University of Southern Queensland

Faculty of Engineering and Surveying

Optimising Condenser Air Extraction To Reduce Greenhouse Gas Emissions At The Loy Yang B Power Station

A dissertation submitted by

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towards the degree of

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ABSTRACT

The theory of global warming due to the greenhouse effect has become widely accepted in modern political circles. There is now a raft of initiatives and proposed legislation aimed at to either containing or reducing greenhouse gas emissions. The electrical power industry is very sensitive to the fact it is a major greenhouse gas emitter and has proactively sought to reduce its emissions voluntarily. The Loy Yang B Power Station (LYB) is no exception and has signed on to the generator efficiency standards which mandate continuous improvement in greenhouse gas emissions.

Two approaches are available to reduce greenhouse gas emissions at a power station. One approach is to improve the steam cycle thermal efficiency, so less coal is burnt for the same amount of electricity sent out. The other approach is to reduce the parasitic in-house energy consumption that is required to run the power station drives, so more electricity is sent out for the same amount of coal burnt. This effectively lowers the greenhouse gas emission, per kW, of electricity sent out to consumers. This project will adopt the latter approach without impacting on the former.

The current steam condenser, air extraction system is designed to handle a wide range of turbine operating conditions. It was proposed, that by retrofitting a smaller capacity unit designed specifically for base load operation, significant energy reductions would result. Investigations found a change in the boiler water chemical treatment regime had reduced the air extraction system over-venting requirements to one third of the original design. Actual air in-leakage rates were measured and also found to be substantially lower than original plant design specifications. An investigation into the energy

efficiency of the current system, based on current air in-leakage rates, showed that it was grossly inefficient at turbine base load operation.

Research on various designs of air extraction systems indicated there were four proven designs widely used within the industry. The operating performance, cost and energy efficiency of these designs were compared against one another. The ultimate choice was to select a 2-stage Liquid Ring Vacuum Pump (LRVP), with the innovation of a Variable Frequency Drive (VFD) controlling its flow capacity to prevent cavitation and reduce energy consumption.

Based on the selected design, a new air extraction system that is sized for optimum capacity is predicted to reduce the in-house energy consumption across the power station by 342 kW per annum. This will result in an effective equivalent Carbon Dioxide emission reduction of 3,710 tonnes per annum. The Net Present Value of this project is predicted to return $616,700 (after tax) over 18-years of operation based on a required rate of return of 12% and an electricity cost of $30/MWh.

The presence of air in the steam condenser can lead to condensate being sub-cooled below its saturation temperature. This sub-cooling can result in significant loss of steam cycle thermal efficiency because the heat removed has to be re-added by the boiler and feed heating plant. Research was conducted on the possible causes of sub-cooling. The steam condensers historical performance data was assessed to determine if any steam cycle thermal efficiency improvements could be made in this area. Evaluation of this data showed no significant condensate sub-cooling is occurring within the condenser, and no further action is warranted in this regard.

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Table of Contents

Page No.

List of Figures

List of Tables

List of Appendixes

List of Abbreviations

Abs .... Absolute

ACW .... Auxiliary cooling water

AEP .... Air extraction pump (LRVP)

AVT .... All volatile treatment (Boiler water chemical treatment)

CO_2 .... Equivalent carbon dioxide (Carbon dioxide, Nitrous Oxide and Methane)

CW .... Circulating water (Condenser cooling water)

DAE .... Dry air equivalent (Combined air and water vapour equivalent)

EHG .... Electro-Hydraulic Governor (Turbine load/speed control)

HEI .... Heat Exchanger Institute (USA)

HP .... High pressure

kV .... kilo-Volt

kW .... kilo-Watt

LP .... Low pressure

LRVP .... Liquid ring vacuum pump

LYB .... Loy Yang B Power Station

MCR .... Maximum continuous rating

MW .... Mega-Watt

MWh .... Mega-Watt hour

NC .... Non-condensable

NEM .... National electricity market

NEMCO .. National Electricity Market Company (Operating authority)

NPV .... Net present value

OT .... Oxygenated treatment (Boiler water chemical treatment)

ppb .... Parts per billion

scfm .... Standard cubic feet per minute (Dry air at 20˚C and 101.3 kPa)

SGS .... Scientific Gas Services

TTD .... Terminal Temperature Difference

VFD .... Variable Frequency Drive

Chapter 1. INTRODUCTION

The two generators at Loy Yang 'B' Power Station (LYB) generate an output of 1020 MW's of electricity at base load operation, with about 74 MW's of this electricity being used to power the station's drive motors and electricity needs. This internal electricity use costs around $19.4 million per annum (p.a.) based on the wholesale cost of electricity at $30/MWh, and produces roughly 641,750 tonnes of equivalent Carbon Dioxide (CO_2) emissions p.a. based on 1.2375 tonnes of equivalent CO_2/MWh of electricity generation.

The approximate power consumption of various boiler and turbine plant drives for each generating unit is shown in Table 1.

Table 1 – In-House Power Consumption per Generating Unit

Drive description	**MW Totals**
Induced Draft Fans (2 off)	6.95
Forced Draft Fans (2 off)	2.51
Pulverized Fuel Mills (6 off)	7.29
Boiler Feed Pumps (2 off)	13.93
Circulating Water Pumps (2 off)	3.14
Auxiliary Cooling Water Pump (1 off)	0.30
Air Extraction Pump (1 off)	0.17
Condensate Extraction Pump (1 off)	0.97
Boiler Circulation Pump (1 off)	0.64
Miscellaneous Power Use	1.10
Total Generating Unit In-House Power Use	**37.00**

This project is one of many aimed at reducing the power station's in-house energy consumption and aims specifically at reducing the energy consumption of the steam condenser air extraction systems. The nominal power consumption of the air extraction systems is 200 kW per generating unit. This represents a station electricity cost of $105,190 p.a. and generates 4340 tonnes of equivalent CO_2 emissions p.a.

This project came about when an operator noticed the air extraction system discharge airflow indication, on each generating unit, showed a nominal flow rate of 0.7-1.0 m^3/min. It was suggested this seemed to be a very small airflow when compared to the nominal, 200 kW power rating for the system.

The task then, was to investigate the efficiency of the current air extraction system and to determine the viability of installing a much smaller 'holding' system specifically designed for base-load operation. While the over-riding goal of this project is to keep the in-house energy usage to a minimum, great care needs to be exercised to avoid any possible negative impacts on the steam cycle thermal efficiency, which would result in energy losses of many magnitudes higher than the efficiency gains made.

In the process of studying the condenser and its air extraction system, it was thought appropriate to investigate whether there was any significant condensate 'sub-cooling' occurring within the condenser. Condensate 'sub-cooling' is where the temperature of the condensate falls below the saturation temperature of the steam entering the condenser and this can result in significant losses of steam cycle thermal efficiency. This is because any extra heat removed from the condensate by the condenser must be re-added further along in the cycle by the boiler and feed heating plant.

Chapter 2. BACKGROUND

2.1. Greenhouse Gas Emissions

The greenhouse gases that result from the combustion of Brown Coal are Carbon Dioxide, Methane and Nitrous Oxides. The amount of greenhouse gases that are emitted by the power station are calculated by analysing coal samples on a six hourly basis, and then multiplying the percentage of the constituents by the actual tonnage of coal burned. The total greenhouse gas emission is then reported as an equivalent tonnage of carbon dioxide (CO_2) emitted.

The Loy Yang 'B' Power Station produces 0.99 tonne of equivalent CO2 for every tonne of Brown Coal burnt and requires about 1.25 tonnes of Brown Coal to produce each MWh. This means that 1.2375 tonne of equivalent CO2 is produced per MWh of electricity produced.

There are two ways to reduce the emission of equivalent CO2 in the flue gases discharged by the station. They are:

- Increase the steam cycle efficiency in order to burn less coal for the same generator output.
- Reduce the amount of in-house energy consumption, so that more electrical energy is available to be sent out to consumers, for the same amount of coal burnt, i.e. this effectively lowers the amount of CO2 produced per kW of electricity produced for consumers.

2.2. Types of Steam Turbines

The two main types of steam turbines used to generate electrical power are the backpressure type and the condensing turbine type (Reid & Renshaw, 1988, p.940). The backpressure type is commonly used in process industries where there is the additional requirement to use the exhaust steam for heating in other processes as well as driving the turbine to generate electricity. These other processes include the drying of Brown Coal to make Briquettes and the heating of hospitals and petrochemicals in refineries.

Where the sole purpose of the steam turbine is to generate electricity then condensing turbines tend to dominate because they maximize the power output of the turbine and hence the generator's electrical power output. This arrangement results in a lower overall steam cycle thermal efficiency because the latent heat loss of the condensing steam is given up to the circulating water and eventually rejected to the atmosphere, rather than being used productively for low grade heating purposes. The higher turbine work output, compared to a backpressure turbine, is the result of the higher temperature and pressure drop across the turbine created by the condenser's vacuum at the turbine's exhaust outlet.

2.3. Functions of the Steam Condenser

The main function of the condenser and its associated plant is to maximize the turbine work cycle by producing and maintaining the lowest economic heat rejection pressure and temperature at the turbine exhaust (LYB PDM, 2005, vol.6, sec.2, p.1). This is achieved by cold water circulating through tube bundles and condensing the steam from the turbine exhaust, thereby forming a vacuum within the condenser. It is the relative volume change from steam to condensate that creates this vacuum. For steam entering the condenser at atmospheric pressure this volume ratio is 1,642:1 but at the condenser operating pressure of 9.5 kPa absolute, this volume ratio is 15,436:1.

The effect of differing turbine exhaust pressures can be seen on the T-s Diagram, in Figure 1, where the extra turbine work is represented by the area between the saturation temperature at atmospheric pressure (100 ˚C) and the saturation temperature at 9.5-kPa absolute (44.8 ˚C).

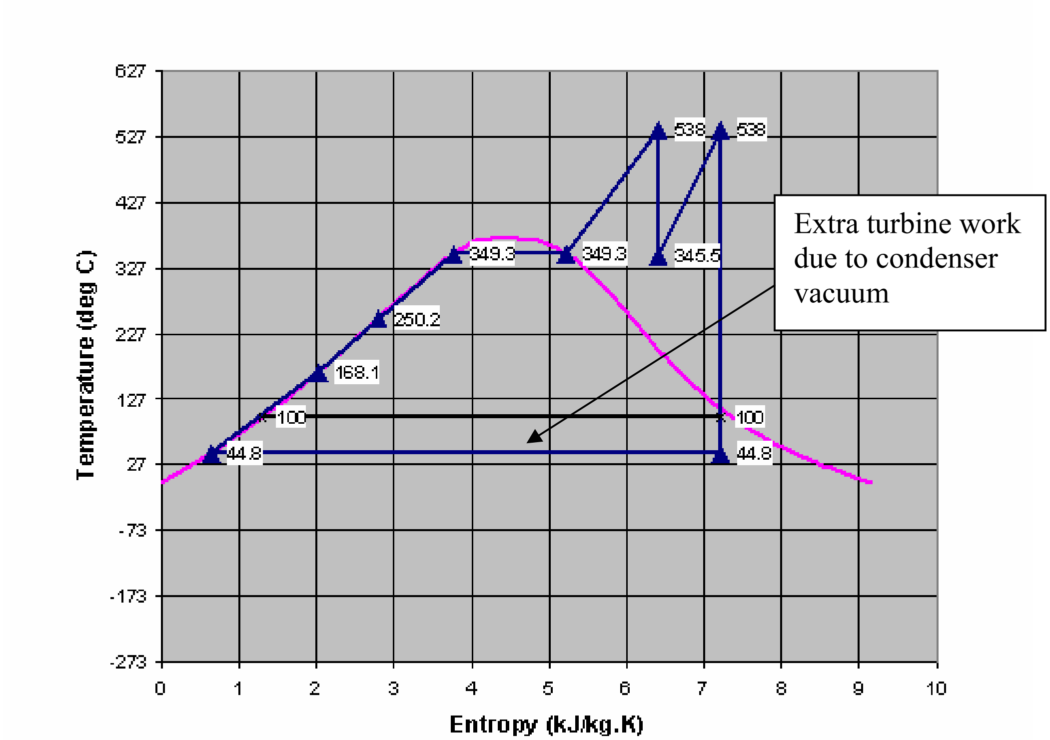

Figure 1 - LYB Rankine Cycle T-s Diagram

The importance of good condenser vacuum on the turbine output should not be underestimated. The following curves, in Figure 2, illustrate the % change in the generating unit's heat rate with changing condenser vacuum and were reconstructed from the LYB Hitachi Manuals.

Heat rate is calculated as the coal energy input divided by electrical energy output. A one percent increase in heat rate will result in an extra 52,500 tonnes of brown coal being burnt p.a. at LYB, costing an extra $236,000 p.a. and producing an extra 52,000 tonnes of equivalent CO_2 p.a.

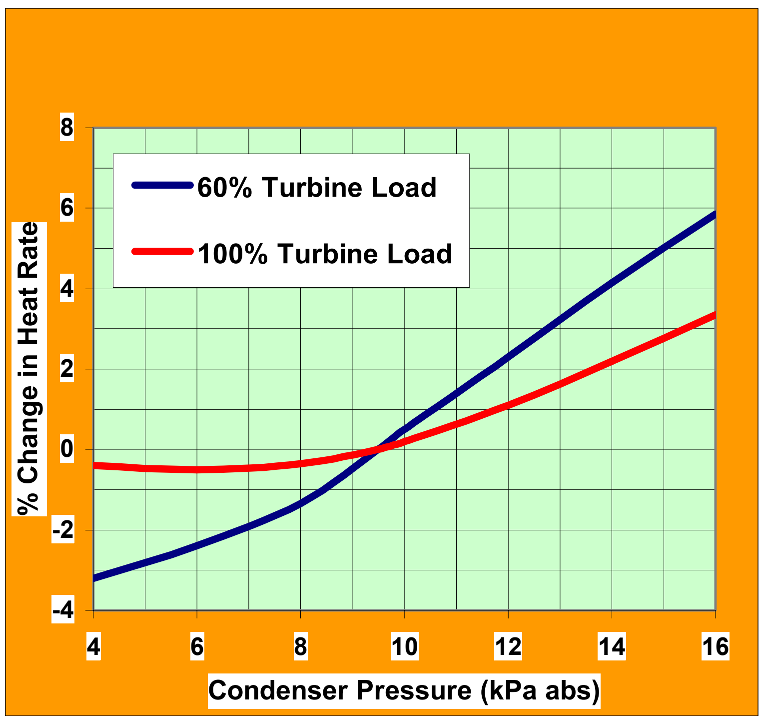

Figure 2 - % Change in Heat Rate Vs Condenser Vacuum

From the curves, it can be seen that at turbine base load (100%) a rise in condenser pressure of just 2.5-kPa would lead to a 1 percent increase in generating unit's heat rate. Insufficient condenser venting could quite easily cause this small rise in condenser pressure, and this is why the operating performance of the air extraction system is so important.

Other primary functions of the condenser are to convert the exhaust steam into water for reuse in the feed cycle, and collect useful residual heat through drainage from the turbine, condensate and feed heating systems. The design of the condenser also provides a net positive suction head for the condensate extraction pumps and facilitates the removal of air and other non-condensable gases from the turbine exhaust steam (LYB PDM, 2005, vol.6, sec.2, pp.1-3).

2.4. Functions of the Air Extraction System

Air, ammonia and other non-condensable gases, resulting from air in-leakage or the decomposition of water treatment chemicals, are present in the turbine exhaust steam and accumulate in the condenser (LYB PDM, 2005, vol.6, sec.2, p.1). The main functions of the condenser air extraction system are to:

- Extract air, ammonia and other non-condensable gases from the condenser to maintain the condenser vacuum.
- Prevent air blanketing of condenser tubing that could dramatically reduce the heat transfer and stop the condensing process.
- Reduce the condensate dissolved oxygen levels that could lead to corrosion of boiler tubing.
- Prevent condensate 'sub-cooling' caused by the presence of air lowering the steam saturation temperature.

Prior to turbine start-up the air extraction system is also used to rapidly raise vacuum by removing air from the condenser shell, turbine casing, steam side of the low pressure (LP) heaters and all other associated steam piping. This initial raising of vacuum is important in order to establish the correct condensate chemical conditions prior to feeding water to the boiler and for protecting the LP Turbine from excessive overheating that could result from 'windage'.

2.5. Effects of Air and NC Gases on the Condenser

The effects of air and other non-condensable (NC) gases may be considered using Dalton's Law of Partial Pressure which states, "Each constituent of a gas mixture exerts that pressure which it would exert, if it alone occupied the containing vessel at the same temperature."

$$\Rightarrow P_{Condenser} = P_{Steam} + P_{Air}$$

Where:
$P_{Condenser}$ = *Condenser Pressure, Pa*
P_{Steam} = *Steam Pressure, Pa*
P_{Air} = *Air Pressure, Pa*

At the turbine exhaust the volume of air is small compared to the steam and the mixture may be regarded as 'pure' steam whose pressure is equal to the condenser pressure, with a corresponding saturation temperature. At the bottom of the condenser, where the volume of steam in the gas mixture has reduced dramatically due to condensation, the volume of air now becomes significant.

If the condenser pressure is assumed to be constant throughout, then the partial air pressure must have increased and the partial steam pressure fallen (when compared to the turbine exhaust steam). This lower steam vapour pressure leads to a lower saturation temperature. As the condensate in the 'hotwell' is in contact with this steam vapour it's temperature can fall due to heat transfer and this can result in sub-cooling (Reid & Renshaw, 1988, p.921). Sub-cooling of condensate and dissolved oxygen pickup can also occur as the condensate falls through localised, accumulated air pockets within the condenser (Harpster, 2002, p.537-539)

Other significant effects of air and non-condensable gases on the condenser is that they can 'blanket' the condenser tubes and greatly reduce the heat transfer, which leads to a loss of vacuum and a corresponding rise in the steam saturation temperature, resulting in reduced turbine work output. This reduction in the rate of heat transfer by 'blanketing' can be significant and if left unchecked could stop the condensing process altogether.

Another effect of air build-up in the condenser is it can lead to high condensate dissolved oxygen levels. If left unchecked this can lead to increased boiler tube corrosion. Likewise, a build-up of ammonia vapour pockets within the condenser can lead to corrosion attack on the copper within the condenser tubes, which can lead to leaks and contamination of the condensate system by the less than pure circulating water.

2.6. The Steam Condenser at Loy Yang 'B' Power Station

The condenser at LYB is a regenerative type and is designed to take the least possible heat out of the steam, which has to be condensed, while maintaining a temperature of the condensate near the temperature of the steam entering the condenser. If excessive heat is removed and the condensate falls below the steam temperature, this is known as ‘sub-cooling’. This type of condenser also facilitates the extraction of air and other non-condensable gases from the turbine exhaust steam, reducing its volume as much as possible before it enters the air pump or ejector (Reid & Renshaw, 1988, p.913).

The LYB condenser consists of three Circulating Water (CW) flow paths, fitted side by side, so as to allow for online maintenance, see figure 3. At turbine base load a minimum of two CW flow paths are required to be in-service. Each condenser CW flow path consists of a 2-Pass cooler arrangement. The CW enters at the top of the condenser water-box, travels across the steam flow path to the return water-box, and then returns across the steam flow path once again to exit at the bottom of the condenser water-box, see Figure 4.

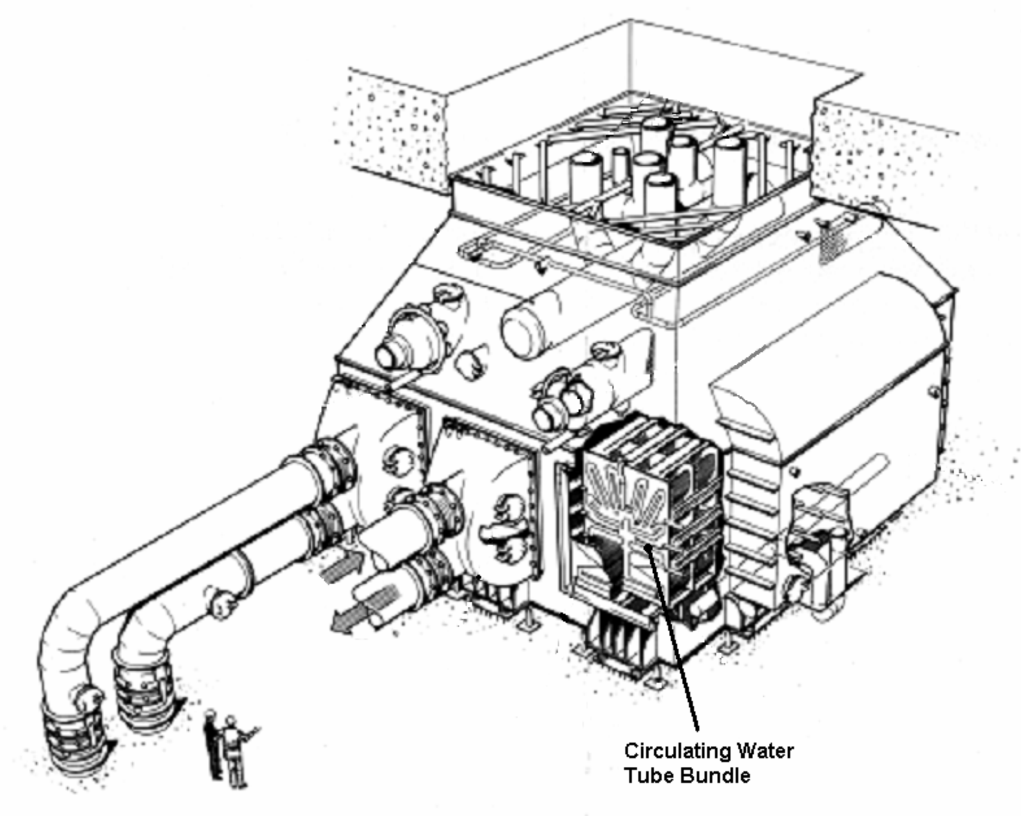

Figure 3 - LYB Circulating Water Flow Arrangement

The various fluid flows for the LYB condenser are shown in figure 4.

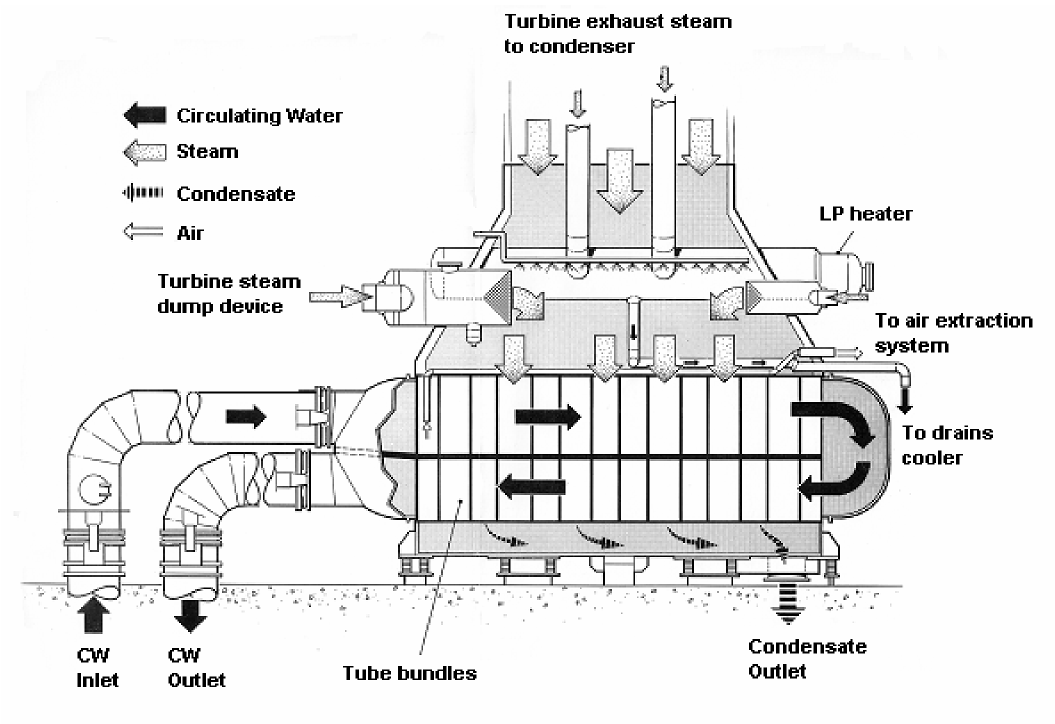

Figure 4 - LYB Condenser Fluid Flow Arrangement

The efficiency of the condenser is dependant on the tube cleanliness, the CW inlet temperature and the CW flow rate. The controlling factors of the CW inlet temperature are the ambient air temperature, humidity, and the efficiency of the cooling tower (LYB PDM, 2005, vol.6, sec.2, p.8).

A cross-section of a circulating water tube bundle for a single circulating water (CW) flow path is shown in Figure 5. Each condenser CW flow path consists of three tube modules positioned to provide maximum exposure to the turbine exhaust steam. The tubes in each module are arranged to create steam lanes with decreasing cross sectional areas. This has the effect of maintaining the steam velocity, maximizing the condensing efficiency through the tube bundles, and eliminating the possibility of air blanketing due to the formation of stagnation zones (LYB PDM, 2005, vol.6, sec.2, p.3).

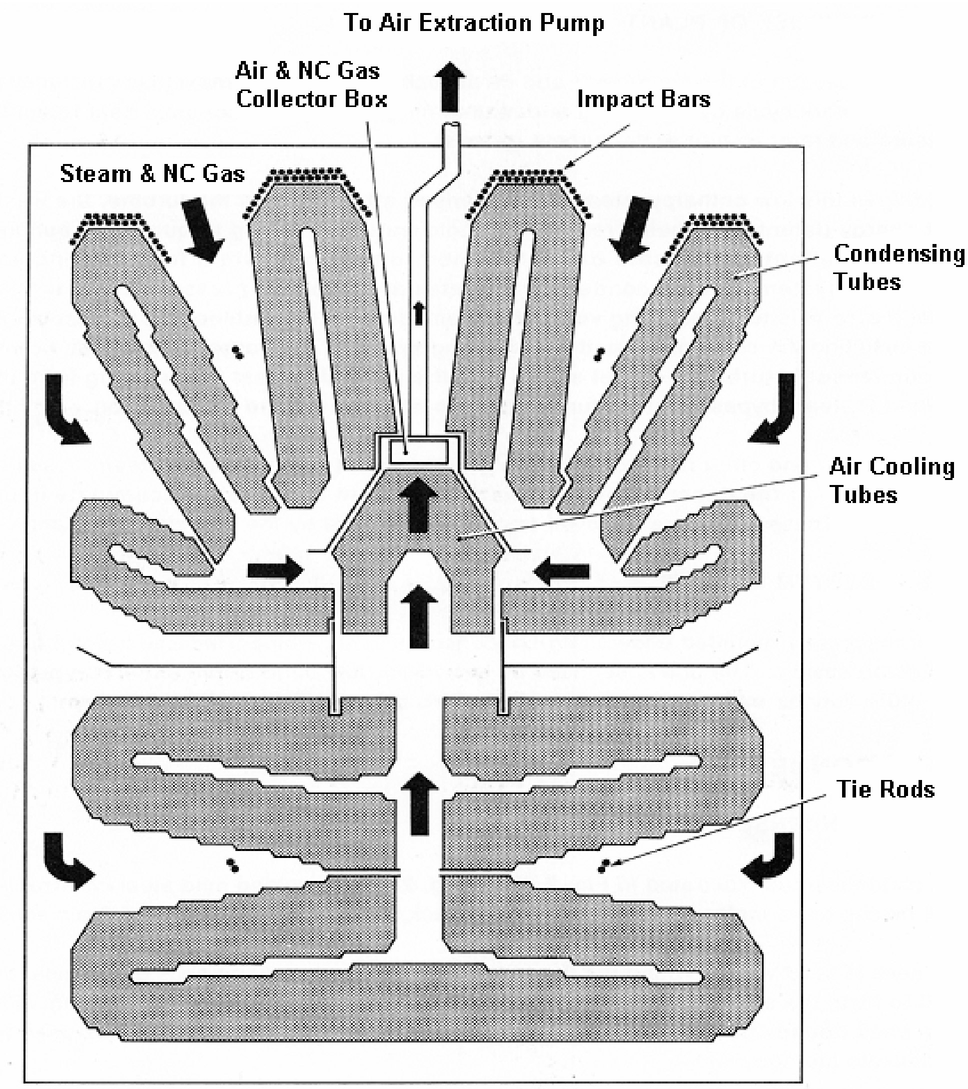

Figure 5 - Circulating Water Tube Bundle Cross-Section

All directional flow of steam and non-condensable gases is directed toward the module centre. Air and non-condensable gases, which are directed to the centre of the module, are drawn up through the air cooler section of tubing and into a box structure extending the full length of each module. This box structure is under the direct suction of the air extraction system.

As the steam condenses on the upper tubes, it falls as condensate to the bottom section of the condenser known as the 'hotwell'. This falling condensate exchanges heat with the steam that it comes in contact with, assisting condensation while maintaining the condensate temperature up to that of the turbine exhaust steam (Reid & Renshaw, 1988, p.914).

Maintaining condensate temperature near the turbine exhaust temperature and eliminating sub-cooling is essential for two reasons. If excessive heat is removed from the condensate then:

- Additional heat must be provided by the feed heating plant and boiler, which will lower the overall thermal cycle efficiency.
- The level of dissolved oxygen in the condensate increases, due to the greater solubility of oxygen at lower temperatures (Harpster & Putman, 2000, pp.3-4) and if left unchecked will eventually lead to increased boiler tube corrosion.

If the condenser pressure rises excessively while the turbine is running, overheating as a result of 'windage' can damage the LP turbine. To protect the turbine, its Electro-Hydraulic Governor (EHG) will progressively unload it as the condenser pressure rises. Turbine unloading from 500 MW will begin to occur at a condenser pressure above 15.5-kPa absolute and continue down to a condenser pressure of 22.5-kPa absolute with a turbine load of 125 MW. If the condenser pressure rises to 25.4-kPa absolute, then a turbine protection trip will automatically initiate. This turbine off-loading versus condenser vacuum is shown in Figure 6.

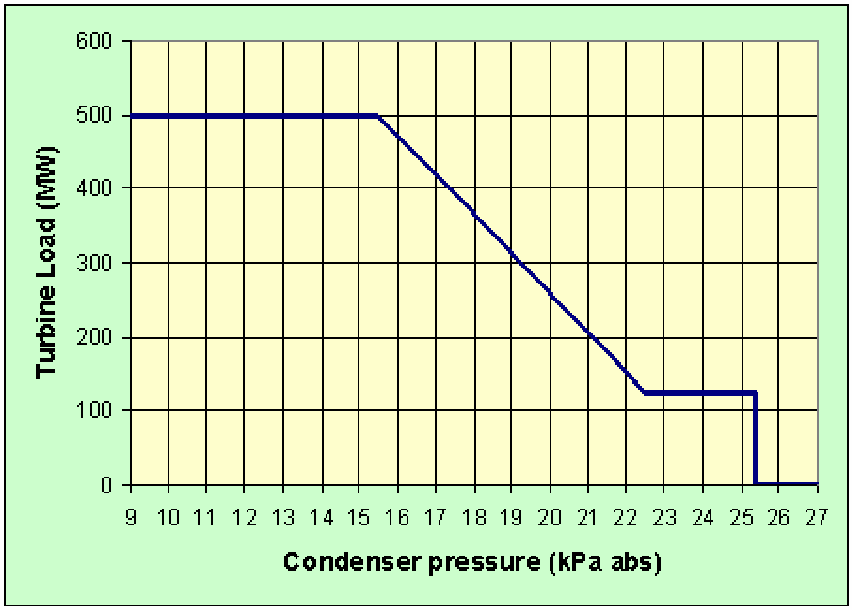

Figure 6 - Turbine Unloading Vs Condenser Vacuum

2.7. The Air Extraction System at Loy Yang 'B' Power Station

2.7.1. System Arrangement

At LYB two 100% duty air extraction pumps (AEP's) are provided to establish the condenser vacuum during turbine start-up and to remove non-condensable gases during normal operation. The AEP's are 2-stage Liquid Ring Vacuum Pumps (LRVP) and are driven by a 3.3 kV, 4-pole electric motor rated at 200 kW through a 2.98:1 reduction gearbox to give a shaft speed of 490 rpm (LYB PDM, 2005, vol.6, sec.1, p.2). There is also an air ejector on the LRVP suction line that is automatically placed into service under normal condenser operating pressures. The system arrangement is shown in Figure 7.

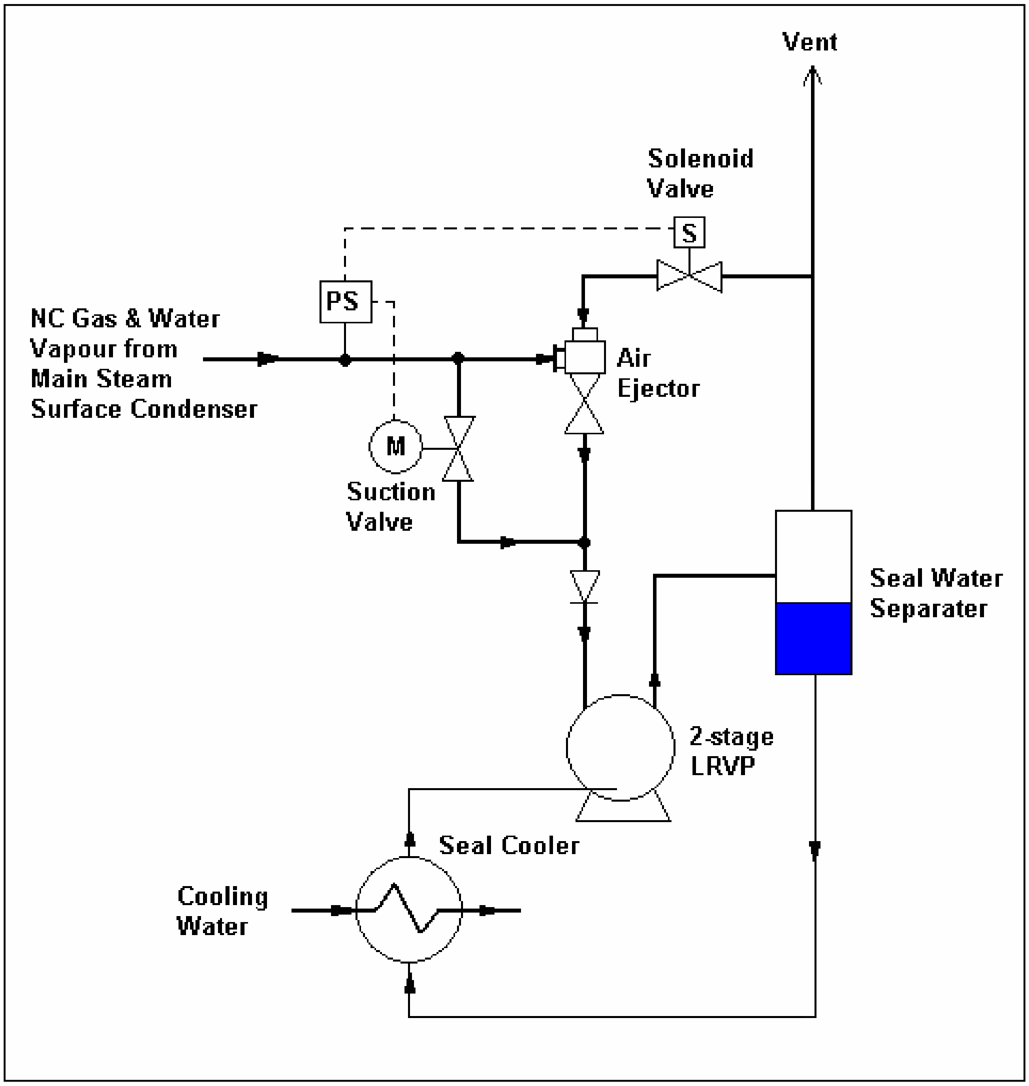

Figure 7 - LYB Air Extraction Pump Arrangement

2.7.2. Modes of Operation

There are two modes of operation for the AEP system, one being a high flow mode and the other a low flow mode. In the high flow mode (known as 'hogging') the air ejector is bypassed by automatically opening the AEP suction valve and closing the air ejector solenoid valve. This mode is activated if the AEP suction pressure rises to 16 kPa absolute and above.

When the AEP suction pressure falls to 15 kPa absolute, the low flow mode (known as 'holding') is activated. This is the normal mode of operation at turbine base load. In this mode the position of the previously mentioned valves are reversed and a large amount of air is recirculated to act as the motive force for the air ejector. It is now the air ejector that extracts air from the condenser.

2.7.3. Range of Operating Conditions

Most air extraction systems are designed to cover a wide range of operating conditions. These operating conditions include:

1. Rapid vacuum raising for turbine start-up – any delay in raising initial vacuum will delay the return to service of the generating unit. Delays could cost up to $4.6 million per hour, if the National Electricity Market (NEM) were trading at its cap price of 10,000/MWh.
2. Turbine partial load operation – reduced turbine loads, particularly in winter, will result in quite low condenser pressures approaching 3.38-kPa.
3. Turbine base load operation – operation around normal design condenser pressure of 9.5-kPa with minimal air in-leakage.
4. Steam dumping to condenser – required to hold the condenser vacuum to acceptable levels with up to 350 MW worth of steam energy being dumped around the turbine and into the condenser.
5. Operation with a significant air leak – it is required to hold condenser vacuum below 15.5-kPa, so that the generating unit can maintain full load until it can be scheduled off for repair.

The design specifications (LYB plant specifications, n.d., sec.3.2) for the current air extraction system at LYB follows this philosophy and states that each pump shall be individually capable of:

- Producing a condenser pressure of 30 kPa absolute in approximately 30 minutes from initial starting conditions.
- Producing a condenser pressure of 10 kPa absolute in approximately 45 minutes from initial start.
- Pumping continuously not less than 0.0544 kg/s of air/water vapour mixture (0.0170 kg/s of dry air) at 21.6 °C and 3.38 kPa absolute of suction pressure.
- Be capable of maintaining the optimum condenser pressure over the full range of normal and abnormal operating conditions.
- Have a design life of 30 years.

While this philosophy ensures the air extraction system is very versatile, in covering all possible operating conditions, it does not place much emphasis on efficiency of the system while operating the turbine at base load under 'normal' conditions. For a base load station, such as LYB, about 99% of turbine operation occurs at full turbine load and this is where the bulk of efficiency gains are to be made.

2.7.4. LYB Air Extraction System Performance Data

The 2-stage LRVP's and Air Ejectors used at LYB are manufactured by SIHI Pumps Australia. The LRVP is a model LPH-10534 and the Air Ejector is a model GPV-15000. The reproduced performance curves for the LRVP and the LRVP/Air Ejector Combination are shown in Figure 8.

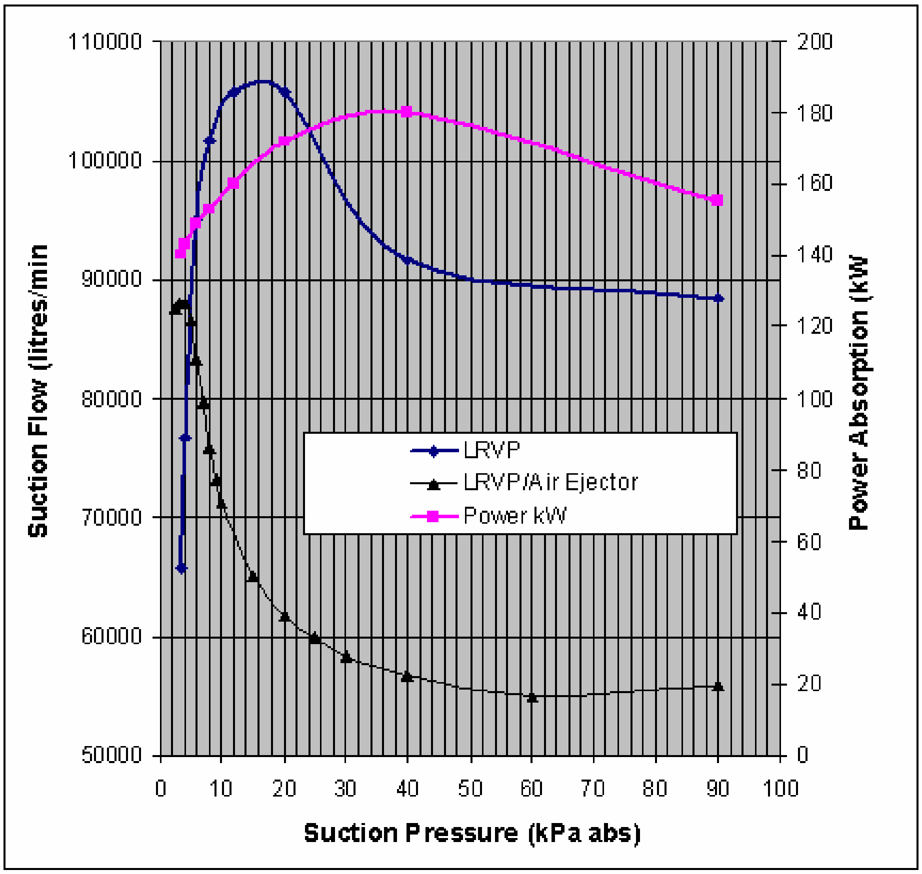

Figure 8 - LRVP LPH-10534 / Air Ejector GPV-15000 Performance Curves
(Reproduced from SIHI performance data, 1990)

The effect of the air ejector in combination with the LRVP can be seen in Figure 8, where the suction capacity has been boosted to 88,000 litres/min at 3-kPa compared to just 66,000 litres/min at 3-kPa for the LRVP alone. It should also be noted from the curves that the LRVP/Air Ejector Combination's capacity drops off rapidly with rising suction pressure, whereas the LRVP actually increases rapidly. This is why there are two modes of operation at LYB, which depend on the condenser pressure at the time.

2.8. Boiler Water Chemistry at Loy Yang B Power Station

There are two different water treatment regimes in use at LYB. These are the traditional all-volatile treatment (AVT) and the newly adopted oxygenated treatment (OT). Without going into too much detail, the AVT regime involves the injection of ammonia hydroxide into the condensate system to control the water's pH within a range of 9.2 – 9.6. It also utilizes condensate de-aeration to maintain low dissolved oxygen levels, typically < 10 parts per billion (ppb).

The OT regime also injects ammonia hydroxide into the condensate, but the water's pH is controlled within the range of 8.0 – 8.5. Instead of de-aerating the condensate to maintain low dissolved oxygen levels, the OT regime actually injects gaseous oxygen in a controlled manner. The aim of this oxygen injection is to control the condensate dissolved oxygen level to a range of 30 – 150 ppb. To prevent removal of gaseous oxygen by de-aeration all feed heater vents and the de-aerator vent to condenser must remain closed during normal operation.

The OT regime is used during normal running because it fosters the formation of more stable oxide layers within the boiler circuit and the lower pH levels reduce the ammonia attack on the condenser tubes. The AVT regime is still used immediately after boiler start-up, and prior to boiler shutdown, to maintain low dissolved oxygen levels and to reduce the solubility of the formed oxide layers by maintaining higher pH levels. This pre-caution ‘conditions’ the boiler water in case of a prolonged generating unit shutdown.

Chapter 3. LITERATURE REVIEW

3.1. Types of Air Extraction Equipment

3.1.1. Steam Jet Ejector System

The steam jet ejector is a type of vacuum pump with no moving parts and is a relatively low cost component, which is easy to operate and requires little maintenance. In a steam jet ejector a steam nozzle discharges a high velocity jet across a suction chamber. This jet stream creates a vacuum, which induces air and NC gas into the suction chamber. The air and NC gas are entrained in the steam and expelled out through a diffuser. The diffuser converts the velocity energy into pressure energy, which helps to discharge the mixture against a pre-determined backpressure (Birgenheier, Butzbach, Bolt, Bhatnagar, Ojala & Aglitz, 1993, p.1).

The steam jet ejector's dimensions fix its capacity, which limits its throughput and the practical limits on the compression it can deliver. To achieve greater compression, multistage steam jet ejectors can be used, arranged in series. Condensers are typically used between successive ejectors in a multistage steam jet ejector system, because they reduce the vapour loading to successive ejectors. This allows smaller ejectors to be used and reduces steam consumption. An after-condenser is sometimes used after the final stage, to condense vapours prior to discharge, although this has no effect on performance (Birgenheier, Butzbach, Bolt, Bhatnagar, Ojala & Aglitz, 1993, p.1). A typical arrangement is shown in Figure 9.

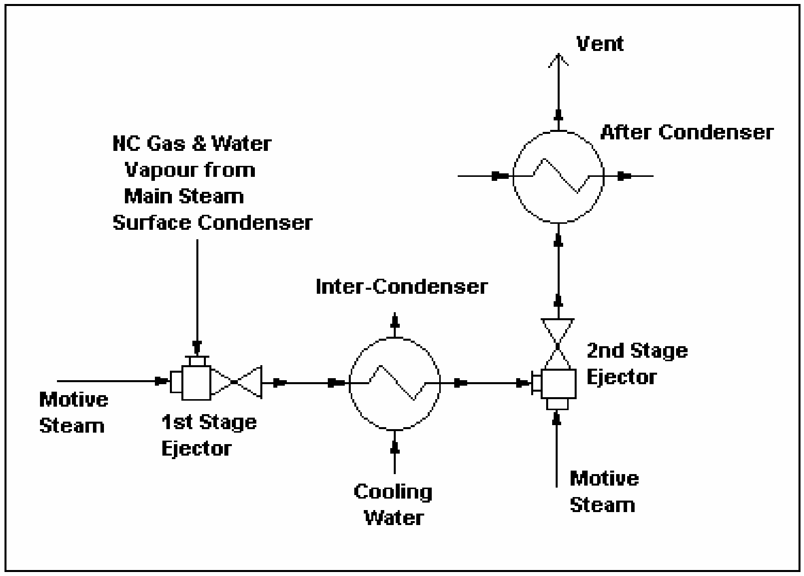

Figure 9 - 2-Stage Steam Ejector System Arrangement

3.1.2. Liquid Ring Vacuum Pump (LRVP) System

The LRVP consists of an eccentrically mounted multi-vane impeller, rotating in a round casing that is partially filled with seal liquid. The seal liquid is thrown to the outside by centrifugal force and the quantity is such that the impeller vane tips are always immersed. Due to the eccentric mounting of the impeller, the volume enclosed between each pair of impeller blades and the liquid ring varies. Air is drawn into the spaces between the impeller vanes at the inlet port, where the volume is increasing, and is then compressed and discharged through the outlet port where the volume is decreasing. A small portion of seal water is constantly lost with the discharge air and must either be constantly made up or recirculated from a seal water separator vessel.

A LRVP may have either a single or multi-stage impeller and it is the vapour pressure of the seal liquid that limits the maximum vacuum obtainable. As the seal liquid absorbs the heat generated by compression and friction, it generally requires cooling to keep it below its saturation temperature. If the seal liquid is allowed to heat up and vaporize, it will take up impeller space and reduce the capacity of the LRVP. If this is allowed to continue, cavitation will occur inside the LRVP, resulting in damage to internal surfaces.

To prevent cavitation the operating vacuum must be limited to 0.85 kPa above the vapour pressure of the seal liquid (Kubik & Spencer, n.d., p.4). LRVP's are positive displacement by nature and if there is insufficient suction load, the suction pressure can fall to the vapour pressure of the seal liquid, causing destructive cavitation to occur. A typical cross section of a LRVP is shown in Figure 10, with a typical system arrangement shown in Figure 11.

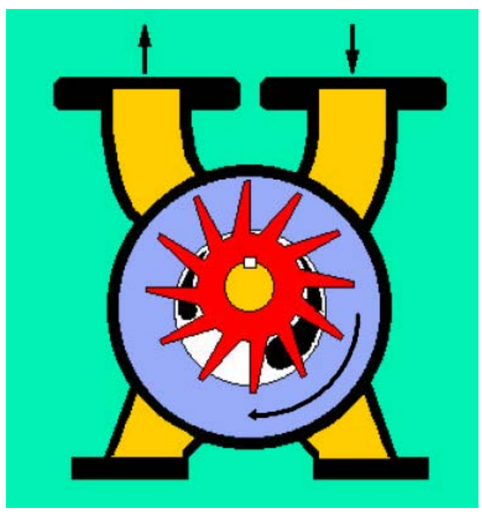

Figure 10 - Cross Section of a LRVP

Some modern LRVP also spray the seal water into the suction line to sub-cool and condense any incoming steam vapour, which reduces the incoming suction volume and increases the LRVP's capacity (Nash, 2005, p.4)

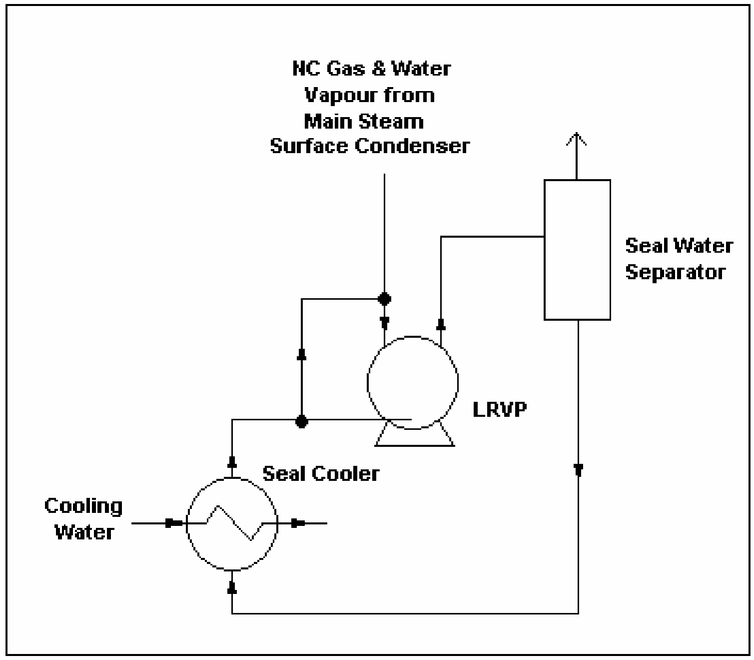

Figure 11 - LRVP System Arrangement

In terms of relative performance, curves for a single-stage steam ejector, a single-stage LRVP and a two-stage LRVP are shown in Figure 12.

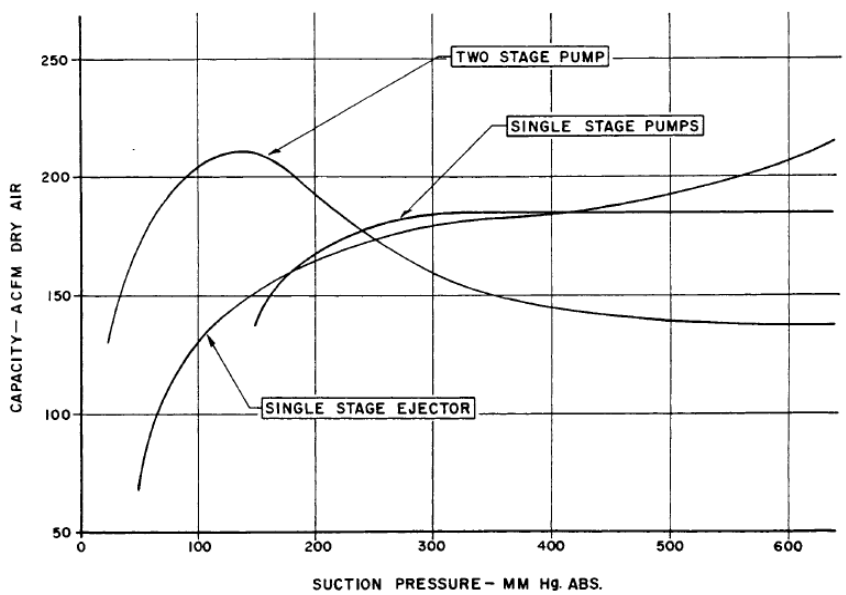

Figure 12 - Steam Ejector Vs LRVP Performance

(Reproduced from Birgenheier & Wetzel, 1988, p.3)

3.1.3. Air Ejector and LRVP System

The air ejector is similar in nature to the steam jet ejector except that it uses atmospheric air as its motive fluid. This air is usually driven by the action of a liquid ring vacuum pump connected to the air-operated ejector as part of an overall system. The advantage of this system is that it raises the LRVP suction pressure so that the LRVP is not prone to cavitation, the system can obtain higher suction vacuums and does not require a steam source (Siemens, 1995, p.5)

The main disadvantages of this system is that it is inefficient because a lot of the air that the LRVP is pumping, is merely being used as the motive force for the air ejector and its suction capacity falls rapidly with a rising suction pressure as would occur if a major condenser air leak developed. The two current air extraction units at LYB are set-up in this arrangement for normal operation when in the low flow mode. A typical air ejector and LRVP system arrangement is shown in Figure 7.

3.1.4. Steam Hybrid System

A steam hybrid system consists of a steam ejector which discharges into a inter-condenser, with the resulting condensate draining back to the main condenser, and the air being extracted from it by a single stage LRVP. The advantages of this system are similar to the air ejector and LRVP system, however there is an additional advantage in the motive steam being condensed and reducing the vapour loading to the LRVP. This allows a smaller capacity LRVP to be used. This system is quite efficient and is common in the United Kingdom where the steam source is generally ‘waste’ de-aerator vent steam (Woodward, Howard & Andrews, 1991, p.383). A typical Hybrid system arrangement is shown in Figure 13.

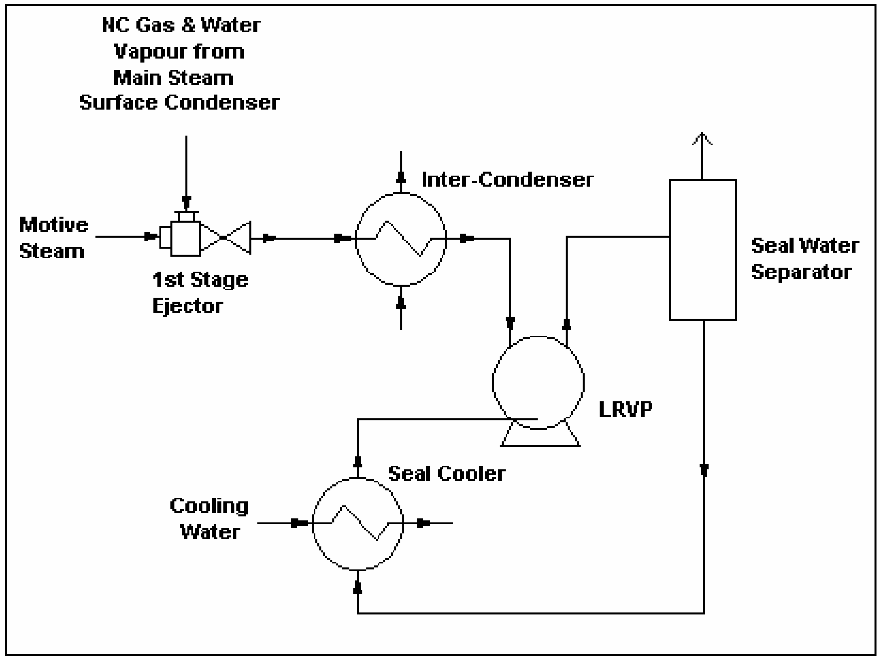

Figure 13 - Steam Hybrid System Arrangement

3.1.5. Other Air Extraction Equipment

There has been a range of compressors and mechanical blowers tried over the years in the power industry, because they offer greater energy efficiency. Unfortunately their maintenance requirements and the susceptibility to water vapour have generally caused operational problems and this has led to them being generally removed from service (Kubik & Spencer, n.d., p.4, Woodward, Howard & Andrews, 1991, p.381).

3.2. Correct Sizing of Air Extraction Systems

Aliasso (n.d., p.1) states that the information needed to accurately size a liquid ring vacuum pump includes:

- Suction pressure
- Suction temperature
- Mass flow rate and the molecular weight of fluid components
- Vapour pressure for each fluid component
- Seal fluid data, if other than water
- Temperature of seal fluid or cooling water
- Discharge pressure

The Heat Exchange Institute of the USA (HEI 1995, p.30) specifies that the condenser venting equipment shall be designed for a suction pressure of 1.0 inch HgA (3.39 kPa abs.), or the condenser design pressure, whichever is lower. This equates to the lower of 3.39 kPa or 9.5 kPa (LYB plant specifications, n.d., sec.3.1) giving a design suction pressure of 3.39 kPa absolute.

The HEI (1995, p.30) also provides guidance on the venting system design suction temperature and states, "that it shall be the steam saturation temperature of the venting equipment's design suction pressure, less the greater of either $0.25(T_S-T_1)$ or 7.5 °F (4.2 °C)", where T_S is the condenser saturation temperature and T_1 is the circulating water inlet temperature.

At LYB, T_S = 44.8 °C at 9.5 kPa abs and T_1 = 27 °C at design specifications (LYB plant specifications, n.d., sec.3.1). Interpolating from Rodgers & Mayhew (1996, p.2), the steam saturation temperature at 3.39 kPa is 26.1 °C. Therefore the design suction temperature of the venting equipment, as specified by the HEI, can be calculated as follows:

$$T = 26.1 - (0.25 \times (44.8 - 27)) = 26.1 - 4.45 = 21.7\ °C$$

When the air extraction system is venting the steam condenser it will draw off not only air and NC gases, but also water vapour. The HEI (1995, p.30) provides the following formula to calculate the water vapour load, to saturate the non-condensable gases:

$$W = \frac{18}{MW_{NC}} \times \frac{P_W}{P_t - P_W}$$

Where:
W = lb. of water vapour per lb. of non-condensable gas
MW_{NC} = molecular weight of non-condensable gas
P_W = absolute "water vapour" pressure corresponding to temperature at condenser vent outlet, in. HgA
P_t = absolute "total" pressure at condenser vent outlet, in. HgA

However, when the non-condensable gas is dry air, W may be obtained directly from the HEI's (1995) "Standards for steam surface condensers"- Appendix E. Using the previously obtained design suction temperature of 21.7 °C and pressure of 3.39 kPa absolute, the water vapour load from Appendix E can be read off as:

W = 2.05 kg water vapour/kg of dry air.

In terms of the ratio of the NC gas load removed to the design capacity of the venting equipment, the HEI (1995, p.12) specifies the following ratios, based on expected condensate dissolved oxygen levels:

Table 2 - HEI Recommended Design Over-Venting

Design Capacity (SCFM)	Actual Load/Design Capacity Ratio	Condensate Oxygen Content (ppb)
20 – 40	0.50	42
	0.24	14
	0.15	7

It can be seen from the HEI Load/Design Capacity recommendations that considerable over-design of the venting system is required in order to achieve very low oxygen levels.

There is a range of international standards available to determine the design capacity of venting systems for steam condensers, in power stations. Some of these standards include, the American HEI standard, the British BEAMA standard and the German VGB guideline. The design capacity of the air extraction system in these standards is based on the mass flow of steam into the condenser at rated load.

Based on the LYB steam flow into the condenser of 283 kg/s at a rated load of 500 MW with two condenser inlet openings, the HEI standard specifies a design capacity of 51.1 kg/h of dry air (HEI, 1995, Table 9). The German VGB guideline however specifies a design capacity of around 19.0 kg/h which, although less than the HEI standard, is still considered a conservative guideline (Siemens, 1995, p.2). Using equation 6, in Appendix B, these capacities equate to 707 litres/min and 263 litres/min of dry air respectively.

The HEI recommended maximum allowable condenser air in-leakage rate is specified as 1 scfm per 100 MW of generating capacity and this figure is endorsed by the Electrical Power Research Institute (EPRI) of the USA as being a suitable limit for maintaining condensate chemical conditions (Dooley, 2002, C-1). Using equation 7, in Appendix B, this equates to 142 litres/min of condenser dry air in-leakage based on 5 scfm for a base load of 500 MW at LYB.

While the intention of this project is to minimize the in-house energy consumption at LYB by optimising the capacity and design of the condenser air extraction system, Kubik & Spencer (n.d., p.6) contend that "it is difficult to understand the obsessive desire to 'undersize' the venting system when the energy consumption of all venting systems is minuscule when compared to the total quantity of energy involved in the equipment serviced."

They then go on to say that, "in fact inadequate condenser venting can result in energy losses in the plant heat rate greater than the parasitic in-house power saved by selecting a smaller capacity venting system." They particularly make the point that under turbine partial load conditions, the capacity of the venting system may drop off, with the venting system limiting condenser vacuum instead of the condenser. This situation can result in substantial plant heat rate losses.

Although some of the argument put by Kubik & Spencer holds true, depending on the design of the air extraction system employed, it is still reasonable to try and attempt to minimize the in-house energy consumption. As with all engineering design decisions, it is essentially a question of arriving at the most economical and feasible design, which also meets all the practical performance criteria.

3.3. Efficiency of Air Extraction Systems

As a benchmark on relative efficiency, Woodward, Howard & Andrews (1991, p.383) mention about an air extraction system that has gained popularity in the United Kingdom (UK) 660 MW market due to its low power requirement. This steam hybrid system consists of a steam jet ejector utilizing de-aerator 'waste' steam to the condenser as its motive force and a LRVP. It is claimed that this system has a "low power consumption of about 0.727 kW/kg/h of air, which is approximately half that of other systems".

As a rough comparison, based on approximately 200 kW of input power and discharge airflow of 0.0170 kg/s, the power consumption for LYB is 3.268 kW/kg/h. This power consumption is approximately 450% higher than the UK system mentioned and tends to suggest that there may be significant room for improvement. It should be noted however that the de-aerator vent to condenser is left shut at LYB, as part of the requirements for the boiler water OT chemical treatment, prohibiting this steam source as an available option for this project.

One reason for this higher power consumption relates to the philosophy of using the same system to cover the full spectrum of operating conditions likely to be encountered. The LRVP is a specific form of rotary positive-displacement pump (Chaudhury, 1996) which requires a minimum load flow, otherwise its suction pressure will continue to fall until its seal liquid reaches its vapour pressure. When vapour pressure is reached the seal liquid will flash off and the pump will cavitate (Lines, Athey & Frens, n.d., p10).

There are several ways of overcoming this problem. Two simple methods are to install an inlet vacuum relief valve or an air bleed valve (Aliasso, n.d., p.1). Both of these methods can waste energy if there is over-capacity at the holding point (Skelton, 1998). Another more innovative approach is to use a variable frequency drive (VFD) for the LRVP to control its speed and flow capacity. This can also reduce energy consumption, if the LRVP is running at over-capacity for the NC gas load (Skelton, 1998).

Yet another common approach is to use a hybrid system that consists of either a steam jet ejector or an air operated ejector as the primary means of extraction followed by a LRVP as the secondary means of extraction. (Kubik & Spencer, n.d., p.4). This system ensures a higher inter-stage pressure for the LRVP suction and reduces the likelihood of the suction pressure falling to the seal liquid vapour pressure, providing the seal liquid is kept cool.

The current air extraction system at LYB is one of these hybrid systems. It utilizes an air-operated ejector in combination with a two stage LRVP when operating in the 'holding' mode. In this mode of operation, most of the LRVP's air load is purely re-circulated air used to provide the motive force for the air-operated ejector. Kubik & Spencer (n.d., p.4) claim that this system is not very efficient in terms of its energy requirements, because of the large volume of air that the LRVP is required to pump, and suggest a hybrid system may not even be necessary for a two stage LRVP.

Compressors and mechanical blowers have been used for condenser exhaust service, but their use has decreased as maintenance requirements have caused operating problems with those systems. These devices have perhaps the lowest energy consumption but have high capital and maintenance costs (Kubik & Spencer, n.d., p.4). A rotary pump (Le Blanc) which uses water as the motive fluid was also once commonly used in 500 MW units but is now of little relevance to modern practice (Woodward, Howard & Andrews, 1991, p.383).

As condenser air extraction is critical to the condenser performance (and the output of the steam turbine/generator) compressors, mechanical blowers and rotary pumps are not considered viable options despite their low power consumption because of potential operating problems associated with high maintenance.

Another type of air extraction system is a multi-stage steam ejector. Multi-stage steam ejectors have a low capital cost, are not limited by condenser pressure or cooling water temperature, have no moving parts, are most reliable and require the least maintenance of all venting systems (Kubik & Spencer, n.d., p.3). The major downside of multi-stage steam ejectors is the large amount of energy used to create the high-pressure motive steam. The relative energy required to run a two-stage steam ejector has been shown to be approximately double of that for a two stage LRVP (Kubik & Spencer, n.d., p.5)

In summary, the main types of air extraction systems used for condenser gas removal in modern power stations are the two-stage steam ejector, the two stage LRVP and hybrid systems utilizing either a steam or air ejector in combination with a LRVP. In terms of energy consumption, Kubik & Spencer (n.d., p.5) provides the following relative comparison, when the equipment is sized to meet HEI recommended capacities:

Table 3 - Comparison of Air Extraction Efficiency Based on Type

Air extraction system	**% of total plant steam**
2 stage steam ejector	0.030 %
2 stage LRVP	0.016 %
Hybrid system fitted with a steam ejector	0.019 %

Unfortunately this comparison does not include a hybrid system fitted with an air ejector, which is the current configuration of the air extraction system at LYB. Birgenheier and Wetzel (1988, p.5) also provide an analysis of steam ejector running costs as compared to the electricity costs required to run a LRVP. These are shown in Figure 14.

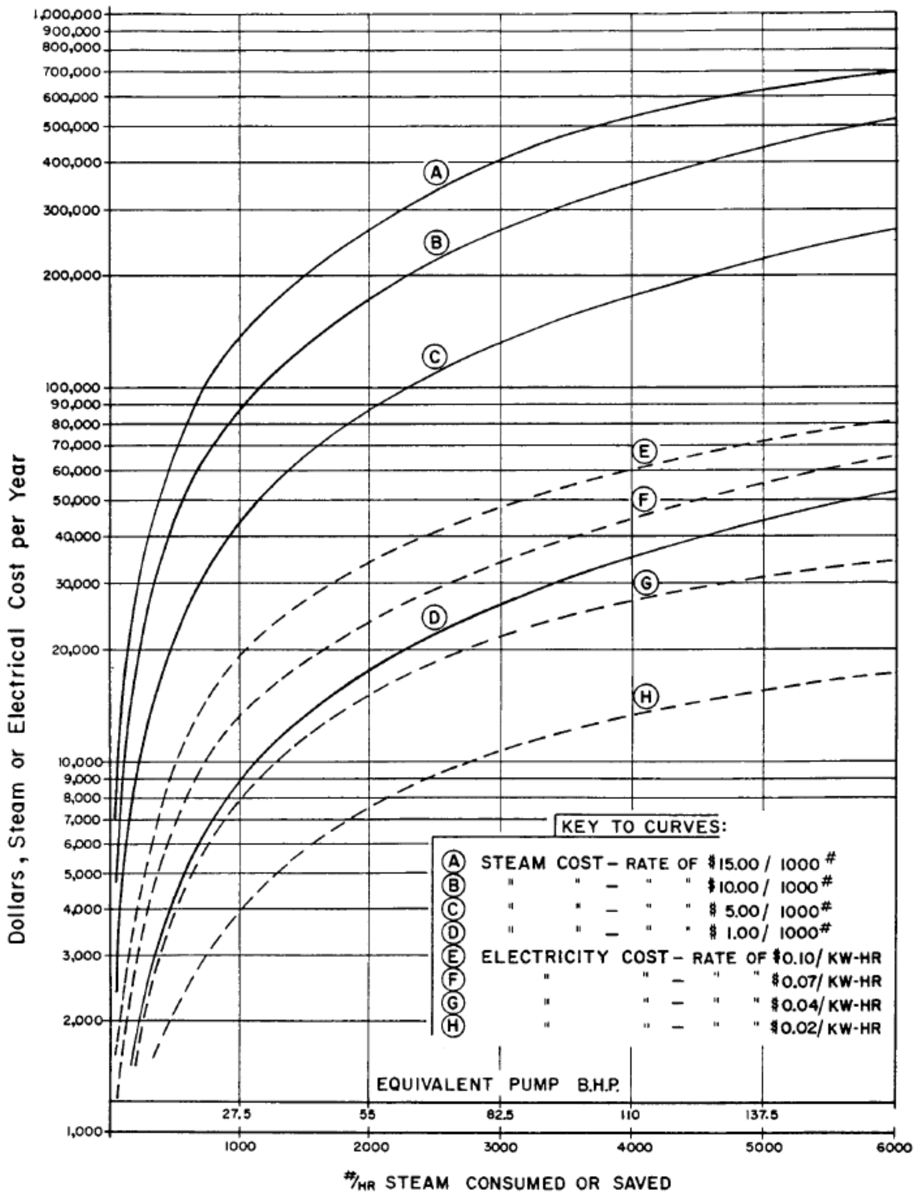

Figure 14 - Steam Vs Electricity Cost of Air Extraction

(Reproduced from Birgenheier & Wetzel, 1988, p.5)

3.4. Condenser Performance and Condensate Sub-Cooling

Condensate sub-cooling, which is also known as ‘condensate depression’, is defined as the saturation temperature corresponding to the vapour pressure of the steam entering the condenser inlet minus the temperature of the condensate leaving the condenser 'hotwell'. Condensate sub-cooling is undesirable because it means that excess heat, which serves no useful purpose, has been removed during the condensation process. This additional heat removed, must then be re-added to the thermal cycle by providing additional boiler firing.

A one-degree drop in the condensate temperature from the specified hotwell temperature of 44.8 °C (LYB plant specifications, n.d., sec.3.1) will result in a heat loss of 4.15 kJ/kg (Rodgers & Mayhew, 1996, p.2). Based on a condensate flow rate at LYB of 298.384 kg/s (Hitachi, 1991, Drg. JD-130-740) this represents a heat loss from the condensate of 1.238 MW/°C. With the overall thermal cycle efficiency at LYB being around 38%, every one-degree of condensate sub-cooling will require an extra 3.258 MW of heat energy to be added by the boiler.

Putman and Harpster (2000, pp.1-2) suggest three main causes of condensate sub-cooling, with these being:

- Lower than design circulating water inlet temperatures, especially during the winter months.
- A 'choked' LP turbine exhaust anulus, with the condenser pressure operating below its pressure at the point of choking.
- In-adequate condenser design.

These conditions are unlikely to be problems at LYB because of the relatively mild winters (compared to Europe and elsewhere) and the generally base-load turbine load profile. Despite this, because of the large energy losses, which can be incurred, an evaluation of condensate sub-cooling will be made during turbine load operation in winter.

An accepted method to measure the condenser performance is to calculate the Terminal Temperature Difference (TTD) because the condenser overall heat transfer co-efficient is related to it by an inverse function (Trela, Butrymowicz, Gluch, Gardzilewicz, Ihnatowicz, & Zieliński, 2000, p.2). Therefore a decrease in the condenser overall heat transfer co-efficient results in a rise of the condenser TTD and vice-versa.

The condenser (TTD) is calculated as the turbine exhaust steam temperature minus the circulating water outlet temperature. At LYB the specified condenser TTD is 3.2 °C, with a cleanliness factor of 85% (LYB plant specifications, n.d., sec.3.2). Factors that will adversely affect the condenser TTD are condenser tube fouling internally on the CW side and condenser tube blanketing externally on the steam side, due to insufficient NC gas removal.

In colder weather the circulating water inlet temperature will drop due to improved cooling tower performance. The circulating water outlet temperature will also drop in response, because the same amount of heat needs to be transferred, and the temperature difference between the circulating water inlet and outlet will remain the same. The lower circulating water inlet temperature will result in improved condenser performance and hence lower condenser pressures and steam saturation temperatures.

If the turbine happens to be operating at partial load, then the amount of heat transfer will be even less with the condenser pressure reducing even further. At lower condenser pressures the specific volume of steam increases rapidly. It is under these conditions that choking of the turbine exhaust anulus can occur due to the increased steam volume passing through it. If the turbine exhaust anulus becomes ‘choked’, and the turbine pressure falls further due to even colder circulating water inlet temperatures, then sub-cooling will result.

As mentioned previously in Chapter 2.5 accumulated air and other NC gases can also cause condensate sub-cooling by lowering the localised steam saturation temperature.

Chapter 4. CURRENT SYSTEM EVALUATION

4.1. LYB Condenser Air In-Leakage Rates

There is an installed paddle wheel type flow meter on the discharge pipe of each generating unit's air extraction system. These meters were reading an erratic 0.7 –1.0 litres/min. There were major doubts about the accuracy of these meters, not only because of the erratic readings, but because the 50 mm paddle wheel was measuring the airflow in a 250 mm discharge pipe with extremely low discharge velocities.

Attempts had been made previously by the engineering group to measure the condenser air in-leakage rates with little success mainly due to the low air extraction discharge flow and the nature of the metering equipment used. Initially it was decided to determine how close the air in-leakage rates were to the air extraction system's specified design flow capacity of 0.017 kg/s. Using equation 6, in Appendix B, this equates to 847 litres/min.

To measure the discharge airflow, all alternative discharge points were blocked off so that the system discharge airflow was forced through a 25 mm vent line and ball type 'Rotometer' (0-150 litres/min) mounted at the turbine floor level. The 'Rotometer' used is shown in Figure 15. A rubber pad and weighted steel plate, held in place with duct tape, was used to cap off the 250 mm discharge pipe on the turbine house roof, as shown in Figure 16.

Wet rags were stuffed up the 50 mm seal water separator tank overflow line to block off this discharge point. The preliminary results were:

- Unit 1 < 10 litres/min
- Unit 2 = 20 litres/min

Figure 15 - Ball Type Rotometer

Figure 16 - Air Extraction Turbine Roof Discharge Cap

These preliminary results seemed to indicate that the air leakage rates were a long way off specification. The method of measurement, however, was very crude and the degree of confidence was rather low. The tests were then repeated with a calibrated rotary type gas flowmeter, kindly loaned from "Scientific Gas Services (SGS) Pty Ltd", shown in Figure 17.

Figure 17 - Rotary Type Gas Flowmeter

It was important to employ positive sealing arrangements in order to guarantee accurate airflow measurements. The capping system on the turbine house roof was retained but was leak tested with soapy water to prove that it was airtight. A metal blank was bolted in place, at a flange on the seal water separator overflow line, instead of using rags. The metal blank is shown prior to insertion in Figure 18.

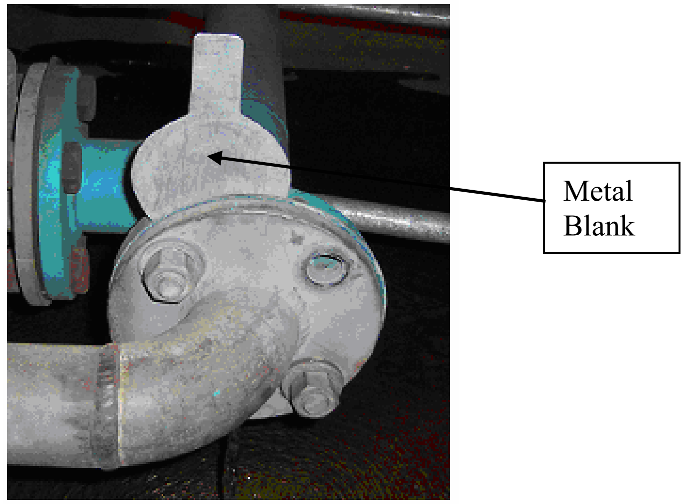

Figure 18- Metal Blank

Readings were taken on the 1st August 2005 on Unit 2 and the 2nd August 2005 on Unit 1, using a calibrated rotary type gas flowmeter with a 2 m^3/h maximum capacity. An electronic stopwatch was used for timing, with readings taken from the meter's counter at one-minute intervals. When the readings were steady, three consecutive results were averaged to obtain a flow measurement. Unit 1 indicated an air in-leakage rate of 49 l/min while Unit 2 indicated an air in-leakage rate of 51 l/min.

To help verify the validity of these readings, the air extraction system was isolated from the condenser on both units to perform a vacuum decay test. Unit 1's vacuum was found to decay at 3.04 kPa over one hour, whereas Unit 2 decayed exactly 1.0 kPa over one hour. Woodward, Howard, & Andrews (1991, p.368) state that expected vacuum decay rates are approximately 6 kPa per hour and that unacceptable vacuum decay rates are 24 kPa per hour.

While this showed that both generating unit's had low air in-leakage rates, it also brought the realisation that the measuring flow meter's capacity of 2 m^3/h (33 l/min) was being exceeded and a larger capacity flowmeter of the same type was required. The original 'Rotometer' (0-150 l/min) flowmeter was then used but found to restrict the airflow excessively. This meant that this meter was also unsuitable for measurement purposes due to its excessive flow restriction.

Another calibrated, rotary type gas flowmeter (5 m3/h maximum flow) was again sourced from SGS. Readings, taken on the 4th August 2005 on Unit 2 and 5th August 2005 on Unit 1, indicated air in-leakage rates of 48 l/min and 85 l/min respectively. Although the maximum flow capacity of 5 m3/h (83 l/min) was just exceeded on Unit 1, these results were deemed acceptable because previously the same type of meter had demonstrated that it could measure up to 50% over-capacity on Unit 2. These results were also more consistent with the condenser vacuum decay tests. See Table 4 for results.

Table 4 - Unit 1 & 2 Air In-leakage Field Measurements

Date	Plant	Discharge flow litres/min.	Condenser Pressure kPa abs	Ambient Temperature deg C	Barometric Pressure hPa
5/08/2005	1-AEP-2	85	7.64	6.12	1026
1/08/2005	2-AEP-1	51	8.87	14.5	1027
4/08/2005	2-AEP-2	48	8.40	10.22	1016

These results need to be corrected to standard air pressure and temperature. The corrected flow rates to 101.3 kPa and 20˚C respectively are determined by equation 1, in Appendix B. The corrected air in-leakage rates equate to 90 litres/min for Unit 1and 50 litres/min for Unit 2 at their respective condenser pressures, at the time of measurement. Condenser air in-leakage will however increase, when the condenser pressure decreases further at turbine low load operation. This is because as the pressure difference between the condenser and atmosphere increases, air will be sucked into the condenser at a greater rate through the source of the leak.

The measurement of the condenser air in-leakage rate at the normal minimum load experienced during operation (300 MW) was therefore highly desirable because, at reduced condenser pressure, the performance of the LRVP would be at its lowest and the air in-leakage rate at its highest. An absence of electricity market off-loading during the period of testing meant that deliberate off-loading would be required in order to perform this test.

Off-loading was estimated to cost $4,400 with a NEM spot price at $11/MWh if the tests were conducted at 4.00 a.m. in the morning. To avoid this cost, it was decided to instead to determine the increased rate of air in-leakage at lower condenser pressures analytically rather than experimentally. A relationship between the condenser to atmospheric differential pressure and the air in-leakage flow rate was derived to provide these estimates. The estimated flow rates derived from equation 2, in Appendix B, are shown in Table 5.

Table 5 - Unit 1 & 2 Estimated Air In-leakage Rates Vs Condenser Pressure

Condenser Pressure	**25.1**	**16.7**	**9.5**	**8.2**	**6.5**	**4.1**	**3.3**	**kPa**
Air in-leakage Unit 1	81.2	85.6	89.1	89.7	90.5	91.7	92.0	l/min
Air in-leakage Unit 2	45.3	47.7	49.7	50.1	50.5	51.1	51.3	l/min

4.2. LYB Current System Energy Consumption

There were three sources of energy consumption identified on the current air extraction system at LYB. These were the electrical power for the duty AEP drive, the electrical power for the anti-condensation heater (on the standby AEP drive) and the pumping power required to supply cooling water through the coolers on both AEP units, as the cooling water is on continuously whether the AEP is in or out of service. The field measurements were obtained with both generators operating at base load of 510 MW. The date of measurement, the results and measuring method used are shown in Tables 6-8, with the system total shown in Table 9:

Table 6 - Anti-Condensation Heater Energy Consumption

AEP Anti-Condensation Heater Power Consumption Assuming load purely resistive and Power Factor =1.0				
Date:	Plant Item:	Volts: Digital Multimeter	Amps: Installed Metering	Power: Calculated
4/9/05	1-AEP-2	240.3 V	1.66 A	0.399 kW
4/9/05	2-AEP-2	240.3 V	1.58 A	0.380 kW
Average energy consumption / heater				0.390 kW

Note: See equation 3, in Appendix B, for energy calculations.

Table 7 - AEP Cooler Pumping Energy Consumption

AEP Cooler Pumping Power Consumed ACW pump efficiency = 0.89 from pump efficiency curves					
Date:	Plant Item:	Inlet Pressure kPa: Digital Pressure Gauge –100 to 2000 kPa	Outlet Pressure kPa: Digital Pressure Gauge –100 to 2000 kPa	Flow litres/sec: Ultrasonic Transit Time Flowmeter	Power: Calculated
5/9/05	1-AEP-2	365 kPa	165 kPa	21.0 l/s	4.719 kW
5/9/05	2-AEP-2	382 kPa	164 kPa	21.0 l/s	5.144 kW
Average energy consumption / cooler					*4.931 kW*

Note: See equation 4, in Appendix B, for energy calculations.

Table 8 - Duty AEP Drive Motor Energy Consumption

Duty AEP 3 Phase Drive Motor Power Consumption Power Factor = 0.853 from motor nameplate				
Date:	Plant Item:	Volts: Installed Metering	Amps: Installed Metering	Power: Calculated
6/9/05	1-AEP-2	3.3 kV	34 A	165.769 kW
6/9/05	2-AEP-2	3.3 kV	34 A	165.769 kW
Average energy consumption / Drive Motor				*165.769 kW*

Note: See equation 5, in Appendix B, for energy calculations.

Table 9 - Current System Total Energy Consumption

Energy Consuming Device	Duty AEP	Standby AEP	Sub-total
Drive Motor	165.769 kW	0 kW	165.769 kW
Anti-condensation heater	0 kW	0.390 kW	0.390 kW
Cooler pumping power	4.931 kW	4.931 kW	9.862 kW
		Total Energy	**176.021 kW**

4.3. LYB Current System Efficiency

In terms of efficiency, the current system is pumping about 90 l/min of standard air on Unit 1. Using a re-arrangement of equation 6, in Appendix B, this equates to 6.615 kg/h of dry air, with the system using 176.021 kW worth of energy. This gives an efficiency of 26.609 kW/kg/h of dry air pumped. Based on the Woodward, Howard & Andrews (1991, p.383) comparative measure of 0.727 kW/kg/h, the existing system is very inefficient at base load operation.

Chapter 5. NEW SYSTEM EVALUATION

5.1. Alternative Designs of Air Extraction

The four alternative designs that were identified as proven in the electrical power industry were:

1. The multi-stage steam ejector system
2. The 2-stage LRVP system
3. The air ejector and 2-stage LRVP system
4. The steam hybrid system

The relative high efficiency of the 2-stage LRVP system initially makes this system appear to be the most attractive. However, like the high efficiency mechanical compressors and blowers that were ruled out because of maintenance and operational problems, the 2-stage LRVP system also has its problems.

Discussions with Mr. Alf Hertaeg of Sinclair Mertz and Knights, who had previously worked for the State Electricity Commission of Victoria, indicated that pump cavitation had been a constant issue at the Yallourn 'W', Newport and Loy Yang 'A' Power Stations. This issue was only resolved by moving to the air ejector and LRVP system at the Loy Yang 'B' Power Station. While this system is effective it is also very inefficient and contrary to the aims of this project.

Based on this information the steam hybrid system was the next obvious choice because it was the next most efficient system and eliminates pump cavitation issues. The steam source, which is the most efficient to use, from the steam cycle perspective, is the de-aerator vent steam because it flows continuously to the condenser rejecting heat to the circulating water as ‘waste’ energy. Unfortunately at LYB, the new OT boiler water chemical treatment demands the de-aerator vent valve be shut during normal operation prohibiting its use as a steam supply.

While it is possible to source an alternative steam supply from somewhere else in the system, such as the LP heater bled steam supply, with a relatively low steam cycle energy this will impact on the overall air extraction efficiency, as this steam energy will be lost from the turbine work cycle.

The multistage steam ejector system has been ruled out due to its higher energy consumption and running costs. Steam jet ejectors must also be sized for a specified flow to obtain maximum efficiency. This means if they are over-sized, then their efficiency will be considerably worse than the earlier figures indicate. Also, if the air extraction load rises along with the condenser pressure, then their capacity may be rapidly exceeded resulting in condenser under-venting and significant unit heat rate losses.

One technological improvement, which has a direct bearing on the decision of the choice of air extraction system, is the advent of cheap and reliable variable frequency speed control units (VFD). Most of the cavitation problems associated with LRVP’s systems stem from the fact that they are positive displacement by nature. When a LRVP 'over-vents' the condenser, steam will be drawn in as well as the non-condensable gases. This steam will condense in the LRVP, as the pressure rises across the pump, rejecting its heat to the seal water. This additional latent heat adds to the temperature rise across the pump and can cause cavitation if the vapour temperature is reached (Skelton, 1999, p.9).

According to Skelton (1998, p.1-2) a VFD to control the speed and flow capacity of a LRVP is a relatively new innovation which can eliminate the need for recycle lines or unreliable vacuum relief valves, while at the same time preventing over-venting and cavitation. As a LRVP is usually sized for over-capacity the VFD option also has the potential to result in additional energy savings.

5.2. New System Arrangement

As the key objective of this project is to reduce in-house energy consumption, the discission was made to select a 2-stage LRVP system, with a VSD unit controlling its flow capacity. There is, however, a limit to the amount of speed reduction possible with a LRVP, because if it is slowed excessively its liquid seal will collapse due to the force of gravity exceeding the centrifugal force imparted by the impellor leading to a total loss of pumping capacity.

As an 'insurance policy' to overcome this problem, a small adjustable recycle valve will be fitted to the system. This valve will be set during commissioning to ensure that the LRVP stays within its operating limits. This valve may introduce some inefficiency but it may not even need to be opened, and if so, only slightly cracked. The proposed system arrangement is shown in Figure 19.

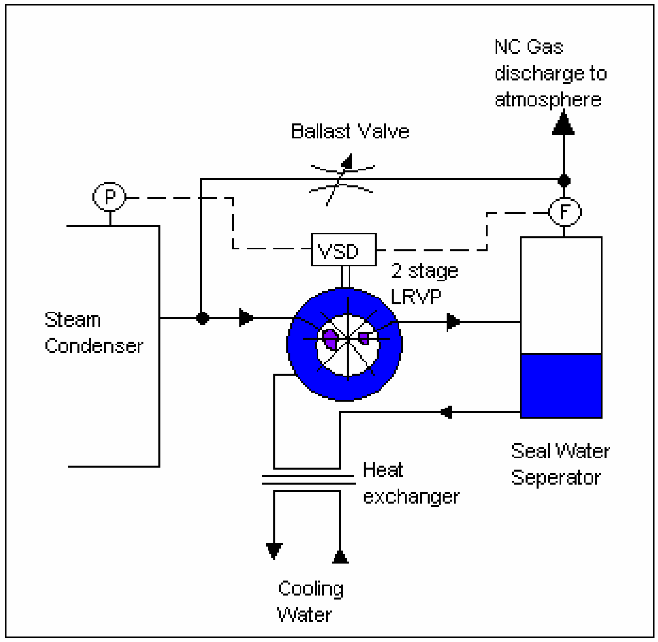

Figure 19 - Proposed New System Arrangement

Another important consideration is the temperature of the seal cooler, cooling water. Skelton (1999, p.9) when discussing condenser over-venting advocates that, “The LRVP should always operate in conjunction with the condenser. To do this, the cooling water to the service liquid heat exchanger on the LRVP system and the condenser water must be the same temperature”. It is uncertain whether this statement is applicable, when a VFD is used to control the LRVP capacity.

The circulating water used to cool the main steam condenser at the Loy Yang B Power Station, ranges in temperature between 18 ˚C in winter to 36 ˚C in summer. Although these temperatures seem to be quite warm for cooling water, when the circulating water temperature rises, so does the condenser pressure and hence the LRVP suction pressure.

Although this may be suitable for cooling purposes under most circumstances, it has been decided to utilize cooling water from the Fire and General Service Water Supply, which has an estimated temperature range of 15 ˚C to 20 ˚C, to increase the LRVP's cavitation resistance. The cooler's water discharge from this supply can be discharged to the Auxiliary Cooling Water Return Conduit to reduce water consumption because it is at a substantially higher pressure. The seal water temperature rise across the LRVP will need to be monitored for excessive temperature rise during commissioning, to ensure cavitation will not occur.

In terms of the new systems control, the VFD speed set point will be derived from the condenser pressure, via a function generator, so as to track the design air extraction rate. The ballast valve will be set to provide the minimum load required to prevent cavitation at the LRVP's minimum speed. System air loading changes are likely to be increases of air in-leakage rather than decreases, and a discharge airflow meter will be used to bias the VFD speed up, in proportion to the increase in measured airflow based on equation 8, in Appendix B. An alarm should be triggered if the discharge air flow meter detects an increase in the discharge airflow of more than 20%.

5.3. Design Air Extraction Rate

The Heat Exchange Institute (HEI 1995, p.30) specifies that the condenser venting equipment shall be designed for a suction pressure of 3.39 kPa absolute or the condenser design pressure, whichever is lower. The power station's data collector indicated that the lowest condenser pressure recorded during winter, at the normal minimum load of 300 MW, was about 4.5-kPa. The condenser pressure readings over the previous year are shown in Figure 20.

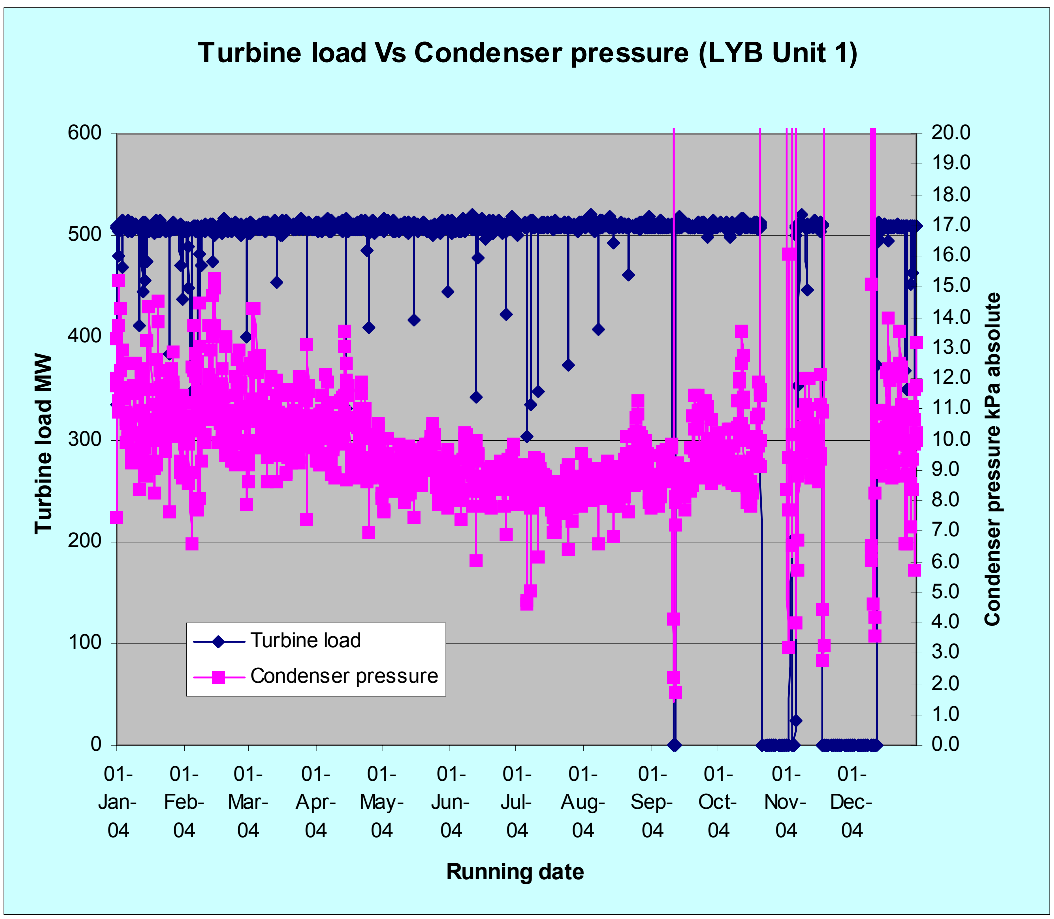

Figure 20 - Condenser Pressure Versus Load

This condenser pressure is not much above the HEI recommended suction pressure of 3.39 kPa absolute, and the HEI values for suction pressure and temperature will be used for design purposes.

In terms of deciding on the design over-venting capacity of the new system the target condensate dissolved oxygen level need to be considered. When based on the AVT chemical target of < 10 ppb dissolved oxygen, a ratio of 0.15 would need to be selected. If however, the OT chemical target of 30-150 ppb were used, then a ratio of 0.50 could be selected because the target dissolved oxygen level of 42 ppb would still allow an adequate control range for the oxygen injection system.

In effect while the current system capacity was designed on an over-venting requirement of six times the actual air in-leakage rate, the new system capacity can be designed on over-venting requirement of just two times the actual air in-leakage rate.

Usually the design air extraction rate would be determined by the HEI standard of 707 litres/min or the VGB guideline of 263 litres/min. However it is assumed that these values allow for significant reserve capacity in case of major air leak developing on the condenser. This is assumption is supported by Siemen's description of the VGB guideline as being 'conservative'. In the case of LYB, the LP Turbine casing is welded to the condenser shell and does not have the usual large and troublesome expansion joint, as do many other power stations.

The new air extraction system will also have the large current system as a backup, in case a major air leak develops.

Because of these factors, and to optimise the capacity of the air extraction system, the design capacity will be based on the HEI maximum recommended air in-leakage rate of 142 litres/min at the condenser design operating pressure of 9.5-kPa absolute. This capacity selection will also reduce the potential for LRVP cavitation developing, as the capacity will be closer to actual air in-leakage rates.

To determine what other NC gases are required to be extracted from the condenser, gas samples were taken from both generating unit air extraction discharge lines and tested in the station's gas chronograph. The test results indicated 99.8% air and 0.2% hydrogen. The hydrogen is believed to be a result of boiler tube oxidation processes. No ammonia was detected in the samples and it is believed that any ammonia vapour is condensed in the LRVP as the pressure rises across the pump. Based on this information, the design extraction rate will be for a discharge flow of 142 litres/min of dry air.

This dry air discharge flow needs to be converted to a saturated mass airflow at the LRVP suction and a Dry Air Equivalent (DAE) found. The design air extraction rate at a condenser pressure of 3.39-kPa can be found by using equation 2, in Appendix B. This gives:

$$\therefore Q = 142 \times \frac{\sqrt{101300 - 3390}}{\sqrt{101300 - 9500}}$$

$$\therefore Q = 146.65\ litres/\min$$

This figure then needs to be converted to an air mass flow rate. Re-arranging equation 6, in Appendix B, we get:

$$\therefore \dot{m} = Q \times \rho \times \frac{60}{1000}$$

$$\therefore \dot{m} = 146.65 \times 1.204 \times \frac{60}{1000}$$

$$\therefore \dot{m} = 10.59\ kg / h$$

Because the mass flow rate is dry air, the value of 2.05 kg of water vapour/kg of dry air may be used, from the HEI's Appendix E. At the air extraction suction temperature of 21.7 ˚C, the specific volume of steam is v_g = 52.4 m^3/kg (Rodgers & Mayhew, 1996, p.2) and it takes 2.05 kg of steam to saturate each kg of dry air. The volume of steam required to saturate the 10.59 kg/h of dry air at 21.7 ˚C is then:

$$\therefore V_{Steam} = 10.59 \times 52.4 \times 2.05 = 1137.578\ m^3 / h$$

The density of the air at the air extraction suction pressure of 3.39 kPa and 21.7 ˚C can be calculated using equation 12, in Appendix B, as:

$$\therefore \rho = \frac{3390}{287 \times 294.85} = 0.040\ kg / m^3$$

The specific volume of the air is then:

$$\therefore v_g = \frac{1}{\rho} = \frac{1}{0.040} = 25\ m^3 / kg$$

The volume of air at the suction is then:

$$\therefore V_{Air} = 25 \times 10.59 = 264.750\ m^3 / h$$

After adding the steam and air volumes at the suction these can then be converted to a the standard dry air equivalent at 20 ˚C and 101.3 kPa by using equation 1, in Appendix B, as follows:

$$\therefore DAE = (1137.578 + 264.750) \times \frac{293.15}{294.85} \times \frac{3390}{101300}$$
$$\therefore DAE = 46.658\ m^3 / h$$
$$\therefore DAE = 778\ litres / \min$$

To achieve a HEI recommended target condensate dissolved oxygen level of 42 ppb an actual load divided by the design capacity ratio of 0.50 must be used.

$$\therefore Q_{Design} = \frac{778}{0.50}$$
$$\therefore Q_{Design} = 1556\ litres / \min$$

It should be noted that the new system is being selected to operate at normal base load conditions and is not expected to establish the initial condenser vacuum during unit start-up. This is because any delay in establishing initial vacuum will delay the return to service of the unit. The current air extraction system has the capacity to reach operating vacuum on start-up in about 25 minutes and will be retained for this purpose.

The steam turbine at Loy Yang B Power Station has the capability of bypassing 70% of its full steam flow around the turbine and dumping this to the condenser. Even though this steam is de-superheated by spraying it down, this steam dumping causes a rapid rise in condenser pressure. The current air extraction system will go into hogging mode under these circumstances and assist in maintaining the condenser pressure below the turbine trip value. The new system is not capable of performing this function and the existing system will also be retained for this purpose. New control logic will be needed to start the current system in this situation.

Excessive air in-leakage can occur as a result of deterioration of flanged joints, expansion joints, corrosion of pipe-work, loss of seal water to valve stems or from various valves being incorrectly left open to atmosphere. The LYB condenser has the distinct advantage of being welded to the turbine casing, rather than having the large and troublesome expansion joint that a lot of other power stations have, which can be a major source of air in-leakage.

Air leaks are a maintenance and operational issue, however the air extraction system must be able to cope with them in the short term to prevent a loss of generation output. With good maintenance and operational practices it is expected that events of excessive air in-leakage should rarely occur. The current system will be retained, as a backup, for when these events occur and logic will need to be provided to start it on rising condenser pressure.

5.4. New System Energy Consumption

Based on a DAE design suction flow rate of 1556 litres/min, at a suction pressure of 3.39-kPa absolute, a "Stirling" 2-stage LRVP Model LPH 3708 was selected. The performance curve of this pump versus the design suction flow for various condenser pressures are shown in Figure 21. The design suction flows were estimated for various condenser pressures by using equation 2, in Appendix B.

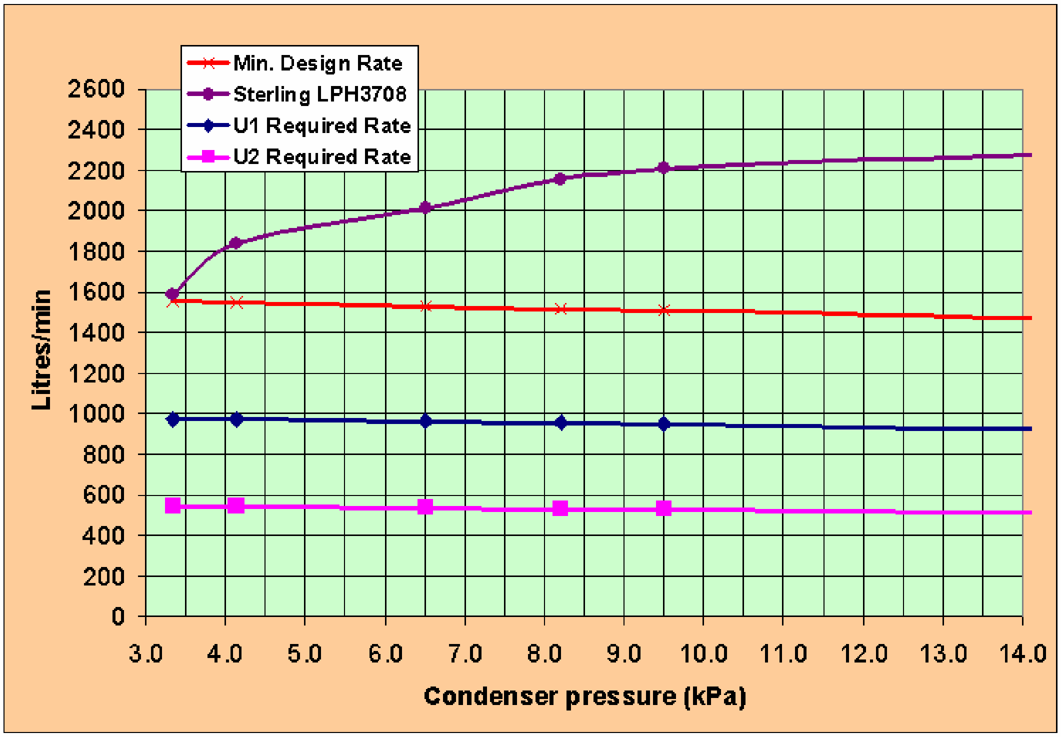

Figure 21 - Stirling LPH 3708 Performance Curve

The LPH 3708 pump is rated at 7.5 HP (5.596 kW) at a full speed of 1750 rpm. At the condenser's design operating pressure of 9.5-kPa absolute the pump will extract about 2200 litres/min, although the design demand is only about 1500 litres/min. Using equation 8, in Appendix B, the required running speed is 1190 rpm. The resulting running power can be determined by equation 9, in Appendix B, to be 6.18 HP (4.61 kW).

In actual practice the VFD will be set to eliminate over-venting. The actual air in-leakage rates at LYB are 63% of the HEI maximum recommended on Unit 1 and 35% on Unit 2. Based on the Assumption that the LPH 3708 pump will run adequately at speeds of 900 rpm (not unreasonable considering it is also available with a motor speed of 1150 rpm) the actual running power is likely to be about 4.012 kW.

To size the cooling requirements for the new system it is assumed that the heat input will be equivalent to the motor power, if the LRVP is pumping in the no load condition. The maximum motor power rating is 4.012 kW, so the design heat input into the cooler is also 4.012 kW. Using equation 11, in Appendix B, the required cooling water flow rate is 0.72 litres/s assuming a 3˚C temperature rise across the cooler as suggested by Kubik (2005, p.3).

Assuming that a plate type cooler is used, with a pressure drop of about 200 kPa across it and a F&GS pump efficiency 0.89, then equation 4, in Appendix B, gives an estimated cooler energy consumption of 162 W.

5.5. New System Efficiency

The energy consumption of the new system with a VFD is likely to be 4.174 kW. Based on pumping the previously mentioned Unit 1 air mass flow rate of 6.615 kg/h of dry air, which gives a pumping efficiency of 0.631 kW/kg/h.

There are two possible options for the system modifications that need evaluating, these being:

- **Option #1** – Install a new LRVP fitted with a VSD drive. This will give a power consumption of 4.012 kW for the drive motor and about 0.162 kW for its cooler pumping energy for a total power consumption of 4.174 kW.

- **Option #2** – Install a new LRVP with a VSD drive and fit electric isolating valves to the current air extraction system coolers, so the cooling water supply to these standby air extraction systems is only available on system start-up. This will give a power consumption of 4.174 kW for the new LRVP system and save 9.862 kW of pumping energy on the current system, when in the standby condition (most of the time).

As the current system will act as a backup for the new system, there will be also a slight increase in power consumption, 0.39 kW, due to an extra motor anti-condensation heater being in service. The decision between these two options will be made on economic considerations and the comparative greenhouse gas reductions that are achievable.

5.6. Capital Cost Recovery

One accepted method to evaluate the capital recovery of a project is to calculate the expected monetary gain or loss by discounting all future cash inflows and outflows to the present point in time, using the required rate of return (Horngren, Datar & Foster, 2003, p.720).

Before Net Present Value can be determined, a number of decisions have to be made on which to base the modelling on. For example: Should the project life be based on the original 30 year design life of the power station or its new 75 year budgeted life? Should the current NEM spot price of electricity ($25/MWh), the previous three-year average price ($30/MWh) or the long run marginal return on investment ($40/MWh) be used in calculations? These decisions require judgement about realistic outcomes so that distortions of the financial model are avoided. On this basis, the following capital recovery modelling assumptions were made:

1. Electricity cost based on a spot price of $30/Mwh, in today's values.
2. Life of project to be 18 years (remaining design life of station).
3. Constant inflation rate of 3% during period.
4. Company tax rate of 30%.
5. Required rate of return of 12%.
6. Depreciation is the straight-line method.
7. Salvage value is $0 at the end of project life.
8. Power prices escalate with inflation.
9. 80% of current maintenance cost of $50,000 p.a. will be saved.

The two options were modelled in Appendix C, with the results shown in Table 10. There was very little difference between the two options economically over the life of the project because although option 2 required the additional capital outlay to fit electric actuator valves, this cost was countered by the expected power savings.

Table 10 - Power Reductions & NPV

Option	Capital Cost	Power Reduction kW	NPV After Tax
1	$68,000	322.410 kW	$617,019
2	$96,800	342.134 kW	$616,709

In terms of risk, it can be seen from the figures in Appendix C that both options will have a capital recovery time of less than one year, based on expected cost savings and capital spending.

5.7. Potential Greenhouse Gas Emission Reduction

At present the station burns about 1.25 tonnes of coal to generate each MWh of electricity into the power grid with each tonne of Brown Coal producing 0.99 tonne of equivalent CO_2. This equates to 1.2375 tonne of greenhouse gas equivalent CO_2 per MWh of electricity sent out to customers.

In terms of reductions claimable under the generator efficiency standards LYB is only able to claim the Victorian average greenhouse gas emission of 0.95 tonne per MWh sent out. This figure is lower because the Victorian generation portfolio includes gas, hydro and wind power. This project will consider the actual reduction of greenhouse gas emitted at the station, in preference to the generator efficiency standard, because the profile of the Victorian grid is constantly changing.

The greenhouse reductions for the two proposed options are calculated in Appendix C and shown in Table 11. The second option shows a greenhouse gas reduction of an additional 214 tonnes p.a. and is the preferred option.

Table 11 - Greenhouse Gas Reductions

Option	Equivalent CO_2 Reduction p.a.
1	3497 tonne
2	3711 tonne

Chapter 6. SUB-COOLING INVESTIGATION

6.1. Condenser Performance Evaluation

To determine whether sub-cooling of condensate was occurring within the condenser, data was extracted from the power station's data collector over the coolest months of the year, being June, July and August 2005 at a one-hour sample rate. These cooler months were chosen because the circulating water temperature is at its lowest, resulting in lower condenser pressures. This can be seen in the data plot shown in Figure 23.

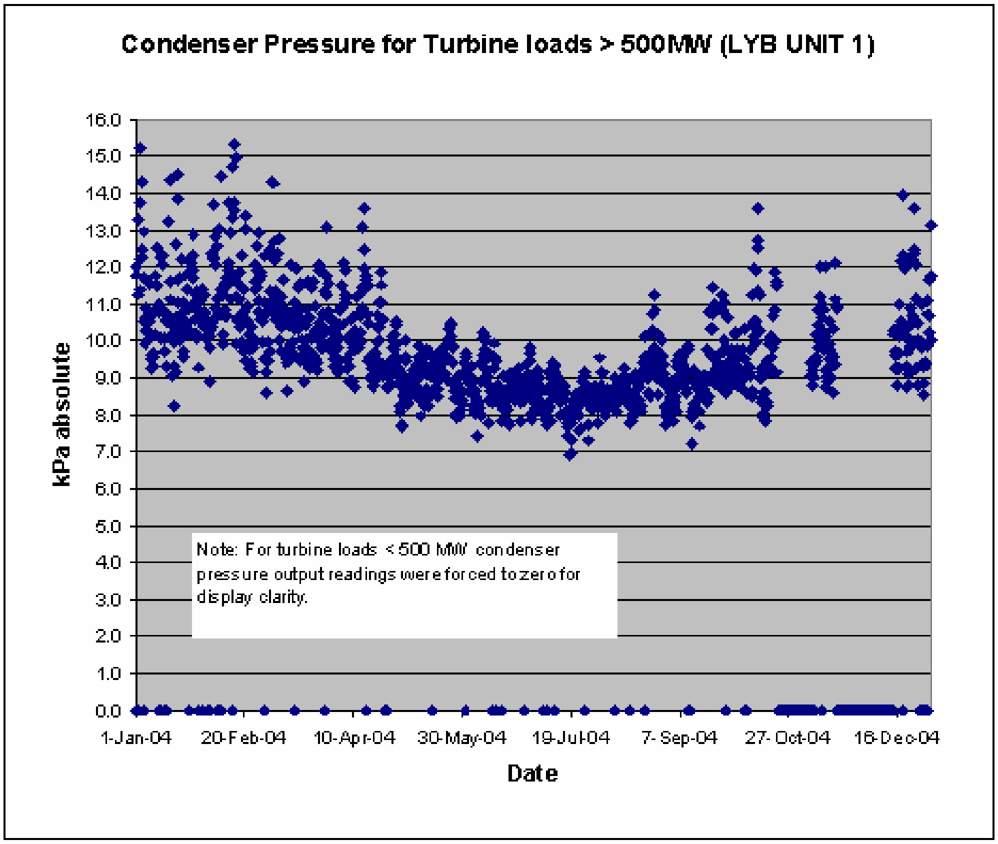

Figure 22 - Condenser Pressure During Year

At these lower condenser pressures, there is both a potential for air extraction under-venting due to drop off in air extraction performance (Kubik & Spencer, n.d., p.2) and turbine exhaust annulus choking due to higher specific steam volumes (Putman & Harpster, 2000, p.3). The summarized results for Unit 1 and 2 are shown in Tables 12 and 13.

Table 12 - Unit 1 Condenser Performance June, July & August 2005

Data Point:	Minimum	Average	Maximum
CW inlet temp ˚C	19.6	24.7	29.4
CW outlet temp ˚C	34.4	39.6	44.7
Condenser TTD ˚C	3.5	4.70	6.9
Turbine exhaust hood temp ˚C	40.4	46.1	49.6
Condensate outlet temp ˚C	38.3	44.3	48.7
Condenser vacuum kPa abs	6.31	8.77	11.31
Turbine load MW	360.5	506.6	530.2
Ambient temperature ˚C	1.2	11.3	24.4
Condenser saturation temp ˚C	37.5	43.4	48.1
Condensate sub-cooling ˚C	-3.7	-0.9	0.2

Table 13- Unit 2 Condenser Performance June, July & August 2005

Data Point:	Minimum	Average	Maximum
CW inlet temp ˚C	18.9	23.8	28.8
CW outlet temp ˚C	33.8	38.9	44.1
Condenser TTD ˚C	3.53	4.33	6.39
Turbine exhaust hood temp ˚C	46.5	51.5	54.4
Condensate outlet temp ˚C	38.3	43.3	48.1
Condenser vacuum kPa abs	6.51	8.47	10.47
Turbine load MW	369.5	508.4	533.5
Ambient temperature ˚C	1.2	11.3	24.4
Condenser saturation temp ˚C	38.1	42.8	46.7
Condensate sub-cooling ˚C	-1.7	-0.5	0.9

6.2. Condensate Sub-Cooling Assessment

From the summarized data it can be seen that the average Unit 1 condensate outlet temp was 0.9 ˚C warmer than the condenser saturation temperature and

Unit 2 it was 0.5 ˚C warmer. A warmer condensate outlet temperature can occur from condensate heating, due to turbine steam drainage returning to the condenser. Closer analysis of the data showed that 99.97% of values recorded on Unit 1 and 92.7% of values on Unit 2 showed no sub-cooling at all. Despite some very slight sub-cooling indicated on Unit 2, 99.88% of all recorded values showed that sub-cooling was less than 0.5 ˚C. This data shows that there was no significant condensate sub-cooling occurring within the condensers at LYB and no further action is warranted in this regard.

It was noted however that the condenser Terminal Temperature Difference (TTD) was 4.70 ˚C on Unit 1 and 4.33 ˚C on Unit 2 against the condenser design specification of 3.2 ˚C with a cleanliness factor of 85% (LYB Plant Specifications, n.d., sec 3.2). This probably indicates that some condenser tube biological fouling is present and this warrants cleaning to restore the condenser heat transfer back to design. The simple condenser 'bake' method would be adequate to assist removal of any biological build-up present.

Chapter 7. CONCLUSIONS

This project investigated four different designs of air extraction, which were proven for use in the electrical power industry. Of these the multi-stage steam ejector was rejected both on energy efficiency and economic grounds. The air ejector and LRVP system, as used at LYB, had shown it was versatile and wasn't prone to cavitation problems. Despite its proven operating performance, this system too was rejected due to its poor energy efficiency. The steam hybrid system was perhaps one of the preferred choices, however the absence of the de-aerator vent steam supply meant that it too was rejected on energy efficiency grounds.

The chosen system was the 2-stage LRVP system because it offered superior efficiency over the alternative designs. Previous inherent cavitation problems of this design could be resolved by using a VFD to control the pump speed and hence its flow capacity. In terms of the air extraction suction design capacity it was found a change from the AVT chemical treatment to OT chemical treatment for boiler water had reduced the required over-venting capacity to one third of the original design. The new system offers the advantage of an efficiency of 0.631 kW/kg/h of dry airflow based on a discharge mass airflow of 6.615 kg/h compared to the current systems efficiency of 26.609 kW/kg/h.

There were two possible options with the new system. The recommended option is to install the LRVP system fitted with a VFD for speed control and to retrofit electric isolating valves to the current system, so as to eliminate the waste cooler pumping energy, while AEP is sitting in the standby mode. This option would require an estimated capital cost of $96,800. However based on a required return of

12%, it would achieve a NPV of $616,700 after tax over its 18-year life. The estimated reduction of equivalent CO_2 greenhouse gases is estimated to be around 3710 tonnes p.a. There may be additional revenue associated with this greenhouse gas reduction, however this has not been considered in the financial modelling.

Investigations into the condenser performance on both units, during the winter months, showed that no significant sub-cooling was occurring in the condensate. It was, however, noted that the TTD measured about 1˚C above the condenser design specification on both units probably due to condenser tube biological fouling. A condenser clean, using the 'bake' method is recommended to return the condenser heat transfer back to original design specifications.

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APPENDIX A

University Of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/4112 RESEARCH PROJECT
PROJECT SPECIFICATION

FOR: Dr. Nigel Hancock

TOPIC: Investigation on ways of improving the efficiency of the condenser air extraction system at the Loy Yang 'B' Power Station located in the Latrobe Valley, Victoria, Australia.

SUPERVISOR: Dr. Ruth Mossad

SPONSERSHIP: International Power / Mitsui LLP - Loy Yang 'B'

PROJECT AIM: This project aims to analyse the existing condenser air extraction efficiency and to investigate possible improvements that could be made to the existing system, in order to reduce in-house electrical usage and greenhouse gas emissions.

PROGRAMME: **Issue B, 13th July 2005**

1. Build the necessary background knowledge for air extraction systems used in power stations.

2. Conduct a literature search for work published on the topic of improving condenser air extraction performance.

3. Describe the existing layout of the condenser air extraction system used at Loy Yang 'B' Power Station.

4. Analyse the current system's efficiency at full and reduced steam turbine load.

5. Investigate whether sub-cooling of condensate is occurring within the condenser at Loy Yang 'B' Power Station and investigate possible design improvements to overcome this (if required) to maximize steam cycle efficiency.

6. Determine the optimum air extraction capacity that is required at Loy Yang 'B' Power Station, for both full and reduced steam turbine load, that will minimize in-house power consumption without negatively impacting on the steam cycle efficiency.

7. Evaluate alternative designs and system improvements on the basis of cost, efficiency and reliability.

8. Conduct an economic evaluation on the recovery of capital cost for the chosen design, based on potential energy savings and efficiency improvements.

9. Estimate the potential greenhouse gas emission reduction as a result of adopting the chosen design.

APPENDIX A

University Of Southern Queensland
Faculty of Engineering and Surveying

As time permits:

10. Determine modifications required to the existing steel structure, piping and control logic to incorporate the chosen design.

11. Determine the required capacity of ancillary service supplies, such as electrical power, cooling water, steam, compressed air, etc.

AGREED: ____________________ (student) _____________________ (supervisor)

(Dated) ____ / ____ / ____

APPENDIX B
Mathematical Formulae

Conversion to Standard Airflow

From Hope (1975, p.102) the Ideal Gas Rule for a given mass is:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Where:
P = Pressure, Pa
V = Volume, m^3
T = Temperature, ˚K

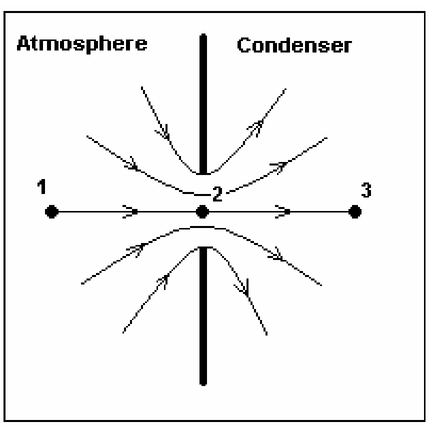
This may be adapted to convert airflow rates to standard conditions of 20˚C and 101.3 kPa by re-arranging as follows:

Equation 1

$$V_{Std} = V_{Measured} \times \frac{293.15}{T} \times \frac{P}{101300}$$

Where:
$V_{Measured}$ = Flow to be converted, l/min
T = Ambient temperature, ˚K
P = Ambient pressure, Pa
V_{Std} = Standard flow, l/min

Estimation of Air In-leakage Rates



Air along the streamline shown above is stationary at point 1 and accelerates as it enters the opening in the condenser at point 2. At this point its density remains unchanged due to the surrounding air.

After the air enters the condenser at point 3, it expands and its density drops. Sometime after point 3 the air temperature increases because of heat exchange with the condenser steam.

If we apply Bernoulli's equation (Hope, 1975, p.67) along the streamline from point 1 to point 2 we get:

$$P_1 + \tfrac{1}{2}\rho v_1^2 + \rho g z_1 = P_2 + \tfrac{1}{2}\rho v_2^2 + \rho g z_2$$

Where:
P = Pressure, Pa
ρ = density, kg/m^3
v = velocity, m/s
g = acceleration due to gravity, m/s^2
z = elevation, m

v_1 = 0 m/s as the atmospheric air is still.
$z_1 = z_2$ as the points are at the same elevation.

$$\therefore P_1 = P_2 + \tfrac{1}{2}\rho v_2^2$$

As ρ is assumed constant then:

$$\therefore v_2^2 \infty (P_1 - P_2)$$

$$\therefore v_2 \infty \sqrt{P_1 - P_2}$$

The air flow rate, $Q = vA$ (Hope, 1975, p.66)

Where:
Q = Flow rate, m^3/s
A = Area, m^2
v = velocity, m/s

As the area of the leak, A is assumed to be constant, then:

$$\therefore Q \infty v$$

And therefore also:

$$\therefore Q \infty \sqrt{P_1 - P_2}$$

APPENDIX B
Mathematical Formulae

Thus if we have a known air in-leakage rate, at a known condenser pressure, we can calculate the new leakage rate at the new pressure by the following relationship:

$$\therefore \frac{Q_{new}}{Q_{known}} = \frac{\sqrt{P_{atmosphere} - P_{new}}}{\sqrt{P_{atmosphere} - P_{known}}}$$

Equation 2

$$\therefore Q_{new} = Q_{knowm} \times \frac{\sqrt{P_{atmosphere} - P_{new}}}{\sqrt{P_{atmosphere} - P_{known}}}$$

Where:
Q_{known} = Measured flow rate, l/min
Q_{new} = Flow rate to be estimated, l/min
P_a = Std atmospheric pressure, Pa
= 101300 Pa (abs)
P_{new} = Desired Condenser pressure to estimate new flow at, Pa (abs)
P_{known} = Condenser pressure at measured flow rate, Pa (abs)

Single Phase Electrical Power

From Hope (1975, p.206) for a.c. electrical power:

Equation 3

$$P = VI \cos\phi$$

Where:
P = Power, W
V = Voltage, V
I = Current, A
cosΦ = Power factor

Cooler Pumping Power

From Hope (1975, p.25):

$$P = Fv$$

Where:
P = Power, W
F = Force, N
v = velocity, m/s

$$\therefore P = p \times A \times v$$

Where:
P = power, W
p = pressure, Pa
A = Area, m^2
v = velocity, m/s

$Q = Av$ (Hope, 1975, p.66)

Where:
Q = Flow rate, m^3/s
A = Cross sectional area of pipe, m^2
v = velocity, m/s

$$\therefore P = pQ$$

Where:
P = Power, W
p = pressure, Pa
Q = Flow rate, m^3/s

The power loss across the cooler, corrected for the pump efficiency, then can then be found by:

Equation 4

$$P = \frac{(p_i - p_o)Q}{\eta}$$

Where:
P = Cooler pump power loss, W
p_i *= Cooling water inlet pressure, Pa*
p_i *= Cooling water outlet pressure, Pa*
Q = Cooling water flow rate, m^3/s
η *= Cooling water pump efficiency*

APPENDIX B
Mathematical Formulae

Three Phase Electrical Power

This well known equation in the power industry is simply equation 3 corrected for the effects of three phases 120˚ apart:

Equation 5

$$P = \sqrt{3} \times VI \cos\phi$$

Where:
P = Power, W
V = Voltage, V
I = Current, A
cosΦ = Power factor

Converting Air Mass Flow to litres/min

This is simply:

Equation 6

$$Q = \frac{\dot{m}}{\rho} \times \frac{1000}{60}$$

Where:

$\dot{m}$ = mass flow of dry air, kg/h
Q = airflow rate, l/min
ρ = density of std. air at 20˚C = 1.204 kg/m^3

Converting Air Flow scfm to litres/min

This is simply:

Equation 7

$$1\ scfm = 28.32\ litres/\text{min}$$

Where:
scfm = Standard cubic feet per minute of air at 20 ˚C and 101.3 kPa.

Calculating air extraction saturated suction, mass flow rates

Equation 8

$$W = \frac{18}{MW_{NC}} \times \frac{P_W}{P_t - P_W}$$

Where:
W = lb. of water vapour per lb. of non-condensable gas
MW_{NC} = molecular weight of non-condensable gas
P_W = absolute "water vapour" pressure corresponding to temperature at condenser vent outlet, in. HgA
Pt = absolute "total" pressure at condenser vent outlet, in. HgA

Note: When the non-condensable is dry air, W may be obtained directly from the HEI's "Standards for steam surface condensers"-Appendix E

For T = 21.7 °C (71.06 °F) &
Pt = 3.39 kPa abs. (1.0 inch HgA)
W = 2.05 kg water vapour/kg of dry air

LRVP Speed Reduction Calculations

Skelton (1998, p.1) provides the following approximations for calculations resulting from speed reductions by a VFD:

Equation 9

$$\frac{Capacity_1}{Capacity_2} = \frac{rpm_1}{rpm_2}$$

Where:
$Capacity_1$ = Design flow rate, l/min
$Capacity_2$ = Pump capacity, l/min
rpm_1 = required running speed, rpm
rpm_2 = maximum running speed, rpm

Equation 10

$$\frac{rpm_1}{rpm_2} = \left(\frac{hp_1}{hp_2}\right)^2$$

Where:
Hp_1 = Horse Power at reduced speed
Hp_2 = Horse Power at maximum speed

APPENDIX B
Mathematical Formulae

Cooling water flow estimate

From Hope (1975, p.86):

Equation 11

$$Heat\ Input = \dot{m}\, c_p \Delta T$$

Where:

$\dot{m}$ = *cooling water mass flow rate, kg/s*
c_p = *specific heat at constant pressure*
= *1.850 kJ/kg. K for water*

ΔT = *cooler temperature rise, ˚C*

Calculation of air density

From Hope (1975, p.103):

Equation 12

$$\rho = \frac{P}{R \times T}$$

Where:
ρ = *Density, kg/m*^{3}
P = *Pressure, Pa*
R = *Gas Constant for Air, J/kg.K*
= *287J/kg.K*
T = *Temperature, ˚K*

APPENDIX C
NET PRESENT VALUE & CO_2 CALCULATIONS FOR OPTION #1

Input Data:

Reduction of in-house load per Unit (kW's)	161.205
Life of project (Years)	18
Predicted inflation rate (%)	3.00%
Company tax rate (%)	30.00%
Spot power price ($/MWh)	30
Power price escalation rate (%)	3.00%
Real rate of return (%)	12.00%
Nominal rate of return inc. inflation (%)	15.36%

Capital Investment per Unit:	**$**
Cost of extraction unit	20000
Additional costs (eg freight, etc)	2000
Other equipment, valves, actuators, VSD	2000
Installation costs	9000
Commissioning costs	1000
Total capital cost	34000
Salavage Value	0
Useful Life (Years)	18
Depreciation p.a. (Straightline method)	1889

Maintenance costs per Unit:	**$**	
Overhaul costs new system p.a.	5000	
Escalation rate (%)	3.00%	
Current system maintenance costs p.a.	50000	
Current system maintenance reduction	80.00%	
Station wide Net present value	**617,019**	**After tax**

Station wide CO2 Reductions:		
MWh reduction p.a.	2826	MWh's
Greenhouse gas tonnes per MWh	1.2375	Tonne/MWh
CO2 Reductions p.a. claimed	**3,497**	**Tonnes p.a.**

Calculation of net present value based on changes in cash flow across station

Year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Initial investment	-68000										
Cash inflows:											
Electricity revenue		84787	87331	89951	92649	95429	98292	101241	104278	107406	110628
Maintenance savings		82400	84872	87418	90041	92742	95524	98390	101342	104382	107513
***less* Cash outflows:**											
Maintenance costs		10300	10609	10927	11255	11593	11941	12299	12668	13048	13439
Taxable cash flows		156887	161594	166442	171435	176578	181875	187332	192952	198740	204702
Income tax		-47066	-48478	-49933	-51431	-52973	-54563	-56200	-57886	-59622	-61411
Tax saving on depreciation		1133	1133	1133	1133	1133	1133	1133	1133	1133	1133
Total net cash flow	-68000	110955	114249	117643	121138	124738	128446	132266	136200	140252	144425

Year	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**
Initial investment								
Cash inflows:								
Electricity revenue	113947	117366	120887	124513	128249	132096	136059	140141
Maintenance savings	110739	114061	117483	121007	124637	128377	132228	136195
***less* Cash outflows:**								
Maintenance costs	13842	14258	14685	15126	15580	16047	16528	17024
Taxable cash flows	210844	217169	223684	230394	237306	244425	251758	259311
Income tax	-63253	-65151	-67105	-69118	-71192	-73328	-75527	-77793
Tax saving on depreciation	1133	1133	1133	1133	1133	1133	1133	1133
Total net cash flow	148724	153152	157712	162409	167248	172231	177364	182651

APPENDIX C
NET PRESENT VALUE & CO_2 CALCULATIONS FOR OPTION #2

Input Data:

Reduction of in-house load per Unit (kW's)	171.067
Life of project (Years)	18
Predicted inflation rate (%)	3.00%
Company tax rate (%)	30.00%
Spot power price ($/MWh)	30
Power price escalation rate (%)	3.00%
Real rate of return (%)	12.00%
Nominal rate of return inc. inflation (%)	15.36%

Capital Investment per Unit:	**$**
Cost of extraction unit	20000
Additional costs (eg freight, etc)	2000
Other equipment, valves, actuators, VSD	16400
Installation costs	9000
Commissioning costs	1000
Total capital cost	48400
Salavage Value	0
Useful Life (Years)	18
Depreciation p.a. (Straightline method)	2689

Maintenance costs per Unit:	**$**	
Overhaul costs new system p.a.	5000	
Escalation rate (%)	3.00%	
Current system maintenance costs p.a.	50000	
Current system maintenance reduction	80.00%	
Station wide Net present value	**616,709**	**After tax**

Station wide CO2 Reductions:		
MWh reduction p.a.	2999	MWh's
Greenhouse gas tonnes per MWh	1.2375	Tonne/MWh
CO2 Reductions p.a. claimed	**3,711**	**Tonnes p.a.**

Calculation of net present value based on changes in cash flow across station

Year	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Initial investment	-96800										
Cash inflows:											
Electricity revenue		89974	92674	95454	98317	101267	104305	107434	110657	113977	117396
Maintenance savings		82400	84872	87418	90041	92742	95524	98390	101342	104382	107513
***less* Cash outflows:**											
Maintenance costs		10300	10609	10927	11255	11593	11941	12299	12668	13048	13439
Taxable cash flows		162074	166937	171945	177103	182416	187889	193525	199331	205311	211470
Income tax		-48622	-50081	-51583	-53131	-54725	-56367	-58058	-59799	-61593	-63441
Tax saving on depreciation		1613	1613	1613	1613	1613	1613	1613	1613	1613	1613
Total net cash flow	-96800	115065	118469	121975	125585	129305	133135	137081	141145	145331	149643

Year	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**
Initial investment								
Cash inflows:								
Electricity revenue	120918	124546	128282	132130	136094	140177	144382	148714
Maintenance savings	110739	114061	117483	121007	124637	128377	132228	136195
***less* Cash outflows:**								
Maintenance costs	13842	14258	14685	15126	15580	16047	16528	17024
Taxable cash flows	217814	224349	231079	238012	245152	252507	260082	267884
Income tax	-65344	-67305	-69324	-71404	-73546	-75752	-78025	-80365
Tax saving on depreciation	1613	1613	1613	1613	1613	1613	1613	1613
Total net cash flow	154083	158658	163369	168222	173220	178368	183671	189132

APPENDIX D
MANUFACTURER'S LRVP DATA

High Vacuum Performance (Saturated Air)

Model Number (Inlet x Outlet Size) measured in inches	Vacuum in Inches Hg.		4	12	20	22.5	25	27.5	28	28.7	28.9	Average Service Liquid Flow USGPM
	Absolute Pressure in Inches Hg.		25.92	17.92	9.92	7.42	4.92	2.42	1.92	1.22	0.98	
	Absolute Pressure in mm Hg.		658	455	252	188	125	61.5	48.8	31	25	
	Speed	Horsepower	CFM	CFM	CFM	CFM	CFM	CFM	CFM	CFM	CFM	
LPH 25003 (1 ¼ x 1¼)	3500	2	13.7	14.9	15.6	15.8	15.9	14.0	14.2	11.1	-	1.4
LPH 25007 (1¼ x 1¼)	3500	3	27.5	30.8	32.4	32.9	34.0	32.3	29.7	19.0	-	1.4
LPH 3704 (1½ x 1½)	1150	3	25	26	26	27	27	25	23	-	-	4
	1750	5	37	37	38	44	46	40	37	32	32	4
LPH 3708 (1½ x 1½)	1150	5	43	43	46	47	48	52	52	-	-	5
	1750	7.5	71	72	73	80	81	76	71	65	56	5
LPH 45312 (1½ x 1½)	1150	5	55	55	60	65	72	83	83	76	78	3
	1750	7.5	90	92	107	117	123	124	123	111	114	3
LPH 45317 (1 ½ x 1½)	1150	5	61	69	77	85	95	100	101	92	89	4
	1750	10	107	108	157	165	164	166	162	142	136	4
LPH 55312 (2 x 2)	1150	7.5	108	109	123	131	136	145	142	130	132	10
	1750	*15	143	145	185	208	218	227	224	205	218	10
LPH 55316 (2 x 2)	1150	10	114	115	141	154	162	158	153	137	141	11
	1750	20	204	205	238	259	272	280	271	235	227	11
LPH 55320 (2 x 2)	1150	15	119	120	154	167	175	174	169	167	153	13
	1750	25	210	211	258	321	345	353	343	289	275	13
LPH 65320 (2½ x 2½)	1150	15	197	199	209	223	241	268	278	276	298	9
	1450	25	270	274	298	303	318	353	349	332	354	9
	1750	30	330	333	345	361	378	390	394	371	391	9