21.6	177.4	21.6	177.4	0	619.7
1.61	169.4	21.2	178.7	21.7	178.7	21.7	178.7	0	625.4
1.62	170.7	21.3	180.0	21.9	180.0	21.9	180.0	0	631.1
1.63	172.1	21.5	181.3	22.1	181.3	22.1	181.3	0	636.8
1.64	173.4	21.7	182.6	22.2	182.6	22.2	182.6	0	642.5
1.65	174.7	21.8	183.9	22.4	183.9	22.4	183.9	0	648.2
1.66	176.1	22.0	185.2	22.6	185.2	22.6	185.2	0	654.0
1.67	177.4	22.2	186.5	22.7	186.5	22.7	186.5	0	659.7

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1.68	178.8	22.3	187.8	22.9	187.8	22.9	187.8	0	665.5
1.69	180.1	22.5	189.1	23.0	189.1	23.0	189.1	0	671.3
1.7	181.4	22.7	190.3	23.2	190.3	23.2	190.3	0	677.1
1.71	182.8	22.8	191.6	23.4	191.6	23.4	191.6	0	682.9
1.72	184.1	23.0	192.9	23.5	192.9	23.5	192.9	0	688.7
1.73	185.5	23.2	194.2	23.7	194.2	23.7	194.2	0	694.6
1.74	186.8	23.3	195.5	23.9	195.5	23.9	195.5	0	700.5
1.75	188.2	23.5	196.8	24.0	196.8	24.0	196.8	0	706.3
1.76	189.5	23.7	198.1	24.2	198.1	24.2	198.1	0	712.2
1.77	190.9	23.8	199.4	24.4	199.4	24.4	199.4	0	718.1
1.78	192.2	24.0	200.7	24.5	200.7	24.5	200.7	0	724.1
1.79	193.6	24.2	202.0	24.7	202.0	24.7	202.0	0	730.0
1.8	194.9	24.4	203.3	24.9	203.3	24.9	203.3	0	735.9
1.81	196.3	24.5	204.6	25.0	204.6	25.0	204.6	0	741.9
1.82	197.6	24.7	205.9	25.2	205.9	25.2	205.9	0	747.9
1.83	199.0	24.9	207.2	25.4	207.2	25.4	207.2	0	753.9
1.84	200.3	25.0	208.5	25.5	208.5	25.5	208.5	0	759.9
1.85	201.7	25.2	209.8	25.7	209.8	25.7	209.8	0	765.9
1.86	203.0	25.4	211.1	25.9	211.1	25.9	211.1	0	771.9
1.87	204.4	25.5	212.4	26.0	212.4	26.0	212.4	0	778.0
1.88	205.8	25.7	213.7	26.2	213.7	26.2	213.7	0	784.0
1.89	207.1	25.9	215.0	26.4	215.0	26.4	215.0	0	790.1

## APPENDIX C - – MATLA POWER STATION

### C.1. OPERATING POINTS

**Table 31: Matla SFP operating data**

Parameter	Design Point	Units
Feed water Tank Pressure	0.913	MPa
Feed water Tank Temperature	176	°C
Feed water Tank Density	891	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	531.93	kg/s
Pressure at Inlet to Booster Pump	1.04	MPa
Manometric Head – Booster Pump	0.961	MPa
Pressure at Outlet to Booster Pump	2.01	MPa
Pressure at Inlet Connection of Main Pump	1.98	MPa
Mass Flow at Delivery Connection of Main Pump	531.93	kg/s
Manometric Head – Main Pump	19.68	MPa
Pressure at Delivery Connection of Main Pump	21.65	MPa
Speed of Main Pump (TOC)	5276	rpm
Power Required at Pump Shaft (TOC)	14400	kW
Main Pump Efficiency (On-site)	81.6	%

**Table 32: Matla EFP operating data**

Parameter	Design Point	Units
Feed water Tank Pressure	0.913	MPa
Feed water Tank Temperature	176	°C
Feed water Tank Density	891	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	265.96	kg/s
Pressure at Inlet to Booster Pump	1.04	MPa
Manometric Head – Booster Pump	1.153	MPa
Pressure at Outlet to Booster Pump	2.193	MPa
Pressure at Inlet Connection of Main Pump	2.15	MPa
Mass Flow at Delivery Connection of Main Pump	265.96	kg/s
Manometric Head – Main Pump	19.51	MPa
Pressure at Delivery Connection of Main Pump	21.65	MPa
Speed of Main Pump (TOC)	6231	rpm
Power Required at Pump Shaft (TOC)	7250	kW
Main Pump Efficiency (On-site)	80.3	%

## C.2. MATLA SFP 2<sup>ND</sup> – LAST STAGE IMPELLER WEAR DATA

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as seen in Figure 88, with diameters of 279.45 mm h7 and 364.45 mm h7 on the eye diameters and 199.3 mm h7 on the neck diameters when manufactured

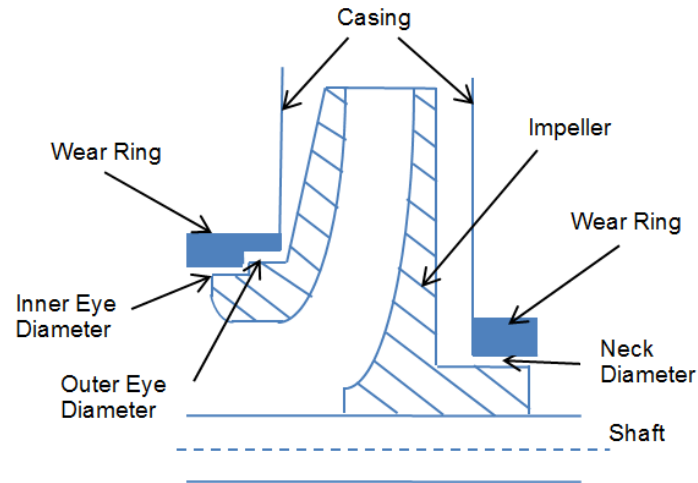


Figure 88: Matla SFP 2<sup>nd</sup> – last stage impeller sectional view

Figure 89 to 91 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.

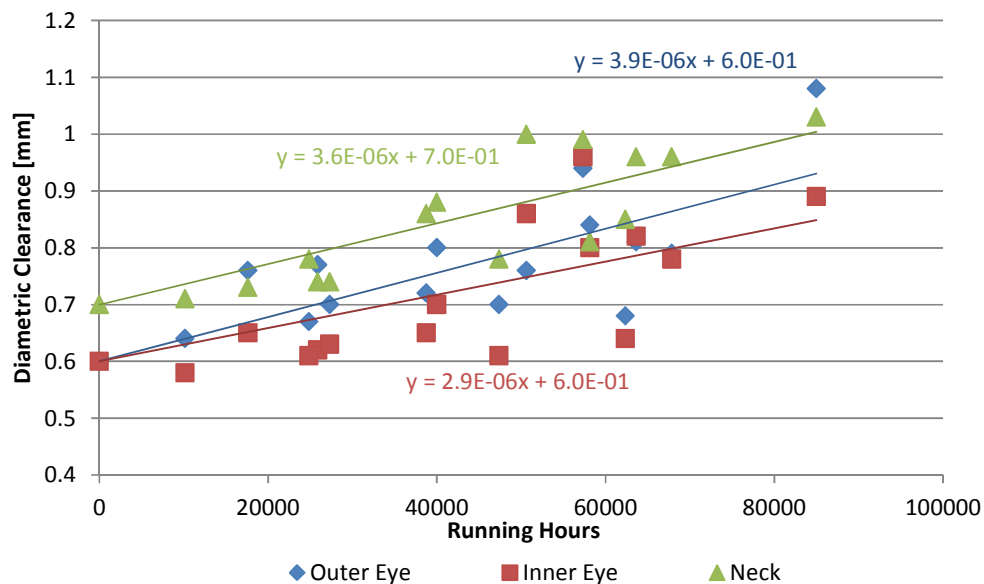


Figure 89: Matla SFP 2<sup>nd</sup> stage impeller wear data

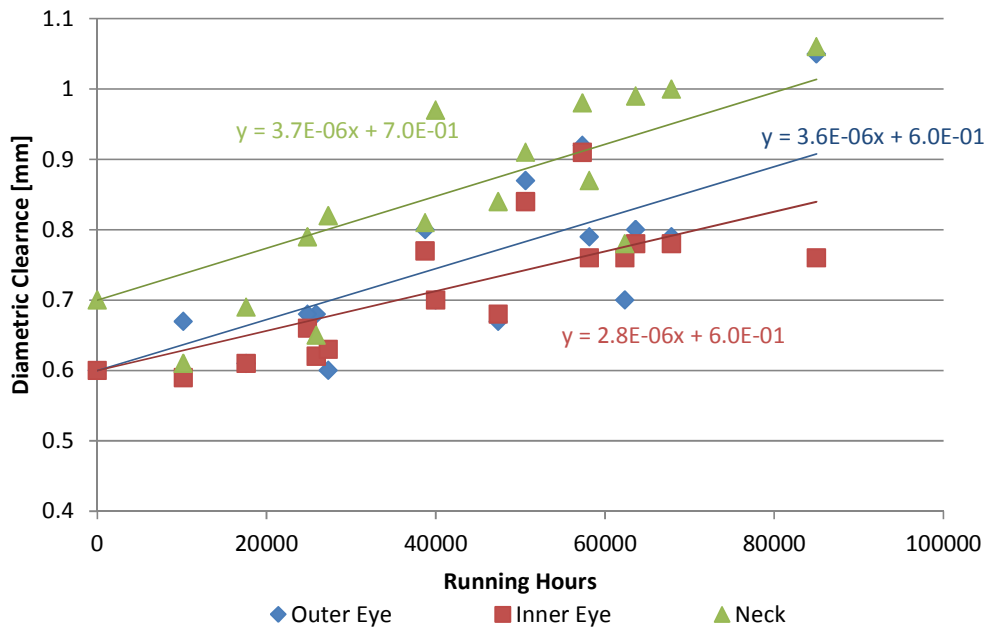


Figure 90: Matla SFP 3<sup>rd</sup> stage impeller wear data

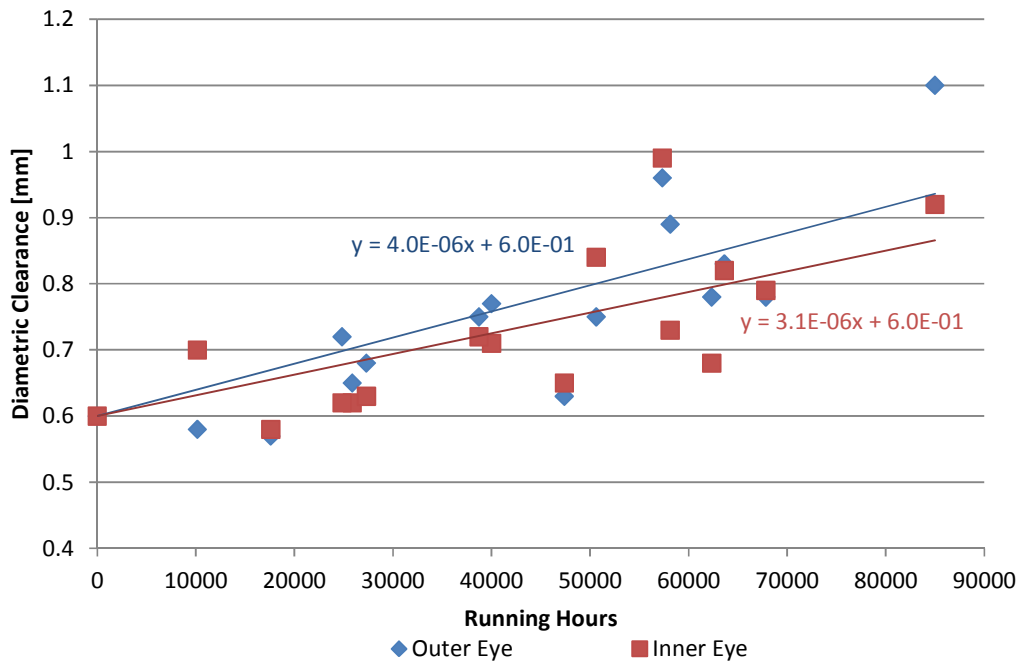


Figure 91: Matla SFP last stage impeller wear data

### C.3. MATLA EFP 2<sup>ND</sup> – LAST STAGE IMPELLER WEAR DATA

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as seen in Figure 92, with diameters of 214.5 mm h7 and 204.5 mm h7 on the eye diameters and 149.37 mm h7 on the neck diameters when manufactured

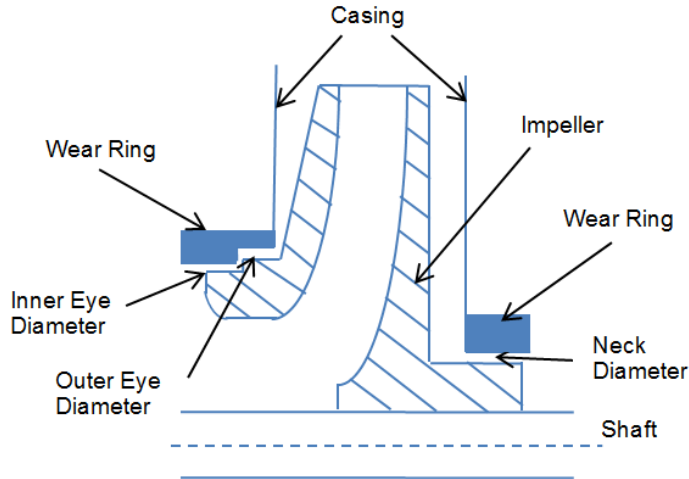


Figure 92: Matla EFP 2<sup>nd</sup> – last stage impeller sectional view

Figure 93 to 96 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.

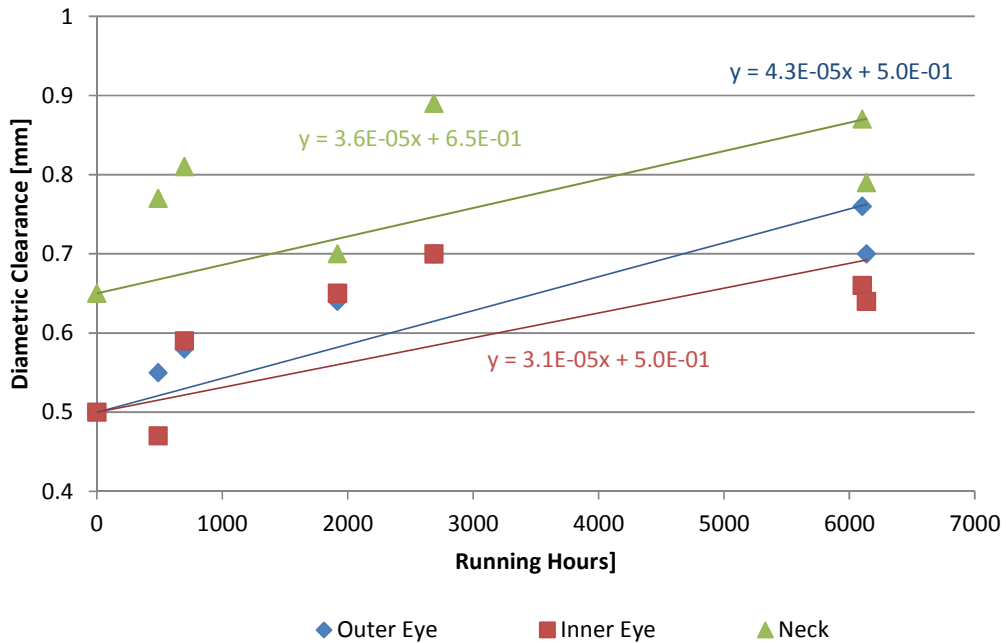


Figure 93: Matla EFP 2<sup>nd</sup> stage impeller wear data

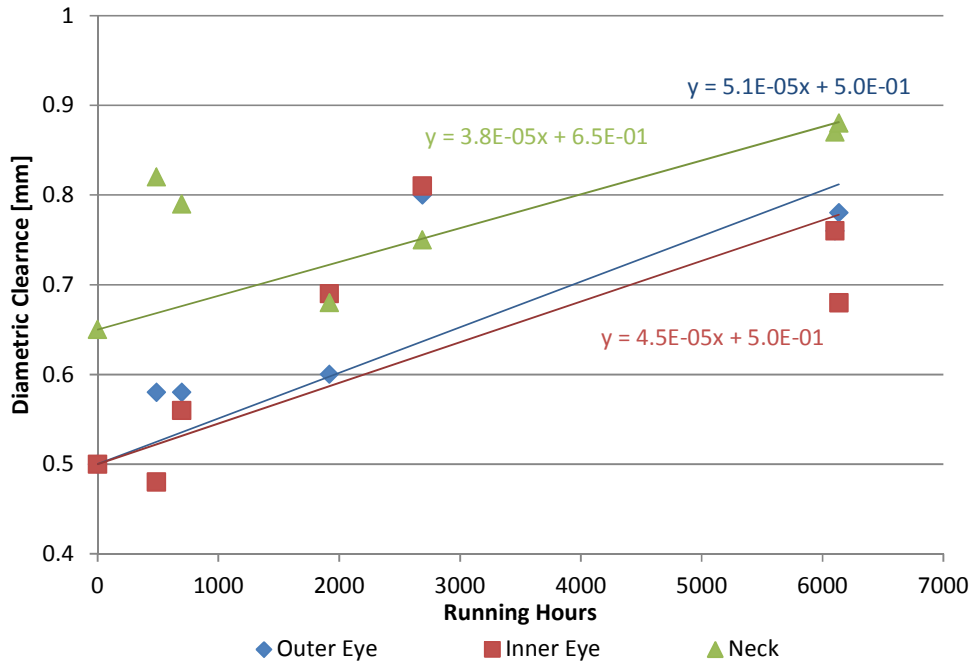


Figure 94: Matla EFP 3<sup>rd</sup> stage impeller wear data

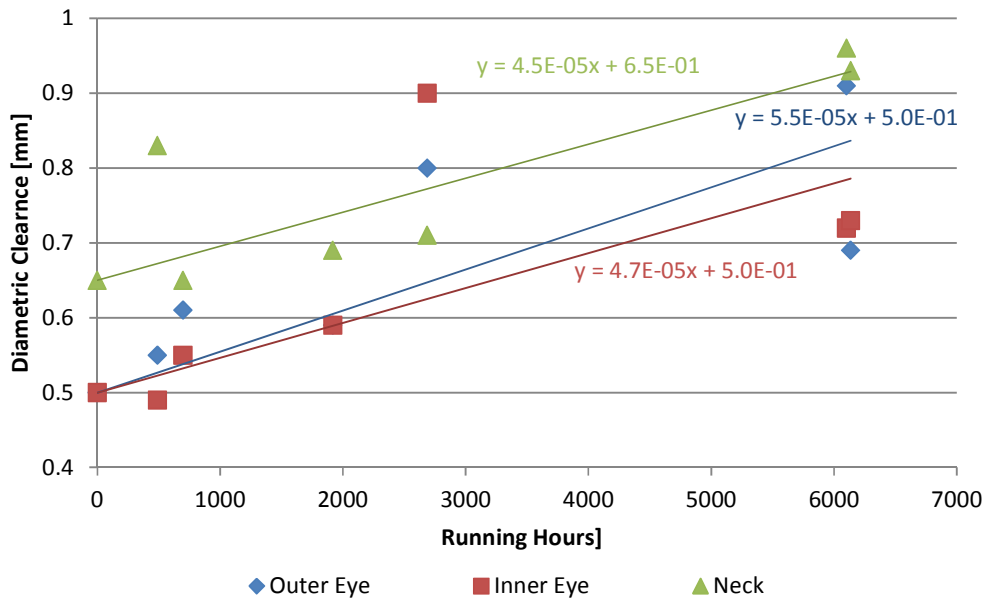
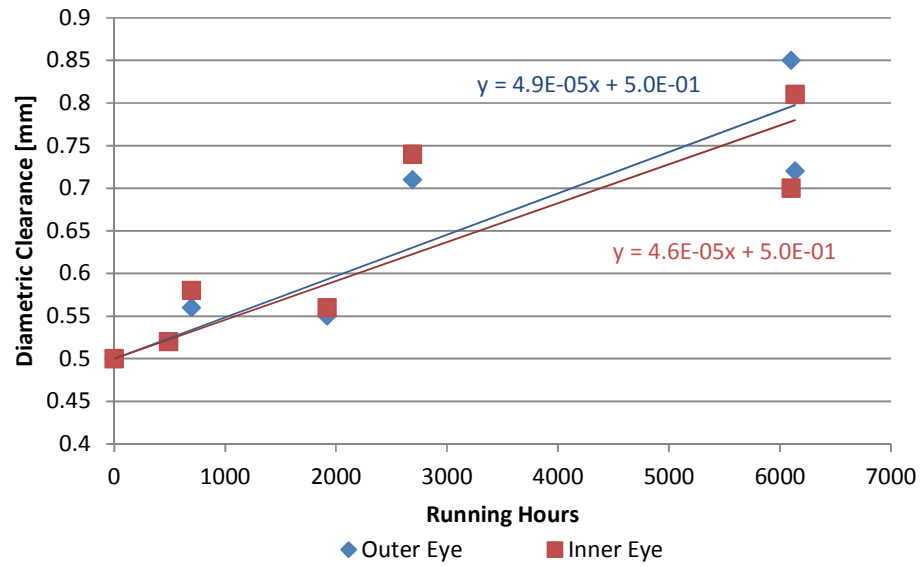


Figure 95: Matla EFP 4<sup>th</sup> stage impeller wear data



**Figure 96: Matla EFP last stage impeller wear data**

### C.4. POWER CONSUMED THROUGH ANNULAR SEALS

**Table 33: Matla SFP – Power consumed through annular seals**

Power Consumed at a Specific Clearance [kW]									
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		Last Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	
0.5	40.89	4.82	34.01	4.85	34.01	4.85	34.01	0	125.94
0.51	42.03	4.95	35.47	4.98	35.47	4.98	35.47	0	129.71
0.52	43.18	5.09	36.92	5.12	36.92	5.12	36.92	0	133.51
0.53	44.34	5.23	38.36	5.26	38.36	5.26	38.36	0	137.34
0.54	45.50	5.37	39.80	5.40	39.80	5.40	39.80	0	141.21
0.55	46.67	5.50	41.24	5.54	41.24	5.54	41.24	0	145.12
0.56	47.85	5.64	42.67	5.68	42.67	5.68	42.67	0	149.05
0.57	49.03	5.79	44.10	5.82	44.10	5.82	44.10	0	153.03
0.58	50.22	5.93	45.52	5.96	45.52	5.96	45.52	0	157.03
0.59	51.42	6.07	46.94	6.11	46.94	6.11	46.94	0	161.07
0.6	52.62	6.21	48.36	6.25	48.36	6.25	48.36	0	165.14
0.61	53.83	6.36	49.78	6.40	49.78	6.40	49.78	0	169.25
0.62	55.05	6.50	51.20	6.54	51.20	6.54	51.20	0	173.38
0.63	56.27	6.65	52.61	6.69	52.61	6.69	52.61	0	177.55
0.64	57.50	6.79	54.03	6.84	54.03	6.84	54.03	0	181.75
0.65	58.73	6.94	55.44	6.98	55.44	6.98	55.44	0	185.98
0.66	59.97	7.09	56.86	7.13	56.86	7.13	56.86	0	190.24
0.67	61.22	7.24	58.27	7.28	58.27	7.28	58.27	0	194.53
0.68	62.47	7.39	59.68	7.43	59.68	7.43	59.68	0	198.85
0.69	63.72	7.54	61.10	7.58	61.10	7.58	61.10	0	203.21
0.7	64.98	7.69	62.51	7.73	62.51	7.73	62.51	0	207.59
0.71	66.25	7.84	63.92	7.89	63.92	7.89	63.92	0	212.00
0.72	67.52	7.99	65.34	8.04	65.34	8.04	65.34	0	216.44
0.73	68.79	8.14	66.75	8.19	66.75	8.19	66.75	0	220.91
0.74	70.07	8.30	68.17	8.35	68.17	8.35	68.17	0	225.41
0.75	71.36	8.45	69.58	8.50	69.58	8.50	69.58	0	229.94
0.76	72.65	8.61	71.00	8.66	71.00	8.66	71.00	0	234.50
0.77	73.95	8.76	72.42	8.81	72.42	8.81	72.42	0	239.08
0.78	75.25	8.92	73.84	8.97	73.84	8.97	73.84	0	243.69
0.79	76.55	9.07	75.26	9.13	75.26	9.13	75.26	0	248.33
0.8	77.86	9.23	76.68	9.29	76.68	9.29	76.68	0	253.00
0.81	79.17	9.39	78.10	9.45	78.10	9.45	78.10	0	257.70
0.82	80.49	9.55	79.52	9.60	79.52	9.60	79.52	0	262.42

0.83	81.81	9.70	80.95	9.76	80.95	9.76	80.95	0	267.17
0.84	83.14	9.86	82.37	9.92	82.37	9.92	82.37	0	271.94
0.85	84.47	10.02	83.80	10.09	83.80	10.09	83.80	0	276.75
0.86	85.80	10.18	85.23	10.25	85.23	10.25	85.23	0	281.58
0.87	87.14	10.35	86.66	10.41	86.66	10.41	86.66	0	286.43
0.88	88.48	10.51	88.09	10.57	88.09	10.57	88.09	0	291.31
0.89	89.83	10.67	89.52	10.73	89.52	10.73	89.52	0	296.22
0.9	91.18	10.83	90.95	10.90	90.95	10.90	90.95	0	301.15
0.91	92.53	10.99	92.39	11.06	92.39	11.06	92.39	0	306.11
0.92	93.89	11.16	93.82	11.23	93.82	11.23	93.82	0	311.09
0.93	95.25	11.32	95.26	11.39	95.26	11.39	95.26	0	316.10
0.94	96.61	11.49	96.70	11.56	96.70	11.56	96.70	0	321.14
0.95	97.98	11.65	98.14	11.72	98.14	11.72	98.14	0	326.20
0.96	99.35	11.82	99.58	11.89	99.58	11.89	99.58	0	331.28
0.97	100.73	11.98	101.02	12.06	101.02	12.06	101.02	0	336.39
0.98	102.11	12.15	102.46	12.22	102.46	12.22	102.46	0	341.52
0.99	103.49	12.32	103.91	12.39	103.91	12.39	103.91	0	346.67
1	104.88	12.48	105.35	12.56	105.35	12.56	105.35	0	351.85
1.01	106.26	12.65	106.80	12.73	106.80	12.73	106.80	0	357.06
1.02	107.66	12.82	108.25	12.90	108.25	12.90	108.25	0	362.29
1.03	109.05	12.99	109.70	13.07	109.70	13.07	109.70	0	367.54
1.04	110.45	13.16	111.15	13.24	111.15	13.24	111.15	0	372.81
1.05	111.85	13.33	112.60	13.41	112.60	13.41	112.60	0	378.11
1.06	113.25	13.50	114.05	13.58	114.05	13.58	114.05	0	383.43
1.07	114.66	13.67	115.51	13.75	115.51	13.75	115.51	0	388.78
1.08	116.07	13.84	116.96	13.92	116.96	13.92	116.96	0	394.14
1.09	117.48	14.01	118.42	14.10	118.42	14.10	118.42	0	399.53
1.1	118.90	14.18	119.87	14.27	119.87	14.27	119.87	0	404.95

## APPENDIX D - KRIEL POWER STATION

### D.1. OPERATING POINTS

**Table 34:Kriel steam feed pump operating data**

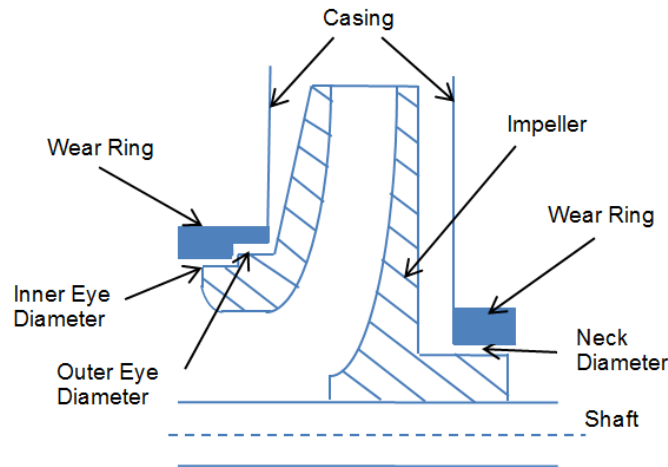
Parameter	Design Point	Units
Feed Water Tank Pressure	0.878	MPa
Feed Water Tank Temperature	174.5	°C
Feed Water Tank Density	893	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	482	kg/s
Pressure at Inlet to Booster Pump	1.086	MPa
Manometric Head – Booster Pump	1.15	MPa
Pressure at Outlet to Booster Pump	2.24	MPa
Pressure at Inlet Connection of Main Pump	2.24	MPa
Mass Flow at Extraction Connection	13.3	kg/s
Pressure at Extraction Connection	9.5	MPa
Mass Flow at Delivery Connection of Main Pump	468.7	kg/s
Manometric Head – Main Pump	23.56	MPa
Pressure at Delivery Connection of Main Pump	25.8	MPa
Speed of Main Pump (TOO)	5280	rpm
Speed of Main Pump (TOC)	5240	rpm
Power Required at Pump Shaft (TOO)	15141	kW
Power Required at Pump Shaft (TOC)	14972	kW
Main Pump Efficiency (On-site)	83.6	%

**Table 35: Kriel electric feed pump operating data**

Parameter	Design Point	Units
Feed Water Tank Pressure	0.878	MPa
Feed Water Tank Temperature	174.5	°C
Feed Water Tank Density	893	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	241	kg/s
Pressure at Inlet to Booster Pump	1.086	MPa
Manometric Head – Booster Pump	0.875	MPa
Pressure at Outlet to Booster Pump	1.961	MPa
Pressure at Inlet Connection of Main Pump	1.96	MPa
Mass Flow at Extraction Connection	6.65	kg/s
Pressure at Extraction Connection	1126.76	MPa
Mass Flow at Delivery Connection of Main Pump	234.5	kg/s
Manometric Head – Main Pump	24.6	MPa
Pressure at Delivery Connection of Main Pump	26.56	MPa
Speed of Main Pump (TOO)	6826	rpm
Speed of Main Pump (TOC)	6805	rpm
Power Required at Pump Shaft (TOO)	7919	kW
Power Required at Pump Shaft (TOC)	8006	kW
Main Pump Efficiency (On-site)	81.6	%

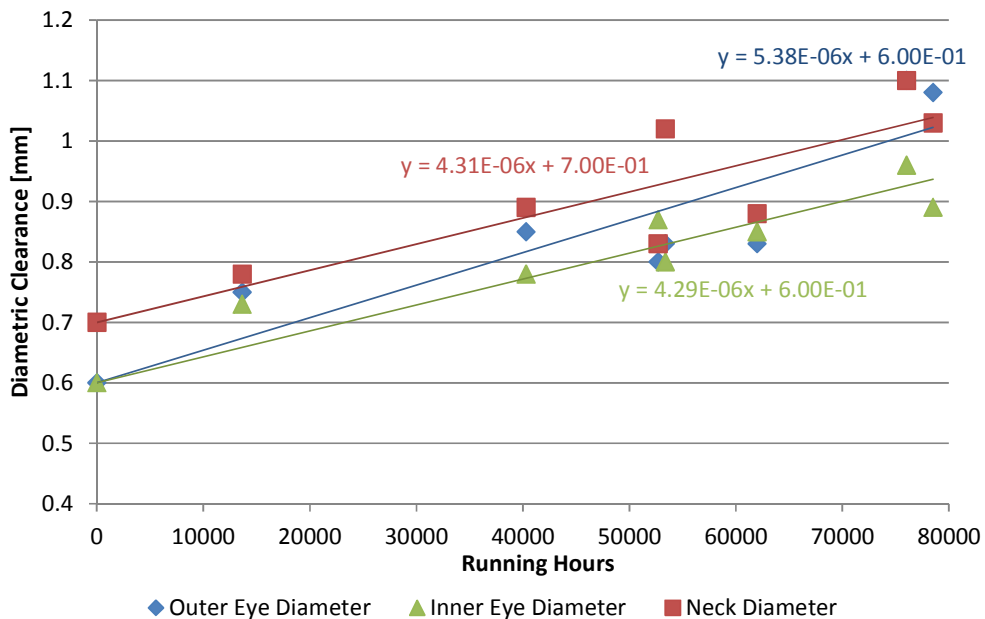
**D.2. KRIEL SFP – 2<sup>ND</sup> TO LAST STAGE IMPELLER**

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as seen in Figure 97, with diameters of 279.45 mm h7 and 264.45 mm h7 on the eye diameters and 199.3 mm h7 on the neck diameters when manufactured.



**Figure 97: Kriel SFP 2<sup>nd</sup> – last stage impeller sectional view**

Figure 98 to 101 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.



**Figure 98: Kriel SFP 2<sup>nd</sup> stage impeller wear data**

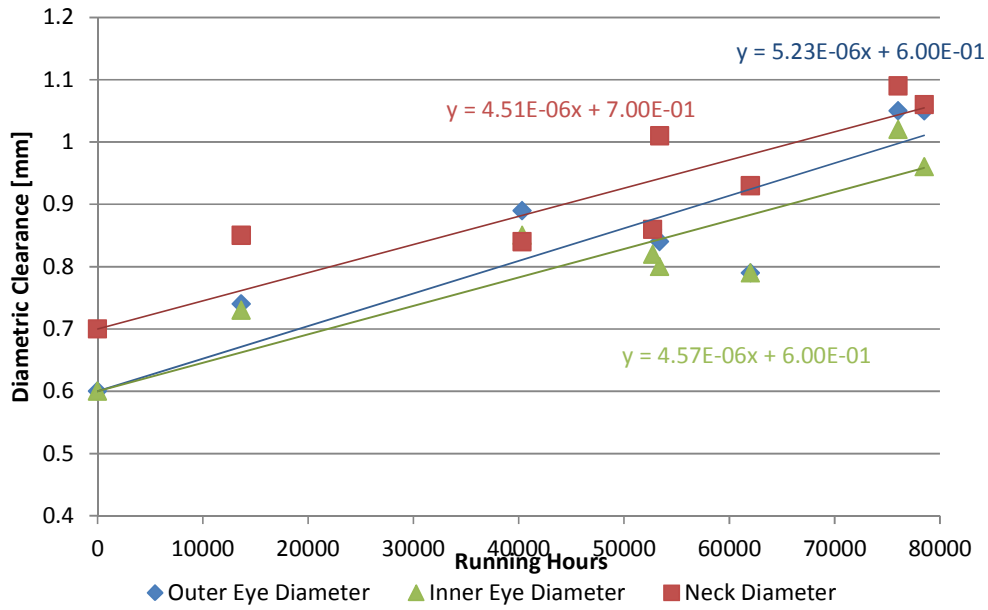


Figure 99: Kriel SFP 3<sup>rd</sup> stage impeller wear data

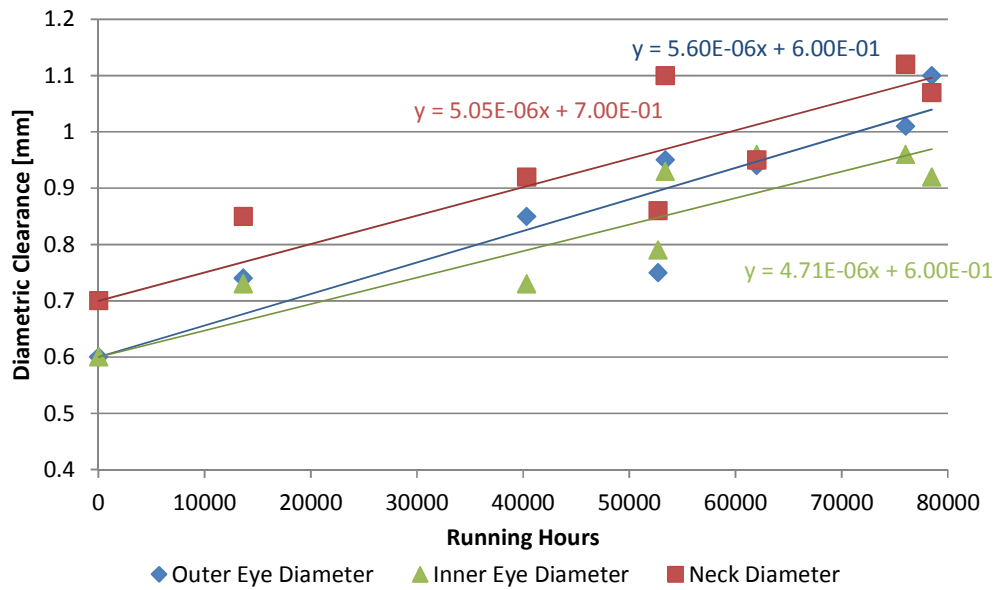
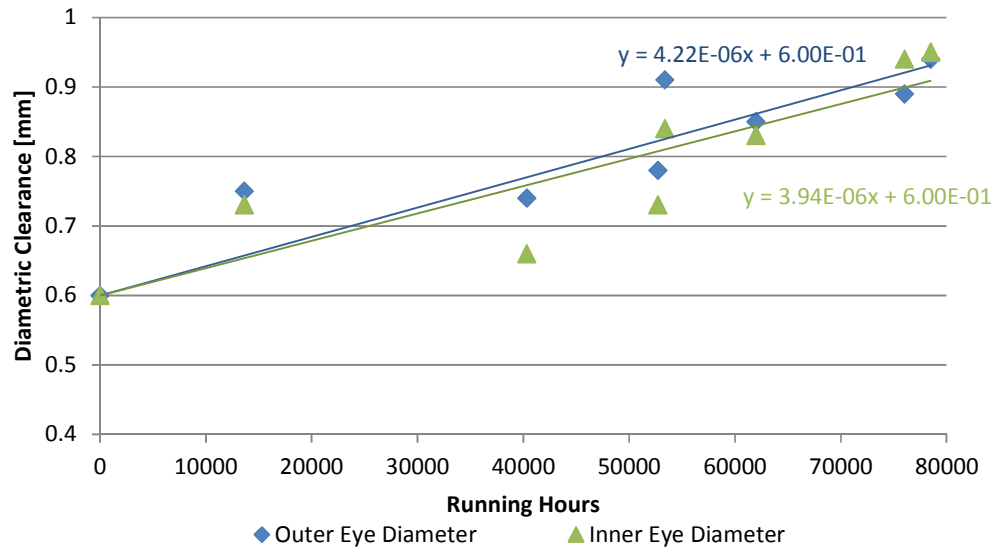


Figure 100: Kriel SFP 4<sup>th</sup> stage impeller wear data



**Figure 101: Kriel SFP last stage impeller wear data**

### D.3. KRIEL EFP – 2<sup>ND</sup> TO LAST STAGE IMPELLER

The 2<sup>nd</sup> to last stage impeller has a stepped annular seal as seen in Figure 102, with diameters of 214.50 mm h7 and 204.50 mm h7 on the eye diameters and 149.35 mm h7 on the neck diameters when manufactured

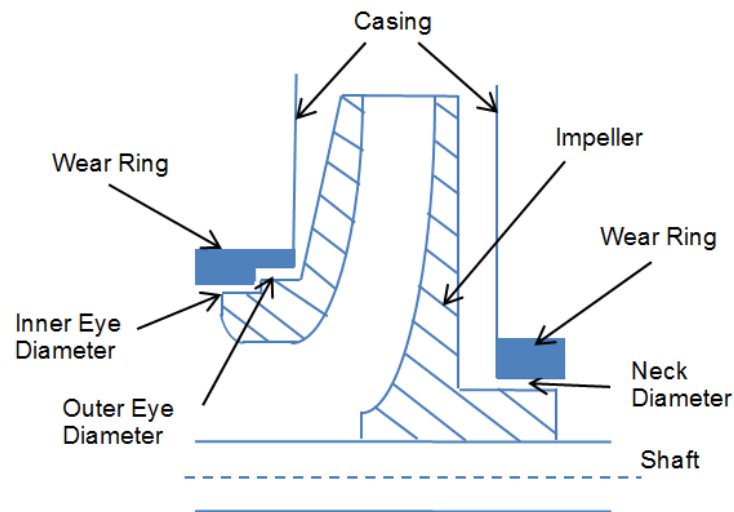


Figure 102: Kriel EFP 2<sup>nd</sup> – last stage impeller sectional view

Figure 103 to 106 detail the second to last stage impeller diametric clearance for refurbishments at various running hours.

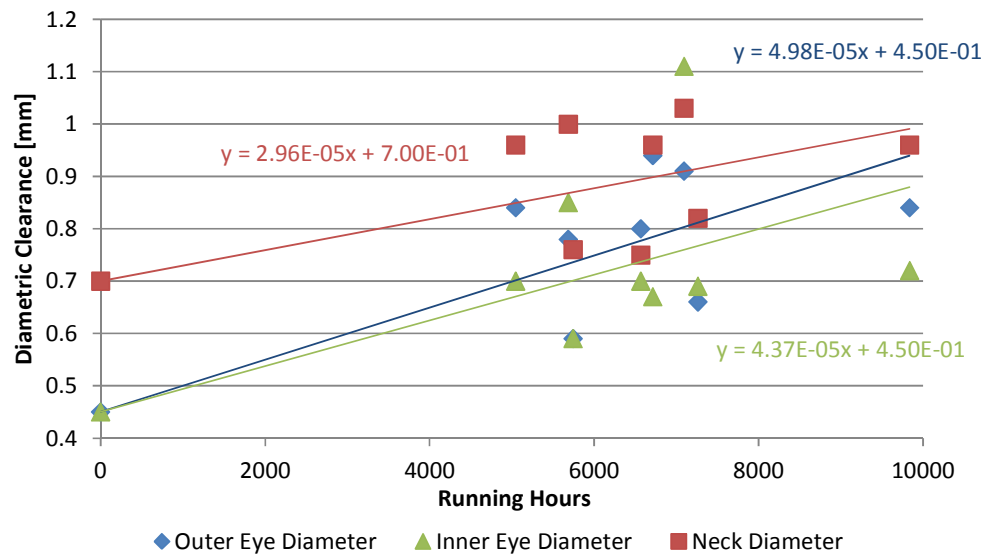
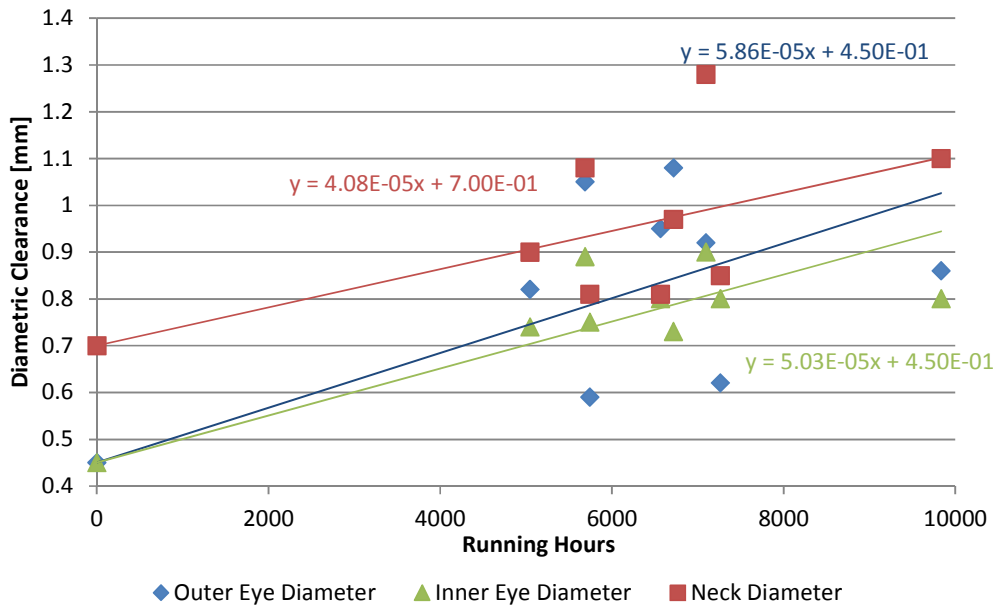
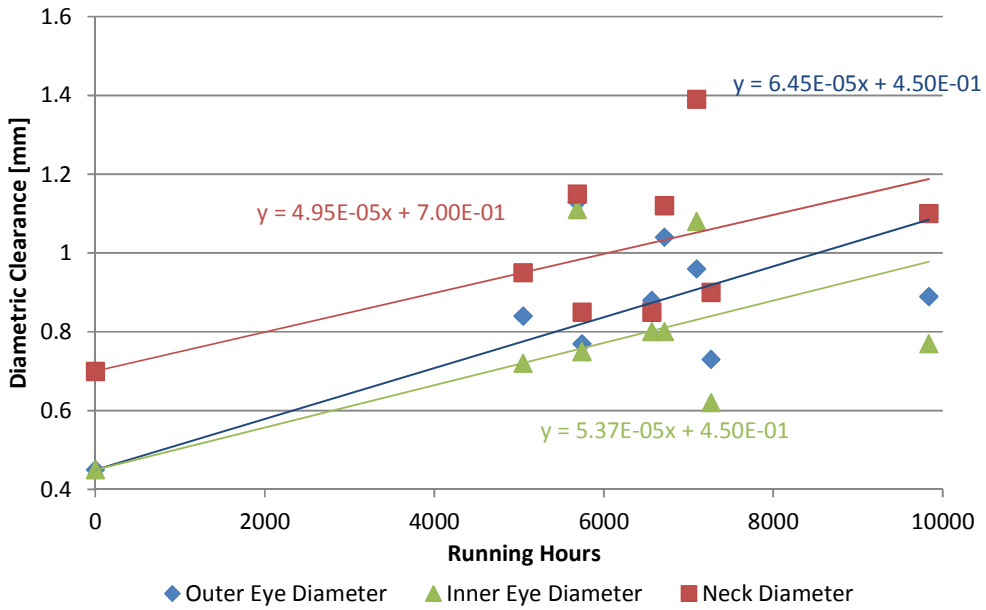


Figure 103: Kriel EFP 2<sup>nd</sup> stage impeller wear data



**Figure 104: Kriel EFP 3<sup>rd</sup> stage impeller wear data**



**Figure 105: Kriel EFP 4<sup>th</sup> stage impeller wear data**

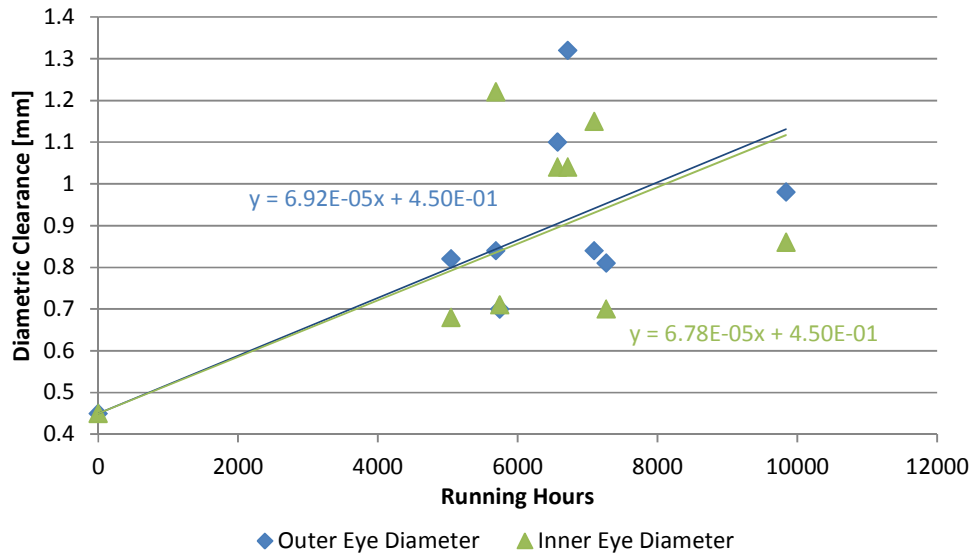


Figure 106: Kriel EFP last stage impeller wear data

#### D.4. POWER CONSUMED THROUGH ANNULAR SEALS

**Table 36: Kriel steam feed pump – Power consumption through annular seals**

Power Consumed at a Specific Clearance [kW]											
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		4th Stage		5th Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	
0.6	66.7	8.2	58.9	8.1	58.9	8.1	58.9	8.1	58.9	0.0	225.0
0.61	68.3	8.3	60.2	8.3	60.2	8.3	60.2	8.3	60.2	0.0	230.6
0.62	69.9	8.5	61.6	8.4	61.6	8.4	61.6	8.4	61.6	0.0	236.3
0.63	71.4	8.7	62.9	8.6	62.9	8.6	62.9	8.6	62.9	0.0	241.9
0.64	73.0	8.9	64.2	8.8	64.2	8.8	64.2	8.8	64.2	0.0	247.7
0.65	74.6	9.1	65.6	9.0	65.6	9.0	65.6	9.0	65.6	0.0	253.4
0.66	76.2	9.3	66.9	9.2	66.9	9.2	66.9	9.2	66.9	0.0	259.2
0.67	77.8	9.5	68.2	9.4	68.2	9.4	68.2	9.4	68.2	0.0	265.1
0.68	79.4	9.7	69.6	9.6	69.6	9.6	69.6	9.6	69.6	0.0	271.0
0.69	81.0	9.9	71.0	9.8	71.0	9.8	71.0	9.8	71.0	0.0	276.9
0.7	82.6	10.1	72.3	10.0	72.3	10.0	72.3	10.0	72.3	0.0	282.9
0.71	84.3	10.3	73.7	10.2	73.7	10.2	73.7	10.2	73.7	0.0	288.9
0.72	85.9	10.5	75.1	10.4	75.1	10.4	75.1	10.4	75.1	0.0	294.9
0.73	87.5	10.7	76.4	10.6	76.4	10.6	76.4	10.6	76.4	0.0	301.0
0.74	89.2	10.9	77.8	10.8	77.8	10.8	77.8	10.8	77.8	0.0	307.1
0.75	90.9	11.1	79.2	11.0	79.2	11.0	79.2	11.0	79.2	0.0	313.3
0.76	92.5	11.3	80.6	11.2	80.6	11.2	80.6	11.2	80.6	0.0	319.5
0.77	94.2	11.5	82.0	11.4	82.0	11.4	82.0	11.4	82.0	0.0	325.7
0.78	95.9	11.7	83.4	11.6	83.4	11.6	83.4	11.6	83.4	0.0	332.0
0.79	97.6	11.9	84.8	11.8	84.8	11.8	84.8	11.8	84.8	0.0	338.3
0.8	99.2	12.1	86.2	12.0	86.2	12.0	86.2	12.0	86.2	0.0	344.7
0.81	100.9	12.3	87.6	12.2	87.6	12.2	87.6	12.2	87.6	0.0	351.1
0.82	102.7	12.5	89.0	12.4	89.0	12.4	89.0	12.4	89.0	0.0	357.5
0.83	104.4	12.7	90.4	12.6	90.4	12.6	90.4	12.6	90.4	0.0	364.0
0.84	106.1	12.9	91.8	12.8	91.8	12.8	91.8	12.8	91.8	0.0	370.5
0.85	107.8	13.2	93.2	13.0	93.2	13.0	93.2	13.0	93.2	0.0	377.0
0.86	109.5	13.4	94.6	13.2	94.6	13.2	94.6	13.2	94.6	0.0	383.6
0.87	111.3	13.6	96.0	13.4	96.0	13.4	96.0	13.4	96.0	0.0	390.2
0.88	113.0	13.8	97.5	13.6	97.5	13.6	97.5	13.6	97.5	0.0	396.9
0.89	114.7	14.0	98.9	13.9	98.9	13.9	98.9	13.9	98.9	0.0	403.6

0.9	116.5	14.2	100.3	14.1	100.3	14.1	100.3	14.1	100.3	0.0	410.3
0.91	118.3	14.4	101.8	14.3	101.8	14.3	101.8	14.3	101.8	0.0	417.0
0.92	120.0	14.6	103.2	14.5	103.2	14.5	103.2	14.5	103.2	0.0	423.8
0.93	121.8	14.9	104.6	14.7	104.6	14.7	104.6	14.7	104.6	0.0	430.6
0.94	123.6	15.1	106.1	14.9	106.1	14.9	106.1	14.9	106.1	0.0	437.5
0.95	125.3	15.3	107.5	15.1	107.5	15.1	107.5	15.1	107.5	0.0	444.4
0.96	127.1	15.5	109.0	15.3	109.0	15.3	109.0	15.3	109.0	0.0	451.3
0.97	128.9	15.7	110.4	15.6	110.4	15.6	110.4	15.6	110.4	0.0	458.3
0.98	130.7	15.9	111.9	15.8	111.9	15.8	111.9	15.8	111.9	0.0	465.3
0.99	132.5	16.2	113.3	16.0	113.3	16.0	113.3	16.0	113.3	0.0	472.3
1	134.3	16.4	114.8	16.2	114.8	16.2	114.8	16.2	114.8	0.0	479.3
1.01	136.1	16.6	116.2	16.4	116.2	16.4	116.2	16.4	116.2	0.0	486.4
1.02	137.9	16.8	117.7	16.6	117.7	16.6	117.7	16.6	117.7	0.0	493.5
1.03	139.7	17.0	119.2	16.9	119.2	16.9	119.2	16.9	119.2	0.0	500.7
1.04	141.5	17.3	120.6	17.1	120.6	17.1	120.6	17.1	120.6	0.0	507.9
1.05	143.4	17.5	122.1	17.3	122.1	17.3	122.1	17.3	122.1	0.0	515.1
1.06	145.2	17.7	123.6	17.5	123.6	17.5	123.6	17.5	123.6	0.0	522.3
1.07	147.0	17.9	125.0	17.8	125.0	17.8	125.0	17.8	125.0	0.0	529.6
1.08	148.9	18.2	126.5	18.0	126.5	18.0	126.5	18.0	126.5	0.0	536.9
1.09	150.7	18.4	128.0	18.2	128.0	18.2	128.0	18.2	128.0	0.0	544.3
1.1	152.6	18.6	129.5	18.4	129.5	18.4	129.5	18.4	129.5	0.0	551.6
1.11	154.4	18.8	130.9	18.6	130.9	18.6	130.9	18.6	130.9	0.0	559.0
1.12	156.3	19.1	132.4	18.9	132.4	18.9	132.4	18.9	132.4	0.0	566.5
1.13	158.1	19.3	133.9	19.1	133.9	19.1	133.9	19.1	133.9	0.0	573.9
1.14	160.0	19.5	135.4	19.3	135.4	19.3	135.4	19.3	135.4	0.0	581.4
1.15	161.9	19.7	136.9	19.5	136.9	19.5	136.9	19.5	136.9	0.0	588.9
1.16	163.7	20.0	138.4	19.8	138.4	19.8	138.4	19.8	138.4	0.0	596.5
1.17	165.6	20.2	139.9	20.0	139.9	20.0	139.9	20.0	139.9	0.0	604.1
1.18	167.5	20.4	141.3	20.2	141.3	20.2	141.3	20.2	141.3	0.0	611.7
1.19	169.4	20.7	142.8	20.4	142.8	20.4	142.8	20.4	142.8	0.0	619.3
1.2	171.2	20.9	144.3	20.7	144.3	20.7	144.3	20.7	144.3	0.0	627.0

**Table 37: Kriel electric feed pump – Power consumption through annular seals**

Power Consumed at a Specific Clearance [kW]											
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		4th Stage		5th Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	Eye	Neck	
0.45	38.0	4.7	33.7	4.7	33.7	4.7	33.7	4.7	33.7	0.0	184.9
0.46	38.9	4.8	34.5	4.8	34.5	4.8	34.5	4.8	34.5	0.0	189.5
0.47	39.8	4.9	35.2	4.9	35.2	4.9	35.2	4.9	35.2	0.0	194.2
0.48	40.7	5.0	36.0	5.0	36.0	5.0	36.0	5.0	36.0	0.0	198.8
0.49	41.6	5.1	36.7	5.1	36.7	5.1	36.7	5.1	36.7	0.0	203.5
0.5	42.5	5.2	37.5	5.2	37.5	5.2	37.5	5.2	37.5	0.0	208.3
0.51	43.4	5.3	38.3	5.3	38.3	5.3	38.3	5.3	38.3	0.0	213.0
0.52	44.3	5.4	39.0	5.4	39.0	5.4	39.0	5.4	39.0	0.0	217.8
0.53	45.2	5.5	39.8	5.5	39.8	5.5	39.8	5.5	39.8	0.0	222.7
0.54	46.2	5.7	40.6	5.7	40.6	5.7	40.6	5.7	40.6	0.0	227.5
0.55	47.1	5.8	41.4	5.8	41.4	5.8	41.4	5.8	41.4	0.0	232.4
0.56	48.0	5.9	42.2	5.9	42.2	5.9	42.2	5.9	42.2	0.0	237.4
0.57	49.0	6.0	42.9	6.0	42.9	6.0	42.9	6.0	42.9	0.0	242.3
0.58	49.9	6.1	43.7	6.1	43.7	6.1	43.7	6.1	43.7	0.0	247.3
0.59	50.8	6.2	44.5	6.2	44.5	6.2	44.5	6.2	44.5	0.0	252.4
0.6	51.8	6.3	45.3	6.3	45.3	6.3	45.3	6.3	45.3	0.0	257.4
0.61	52.7	6.5	46.1	6.5	46.1	6.5	46.1	6.5	46.1	0.0	262.5
0.62	53.7	6.6	46.9	6.6	46.9	6.6	46.9	6.6	46.9	0.0	267.7
0.63	54.6	6.7	47.7	6.7	47.7	6.7	47.7	6.7	47.7	0.0	272.8
0.64	55.6	6.8	48.5	6.8	48.5	6.8	48.5	6.8	48.5	0.0	278.0
0.65	56.6	6.9	49.3	6.9	49.3	6.9	49.3	6.9	49.3	0.0	283.3
0.66	57.5	7.0	50.1	7.0	50.1	7.0	50.1	7.0	50.1	0.0	288.5
0.67	58.5	7.2	50.9	7.2	50.9	7.2	50.9	7.2	50.9	0.0	293.8
0.68	59.5	7.3	51.7	7.3	51.7	7.3	51.7	7.3	51.7	0.0	299.1
0.69	60.4	7.4	52.5	7.4	52.5	7.4	52.5	7.4	52.5	0.0	304.4
0.7	61.4	7.5	53.3	7.5	53.3	7.5	53.3	7.5	53.3	0.0	309.8
0.71	62.4	7.6	54.1	7.6	54.1	7.6	54.1	7.6	54.1	0.0	315.2
0.72	63.4	7.8	55.0	7.8	55.0	7.8	55.0	7.8	55.0	0.0	320.7
0.73	64.4	7.9	55.8	7.9	55.8	7.9	55.8	7.9	55.8	0.0	326.1
0.74	65.4	8.0	56.6	8.0	56.6	8.0	56.6	8.0	56.6	0.0	331.6
0.75	66.4	8.1	57.4	8.1	57.4	8.1	57.4	8.1	57.4	0.0	337.1
0.76	67.4	8.2	58.2	8.2	58.2	8.2	58.2	8.2	58.2	0.0	342.7
0.77	68.4	8.4	59.1	8.4	59.1	8.4	59.1	8.4	59.1	0.0	348.2
0.78	69.4	8.5	59.9	8.5	59.9	8.5	59.9	8.5	59.9	0.0	353.8

0.79	70.4	8.6	60.7	8.6	60.7	8.6	60.7	8.6	60.7	0.0	359.5
0.8	71.4	8.7	61.5	8.7	61.5	8.7	61.5	8.7	61.5	0.0	365.1
0.81	72.4	8.9	62.4	8.9	62.4	8.9	62.4	8.9	62.4	0.0	370.8
0.82	73.4	9.0	63.2	9.0	63.2	9.0	63.2	9.0	63.2	0.0	376.5
0.83	74.4	9.1	64.0	9.1	64.0	9.1	64.0	9.1	64.0	0.0	382.3
0.84	75.5	9.2	64.9	9.2	64.9	9.2	64.9	9.2	64.9	0.0	388.0
0.85	76.5	9.4	65.7	9.4	65.7	9.4	65.7	9.4	65.7	0.0	393.8
0.86	77.5	9.5	66.5	9.5	66.5	9.5	66.5	9.5	66.5	0.0	399.7
0.87	78.6	9.6	67.4	9.6	67.4	9.6	67.4	9.6	67.4	0.0	405.5
0.88	79.6	9.7	68.2	9.7	68.2	9.7	68.2	9.7	68.2	0.0	411.4
0.89	80.6	9.9	69.0	9.9	69.0	9.9	69.0	9.9	69.0	0.0	417.3
0.9	81.7	10.0	69.9	10.0	69.9	10.0	69.9	10.0	69.9	0.0	423.2
0.91	82.7	10.1	70.7	10.1	70.7	10.1	70.7	10.1	70.7	0.0	429.1
0.92	83.7	10.2	71.6	10.2	71.6	10.2	71.6	10.2	71.6	0.0	435.1
0.93	84.8	10.4	72.4	10.4	72.4	10.4	72.4	10.4	72.4	0.0	441.1
0.94	85.8	10.5	73.2	10.5	73.2	10.5	73.2	10.5	73.2	0.0	447.2
0.95	86.9	10.6	74.1	10.6	74.1	10.6	74.1	10.6	74.1	0.0	453.2
0.96	87.9	10.8	74.9	10.8	74.9	10.8	74.9	10.8	74.9	0.0	459.3
0.97	89.0	10.9	75.8	10.9	75.8	10.9	75.8	10.9	75.8	0.0	465.4
0.98	90.1	11.0	76.6	11.0	76.6	11.0	76.6	11.0	76.6	0.0	471.5
0.99	91.1	11.2	77.5	11.2	77.5	11.2	77.5	11.2	77.5	0.0	477.7
1	92.2	11.3	78.3	11.3	78.3	11.3	78.3	11.3	78.3	0.0	483.8
1.01	93.2	11.4	79.2	11.4	79.2	11.4	79.2	11.4	79.2	0.0	490.0
1.02	94.3	11.5	80.0	11.5	80.0	11.5	80.0	11.5	80.0	0.0	496.3
1.03	95.4	11.7	80.9	11.7	80.9	11.7	80.9	11.7	80.9	0.0	502.5
1.04	96.4	11.8	81.7	11.8	81.7	11.8	81.7	11.8	81.7	0.0	508.8
1.05	97.5	11.9	82.6	11.9	82.6	11.9	82.6	11.9	82.6	0.0	515.1
1.06	98.6	12.1	83.4	12.1	83.4	12.1	83.4	12.1	83.4	0.0	521.4
1.07	99.7	12.2	84.3	12.2	84.3	12.2	84.3	12.2	84.3	0.0	527.7
1.08	100.8	12.3	85.2	12.3	85.2	12.3	85.2	12.3	85.2	0.0	534.1
1.09	101.8	12.5	86.0	12.5	86.0	12.5	86.0	12.5	86.0	0.0	540.5
1.1	102.9	12.6	86.9	12.6	86.9	12.6	86.9	12.6	86.9	0.0	546.9
1.11	104.0	12.7	87.7	12.7	87.7	12.7	87.7	12.7	87.7	0.0	553.3
1.12	105.1	12.9	88.6	12.9	88.6	12.9	88.6	12.9	88.6	0.0	559.8
1.13	106.2	13.0	89.5	13.0	89.5	13.0	89.5	13.0	89.5	0.0	566.3
1.14	107.3	13.1	90.3	13.1	90.3	13.1	90.3	13.1	90.3	0.0	572.8
1.15	108.4	13.3	91.2	13.3	91.2	13.3	91.2	13.3	91.2	0.0	579.3
1.16	109.5	13.4	92.1	13.4	92.1	13.4	92.1	13.4	92.1	0.0	585.9
1.17	110.6	13.5	92.9	13.5	92.9	13.5	92.9	13.5	92.9	0.0	592.4

**D.5. DATA FROM PERFORMANCE TESTING ON-SITE**

	Unit 2 Test 2012 (450MW)	Unit 3 Test 2012 (420MW)	
<b>Main Flow</b>			
Density @ 21536.215kPa and 176.002degC	904	912	kg/m <sup>3</sup>
Flow	353	337	kg/s
Flow (l/s)	390	370	l/s
<b>Reheater Spray Flow</b>			
Density @ 8821.831kPa and 172.05degC	900	908	kg/m <sup>3</sup>
Calculated Diameter	54	54	Mm
Flow attemperator	10	3	kg/s
Flow attemperator	11	4	l/s
<b>Boiler Feed Pump Turbine</b>			
Heat Rejected to cooling water	51826	46926	kJ/s
Enthalpy (Condensate)	256	247	kJ/kg
Heat @ Exhaust	57381	51841	kJ/s
Enthalpy @ turbine exhaust	2644.0	2601.3	kJ/kg
Quality @ turbine exhaust	1.00	1.00	
Enthalpy @ turbine inlet (before Valve)	3114	3116	kJ/kg
Entropy (before Valve)	7.266	7.330	kJ/kgC
Enthalpy @ turbine inlet (aft Valve)	3117	3116	kJ/kg
Entropy (after Valve)	7.385	7.335	kJ/kgC
Isentropic Enthalpy (Cylinder)	2401	2409	kJ/kg
Isentropic Enthalpy (Blading)	2441	2410	kJ/kg
Cylinder Efficiency	65.9	72.8	%
Blading Efficiency	70.0	73.0	%
Expansion Power (Turbine)	10198	10261	kW
<b>Pumps</b>			
Delta P booster	506	635	kPa
Delta P (inlet Main Pump to attemperator tap off)	7089	7254	kPa
Density (attemperator spray)	900	908	kg/m <sup>3</sup>
Delta P Main Pump	19804	19373	kPa
Flow (booster and Main pump in)	401	374	l/s
Flow reheater spray	11	4	l/s
Flow (Main pump out)	390	370	l/s

Density @ booster inlet	896	903	kg/m <sup>3</sup>
Density @ main pump outlet	904	912	kg/m <sup>3</sup>
Head (booster pump)	58	72	m
Power (Hydraulic - booster pump)	203	237	kW
Head (Main Pump inlet to attemperator spray)	803	814	m
Head (Main Pump, flow less reheater spray flow)	2233	2165	m
Power Hydraulic (Main Pump inlet to attemperator spray)	80	27	kW
Power Hydraulic (Main Pump, flow less R/H Spray Flow)	7723	7168	kW
Power Hydraulic (Main Pump)	7804	7195	kW
<b>Main Pump Efficiency<sup>66</sup></b>	<b>78.57%</b>	<b>83.61%</b>	<b>%</b>
Estimated Shaft Power <sup>67</sup>	9932	8605	kW
Total Power Hydraulic	8007	7432	kW
<b>Turbine/Pump Set</b>			
Efficiency (Pump Set)	78.5	72.4	%

<sup>66</sup> Testing was done by Tuthukani Performance and Testing and facilitated by Turbine Engineering Department

<sup>67</sup> Mass flow multiplied by change in enthalpy

## APPENDIX E - MATIMBA POWER STATION

### E.1. OPERATING PARAMETERS

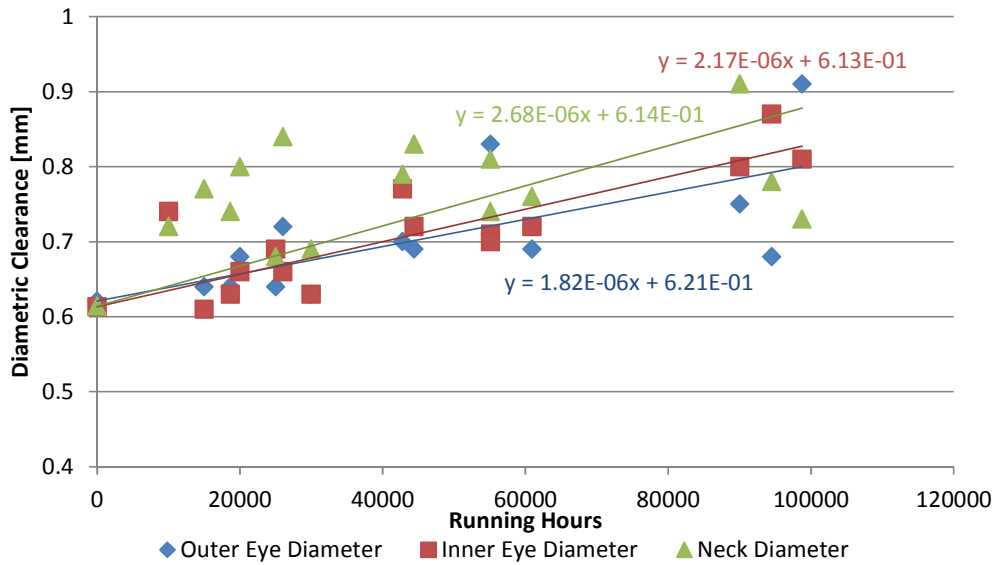
**Table 38: Matimba electric feed pump operating parameters**

Parameter	Design Point	Units
Feed Water Tank Pressure	0.8017	MPa
Feed Water Tank Temperature	170.5	°C
Feed Water Tank Density	896.9	kg/m <sup>3</sup>
Mass Flow at Inlet to Booster Pump	293.86	kg/s
Pressure at Inlet to Booster Pump	0.9142	MPa
Manometric Head – Booster Pump	1.126	MPa
Pressure at Outlet to Booster Pump	2.04	MPa
Pressure at Inlet Connection of Main Pump	2.01	MPa
Mass Flow at Extraction Connection	5.46	kg/s
Pressure at Extraction Connection		MPa
Mass Flow at Delivery Connection of Main Pump	288.4	kg/s
Manometric Head – Main Pump	21.095	MPa
Pressure at Delivery Connection of Main Pump	23.1	MPa
Speed of Main Pump (TOO)	5565	rpm
Power Required at Pump Shaft (TOO)	8510	kW
Main Pump Efficiency (On-site) – TOO	80.4	%

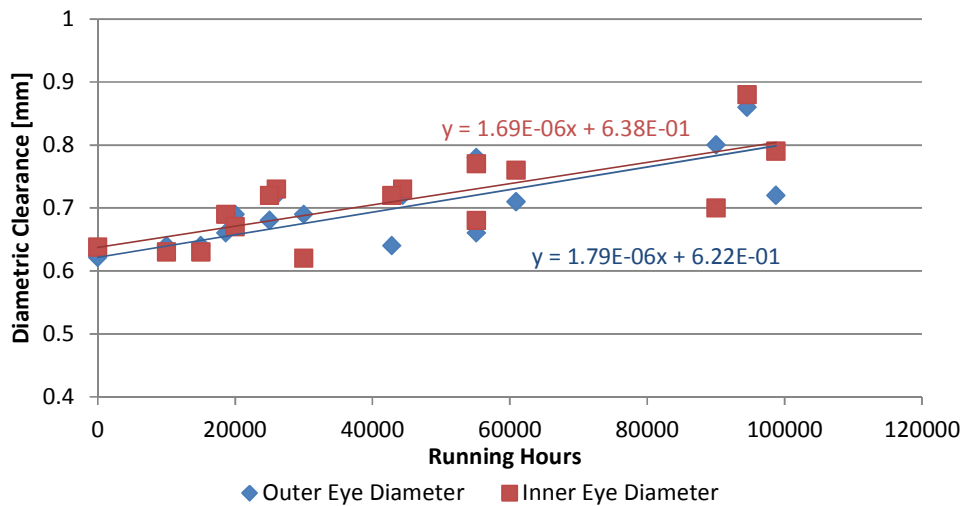
**E.2. 2<sup>ND</sup> AND LAST STAGE IMPELLERS WEAR DATA**

The 2<sup>nd</sup> and last stage impeller has a stepped annular seal, with diameters of 276.39 mm h7 and 269.39 mm h7 on the eye diameters and 182.00 mm h7 on the neck diameters when manufactured

Figure 107 and 108 details the second to last stage impeller diametric clearance for refurbishments at various running hours.



**Figure 107: Matimba 2<sup>nd</sup> stage impeller wear data**



**Figure 108: Matimba last stage impeller wear data**

### E.3. POWER CONSUMED THROUGH ANNULAR SEAL LEAKAGE

**Table 39: Matimba EFP power consumed through annular seals**

Power Consumed at a Specific Clearance [kW]							
Diametric Clearance [mm]	Suction Impeller		2nd Stage		3rd Stage		Balance Device
	Eye	Neck	Eye	Neck	Eye	Neck	
0.6	75.8	11.3	75.8	11.3	77.1	0.0	181.1
0.61	77.5	11.6	77.5	11.6	78.9	0.0	185.6
0.62	79.2	11.8	79.2	11.8	80.6	0.0	190.1
0.63	80.9	12.1	80.9	12.1	82.3	0.0	194.7
0.64	82.6	12.3	82.6	12.3	84.1	0.0	199.3
0.65	84.3	12.6	84.3	12.6	85.8	0.0	203.9
0.66	86.1	12.9	86.1	12.9	87.6	0.0	208.6
0.67	87.8	13.2	87.8	13.2	89.4	0.0	213.3
0.68	89.5	13.4	89.5	13.4	91.1	0.0	218.1
0.69	91.3	13.7	91.3	13.7	92.9	0.0	222.8
0.7	93.0	14.0	93.0	14.0	94.7	0.0	227.6
0.71	94.8	14.2	94.8	14.2	96.5	0.0	232.5
0.72	96.5	14.5	96.5	14.5	98.3	0.0	237.3
0.73	98.3	14.8	98.3	14.8	100.1	0.0	242.2
0.74	100.0	15.1	100.0	15.1	101.9	0.0	247.2
0.75	101.8	15.4	101.8	15.4	103.7	0.0	252.1
0.76	103.6	15.6	103.6	15.6	105.5	0.0	257.1
0.77	105.4	15.9	105.4	15.9	107.3	0.0	262.2
0.78	107.1	16.2	107.1	16.2	109.1	0.0	267.2
0.79	108.9	16.5	108.9	16.5	111.0	0.0	272.3
0.8	110.7	16.8	110.7	16.8	112.8	0.0	277.4
0.81	112.5	17.1	112.5	17.1	114.6	0.0	282.6
0.82	114.3	17.3	114.3	17.3	116.5	0.0	287.7
0.83	116.1	17.6	116.1	17.6	118.3	0.0	292.9
0.84	117.9	17.9	117.9	17.9	120.2	0.0	298.2
0.85	119.7	18.2	119.7	18.2	122.0	0.0	303.4
0.86	121.5	18.5	121.5	18.5	123.9	0.0	308.7
0.87	123.4	18.8	123.4	18.8	125.7	0.0	314.1
0.88	125.2	19.1	125.2	19.1	127.6	0.0	319.4
0.89	127.0	19.4	127.0	19.4	129.5	0.0	324.8
0.9	128.8	19.7	128.8	19.7	131.4	0.0	330.2
0.91	130.7	20.0	130.7	20.0	133.2	0.0	335.6
0.92	132.5	20.3	132.5	20.3	135.1	0.0	341.1
0.93	134.4	20.6	134.4	20.6	137.0	0.0	346.6

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0.94	136.2	20.9	136.2	20.9	138.9	0.0	352.1
0.95	138.0	21.2	138.0	21.2	140.8	0.0	357.7
0.96	139.9	21.5	139.9	21.5	142.7	0.0	363.2
0.97	141.8	21.8	141.8	21.8	144.6	0.0	368.8
0.98	143.6	22.1	143.6	22.1	146.5	0.0	374.5
0.99	145.5	22.4	145.5	22.4	148.4	0.0	380.1
1	147.3	22.7	147.3	22.7	150.3	0.0	385.8
1.01	149.2	23.0	149.2	23.0	152.2	0.0	391.5
1.02	151.1	23.3	151.1	23.3	154.1	0.0	397.2
1.03	152.9	23.6	152.9	23.6	156.0	0.0	403.0
1.04	154.8	23.9	154.8	23.9	157.9	0.0	408.8
1.05	156.7	24.2	156.7	24.2	159.9	0.0	414.6
1.06	158.6	24.5	158.6	24.5	161.8	0.0	420.4
1.07	160.4	24.8	160.4	24.8	163.7	0.0	426.3
1.08	162.3	25.1	162.3	25.1	165.6	0.0	432.1
1.09	164.2	25.4	164.2	25.4	167.6	0.0	438.0
1.1	166.1	25.7	166.1	25.7	169.5	0.0	444.0
1.11	168.0	26.1	168.0	26.1	171.4	0.0	449.9
1.12	169.9	26.4	169.9	26.4	173.4	0.0	455.9
1.13	171.8	26.7	171.8	26.7	175.3	0.0	461.9
1.14	173.7	27.0	173.7	27.0	177.3	0.0	468.0
1.15	175.6	27.3	175.6	27.3	179.2	0.0	474.0
1.16	177.5	27.6	177.5	27.6	181.2	0.0	480.1
1.17	179.4	27.9	179.4	27.9	183.1	0.0	486.2
1.18	181.3	28.3	181.3	28.3	185.1	0.0	492.3
1.19	183.2	28.6	183.2	28.6	187.0	0.0	498.5
1.2	185.1	28.9	185.1	28.9	189.0	0.0	504.6