University of Southern Queensland

Faculty of Engineering and Surveying

Induced Draft (ID) Fan Lubrication System Design Review and Proposed Modification Upgrade at Callide C Power Station

A dissertation submitted by

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in the fulfilment of the requirements of

Courses ENG4111/4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

28 October, 2010

#Abstract

Australia's coal fired power stations are the most efficient form of providing bulk base load power generation (i.e. electricity) to consumers. This is due to Australia having an abundance of thermal coal reserves, which is the fuel used in coal fired power stations. Therefore it is extremely important that these power stations operate at maximum availability and reliability to ensure the consumer receives cost effective and uninterrupted electricity.

Callide C Power Station in Biloela Queensland is a 900 Megawatt (MW) coal fired power station that was commissioned in 2001. Unfortunately Callide C Power Station has been plagued with continuous operational and reliability problems caused from the induced draft (ID) fans since initial commissioning. The ID fan problems have arisen from the bearing lubrication system which provides oil recirculation to the induction motor bearings and fan main shaft bearings. Consequently these issues have caused half-load unit (225 MW) run-backs and full unit (450 MW) trips over the past decade.

This project's aim is to analyse the ID fan lubrication system and then identify and define all root causes and their associated failure modes. Once all root causes are identified through Root Cause Analysis (RCA) process, effective design solutions can be researched and evaluated so a proposed modification design project can be finalised. This final design proposal will be used to justify capital expenditure so implementation can occur in the near future.

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ENG4111 & ENG4112 *Research Project*

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Date

Acknowledgements

This project was conducted under the principle supervision of Mr Bob Fulcher, Faculty of Engineering and Surveying, University of Southern Queensland.

I would also like to acknowledge and thank the technical engineering staff and Callide C maintenance staff at Callide Power Station for their continuous support, mentoring and assistance.

Stuart D. Baker

Glossary

AC:	Alternating Current
AS:	Australian Standards
ASME:	American Society of Mechanical Engineers
CNC:	Computer Numerical Control
CO_2:	Carbon Dioxide
cSt:	Centistokes
DC:	Direct Current
DE:	Drive end
DN:	Nominal Diameter
DNB:	Departure from nucleate boiling
EP:	Extreme Pressure
FD:	Forced Draft
HHI:	Hyundai Heavy Industries
HV:	High Voltage
ICMS:	Integrated Control Management System
ID:	Induced Draft
IHI:	Ishikawajima – Harima Heavy Industries
kg/s:	Kilogram per second
kPa:	Kilopascal = 1000 Pascals
kV:	Kilovolt = 1000 volts
kW:	Kilowatt = 1000 watts
L/min:	Litre per minute

m:	metre
M:	Metric
ml:	Millilitre
mm:	Millimetre
mm^2/s:	Millimetre squared per second
m/s:	Metre per second
m^3/s:	Metre cubed per second
MPa:	Megapascal = 1000000 Pascals
MRC:	Maximum Continuous Rating
MW:	Megawatt = 1000000 watts
MWh:	Megawatt hour = 1000000 watts for one hour
NDE:	Non-drive end
OEM:	Original Equipment Manufacturer
PA:	Primary Air
P&ID:	Piping and Instrumentation Diagram
PF:	Pulveriser Fuel
RCA:	Root Cause Analysis
rpm:	Revolution per minute
V:	Volt
VI:	Viscosity Index
µm:	Micrometre (micron) = 1×10^{-6} m

Symbology

A	Cross-sectional area	[m^2]
D	Pipe inside diameter	[mm]
D_1	Orifice inside diameter (DE)	[mm]
D_2	Orifice inside diameter (NDE)	[mm]
f	Friction factor	
F	Voltage frequency	[Hz]
g	Gravitational acceleration	[m/s^2]
h	Potential head	[m]
h_1	Major head loss	[m]
h_{1m}	Minor head loss	[m]
H	Total head loss	[m]
K	Loss coefficient factor	
L	Length of pipe	[m]
n	Revolutions per minute	[rpm]
p	Pressure	[kPa]
P	Number of motor winding poles	
Q	Volumetric flow rate	[L/min]
Re	Reynolds number	
$Srpm$	Synchronous revolutions per minute	[Srpm]
v	Kinematic viscosity	[m^3/s]

V	Displacement per revolution	[cm^3/rev]
$\dot{V}$	Fluid flow rate	[m^3/s]
$\overline{V}$	Fluid velocity	[m/s]
η	Volumetric efficiency	[%]

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Chapter 1 – Introduction

1.1 Project Objectives

The aim of this project is to conduct an engineering appraisal on the operational and reliability issues in the existing Callide C Power Station induced draft (ID) fan lubrication system, and recommend a cost effective and reliable design for a future modification project. Project objectives are:

1. To describe the general operation of Callide C Power Station with special attention to boiler section pertaining to induced draft (ID) fan operations.

2. Research theory on lubrication and bearing systems in rotating equipment.

3. Investigate further information on the different types of bearings and lubrication used in heavy electric drives and axial fans.

4. Identify the operational and reliability failure modes in the existing ID fan lubrication system which is causing protection alarms and trips.

5. Research and evaluate effective design solutions to prevent reoccurring operational and reliability issues in the existing ID fan lubrication system.

6. Prepare a modification design recommendation that will make certain the lubrication system is designed for correct functionality and long-term reliability.

7. Submit an academic dissertation on the engineering research conducted and proposed design modification to the ID fan lubrication system.

1.2 Background

The rapid technological advances in todays electrical hardware and software has lead to our civilization becoming increasing dependant on electrical power, also commonly known as electricity. This increasing dependence on electricity is forcing power generation companies all throughout Australia to produce reliable, cost effective and safe electricity for their consumers. Electricity in Australia is commonly produced in large scale using a four types of electrical power generation methods, these include:

- Coal fired power stations
- Gas fired power stations
- Hydro driven power stations
- Large wind turbine farms

Of the four types listed above, coal fired power stations are the most favoured in Australia as they are the most efficient method of bulk base load power generation. Australia is also very fortunate to have an abundance of thermal coal reserves, which is the fuel used in coal fired power stations. In Australia, base load coal fired power generation units range in size from 30 megawatt (MW) units which were built in 1960's all the way to 750 MW units built in recent years.

Bulk base load power generation is paramount in sustaining a stable electrical network (i.e. grid) system as it ensures continuous uninterrupted supply of electricity to the consumer. This stability is built from base load power stations which are designed to run at constant operating output, all day and every day with the exception of planned outages.

A major disadvantage when building electrical network systems on large base load units ranging from 350 MW to 750 MW is when a power generation unit unexpectedly trips offline to the grid, it leaves a big gap in the transmission supply which can lead to voltage and frequency instability. Such events can cause forced network load shedding, where electricity load is tripped from the grid.

An example of this, take Queensland's 2009 peak electricity consumption of 8699 MW (Department of Mines and Energy 2010) and Kogan Creek Power Station in Chinchilla Queensland, a single 750 MW base load unit. When 750 MW is tripped offline from the state grid of 8699 MW, approximately 9% of available supply electricity is lost and this gap must be instantaneously supplied from other power stations connected to the grid. If such an event occurs simultaneous with multiple power stations tripping offline, the entire grid can become unstable which can lead to regional and state blackouts. To avoid grid instability and power blackouts it is crucial that power stations are designed, maintained and operated to run at maximum availability and reliability.

However, all power generation stations from time to time suffer from reliability issues mainly caused from inadequate design factors, lack of maintenance procedures and skills, and operator error. To combat such factors that cause reliability issues in large base load power stations. It is generally the responsibility of the technical engineering staff to adopt a continuous improvement culture which investigates and analyses key performance and reliability issues. Such investigations, better known as Root Cause Analysis (RCA) are very popular in today's industry as this process identifies root causes in machinery failure and poor reliability. Once the root causes are identified and understood, the technical engineering department is able to proceed with modification concepts and designs to prevent reoccurrence of the problem causing poor reliability.

1.3 Reliability Issues with Callide C Power Station

Callide C Power Station is situated 18 kilometres east from the township of Biloela, approximately 120 kilometres west of Gladstone Queensland. Callide C Power Station is quite famous as it was the first supercritical coal-fired power station built and commissioned in Australia, in the year 2001 (Power Technology 2010). Callide C has a total generation capacity of 900 MW which consist of 2 x IHI supercritical power generation units capable of 450 MW per unit. This amount of continuous generation is the basis for Callide C being an integral base load power generation station in Queensland's state electricity grid.

Each of the two 450 MW units consists of two Howden Variax Dual Stage Axial Flow Induced Draft (ID) Fans. Each fan is powered via an alternating current (AC) 6600 volt - 3 phase 6 pole synchronous induction motor manufactured by Hyundai Heavy Industries (HHI). The four ID fans have suffered from reliability and operational issues from commissioning stage in 2001, which has caused half load unit run-backs and full load unit trips. A half load 225 MW unit run-back occurs when one ID fan trips and a full unit 450 MW trip occurs when both ID fans trip. This amount of generation capacity tripping off the grid unexpectedly has the potential to cause frequency and voltage instability of the state grid at peak times.

Figure 1.1: Callide C ID fan and induction motor

The ID fan problems have arisen from the bearing lubrication system which supplies oil recirculation to the induction motor bearings at 15 – 16 L/min and fan main shaft bearings at 15 - 18 L/min. The lubrication system has 2 x 100% duty cycle lubrication pumps, so one pump is always in standby. Prior to 2004, there were two flow transmitters on each lubrication line (refer to Figure 1.2) that were used to provide alarm and trip values for equipment protection. These values were used by the Integrated Control Management System (ICMS) for the following parameters:

- Low oil flow alarm
- Oil flow for fan start condition
- Low oil flow fan trip
- Low oil flow start stand-by lubrication pump

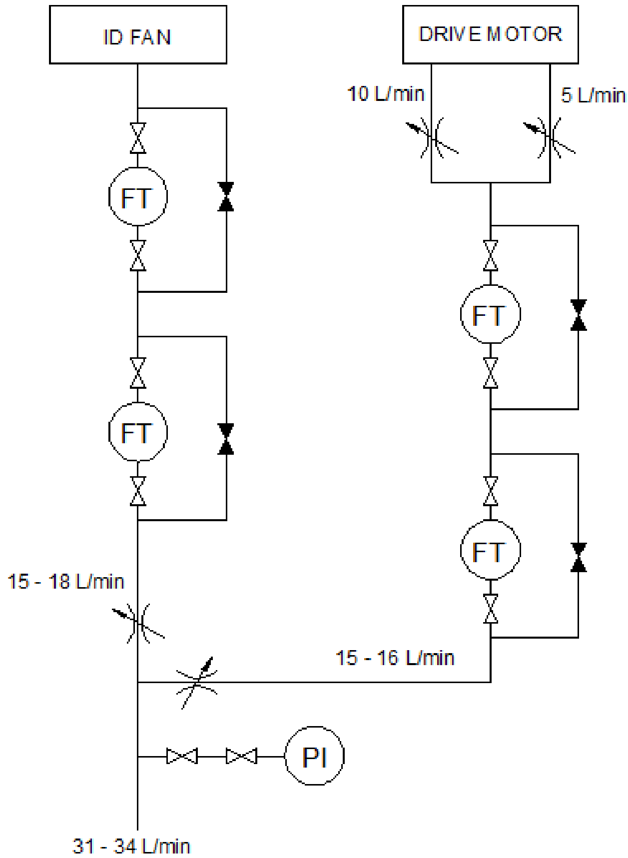


Figure 1.2: Lubrication system P&ID

1.3.1 ID Fan Lubrication System Reliability Issues

To balance the required oil flow rates between the induction motor bearings and fan shaft bearings, two throttle valves and two adjustable orifices are used to obtain required flow rates. Due to equipment and personnel safety factors, the oil flow balancing set-up can only be performed when the lubrication system is off-line. This is when the oil is at ambient temperature 25℃, not operating temperature between 50 – 60℃. Shell Tellus S46 is the oil used in the ID fan lubrication system with a kinematic viscosity of 100 mm^2/s at 25℃ and 20 mm^2/s at 60℃. This shows the oil viscosity-to-temperature relationship is logarithm and oil viscosity is a factor of 5 times less from ambient temperature to operating temperature.

The problems became apparent after the oil flow balancing set-up was completed and the ID fan returned to service and eventually oil temperature reached 60℃ and even higher on hot summer days. The oil viscosity and density would decrease causing the flow transmitters to read a lower value then previous for oil flow rate. This would then create a low oil flow alarm in the ICMS and then start the stand-by pump in an attempt to increase oil flow rate to above alarm value. However, due to poor design factors the stand-by pump was unable to self-prime and make system pressure. Therefore no additional oil flow was created and eventually the ID fan would trip on low oil flow.

To combat this issue, the operation and maintenance teams would eventually get the stand-by pump to prime and return the ID fan back to service with the both lubrication pumps running simultaneously. Having two pumps running ensured oil flow rate was much greater than alarm and trip values, therefore preventing future alarms and trips. However this solution created another problem, if a lubrication pump dropped performance or completely failed, the oil flow rate would again drop to alarm and trip value, eventually tripping the ID fan.

The flow transmitters also suffered from other failure modes such as: (1) oil leaking internally into the electrical circuitry causing loss of analog output signal to the ICMS, (2) could not be site calibrated to suit different fluid viscosity and density, and (3) contamination of small particles in the oil caused sticking of the internal components. To rectify this problem, in 2004 an engineering modification was implemented which replaced the four flow transmitters with four pressure transmitters in an attempt to reduce the number of ID fan alarms and trips. The alarm and trip pressure protection values were supplied from the fan supplier, Howden Australia.

This modification unfortunately suffered from the same fate as the flow transmitters. As the oil viscosity and density decreased when oil reached operating temperature, the flow resistance also decreased and therefore reducing line pressure to both the induction motor bearings and fan main shaft bearings. This drop in line pressure would again create low oil pressure alarms and trips, eventually tripping the ID fan. So unfortunately this engineering modification did not prove successful and did not eliminate the root causes.

Some work was recently done to decrease the pressure alarm and trip values which did prevent oil pressure alarms and trips. However this solution still did not address the issue of the stand-by pump not making system pressure when started. This aspect is very important as operations are required to change-over lubrication pump from duty cycle to stand-by cycle every month to prove pump performance and integrity. So every month when this operational activity occurs, the stand-by pump can not make system pressure which leads to a critical machine operating with no stand-by lubrication pump.

1.3.2 Pressure Monitoring versus Flow Monitoring

Pressure is not the desired online monitoring parameter for the ID fan lubrication system as the 'Original Equipment Manufacturer' (OEM) plant manual gives oil flow rates for operation conditions, not oil pressures. The reason for oil flow rate as the preferred monitoring parameter is because the induction motor bearings and fan main shaft bearings are oil bath configuration. Therefore oil flow to the bearing inlet is only slightly above atmospheric pressure and the only 'head' resistance created is from dynamic friction of pipes, fittings and orifices. This means pressure will dramatically vary with oil temperature however flow rate will always remain constant.

It must also be further emphasised when pressure is increased, the corresponding flow does not necessary increase. If the cross-sectional area of an oil line is decreased upstream of the pressure transmitter by reducing orifice size, then line pressure will increase due to an increase in ‘head’ resistance. However, oil flow rate will not increase, instead oil flow rate can actually decrease causing possible bearing overheating and failure. The actual decrease in oil flow rate is dependant on system design, pump type and pump performance.

Chapter 2 – Literature Review

2.1 General operation of Callide C Power Station

Callide C Power Station utilises 2 x 450 MW supercritical coal-fired boilers, which achieves higher thermal efficiency than conventional subcritical drum-type coal-fired technology. Callide C obtains coal from the adjacent Anglo Coal Callide Mine and receives water from adjacent Callide Dam which draws water via an overland pipeline from Awoonga Dam near Gladstone. General information for Callide C Power Station is shown below:

<u>GENERAL</u>	
Commissioned	2001
Capacity	900 MW
Units	2 X 450 MW
Transmission	275 kV
Fuel	Black Thermal Coal
<u>TURBINE</u>	
Type	Steam
Manufacturer	Toshiba
<u>BOILER</u>	
Manufacturer	IHI (Tokyo, Japan)
Height	42 m
Operating Temperature	1400℃
Steam Pressure	25 100 kPa (~250 bar)
Steam Temperature	566℃
<u>CHIMNEY</u>	
Height	230 m
Flue Gas Temperature	135℃

2.2 Characteristics of Supercritical Boilers

Supercritical boiler technology was introduced to the power industry in the early 1960's. Since this time, there have been many innovative boiler design configurations and features introduced to reduce capital and operating costs, simplify operation and maintenance, and increase reliability.

2.2.1 Supercritical versus Subcritical

Supercritical 'once-through' boilers are constructed differently from the conventional subcritical 'drum' boilers. Primarily in two areas which is the boiler furnace and boiler drum (refer to Figure 2.1). For comparison, in a drum type boiler which operates at subcritical pressures and temperatures, large diameter furnace tubes are used to minimise flow resistance so that sufficient amount of stream and water can flow through the tubes by natural circulation. For that reason, drum type boilers by design permit high circulation rates so the water passing through the tubes at any stage never completely evaporates to steam, so it remains as saturated steam (steam-water mixture) with latent heat addition. This ensures a liquid film (wet wall flow) is maintained on the tube wall inner diameter so that the departure from nucleate boiling (DNB) and/or dryout (dry wall flow) does not occur in all conditions of operations. Nucleate boiling is vital as it provides a high heat transfer coefficient so all furnace tubes remain at essentially the saturation temperature for the operating temperatures of the boiler, preventing tube overheating and puncture.

Whereas a supercritical boiler does not use a head drum to promote a natural circulation circuit required for latent heat addition. Steam flow is forced by the boiler feedwater pump as it does not rely on the density difference between steam and water to provide proper circulation and cooling of the furnace tubes. As a result, it can be operated at supercritical [>220 bar] pressures.

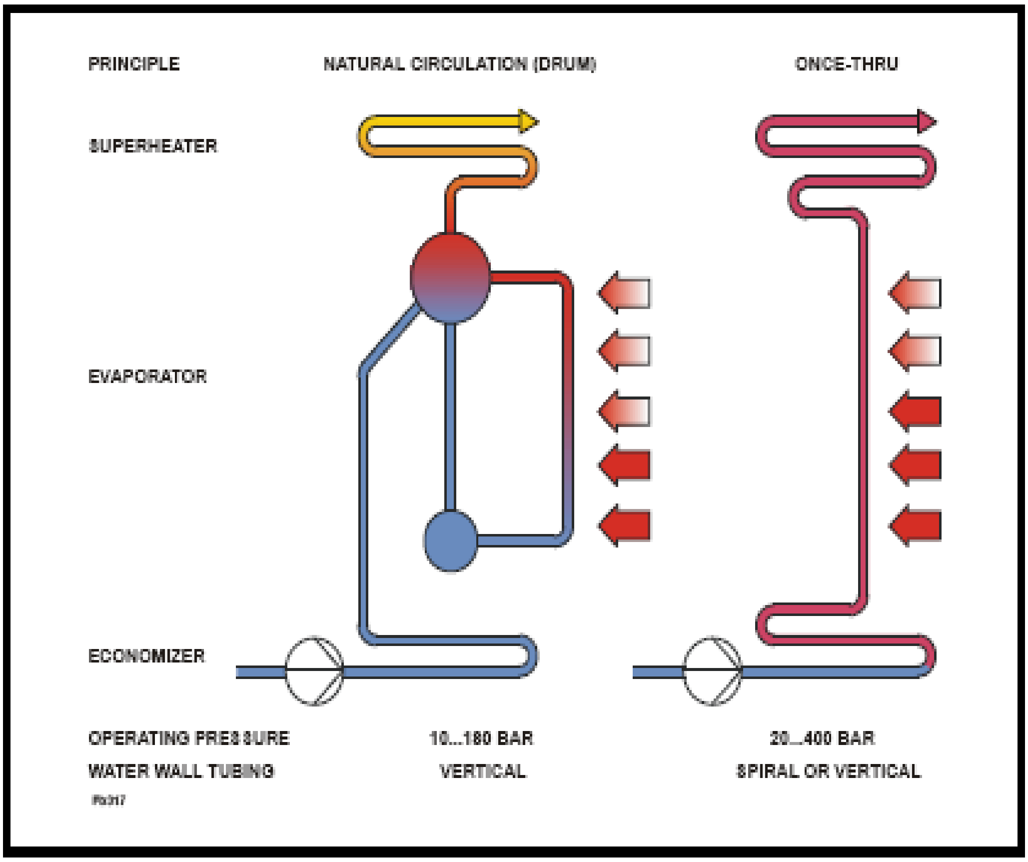


Figure 2.1: Boiler circulations methods

In the absence of natural circulation and water progressively converting to steam without boiling, all the water turns into superheated steam in the furnace tubes of a supercritical boiler. However, small separating vessels in stages along the furnace tubes are required to separate the water fraction and recirculate the water when the unit operates at different loads and pressures. As illustrated in Figure 2.2, the furnace tubes in a supercritical boiler can also be smaller in diameter as natural circulation is not needed. This reduces liquid inventory in the furnace tubes which increases boiler load dynamic response.

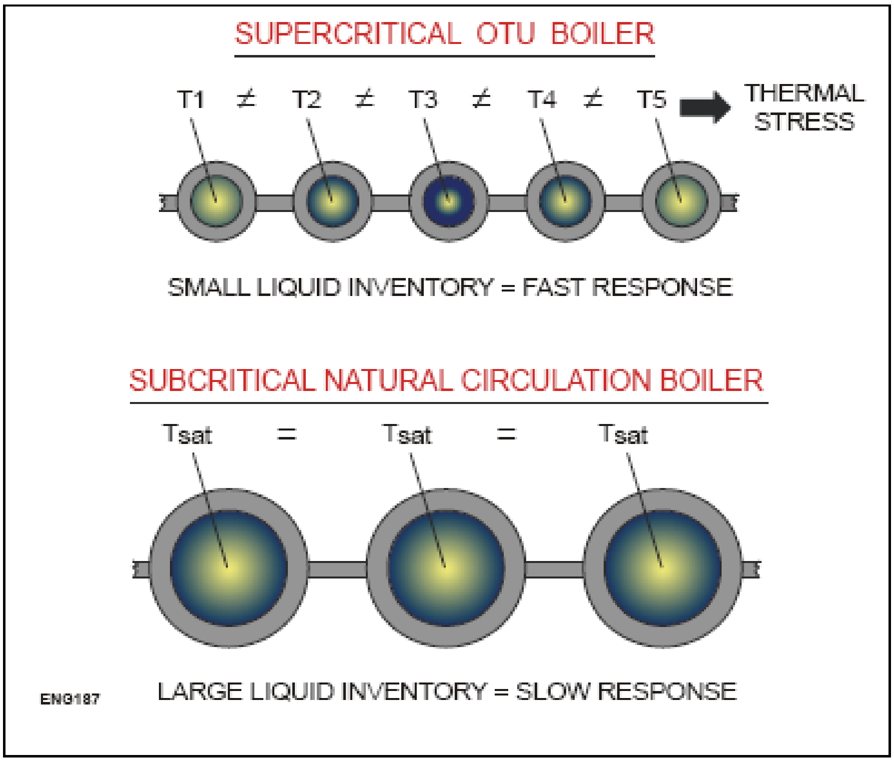


Figure 2.2: Furnace tube size and temperature comparison

In a supercritical boiler, operating at supercritical pressures, there is no distinction between liquid and vapour phases and there is a continual increase in fluid temperature (refer to Figure 2.3). This can lead to unbalances in heat absorption between furnace tubes and if temperature unbalance is not limited, high thermal stresses will result which can lead to tube failures. For this reason, it is extremely important furnace tube circuitry design must meet the following requirements:

- Provide a means to accommodate heat absorption variations from tube to tube so that the temperature difference between adjacent tubes is limited.
- Provide adequate tube cooling to prevent the departure of nucleate boiling (DNB) and suppress dryout so peak tube metal temperatures are minimised.

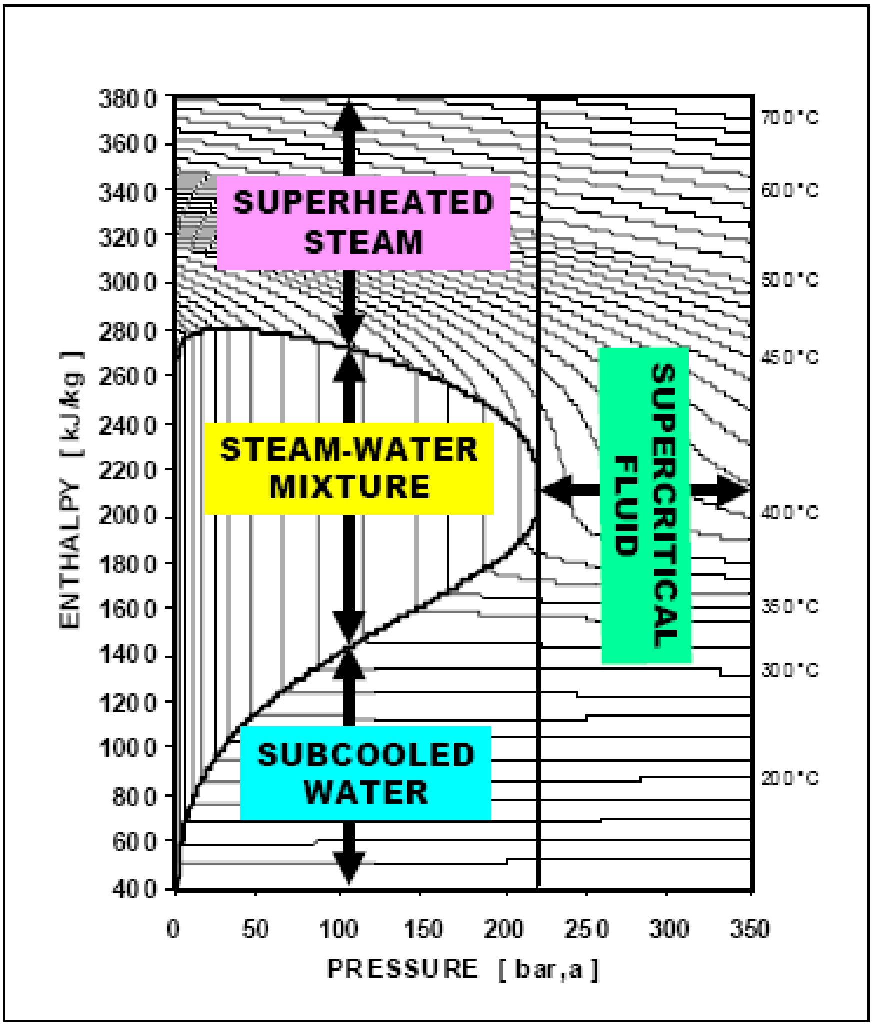


Figure 2.3: Subcritical versus supercritical steam

In the American Society of Mechanical Engineers (ASME) steam tables, supercritical condition is at 220.7 bar and 374.1°C when water progressively converts to steam without boiling and with no latent heat addition. The overall cycle efficiency of a supercritical boiler increases as the pressure and temperature are increased, which results in economic and environmental benefits that result from firing less coal for same power output. Today's most advanced supercritical boilers are operating up to 315 bar and 620°C, while those up to 350 bar and 700°C are in planning stage (Rayaprolu 2009, p.399). The benefits of supercritical boilers over subcritical boilers are:

- Fuel savings due to higher cycle efficiency
- Reduction of CO_2 emissions due to lower fuel input for the same energy
- Superior load dynamics as the furnace tubes have smaller liquid inventory and there is no water in circulation
- Part load performance improves with variable/sliding pressure operation

2.3 Callide C Boiler Draft System

The boiler draft operation at Callide C Power Station is commonly known as "balanced draft pulveriser fuel (PF) boiler" operation. This is when a mixture of hot and cold combustion air is forced into the coal pulveriser fuel mills (PF mills) via primary air (PA) fans. Once the PF mills have ground the coal into PF, a mixture of hot air and PF is directed to the boiler burners where additional hot combustion air is forced into the burner via forced draft (FD) fans. After combustion inside the boiler at approximately 1400℃, the air and PF turns into gas and must be evacuated by means of induced draft (ID) fans where the gas is sent to the stack (or chimney) and released to atmosphere . This balance draft operation is when there is pushing of air and pulling of gas and the furnace is maintained at near or below atmospheric condition. Hence, describing the term "balancing of drafts", as depicted in Figure 2.4 and Figure 2.5.

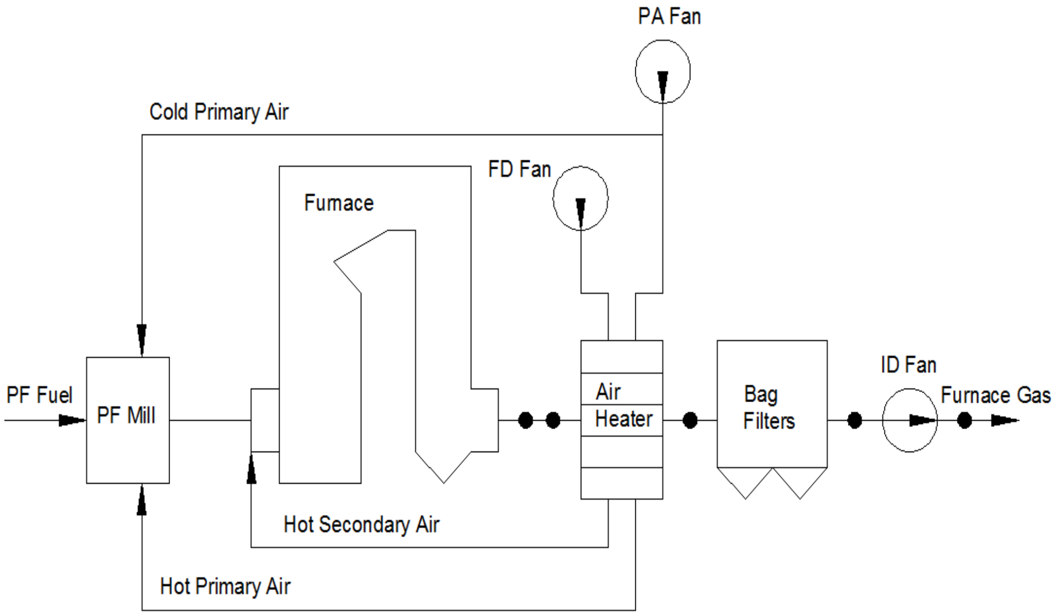


Figure 2.4: Callide C – Balanced draft PF boiler

Figure 2.5: Typical draft-loss pressure diagram for a PF Boiler

2.4 Boiler Draft Efficiency and Design

Fans in a boiler plant are perhaps the most important of all auxiliary equipment because they affect boiler performance, auxiliary power consumption and boiler dynamics. Design factors such as fan margin and fan type have a major role in boiler design and efficiency. It is important thorough engineering must be sought when selecting boiler draft fans.

2.4.1 Fan Margins

Fan margin calculations are the most critical design aspect when selecting correct fan size. Margins on volume, head, operating and ambient temperatures are included into fan design conditions for calculating boiler maximum continuous rating (MCR). Boiler draft fan margins are essential for the following reasons:

- Safety margins for pressure and draft losses through the boiler draft system are vital in ensuring MCR is always available.
- Ambient temperature variation of -5℃ to +50℃ in the tropics and -20℃ to +35℃ in temperate climates, translates to a variation of approximately 20% in the specific volume of air.
- Unpredictable fouling of surfaces increases pressure head, particularly with fuels such as coal.
- Differences in boiler construction geometry compared with design calculations due to inaccuracies in manufacturing and erection.
- Boiler systems that need over-firing to catch up with rapid load ramps needs more power from the fans.
- Fan manufacturers have negative tolerances on head and volume specifications that must be compensated for if all parameters are too met in practice.

Fan margin figures are continuously being refined by design engineers to avoid unnecessarily large margins that can lead to high capital costs and excessive auxiliary power consumption. The general industry consensus for draft fan margins in a coal fired boiler should be around (Rayaprolu 2009, p.313):

- 20% for volume
- 44% for variable pressure head
- 20% for operating temperature

2.4.2 Fan Types

Boiler draft fans are predominately divided into two fan types, centrifugal fans and axial fans. Both fans are different in terms of capacity and pressure, degree of control, resistance to wear and corrosion. Centrifugal fans move air and gas perpendicular to the fan shaft (refer to Figure 2.6). Whereas axial fans move air and gas along the fan shaft axis (refer to Figure 2.7).

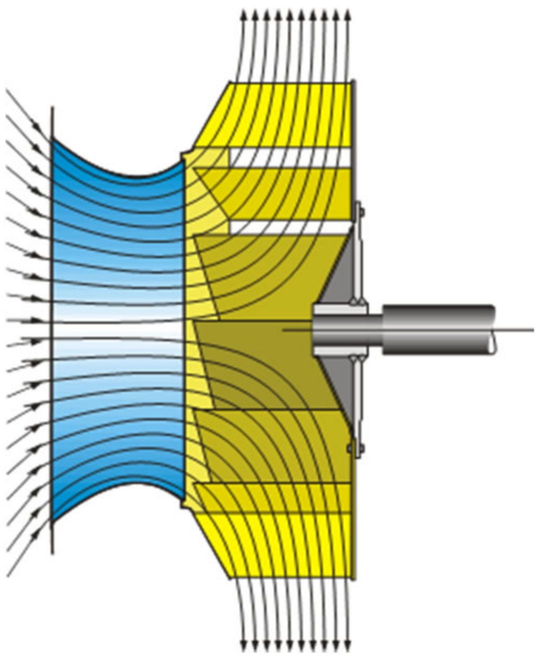


Figure 2.6: Centrifugal fan flow path

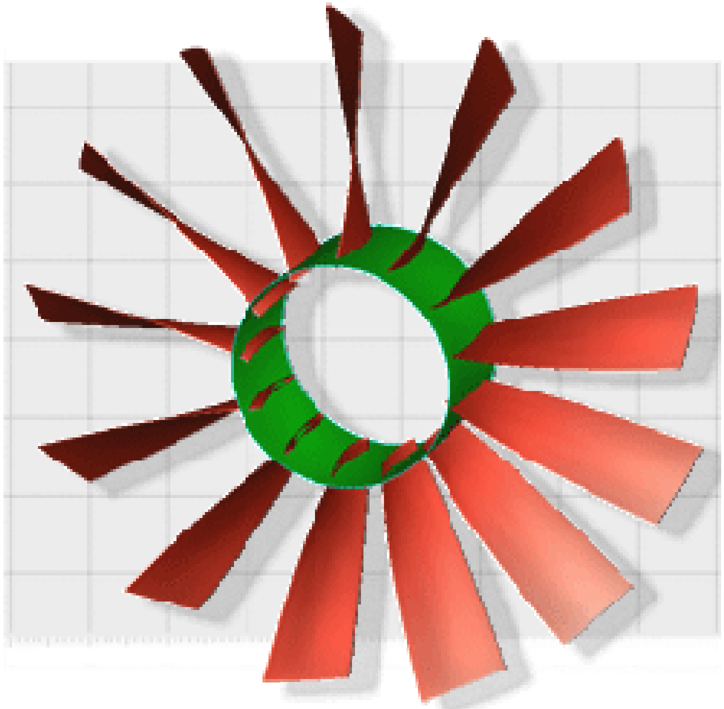


Figure 2.7: Axial fan flow path

Centrifugal fans are divided on the basis of the blade shape used in the impeller, which gives different characteristics depending on specific fan application. The control of flow and pressure for centrifugal fans is governed using two methods: (1) inlet or outlet vane control at constant fan speed and (2) variable fan speed. There are three types of blades used in centrifugal fans:

1. Backward curved (backward bladed)
2. Radial
3. Forward curved (forward bladed)

Axial fans are available in two forms: single stage and two stage, this depends on the number of stages of blades on the rotor. Axial fans control flow and pressure by varying the blade angle. The advantage of axial fans is they provide high efficiency and self-limiting characteristics. As shown in Figure 2.8, axial fans give higher efficiencies and reduced power consumption at various boiler loads compared to centrifugal fans with inlet or outlet vane control. However, axial fans are more expensive in capital and on-going maintenance because of:

- Blade angle movement mechanisms mean an increase in the number of complex parts.
- Superior manufacturing techniques and higher quality materials are required for axial fan components, especially the fan blades.
- Blade erosion can occur when abrasive ash is mixed with the air or gas, particularly with ID fan operating in flue gas mixed with fly ash.

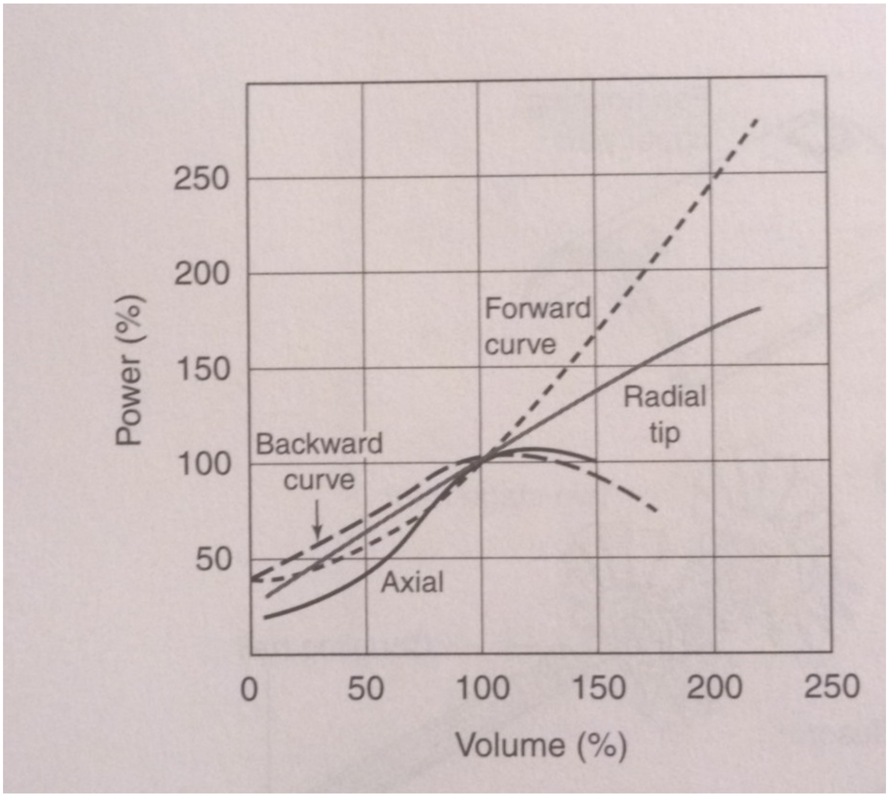


Figure 2.8: Efficiency comparison of various fan types

2.5 Tribology of Rotating Machines

The theory of lubrication and bearing design in rotating machines falls under the broader context of tribology. The term ‘tribology’, is the science of mechanisms of friction, lubrication, and wear of interacting surfaces that are in relative motion. Tribological components are those which carry all the relative movements in rotating machines and their performance is a critical contribution to reliability and efficiency. These components are the local points where high forces and rapid movements are transmitted simultaneously. They are also the components most likely to fail because of the high concentration of energy that is carried at this point during normal operation and particularly start-up. These components are the mechanical fuses if a machine fails in service, therefore indicating a failure mode exists.

2.5.1 Bearing Design

Bearings in heavy rotating machines are divided into two categories: (1) anti-friction bearings, better know as rolling element bearings and (2) fluid film bearings, more commonly know as journal bearings or sleeve bearings. Within these two categories, bearing load on the shaft is categorised by the load direction: radial load, axial load and combination radial and axial load. The axial load component (also referred to as thrust load) is in the direction of the shaft axis, while the radial load component is in the direction normal to the shaft axis (Harnoy 2003, p.3).

Rolling Element Bearings

Rolling element bearings use a set of rolling elements (either balls or rollers) that roll in an annular space between two races (see Figure 2.9), known as the inner race and outer race. The rolling elements are maintained equally spaced via a separator, also known as cage or retainer. The two broad categories for rolling element bearings are ball bearings and roller bearings. The most common types of ball bearings are:

1. Deep-groove ball bearing
2. Self-aligning ball bearing
3. Double-row deep-groove ball bearing
4. Angular contact ball bearing

The most common types of roller bearings are:

1. Cylindrical roller bearing
2. Tapered roller bearing
3. Self-aligning spherical roller bearing
4. Needle roller bearing
5. Multirow tapered roller bearing

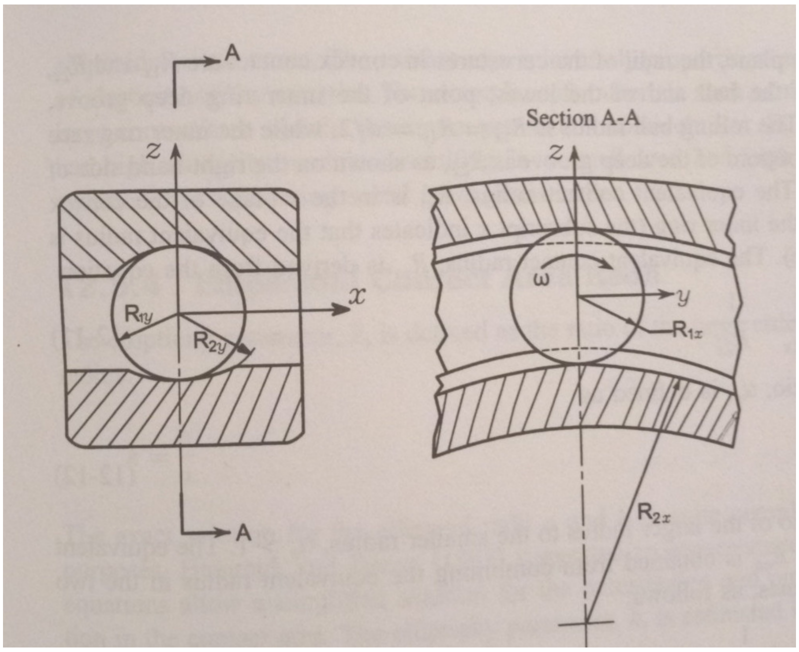


Figure 2.9: Typical ball bearing cross sections

Rolling elements bearing are becoming more common in today's industries mainly due to precision CNC machining capabilities and advances in material metallurgy. These bearings offer the following advantages when compared with journal bearings: (1) less friction at starting and general operation, (2) less sensitive to lubrication interruptions, (3) suitable for high precision applications and (4) more suitable for supporting combination radial and axial loads. However, the disadvantages are: (1) require more space in the radial direction and (2) repeated cycle stresses at ball-to-race contact creates a finite fatigue life.

Journal Bearings

A journal bearing (or tilting pad bearing) is where the journal is rotating inside the bore of a sleeve with a thin clearance. There is a continuous fluid film running in this clearance which separates the solid surfaces and provides load-carrying capacity, hence the terminology fluid film bearings.

There are two principal methods of creating and maintaining a load-carrying film between solid surfaces in relative motion (Szeri 1998, p.33). A journal bearing that creates self-acting lubrication operates in ‘hydrodynamic’ mode. This is when the fluid film is generated and maintained by a wedge of viscous lubricant drawn into the clearance between the two converging surfaces. On the other hand, a journal bearing that is externally pressurized for lubrication operates in ‘hydrostatic’ mode. This is when the fluid film is created and maintained by an external pump that forces high-pressure lubricant between the two surfaces. The lubricant is fed from a pump into the journal bearing via several recesses around the bore of the bearing.

Journal bearings are frequently used in high speed and high load applications where rolling element bearings have shorter life and higher vibration/noise. Figures 2.10 and 2.11 illustrate the cross section geometry of a typical hydrodynamic and hydrostatic bearing.

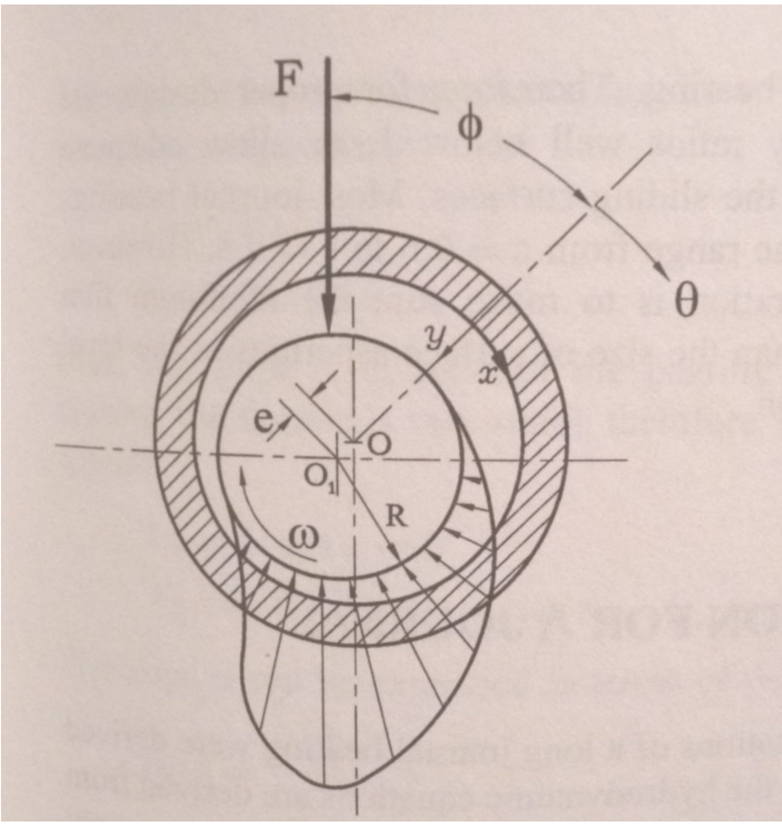


Figure 2.10: Hydrodynamic journal bearing cross section

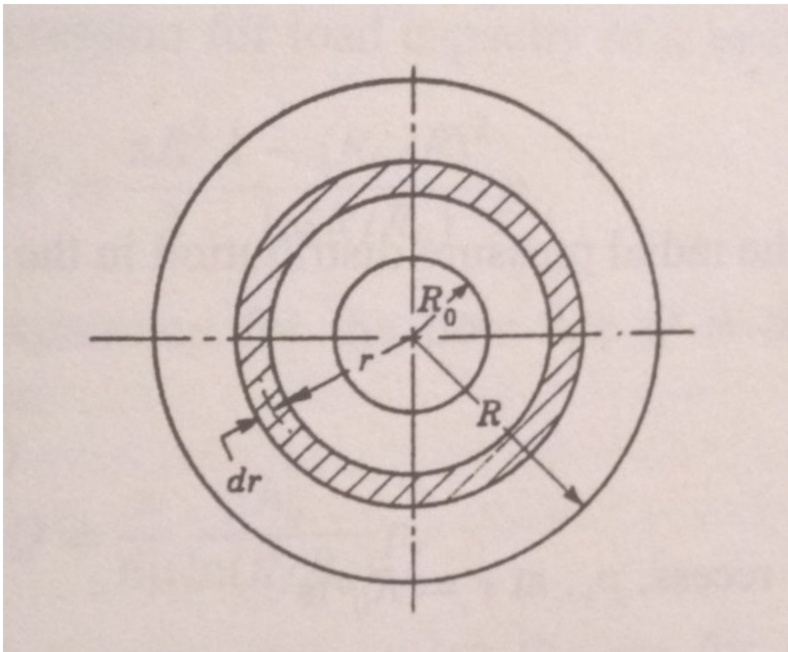


Figure 2.11: Hydrostatic journal bearing cross section

Hydrodynamic bearings are very popular in heavy rotating machines due to their simplistic design and low costs as there is no external high-pressure pump system to install and maintain. One of major disadvantages of hydrodynamic bearings is that a certain minimum angular velocity is required to generate a complete fluid film that completely separates the two surfaces. For this reason, these bearings are not suitable for variable speed drives and excessive stopping and starting applications. In particular, hydrodynamic bearings undergo severe wear during start-up, when they accelerate from zero speed, because static friction is higher than dynamic friction (Harnoy 2003, p.9).

To eliminate the wear problems associated with hydrodynamic bearings, externally pressurized hydrostatic bearings ensures a complete fluid film separates the two surfaces at all speeds, including zero speed. This makes hydrostatic bearings well suited for rotating machines requiring frequent stopping and starting cycles and variable speed. Another advantage of hydrostatic bearings is their good stiffness to radial loads due to the constant pressure fluid film between two solid surfaces. This high stiffness of radial displacement makes this bearing suitable for precision machines, such as precise machine tools. The disadvantages of hydrostatic bearings when compared with hydrodynamic bearings are: (1) higher initial cost, (2) requirement for high-pressure external pump and (3) greater risk of lubrication failure because of failure of external pump.

2.5.2 Callide C ID Fan and Electric Motor Bearings

The Callide C ID fans and electric induction motors utilize both rolling element bearings and journal bearings. The induction motors use a split-sleeve hydrodynamic bearing on both drive end (DE) and non-drive end (NDE). This bearing type is ideal for heavy electric drive applications due to high radial loads, high speeds, simplistic design, constant speed and infrequent stop and start cycles.

The fan main shaft assembly uses single-flange cylindrical roller bearings on both DE and NDE for high radial loads created from the fan rotor and blades. This bearing type is designed for high radial load and low axial load carry capacity. A spherical roller thrust bearing is also used on the NDE for high axial loads created from the axial fan thrust. This bearing type is designed for high axial load and low radial load carry capacity. The fan main shaft bearings provide precision alignment of critical rotating components and can withstand a combination of high radial and axial loads.

2.5.3 Lubrication Theory

A lubricant is a gas, liquid, or solid that is used to prevent contact of parts in relative motion, reducing friction and wear (Mobley 1999, p.55). Lubricants also perform other important functions such as: machine cooling, rust prevention, preventing the deposition of solids on close-fitting parts, and power transmission.

Lubrication is an important part in the efficiency and reliability of rotating machines which is under ever increasing scrutiny in today's economic climate. Engineers are continuously improving lubrication systems in an attempt to increase productivity, more cost effective. Correct lubricant selection and lubrication system design is vital in achieving a cost effective and reliable machine.

Of the three lubricant types listed above, liquid lubricant is the most commonly used in heavy rotating machines. Liquid lubricants have many advantages over solid and gas lubricants. The most important advantages of liquid lubricants are the formation of hydrodynamic films, the cooling of the bearing by effective convection heat transfer, and finally their relative convenience for use in bearings (Harnoy 2003, p.47). The most common liquid lubricants are minerals oil, which are produced from petroleum (Harnoy 2003, p.47). Mineral oils typically contain greater than 90% base oil and less than 10% additives. Oil additives are used to improve base oil characteristics, such as:

- Minimise lubricant oxidation
- Maintain thermal stability at high temperatures
- Provide good water separation properties
- Protect contact surfaces against corrosion
- Improve viscosity index (VI)
- Improve low-temperature characteristics
- Provide a wear-resistance film on all contact surfaces
- Maintain cleanliness of components
- Reduce air entrainment and foaming
- Improve wear resistance under extreme pressure (EP)

2.5.4 Fluid Film Lubrication

Fluid film lubrication in bearings is commonly divided into two categories: thin-film lubrication and thick-film lubrication. A thick-film lubrication condition is when the coefficient of friction between the two surfaces is small and depends on no other material property of the lubricant than its bulk viscosity. In many respects, this type of lubrication is the simplest and most desirable as friction coefficient is very low (i.e. 0.0001) and there is complete separation between the two solid surfaces. Thus, this form of lubrication is encountered in both hydrodynamic and hydrostatic bearings. A thin-film lubrication condition is when the fluid film thickness is 1 µm or less, usually encountered in rolling element bearings where there is counter-formal contact of the rolling element and race. This type of lubrication has a higher friction coefficient (i.e. 0.05 – 0.15) which depends on the surface roughness, bearing material properties and lubricant properties.

2.5.5 Callide C ID Fan Lubrication

As highlighted above, the importance oil selection with the proper additives is crucial in all lubrication systems. The oil used in Callide C ID fan lubrication system is Shell Tellus S 46, which is a zinc-free hydraulic oil used for severe conditions. The Shell Tellus S oil range is a 'top-tier' anti-wear hydraulic oil formulated to provide exceptional performance in hydraulic fluid power transmission subjected to severe duty. One of the most important performance benefits is long life due to its excellent oxidation properties, between two to four times that of other anti-wear hydraulic oils. Extended oil life depends on its ability to resist oxidation due to heat in the presence of air, water and metal catalysts such as copper (Shell Product Data Sheet 2009). Refer to Table 2.1 for Shell Tellus S 46 oil properties.

Table 2.1 – Shell Tellus S 46 oil properties

Shell Tellus Oil		**S 46**
Viscosity Grade (ISO 3448)		46
ISO Oil Type		HM
Kinematic Viscosity		
@ 0°C	cSt	576
20°C	cSt	135
40°C	cSt	46
100°C	cSt	6.8
(IP 71)		
Viscosity Index (IP 226)		98
Density @ 15°C (IP 365)	kg/l	0.876
Flash Point (IP 34) (PMCC)	°C	218
Pour Point (IP 15)	°C	-30

Chapter 3 – Methodology

3.1 Further Research and Appraisal

Before investigation effort is conducted into the operational and reliability failure modes in the existing ID fan lubrication system, further research and appraisal is required on the different types of bearings and lubrication used in large electric drives and axial fans. This process is crucial in identifying any inherent bearing/lubrication design defects in the ID fans and electric induction motors.

3.1.1 Bearing Selection

The decision between which bearing type to use in a particular application is not always easy or even obvious. The 'best' bearing decision depends on the details of the particular application and weighing up the advantages and disadvantages of journal bearings and rolling element bearings. The choice of bearing arrangement is based on the following key aspects:

- Load carry capacity in the axial and radial direction
- Rotating speed
- Over speed and duration
- Operating temperature
- Environmental conditions
- Lubrication selection
- Bearing operational hours

The main shaft (rotor) assembly for both axial fans and inductions motors can be purchased with either rolling element or journal bearings. Both bearing systems have proven their valve and reliability in many installations over the years, therefore the decision between journal bearing and anti-friction bearing is often philosophical. Both designs have different performance characteristics and inherent maintenance and reliability differences. The initial cost of the journal bearing is much greater than the rolling element bearing design, but this may not be as significant when you take into account the total life cycle cost (Finley & Hodowanec 2001).

Journal bearings, either split-sleeve or tilting pad configuration are commonly used in heavy rotating machines were large diameter rotors are sought. As larger rotor sizes are encountered, larger shaft diameters will have to be used to keep torsional shaft stresses within acceptable levels. However, as shaft size (and thus bearing size) goes up, the bearing speed limit goes down. Eventually the required shaft size will result in anti-friction bearings that have a lower bearing speed limit than operational speed of the machine (Finley & Hodowanec 2001). This is the transition point where a journal bearing is required for this particular application.

In small induction motors 150 kW or less, there is no choice in bearing selection as only rolling element bearings are readily available for small shaft diameters. Likewise, a choice does not always exist on larger motors above 1500 kW as various design requirements leave only the journal bearing (or tilting pad bearing) as a viable option. As a result, a suitable bearing choice for intermediate size motors between 150 kW and 1500 kW is often made by the design engineers. The bearing selection graph shown in Figure 3.1 gives a guide into correct bearing selection for different motor size and operating speed (Hoppler & Errath 2007).

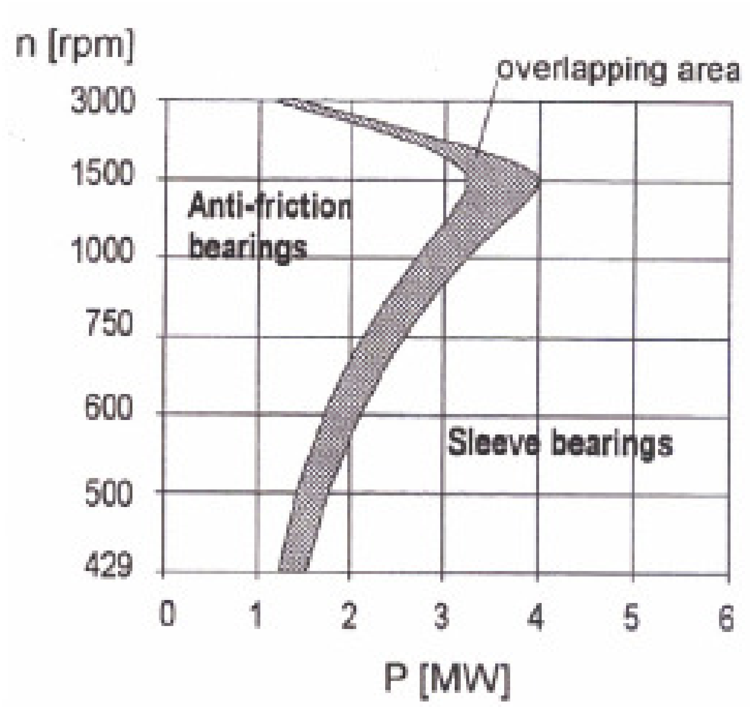


Figure 3.1: Induction motor bearing selection graph

ID Fan Induction Motors

The Callide C ID fan induction motors are high voltage (HV) induction motors and rated voltage is 6600 V or 6.6 kV. These large motors have a rated output of 3,810 kW or 3.8 MW, with a rotor weight of 5,000 kg and total weight of 22,500 kg. They are a 6 pole motor with synchronous output speed 1000 rpm and rated output speed 995 rpm, thus rated slip is 0.5%. The synchronous speed is governed by the following equation:

$$Srpm = \frac{120 \times F}{P} = \frac{120 \times 50}{6} = 1000$$

where Srpm = synchronous revolutions per minute

120 = constant

F = supply voltage frequency (Hz)

P = number of motor winding poles

As mentioned earlier in Chapter 2, the ID fan induction motors use a split-sleeve journal hydrodynamic bearing on the DE and NDE. Due to the large size and heavy weight of these motors, the existing bearing arrangement is correct and fit-for-purpose. This is further justified using Figure 3.1, as it shows a 3.8 MW electric motor running at 995 rpm is recommended to use sleeve bearings (also known as journal bearings).

Another important aspect of split-sleeve journal bearings is their user-friendliness for maintainability and replacement. Split-sleeve means the bearing is manufactured as two identical semicircle pieces which form together to provide a full 360 degree bearing. The biggest advantage of this feature when compared with one-piece journal bearings or rolling element bearings is that the shaft does not need to be separated from the machine for bearing inspection or replacement. This allows all bearing maintenance to be conducted in-situ, which is a huge time and cost saving benefit for large and heavy rotating machines.

ID Fan Main Shaft

The Callide C ID fans are Howden Variax dual-stage axial fans which are designed to operate at 88% efficiency at design volume flow rate 350 m^3/s. These fans are quite large in physical size with an impeller outer diameter 2700 mm and hub outer diameter 1500 mm.

As stated in Chapter 2, the ID fan main shaft bearings consist of two single-flange cylindrical roller bearings and a spherical roller thrust bearing. This type of rolling element bearing arrangement offers distinguishing advantages compared to journal bearings in the same application. Rolling element bearings provide superior emergency operation in the event of lubrication interruptions to the bearing sump. They also permit precision tolerances for critical rotating parts and are more suitable for supporting high axial and radial loads. Given this information, the current bearing arrangement for the ID fan main shaft is also correct and fit-for-purpose.

3.1.2 Lubrication Selection

Correct lubrication selection is just a vital as correct bearing selection. Lubrication for rolling element bearings and journal bearings in axial fans and induction motors is commonly divided into two categories: (1) oil lubrication and (2) grease lubrication. Oil is classed as a liquid lubricant and grease is defined as a semi-fluid lubricant. Basically, a grease lubricant is a liquid lubricant that is mixed with another thickener substance (i.e. soap) to form a semi-fluid. This gives grease a higher viscosity than oil, which is very useful for infrequent lubrication applications where oil lubrication is neither practical nor feasible. The decision between oil or grease lubricant depends on the details of the particular application and weighing up their advantages and disadvantages. The choice of lubricant type is based on the following key aspects:

- Bearing type and speed limit
- Bearing operating temperature
- Required bearing cooling rate
- Bearing operational hours
- Environmental conditions

ID Fan Induction Motors

Because the Callide C ID fan induction motors currently run with hydrodynamic journal bearings, there is no choice other than oil lubrication for this application. These bearings require oil lubrication to provide a fluid film which separates the solid surfaces and provides load-carrying capacity. Hydrodynamic journal bearings usually require flood (or force fed) lubrication systems as this ensures oil circulation is maintained to provide cooling of bearing. The flood lubrication system must be able to deliver a predetermined volume flow rate of oil at a certain pressure to ensure maximum bearing life. Oil rings are oftentimes used to provide oil circulation in the bearing sump in the event that the flood lubrication system fails. This redundant feature will enable the induction motor to safety coast down shaft speed to a stand still without causing severe bearing damage.

The lubrication design on the ID fan induction motors uses all the aspects as listed above, therefore proving current design is correct and fit-for-purpose. The current design is oil flood lubrication via an external lubrication system and oil rings are also used for redundancy. The only key component that is absent is no oil flow monitoring in the current system to ensure volumetric flow rate is maintained as per OEM recommendations.

ID Fan Main Shaft

Unlike the induction motor journal bearings which must employ oil lubrication, rolling element bearings have the choice between oil and grease lubrication. The choice between oil and grease lubrication in axial fans with rolling element bearings is dependant fan application.

Axial fans with grease lubricated bearings have a lower initial cost compared to oil lubricated bearings. However grease lubrication requires more ongoing routine maintenance and can be very problematic. Common problems are over-lubrication, under-lubrication and lack of cleanliness during re-lubrication, which all impact on reliability and maintenance costs.

Grease lubrication is also very limited in providing cooling of the bearing as there is no volumetric flow of fluid to permit effective convection heat transfer. For this reason, axial fans operating with grease lubrication are primarily designed for ambient temperature applications as the bearings are located in the gas flow path. This is typical in FD fan and PA fan applications where an axial fan is used to supply ambient temperature air to the boiler.

On the other hand, axial fans that are designed for applications which operate above ambient temperatures must have some form of bearing cooling. The most common form of bearing cooling is oil flood lubrication as volumetric flow of oil provides cooling of the bearing. This is typical in ID fan applications where an axial fan is designed to remove hot gas between 120℃ to 150℃ from the boiler. Callide C ID fan main shaft bearings are running oil flood lubrication for this reason, therefore proving the current design is correct and fit-for-purpose. Again, as explained previously, the main factor that is absent is no oil flow monitoring in the current system to ensure volumetric flow rate is maintained as per OEM recommendations.

3.2 Root Cause Analysis Methodology

Root Cause Analysis (RCA) is a tool used to identify and define root causes and their associated failure modes. This method is critical as it ensures the correct failure modes are determined so the root causes can be eliminated. Close adherence to this process avoids "solving the wrong problem", which is evident when the flow transmitters were replaced with pressure transmitters back in 2004. The Apollo RCA process developed by Apollo Associated Services will be adopted for this project as it is CS Energy's preferred method. The Apollo RCA process is a 4-step method (Gano 2003, p.55):

1. Define the Problem
2. Create an Apollo Cause and Effect Chart
3. Identify Effective Solutions
4. Implement the Best Solutions

The first step to make certain the RCA process obtains common agreement on the root causes is to define a "problem statement". A problem should contain (Gano 2003, p.63):

- *What* is the problem?
- *When* did it happen?
- *Where* did it happen?
- What is the *significance* of the problem?

This gives

- What – Lubrication system trips ID fan
- When – Since commissioning in 2001
- Where – Callide C ID fans
- Significance – Has caused 11 ID fan trips and a loss of 7800 Megawatt hours (MWh) from 2004

Problem Statement: *"Since commissioning in 2001 the lubrication system on Callide C ID fans has resulted in 11 ID fan trips and a loss of 7800 MWh from 2004"*

The next stage is to analyse the 'cause and effect' relationships, and then the findings using a "Causal Tree Analysis". The key benefits to this process are:

- Ideal for solving multi-faceted problems
- Displays the sequence of a series of facts, conditions and actions leading to an event
- Graphically displays the known's and unknown's, therefore identifying what questions to ask to follow the path to the root cause
- A systematic approach in organising and presenting investigation data

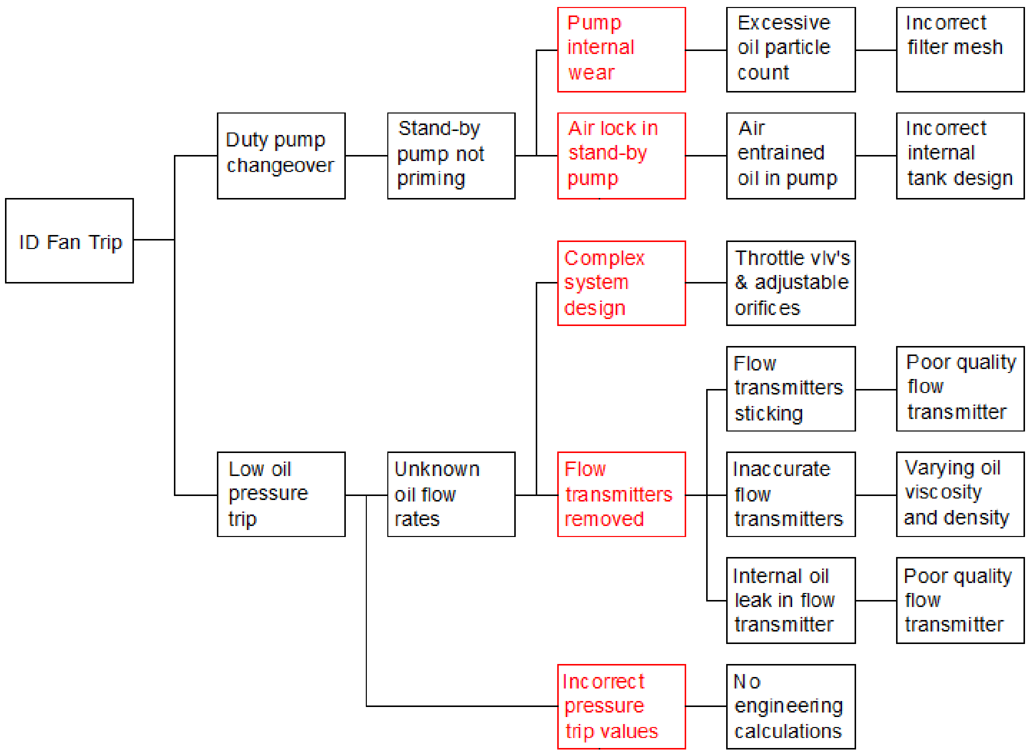


Figure 3.2: Causal Tree Analysis

Figure 3.2 highlights all failure modes and root causes which have been causing problems in the ID fan lubrication system. The causal tree analysis also shows the complex nature of identifying all problems which contribute to the root causes. The five primary causes are:

1. Pump internal wear
2. Airlock in stand-by pump
3. Complex system design
4. Flow transmitters removed
5. Incorrect pressure trip values

3.2.1 Pump Internal Wear

The Callide C ID fan lubrication pumps are a Bosch Rexroth External Gear Pump Series S, (Silence) part number 0 517 725 002. Refer to Figure 3.3 for exploded assembly drawing. The key task of external gear pumps is to convert mechanical energy (torque and rotation speed) into hydraulic energy (flow and pressure). This unit is a fixed displacement pump with a displacement [V = 25 cm^3/rev]. To convert displacement into volumetric flow rate [Q], refer to following steps:

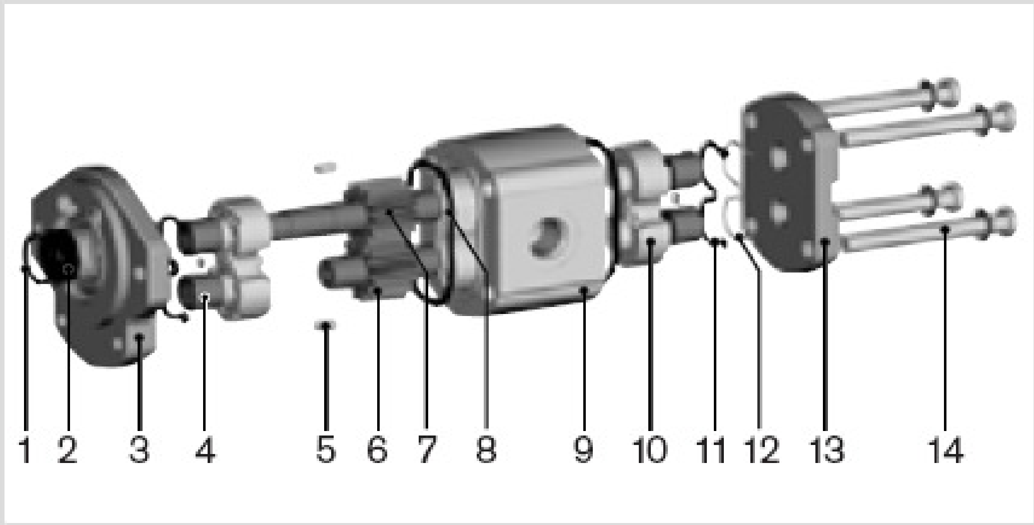
$$Q = \frac{V \times n \times \eta}{1000}$$

where V = 25 cm^3/rev

n = 1440 rpm

η = 95%

$$Q = \frac{25 \times 1440 \times 0.95}{1000} = 34.2 \approx 34 \ L/\min$$



1 Retaining ring	8 Case seal
2 Shaft seal ring	9 Pump case
3 Front cover	10 Bearing
4 Slide bearing	11 Axial zone seal
5 Centering pin	12 Support
6 Gear	13 End cover
7 Gear (frictional)	14 Fixing screws

Figure 3.3: Exploded assembly drawing of gear pump

Table 3.1 – Comparison of gear pump specifications

	Gear Pump Design	**Operation Conditions**
Kinematic Viscosity	20 to 100 mm^s/s	20 to 100 mm^s/s
Maximum rotation speed	3000 rpm	1440 rpm
Maximum pressure	22.5 MPa	2 MPa
Max fluid temperature	80℃	25 to 60℃
Maximum oil contamination class to AS4002	20/18/15	22/20/17

Table 3.1 compares the gear pump design specification against the operation conditions. This shows operating conditions of kinematic viscosity, rotation speed, pressure and fluid temperature are within the gear pump design parameters. However the operating oil cleanliness code 22/20/17 to AS4002.1 exceeds the gear pump maximum allowable cleanliness code 20/18/15. The cleanliness code for operating conditions was conducted using oil samples in June 2010. The samples were sent to ASL Tribology for results.

AS4002.1 is an Australian Standard: "Hydraulic fluid power – Particulate contamination of systems, Part 1: Method for coding the level of contamination". The code for contamination levels using automatic particle counters comprises of three scale numbers, which permit the differentiation of the dimension and the distribution of the particles as follows (Australian Standards, 2001):

1. the first scale number represents the number of particles equal to or larger than 4 µm per millilitre of fluid;
2. the second scale number represents the number of the particles equal to or larger than 6 µm per millilitre of fluid;
3. the third scale number represents the number of the particles equal to or larger than 14 µm per millilitre of fluid.

The scale numbers are allocated according to the number of particles counted per millilitre of the fluid sample, refer to AS4002.1 page 3.

EXAMPLE: A code 22/20/17 signifies that there are more than 20,000 and up to and including 40,000 particles equal to or larger than 4 µm, more than 5,000 and up to and including 10,000 particles equal to or larger than 6 µm and more than 640 and up to and including 1,300 particles equal to or larger than 14 µm in 1 millilitre (ml) of a given fluid sample.

Table 3.2 – Comparison of oil particle contamination

AS4002.1	**Gear Pump Design (20/18/15)**	**Operation Conditions (22/20/17)**
First scale number ≥ 4 µm	5,000 to 10,000 particles	20,000 to 40,000 particles
First scale number ≥ 6 µm	1,300 to 2,500 particles	5,000 to 10,000 particles
First scale number ≥ 14 µm	160 to 320 particles	640 to 1,300 particles

Table 3.2 gives a comparison of particle contamination with gear pump design versus current operation conditions. This shows the current oil particle contamination is a factor of four times greater than allowable, which dramatically decreases gear pump life and increases internal wear. To determine the extent of wear in the gear pumps, all Callide C ID fan lubrication pumps were replaced in June/July 2010 and disassembled for internal inspection.

Upon disassembly, it was evident the pumps had suffered from internal abrasion wear. This is when particles bridge a critical clearance and scrape along one or both surfaces during relative movement. As illustrated in Figure 3.4 and 3.5 the abrasion wear is on the pump outer casing and shaft bearings. The cause for this wear at these locations is due to the aluminium outer casing and shaft bearing having a softer material than the hardened steel pump gears. The oil running through the pump maintains a fluid film to separate the relative moving components, however due to excessive particle contamination in the oil the softer aluminium material is removed via a cutting action from the pump gears.

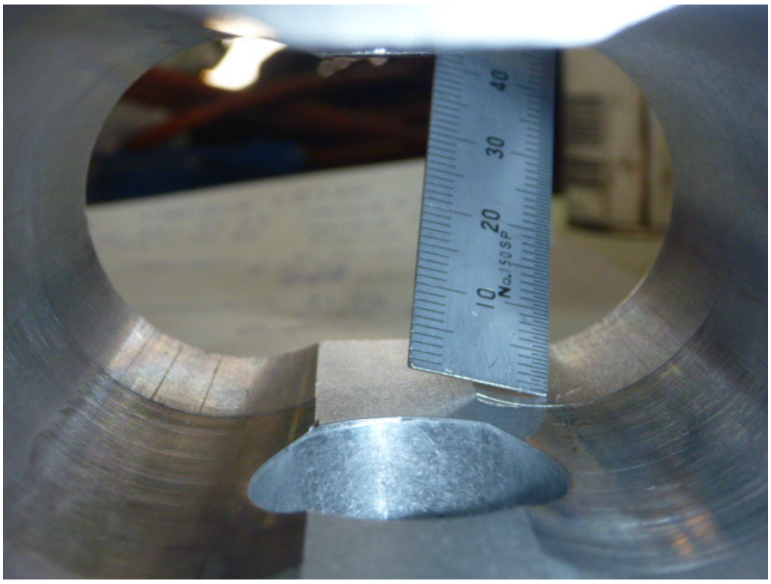


Figure 3.4: Abrasion wear on gear pump outer casing

Figure 3.5: Abrasion wear on gear pump shaft bearing

Precision measuring tools such as micrometers and telescopic gauges were used to measure the addition wear on the aluminium components. The addition wear on pump outer casing inner diameter is 50 µm and shaft bearings is also 50 µm. This amount of additional wear in a positive displacement pump seriously affects its volumetric performance and self priming capability.

3.2.2 Airlock in Stand-by Pump

From commissioning in 2001, the Callide C ID fan lubrication system has always suffered from the stand-by pump unable to prime (or make system pressure). This situation poses a huge risk to the operation and reliability of the ID fans as the stand-by pumps are required for emergency back-up situations.

It has been assumed that the cause of the stand-by pump inability to prime is due to an airlock in the pump internals. This assumption in the past has been a little difficult to prove as there is no bleed-off line at the pump discharge to vent air to atmosphere. Also, there is no provision for individual pump pressure testing as the system pressure [700 kPa] indication is taken from the common line (see Figure 3.6) after non-return valves and isolation valves.

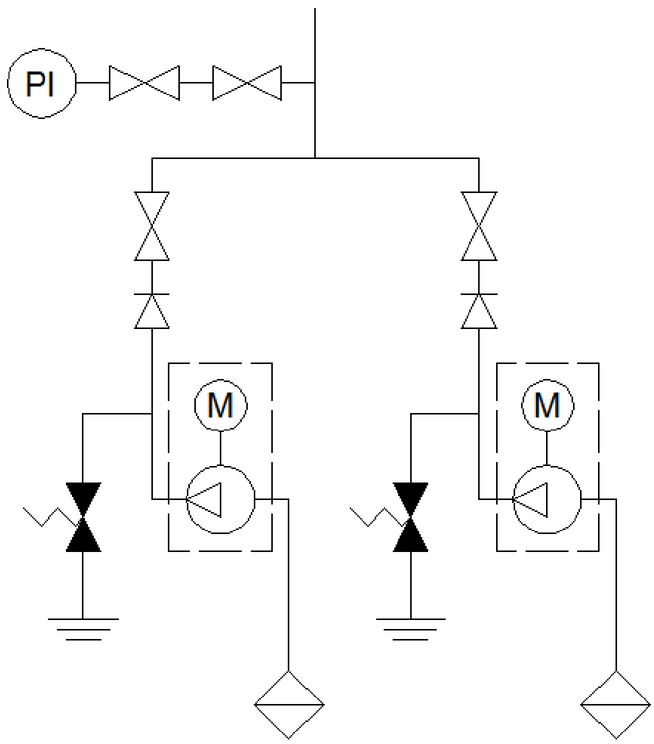


Figure 3.6: Lubrication pump P&ID

So to prove the airlock theory, a special procedure was developed to allow the removal of the stand-by pump pressure relief valve and install a custom machined test plug with a flexible hose and local pressure gauge (see Figure 3.7). This procedure was very beneficial for three reasons: (1) allows any airlock in the pump internal to bleed to atmosphere, (2) gives individual pump pressure indication to diagnose if stand-by pump can create system pressure to open the non-return valve and (3) most importantly this procedure allowed the ID fans to remain in service without any load reduction.

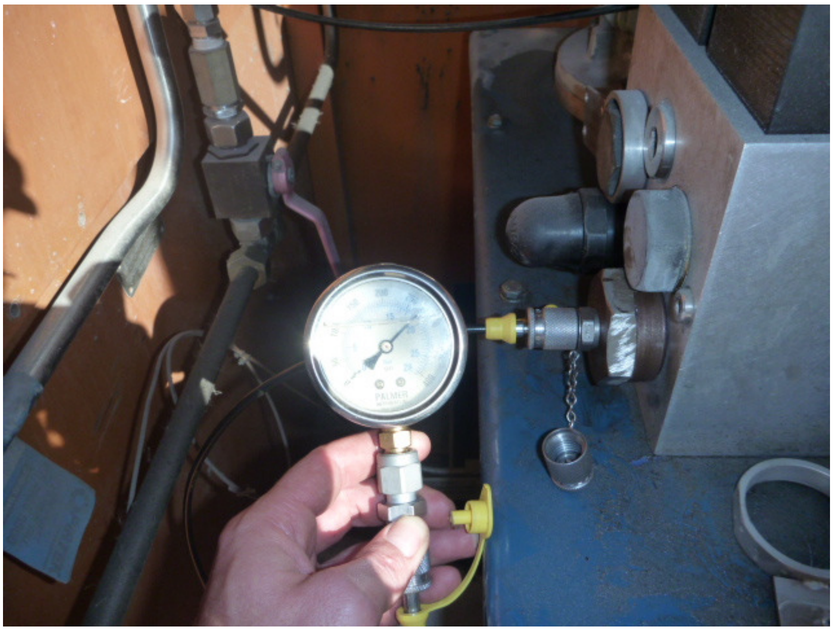


Figure 3.7: Individual pump pressure test

The outcome of this procedure was successful and proved the airlock theory. When the pressure relief valve was removed, it allowed trapped air to vent to atmosphere and this allows the tank oil level to self prime the stand-by pump. The pressure testing equipment was then installed and the stand-by pump was started to confirm whether it could make system pressure and lift the non-return valve, refer to Figure 3.8. Once the stand-by pump started, it actually doubled system pressure [1400 kPa] as expected with two pumps running simultaneously.

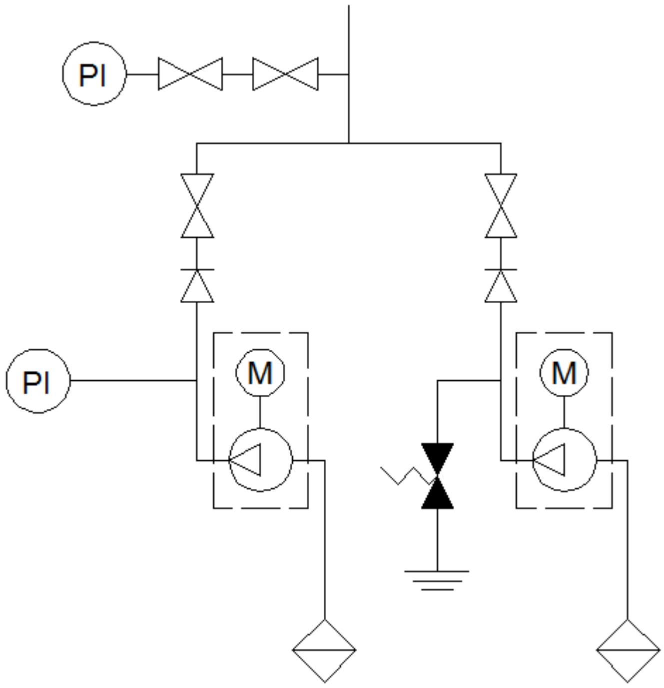


Figure 3.8: Lubrication pump P&ID with local test pressure gauge

The next step was to determine how air is entering the pump internals while in stand-by mode. As illustrated in Figure 3.9, the ID fan lubrication tank internal arrangement has two oil return lines facing downwards at the tank rear wall. The pump suction strainers are within very close proximity to the flow path created from the return lines. Unfortunately this design allows air entrained return oil from the fan and induction motor bearings to recirculate directly into the pump suction strainers. Once this oil comes into contact with the suction strainer, the air separates from the oil and then air floats up the suction pipe into the pump internals. Basically the suction strainers are acting as a de-aeration screen which is causing airlocks in the stand-by pumps.

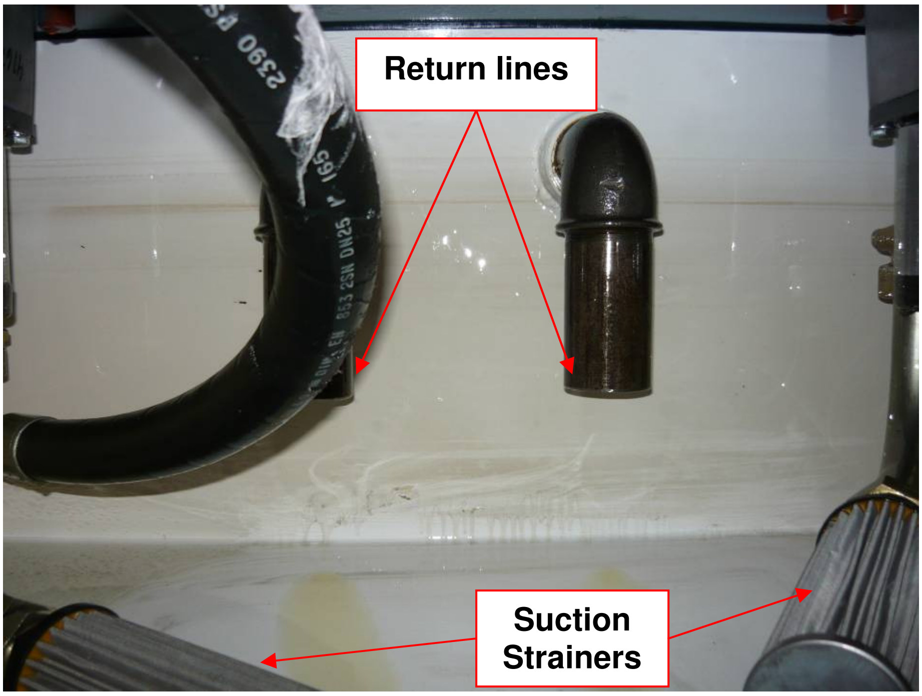


Figure 3.9: ID fan lubrication tank internal arrangement

3.3.3 Complex System Design

As mentioned earlier in Chapter 1, the lubrication oil flow rates to the ID fan and induction motor bearings are adjusted using two throttle valves and two adjustable orifices (see Figure 3.10). This method can be sustainable with accurate and reliable flow transmitters. However, since the removal of the flow transmitters this method becomes very complex and unachievable.

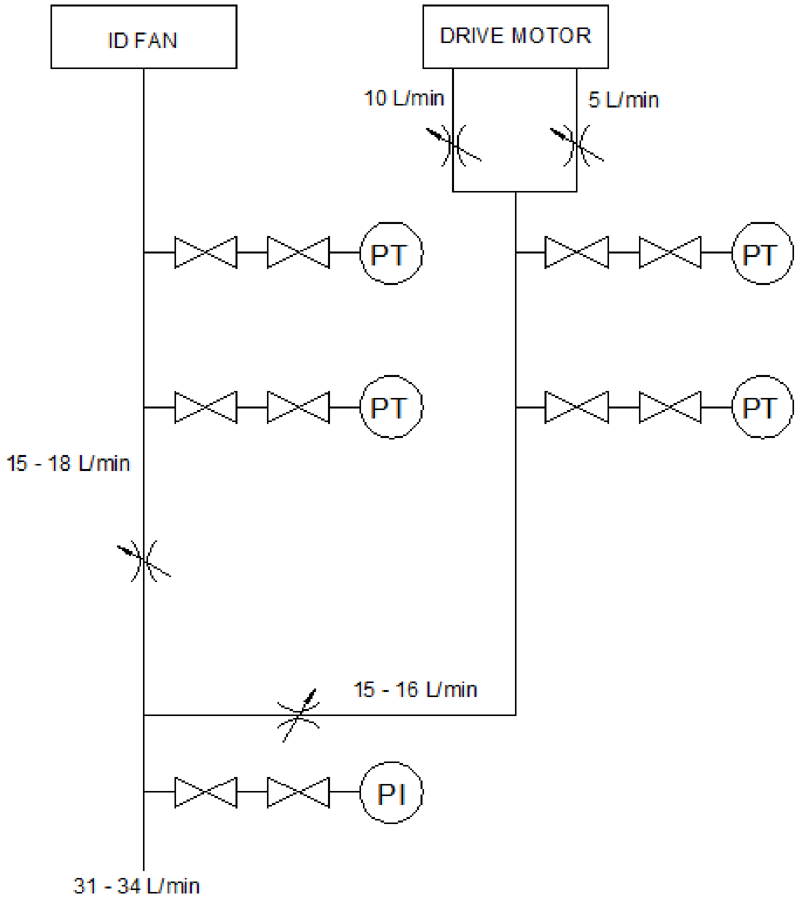


Figure 3.10: Lubrication system P&ID

The two throttle valves located inside the lubrication system cabinet (see Figure 3.11) are used to attain required flow rates to induction motor bearings at 15 – 16 L/min and fan main shaft bearings at 15 - 18 L/min. Currently these flow rates are not known as there is no flow monitoring, leaving the system vulnerable to incorrect set-up. The current pressure transmitters which were installed in 2004 do not give any valid assistance during the flow rate set-up process.

Figure 3.11: Two throttle valves located inside cabinet

The two adjustable orifices located at the inlet points to the induction motor bearings (see Figure 3.12) are used to attain required flow rates to DE bearing at 10 L/min and NDE bearing at 5 L/min. The main problem with this particular orifice is that there is no provision for cross-sectional area calibration between the each orifice. This makes accurate oil flow calibration between the two orifices seem almost impossible and very unpractical. Another issue with this type of adjustable orifice is they can not be adjusted while there is pressure in the system as high pressure oil will leak to atmosphere when the inspection plug is removed. This means calibration can not be done while the lubrication system is running and also poses an environmental and safety issue if high pressure oil leaks to atmosphere.

Figure 3.12: Adjustable orifice

3.3.4 Flow Transmitters Removed

The OEM Hedland Flow Transmitter (see Figure 3.13) is a piston type, ‘variable area’ flowmeter with a precision machined sharp edged orifice located within the piston assembly. The ID fan lubrication design was installed with 4 x Hedland Flow Transmitters (Part No. HP602A-010ELS10) which provided continuous monitoring of oil flow rates to the induction motor and ID fan main shaft bearings. For redundancy purposes there were two flow transmitters for each line (induction motor and ID fan main shaft) with isolation valves on both inlet and outlet with a bypass line (see Figure 1.2). This allowed repair and/or replacement of a flow transmitter without having to remove the ID fan from operation, which is a very important redundancy feature.

Figure 3.13: OEM Hedland flow transmitter

As mentioned in Chapter 1 and further emphasised in the Causal Tree Analysis, the flow transmitters were unfortunately removed in 2004 due to three failure modes. The issues with the sticking components and internal oil leaks were caused from poor quality flow transmitters. The flow rate accuracy problem was due to differences in flow transmitters design parameters and actual lubrication system operating conditions.

Table 3.3 – Comparison of flow transmitter specification

	Flow Transmitter Design	**Operation Conditions**
Kinematic Viscosity	4.2 to 108 mm^s/s	20 to 100 mm^s/s
Specific Gravity (Density)	0.876	0.846 to 0.870
Volumetric Flow Rate	0 to 36 L/min	10 to 18 L/min
Operating Pressure	0 to 6.9 MPa	0 to 2 MPa
Fluid Velocity	0 to 7.6 m/s	0 to 1.5 m/s

As shown in Table 3.3, the operating parameters of kinematic viscosity, volumetric flow rate, operating pressure and fluid velocity are within the flow transmitter design parameters. However, oil specific gravity range 0.846 to 0.870 is the only operating parameter which does not fall within design value 0.876. The flow transmitter is designed for petroleum based fluids with a specific gravity of 0.876. Shell Tellus S 46 only maintains specific gravity 0.876 @ 15℃ which is not within oil operating temperature. This inaccuracy would have been a contributing factor to any flow rate errors.

Also, the volumetric flow rate range 0 – 36 L/min for the flow transmitter is double the operating condition range 10 – 18 L/min. This excessive range means the flow transmitter is only operating in the lower half range which means the incremental device range can be too coarse for required accuracy. A flow transmitter with a range say of 2 - 22 L/min would have given improved accuracy for this particular application.

3.3.5 Incorrect Pressure Trip Values

In 2004, an engineering consultancy was approached by CS Energy to provide an engineering solution for the on-going issues which plagued the operation and reliability of Callide C ID fans. From this, the design and implementation of an engineering modification project was completed in 2004. The project involved removing the 4 x flow transmitters from each ID fan lubrication system and replacing with 4 x pressure transmitters. The ICMS alarm and trip parameters were also modified from oil flow rate to oil pressure, as shown in Table 3.4.

Table 3.4 – Comparison of flow rate and pressure protection values

Description	**Alarm**	**Start Condition**	**Trip**	**Start stand-by pump**
Prior to 2004 engineering modification				
Fan Oil Flow	x < 12 L/min	x > 12 L/min	x1 < 10 L/min	x < 12 L/min
Motor Oil Flow	x < 12 L/min	x > 12 L/min	x1 < 10 L/min	x < 12 L/min
After 2004 engineering modification				
Fan Oil Pressure	x < 200 kPa	x1 > 250 kPa	x2 < 150 kPa	x1 < 250 kPa
Motor Oil Pressure	x < 105 kPa	x1 > 130 kPa	x2 < 80 kPa	x1 <130 kPa

Unfortunately after this modification project was implemented, the ID fans still suffered from continuous alarms and trips which caused half load and full load unit trips. Again, the main issue was when the oil temperature increased to a maximum 60℃ on a hot day, oil viscosity and density would decrease causing a drop in line pressure. The drop in line pressure would eventually activate "Start stand-by pump" and when the stand-by pump did not prime and make system pressure, the pressure would then drop further and activate "Alarm" and subsequently "Trip" conditions. A secondary issue worth mentioning, degrading pump performance from internal wear also contributed to pressure drops which further influenced this problem.

A full review of the modification file was recently conducted to determine the process of selecting the modified oil pressure values for ICMS. The file did not contain any relevant OEM correspondence or engineering calculations to prove validity of the new values. Eventually the author contacted the Howden Australia to investigate the integrity of the parameters used in the modification project. It was then discovered that Howden Australia supplied these values as only 'guideline values' to assist with the commissioning. Engineering appraisal should have been conducted at commissioning stage to 'fine-tune' the oil pressure protection values to suit all temperature and pressure fluctuations.

Due to problems created from the new pressure values, in 2009 CS Energy engineers were responsible for modifying the parameters to avoid continuous alarms and trips. During this process it was also discovered that the "Start stand-by pump" value is greater than the "Alarm" value. This arrangement is not recommended as the stand-by pump starts before an alarm is activated to the unit operator. A more suitable arrangement is to have "Start stand-by pump" value equal to or less than "Alarm" value [$x \geq x1$]. This ensures the unit operator is made aware of a lubrication system fault with only the duty pump in service and therefore appropriate corrective maintenance can be planned and executed.

Chapter 4 – Effective Design Solutions

4.1 Research and Evaluation of Solutions

As highlighted in the previous chapter, five primary causes were identified using the RCA process. From this information, all possible design solutions now can be researched and evaluated to decide a final design proposal for Callide C Power Station.

4.1.1 Particle Contamination

The internal wear discovered in the gear pump is caused from excessive particle contamination in the lubrication oil. The only method of decreasing particle contamination is by using finer filtration compared to the current filtration unit. The current ID fan lubrication filtration unit is a duplex change-over unit (i.e. duty and stand-by operation) capable of 60 L/min. The current filter elements are glass fibre 25 µm with a beta efficiency ratio (β_{25}) = 100 (99%). This means the filter elements are 99% efficient of particle separation of particle size equal to or greater than 25 µm. Beta ratio and efficiency derivation is given by Rohner (1995):

Beta Ratio Derivation

$$Beta\ Ratio\,(\beta) = \frac{particle\ count\ upstream}{particle\ count\ downstream}$$

Beta Efficiency Derivation

$$Beta\ Efficiency\,(\%) = \frac{particle\ count\ upstream - particle\ count\ downstream}{particle\ count\ upstream}$$

From the oil samples conducted in June 2010, it is very obvious the current 25 µm filter elements can not meet an oil cleanliness code 20/18/15 or better to match Bosch Rexroth recommendation. As the current filter elements already have high beta efficiency of 99%, the filter fineness needs to lessen to meet the required oil cleanliness code. This particular model of filtration unit is suitable for a variety of filter element sizes: 3 µm, 6 µm, 10 µm, 16 µm and 25 µm.

To select the correct the filter fineness size the OEM plant manual was studied to obtain any recommendation. It was discovered the OEM recommends using a 10 µm filter element with a beta efficiency ratio (β_{10}) = 100 (99%). CS Energy's Lubrication Standard was also used to determine a recommended oil cleanliness which specified an oil cleanliness code 18/16/13 for hydraulic oil which is much cleaner than 22/20/17. In fact, it is approximately a factor of sixteen times cleaner which is massive variation. A 10 µm filter element will achieve both CS Energy and Bosch Rexroth oil cleanliness recommendations. Refer to Table 4.1 for exact figures of particle contamination to AS4002.1.

Table 4.1: Comparison of cleanliness code

AS4002.1	**CS Energy's Lubrication Standard (18/16/13)**	**Operation Conditions (22/20/17)**
First scale number ≥ 4 µm	1300 to 2500 particles	20,000 to 40,000 particles
First scale number ≥ 6 µm	320 to 640 particles	5,000 to 10,000 particles
First scale number ≥ 14 µm	40 to 80 particles	640 to 1,300 particles

A 10 µm filter element which will achieve an oil cleanliness code 18/16/13 would dramatically decrease gear pump internal wear and therefore improve long-term reliability. To change from 25 to 10 µm filter element only requires a change to the material specification which is a very simple change at minimal cost.

4.1.2 Internal Tank Design

The ID fan lubrication tank is suffering from poor internal tank design which permits air to enter the stand-by pump suction line and causing inability of pump to prime. Neale (1995, p.C20.1 – 20.2) has a section of his book "The Tribology Handbook" which shows correct design of lubrication storage tanks. There are four key aspects for correct internal tank design to prevent air entering pump suction line:

- Return line
- Suction line
- Baffles and weirs
- De-aeration screen

Return Line

As shown in Figure 3.9, the tank return lines have 90° elbows and DN50 mm vertical straight sections which direct oil flow downwards at one-third tank level. This allows air entrained oil to circulate directly into pump suction line before the air has been separated from the oil. Neale (1995, p.C20.1) recommendations for tank return lines:

- *Location*: at or just above running level
- *Size and slope*: chosen to run less than half full to let foam drain, and with least velocity to avoid turbulence.
- *Refinement*: perforated tray at exit point prevents aeration due to plunging

If the two 90° elbows and straight sections were removed from the tank internals, the return line design would be at the correct location, size and slope. This would prevent air entrained oil to circulate directly into pump suction line and is a straightforward modification. A perforated tray could be easily installed for further refinement if deemed necessary.

Suction Line

The pump suction lines currently have a 130 µm stainless steel strainer located at the bottom of the tank. A suction strainer is a coarse filter used to prevent large foreign particles entering the pump that would cause catastrophic failure. Neale (1995, p.C20.1) recommendations for tank suction lines:

- *Location*: remote from return line
- *Inlet depth*: between tank bottom and one-third of running level
- *Refinement*: foot-valve suction strainer to maintain fluid in pump and prevent air entering pump

The current suction line location and inlet depth is adequate as per recommended design. However the suction strainers do not have a built-in foot valve. The installation of a foot valve suction strainer is simple and inexpensive. However pressure losses through the foot valve must be carefully considered to avoid insufficient suction pressure to pump which may cause cavitation and/or inability to prime.

Baffle and Weirs

The current internal tank design does not have baffles and weirs. Baffles and weirs serve three primary purposes: (1) provide structural stiffening of tank walls, (2) inhibit sloshing in mobile systems and (3) prevent direct flow between return and suction (Neale 1995, p.C20.1). As the ID fan tank system does not require additional stiffening or reduced sloshing, the addition of baffles and weirs would only be applicable to prevent direct flow between return and suction. To install a series of baffles in the existing lubrication tank design will be very difficult and expensive due to tank configuration and large amount of workshop work required.

De-aeration Screen

The existing internal tank design does not have a de-aeration screen for the removal of entrained air in oil. De-aeration screens are typically 100 mesh (150 µm) and there sole purpose is to separate air from oil to prevent recirculation of air entrained oil (Neale 1995, p.C20.2). Correct design is to completely immerse the screen as surface foam can penetrate the finest screen.

The installation of a suitable de-aeration screen into the current tank design would require site dimensions, design drafting, fabrication and installation. Careful thought into the design is required so it can be installed without using welding and/or flame cutting to minimise cost and installation time. The design must be simple enough so all fabrication is done in a workshop and the only site installation work required is loosen/tighten of mechanical fasteners.

4.1.3 Simplify System Design

The existing throttle valves and adjustable orifices in the ID fan lubrication system are over-complicating the oil flow rate set-up process. Too simplify lubrication design and set-up process, many lubrication systems are designed with fixed-size orifices. The cross-sectional area of a fixed-size orifice is calculated so correct flow rate and constant system pressure is maintained. This avoids any human intervention which can lead to in-correct set-up during commissioning and on-going maintenance.

If the two adjustable orifices at the inlet to the induction motor bearings were replaced with precisely machined fixed-size orifices, the two throttle valves could be removed or made redundant and only used for fine tuning system flow rates. The cross-sectional area of the fixed-size orifices will need to be calculated so flow rate to induction motor DE and NDE bearings is 10 L/min and 5 L/min respectively. The remaining system flow rate will be directed to the fan main shaft bearings.

To minimise cost and simplify installation of fixed-size orifices, the design must permit the external dimensions of existing adjustable orifice. Therefore no pipe work modification will be required and site installation is straightforward.

4.1.4 Suitable Flow Transmitters

As a result of the Hedland Flow Transmitters removed from service in 2004, the flow rates to the induction motor bearings and fan main shaft bearings have been unknown. This poses a huge risk to in-correct system set-up and no flow rate monitoring for machine protection. The only method of continuously monitoring flow rate in the ID fan lubrication system is to install good quality and accurate flow transmitters. Thorough research and design must be considered when selecting suitable flow transmitters to avoid the failures which plagued the OEM Hedland Flow Transmitters.

There are many different styles and types of flow transmitters available in today's industry. Each has their advantages and disadvantages for specific applications throughout a wide range of industries. The most common models available are:

- Differential Pressure Flowmeter
- Variable Area Flowmeter
- Positive Displacement Flowmeter
- Turbine Flowmeter
- Vortex Flowmeter
- Electromagnetic Flowmeter
- Ultrasonic Flowmeter
- Coriolis Mass Flowmeter

To conduct thorough research and analysis into all eight different flow transmitter models would take many hours. To expedite this process to ensure project schedule is not delayed, a technical sales consultant from Yokogawa was contacted to provide assistance. Yokogawa is a well known company that provides an extensive range of industrial field instruments and provides much technical support to Callide Power C Station. After discussions with Yokogawa technical sales in Gladstone, it was determined the most suitable flow transmitter models for this application are: (1) variable area flowmeter, (2) positive displacement flowmeter and (3) coriolis mass flowmeter.

Variable Area Flowmeter

The variable area flowmeter is one of oldest and basic principles of flow measurement. A float is guided inside a conically (tapered) shaped tube. The float hovers at a point in the tube depending on rate of flow coming up through the tube, as well as liquid viscosity and density. The float rises within the tube as flow increases due to upward force acting on float (see Figure 4.1). However, the float rises to a widening area which means the clearance between the float and inside diameter of the tube will increase, hence the terminology “variable area flowmeter”.

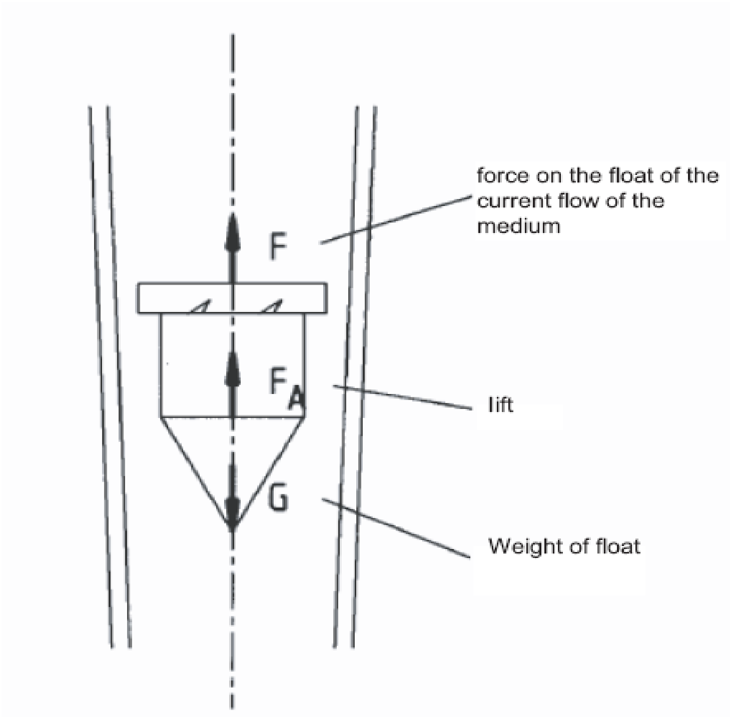


Figure 4.1: Variable area measurement principle

Table 4.2 – Advantages and disadvantages of variable area flowmeters

Advantages	Disadvantages
• Lowest cost flowmeter. • 2-wire loop powered transmitter, so no external power required. • Simple and reliable construction. • Suitable for slightly dirty liquids. • Suitable for pressures up to 200 bar and temperatures up to 400℃. • Simple on-going maintenance. • Low pressure losses across flowmeter.	• Accuracy between 1 – 2% of full scale reading. • Flowmeter is calibrated to a predetermined liquid viscosity and density, so any change will affect flowmeter accuracy. • Not suitable for high viscosity liquids. • Liquid can not contain large concentration of solids. • Must be installed in vertical position with liquid flowing upwards.

Figure 4.2: A typical Yokogawa variable area flow transmitter

Positive Displacement Flowmeter

This type of flowmeter measures total volumetric flow rate with great accuracy because exact quantities of liquids are conveyed between gear teeth. This flowmeter operates very similar to a gear pump with the exception being that the rotors (i.e. gears or ovals) are rotated by the conveyed liquid rather than by a drive shaft. The volume of 'free-space' occupied by the cavities created by the rotors coming in-and-out of mesh per revolution is fixed. Therefore, for every revolution of the rotor, a predetermined volume of liquid has passed through the flowmeter. Volumetric flow rate is then calculated by measuring revolutions of rotor per time period. Figure 4.3 depicts the operating principle of an Oval Gear Flowmeter.

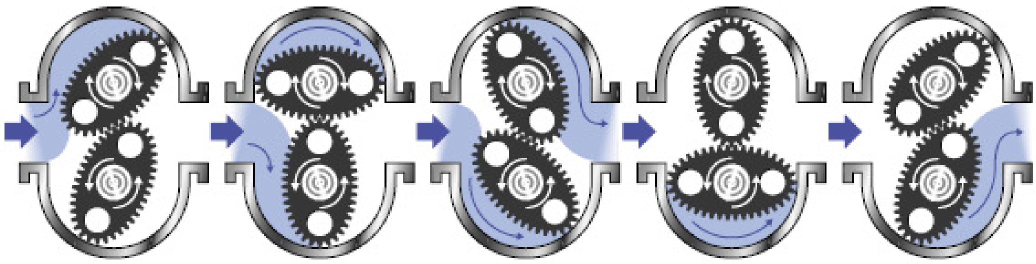


Figure 4.3: Operation principle of an oval gear flowmeter

Table 4.3 – Advantages and disadvantages of positive displacement flowmeters

Advantages	Disadvantages
• Extremely accurate flow measurement (0.1 – 1.0%). • Accuracy improves as liquid viscosity increases. • 2-wire loop powered transmitter, so no external power required. • Very reliable when used with fuels and oils as this lubricates rotors. • Very accurate for a wide range of liquid viscosities between 5 to 1000 cSt. • Suitable for pressures up to 200 bar • Can be installed in any direction as long as rotor shafts are horizontal.	• Can not be used with dirty liquids containing solid particles, due to wear and clogging. • Upstream strainers (typically 100 to 200 µm) required to filter solid particles. • Only suitable for temperature up to 120℃ because of tight clearance between rotors. • Moderate pressure losses across flowmeter. • Some makes and models may cause pulsating flow. • Rotating elements will eventually wear over time and will require replacement.

Figure 4.4: A typical Macnaught positive displacement flow transmitter

Coriolis Mass Flowmeter

The main feature of coriolis mass flowmeters is that the primary variable being measured is mass flow rate (kg/s) of the liquid rather than volumetric flow rate (m^3/s). They also have multi-variable sensors which allow them to measure temperature and density, and as a result volumetric flow rate can be calculated by the instrument on-board control system.

As shown in Figure 4.5, coriolis mass flowmeters have dual measurement tubes located side-by-side. When either a gas or liquid passes through the tubes electromagnetic forces cause resonant vibrations of the measurement tube. Coriolis forces acting on the medium flowing through the tubes will cause them to twist and therefore alter their resonant vibrations. The relationship between the resonant vibrations of the tubes and the small deviations caused by the Coriolis forces results in a small phase shift which is detected by two electromagnetic sensors. The corresponding phase shift is a measure of mass flow rate and the change in resonant frequency is a measure of fluid density.

When these variables are combined with modern digital technology, the Coriolis Mass principle provides unsurpassed accuracy for a wide range of parameters, such as flow rate, viscosity and density.

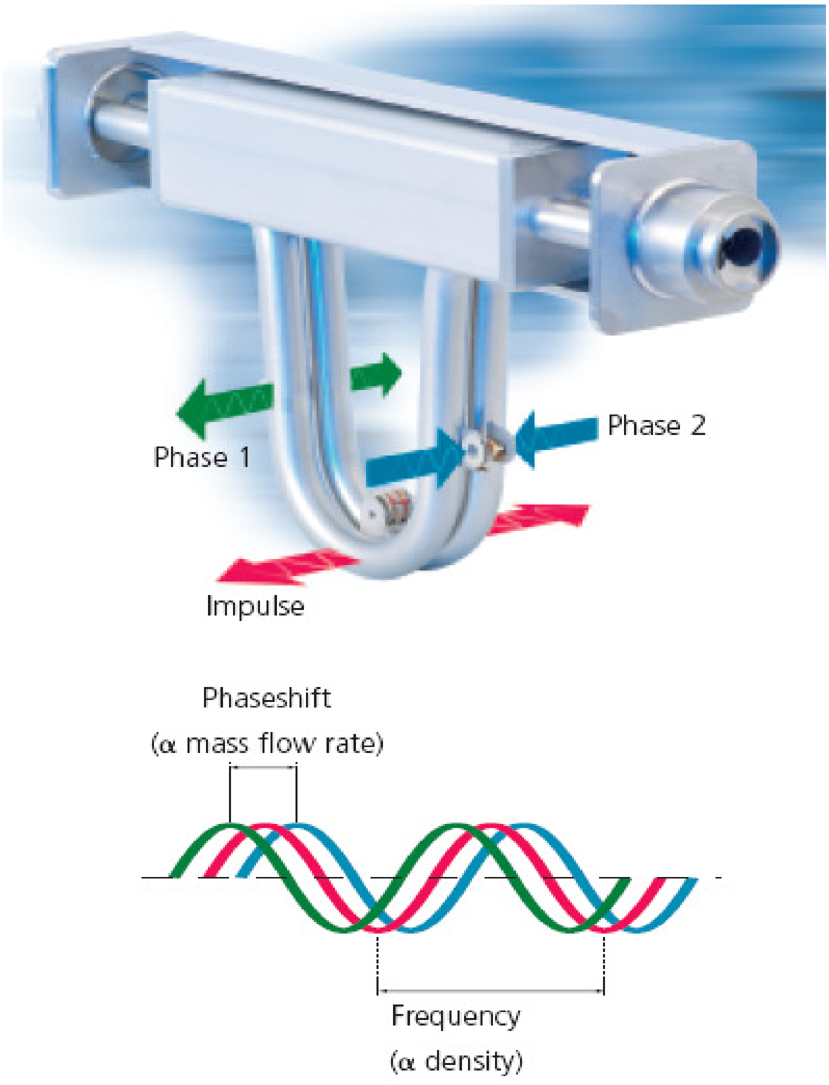


Figure 4.5: Coriolis mass operating principle

Table 4.4 – Advantages and disadvantages of coriolis mass flowmeters

Advantages	Disadvantages
• Extremely accurate of measuring mass flow (0.1 – 1.0%). • Mass flow measurement is independent of density and viscosity. • Suitable for pressures up to 250 bar and temperatures up to 350℃. • Not affected by flow profile, so flowmeter can be installed in any location and direction. • Suitable for both conductive and non-conductive liquids	• Very expensive. • 4-wire power loop and signal loop transmitters, therefore requires external AC or DC power supply. • Not suitable of very low and high temperatures. Typically limited to -50 to 350℃. • Very heavy compared to other flowmeter types. • Can be prone to blockages when used with dirty or highly viscous fluids.

Figure 4.6: A typical Yokogawa coriolis mass transmitter

4.1.5 Pressure Alarm and Trip Values

Due to in-correct pressure alarm and trip values which did not allow for fluctuating oil viscosity and system pressure, the Callide C ID fans were always going to cause operational issues, especially on hot summer days. There are two options which can be considered to correct this problem: (1) disable the alarm and trip values from the ICMS, or (2) use fluid mechanic equations to calculate correct values.

Disabling the pressure alarm and trip values from the ICMS would be a feasible idea only if accurate flow transmitters were to be reinstated with a back-up transmitter on each line for emergency redundancy. Currently, the only form of lubrication system monitoring is via the pressure transmitters and if this protection is removed, the last form of protection for the ID fans is bearing metal temperature. Having bearing metal temperature as the only form of equipment protection poses increased risk to the future reliability of the ID fans. Bearing metal temperature protection should only be the last form of protection to avoid major equipment failure and associated costs.

However, the other alternative of using fluid mechanic equations to calculate correct protection values can be easily achieved and implemented into the ICMS. The values will need to be calculated using “worst case scenario” parameters to account for maximum oil temperature and minimum system pressures.

Chapter 5 – Modification Design Proposal

5.1 Design Considerations

As discussed in Chapter 4, there is a wide variety of possible solutions for the Callide C ID fan lubrication systems. However carefully consideration must be sought when deciding which solutions are best to incorporate into the final design proposal. The key factors which must be considered are:

- Initial capital and on-going maintenance cost
- Simple and user-friendly design for future maintenance
- Easy installation methodology to minimise possible errors
- Do not introduce potential failure modes into the system

5.1.1 Filter Element Selection

To ensure lubrication oil cleanliness code is 20/18/15 or better, the only viable solution is to replace the existing filter elements from 25 to 10µm. This solution is very easy and inexpensive. Once all filters across all four ID fans have been replaced, the CS Energy stock material specification will require updating to incorporate the difference in filter fineness.

5.1.2 Internal Tank Modifications

As highlighted in Chapter 4, the internal tank design of the ID fan lubrication system suffers from incorrect design which is allowing air entrained oil to enter the pump suction strainer. The easiest and most cost effective way to correct this solution is to remove the two 90° elbows and DN50 mm pipe straight sections from the tank internals. This will require removing the tank lid via loosening the fasteners and cutting the pipe section at the tank running level.

A custom designed de-aeration screen will also provide further refinement to prevent recirculation of air entrained oil. The de-aeration screen design will incorporate the following factors:

- Localise oil turbulence
- Trap foam and floating contaminants
- 100 mesh (150 µm) stainless steel mesh
- Simple fabrication and installation design to minimise cost

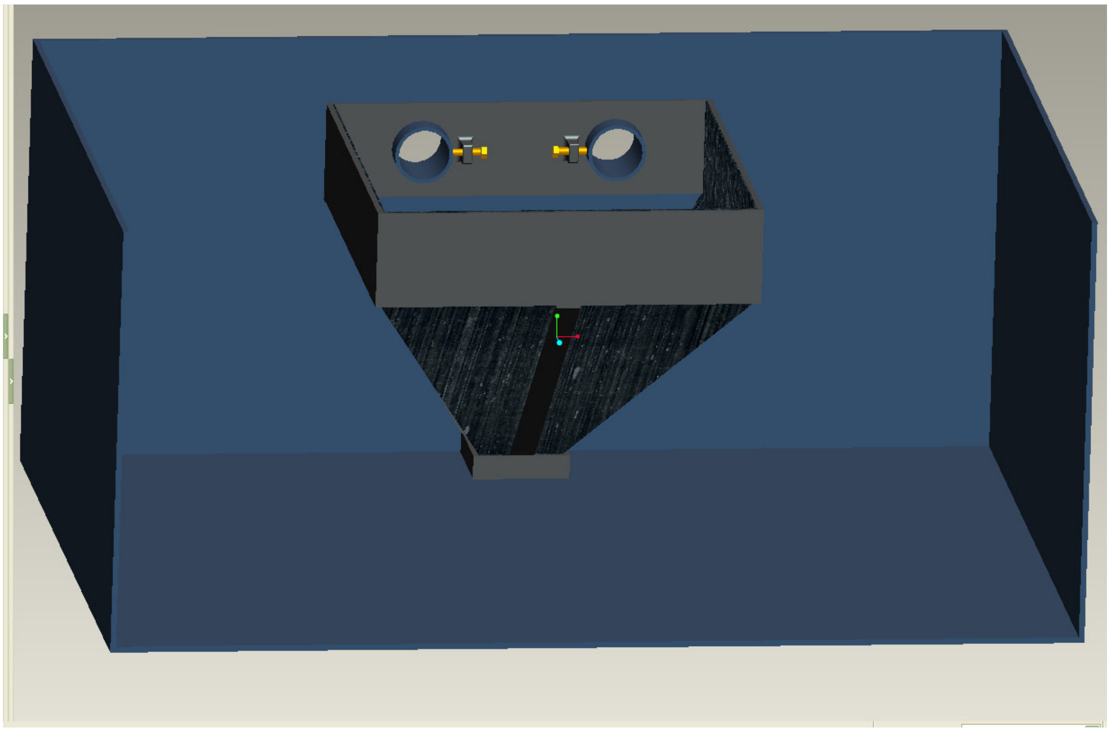


Figure 5.1: Proposed de-aeration screen design

As shown in Figure 5.1, the proposed de-aeration screen design will neatly fit into the existing tank internals. Once the two 90° elbows and DN50 mm pipe straight sections have been removed, the de-aeration screen will clamp directly to the return pipes via two M8 fasteners. All components will be manufactured from 316 stainless steel to prevent the introduction of another failure mode (i.e. corrosion). Refer to Appendix B for detailed engineering drawing.

5.1.3 Fixed-size Orifices

To simplify the current lubrication system design, the existing two throttle valves and two adjustable orifices must be replaced with fixed-size orifices. The system will require two fixed-size orifices to replace the existing adjustable orifices at the induction motor bearings (see Figure 5.4). This design will also allow the removal of the existing two throttle valves located inside the lubrication cabinet. To calculate correct cross-sectional area and diameter for the two fixed-size orifices, the following calculations are done.

Givens

Flow rate split to DE bearing is 10 L/min (2/3 of 15 L/min) and NDE bearing is 5 L/min (1/3 of 15 L/min)

Maximum inside diameter of existing adjustable orifice = 8.3 mm

Maximum cross-sectional area of adjustable orifice

$$A = \frac{\pi D^2}{4} = \frac{\pi \times 0.083^2}{4} = 5.41 \times 10^{-5} \ m^2$$

DE bearing fixed-size orifice cross-sectional area

$$A_1 = 5.41 \times 10^{-5} \frac{2}{3} = 3.6 \times 10^{-5} \ m^2$$

DE bearing fixed-size orifice inside diameter

$$D_1 = \sqrt{\frac{3.6 \times 10^{-5} \times 4}{\pi}} = 6.8 \ mm$$

NDE bearing fixed-size orifice cross-sectional area

$$A_2 = 5.41 \times 10^{-5} \frac{1}{3} = 1.8 \times 10^{-5} \ m^2$$

NDE bearing fixed-size orifice inside diameter

$$D_2 = \sqrt{\frac{1.8 \times 10^{-5} \times 4}{\pi}} = 4.8\ mm$$

This process gives D_1 = 6.8 mm and D_2 = 4.8 mm for the inside diameter of two fixed-size orifices. This will ensure a 2/3:1/3 ratio split of flow rate into the induction motor bearings at all times. The above calculations are based on existing adjustable orifice maximum diameter 8.3 mm to minimise pressure losses and additional heat. Refer to Appendix B for detailed engineering drawing of fixed-size orifices.

Note: When commissioning the lubrication system with the fixed-size orifices, the flow rates to the fan main shaft bearings and induction motors bearings must be confirmed to be within OEM specifications. If the flow rates are not correct, the above calculations must be repeated using smaller or larger orifice diameters. If the flow rate to induction motor bearings is too high, then fixed-size orifice diameter is to be reduced and vice-versa if flow rate is too low. A Matlab program (see Figure 5.3) has been developed to simplify this process and ensure 2/3:1/3 ratio split is always maintained.

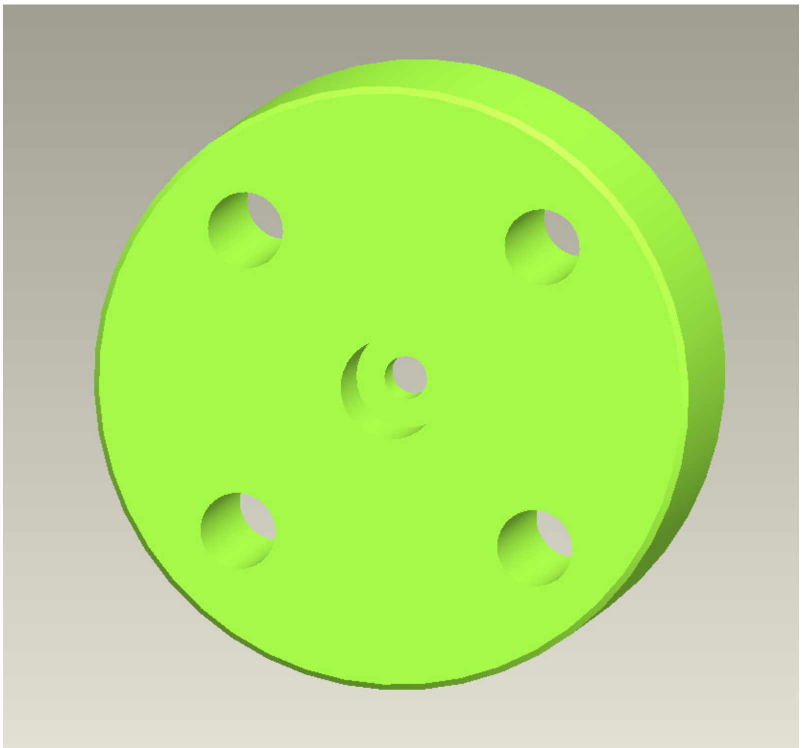


Figure 5.2: Fixed-size orifice 3D model

```
%Program: Fixed_Orifice
%Written by Stuart Baker
%Last revision date 08/09/09
%This program has been developed to calculated orifice diameters
%for both DE and NDE induction motor bearings
clc
clear

Orifice = inputdlg...
    ({'Select Nominal Inside Diameter of Induction Motor Orifice
(mm)'},...
    'Orifice Calculation Program',1,{'8.3'});

Input = str2num(Orifice{1,1});
Orifice_area = (pi*Input^2)/4;
DE_area = Orifice_area*(2/3);
NDE_area = Orifice_area*(1/3);
DE_dia = sqrt(DE_area*4/pi);
NDE_dia = sqrt(NDE_area*4/pi);

fprintf('\n');
fprintf('DE Fixed-size Orifice Diameter = %2.1f mm\n', DE_dia);
fprintf('\n');
fprintf('NDE Fixed-size Orifice Diameter = %2.1f mm\n', NDE_dia);
%EOF
```

Figure 5.3: Fixed-size orifice Matlab script

5.1.4 Accurate Flow Transmitters

The previous research conducted on suitable flow transmitters for the ID fan lubrication system highlighted two possible alternatives: (1) positive displacement flow transmitter and (2) coriolis mass transmitter. Both models offer excellent accuracy with varying oil viscosity, density and pressure. The decision of which model to choose will come down to purchase and installation cost.

The approximate cost of a suitable positive displacement flow transmitter for this particular application is $2,500. Whereas, the approximate cost of a suitable coriolis mass transmitter is much greater at $10,000. The coriolis mass flow transmitter will also require an external power source as it is a 4-wire configuration. Providing an external AC/DC power source will dramatically increase installation cost. However, the positive displacement flow transmitter is a 2-wire configuration and does not require an external power source. Given this information, the positive displacement flow transmitter is the most cost effective and reliable model for the ID fan lubrication system.

After discussion with Yokogawa technical sales in Gladstone, it was confirmed the best suited positive displacement flow transmitter for this application is a Macnaught M4 – ½" pulse and LC display flow meter, refer to Appendix C for specifications. This particular model is ideal for this application because:

- Compact size and easy installation
- Can be mounted horizontal or vertical
- Low pressure drop (15 to 20 kPa) across flow meter allows for economical pump performance and negligible additional heat
- Flow meter accuracy is ±0.5%
- Basic design to minimise the number of wearable and replacement parts

Table 5.1 – Comparison of Macnaught flow transmitter specifications

	Flow Transmitter Design	**Operation Conditions**
Kinematic Viscosity	5 to 1000 mm^s/s	20 to 100 mm^s/s
Volumetric Flow Rate	2 to 30 L/min	10 to 18 L/min
Operating Pressure	0 to 5.5 MPa	0 to 2 MPa
Max Operating Temperature	80℃	60℃
Recommended Filter Fineness	250 µm	25 µm

As shown in Table 5.1, the Macnaught flow transmitter design parameters are all within the operation conditions of ID fan lubrication system. To ensure flow transmitter design has sufficient redundancy for emergency operation, there will be two positive displacement flow transmitters on each line. By-pass lines will be included on each flow transmitter (see Figure 5.4) to allow repair and/or replacement of flow transmitter with the ID fan in-service. The original OEM values will be used in the ICMS for flow rate protection.

Table 5.2 – OEM flow rate protection values

Description	**Alarm**	**Start Condition**	**Trip**	**Start stand-by pump**
Fan Oil Flow	x < 12 L/min	x > 12 L/min	x1 < 10 L/min	x < 12 L/min
Motor Oil Flow	x < 12 L/min	x > 12 L/min	x1 < 10 L/min	x < 12 L/min

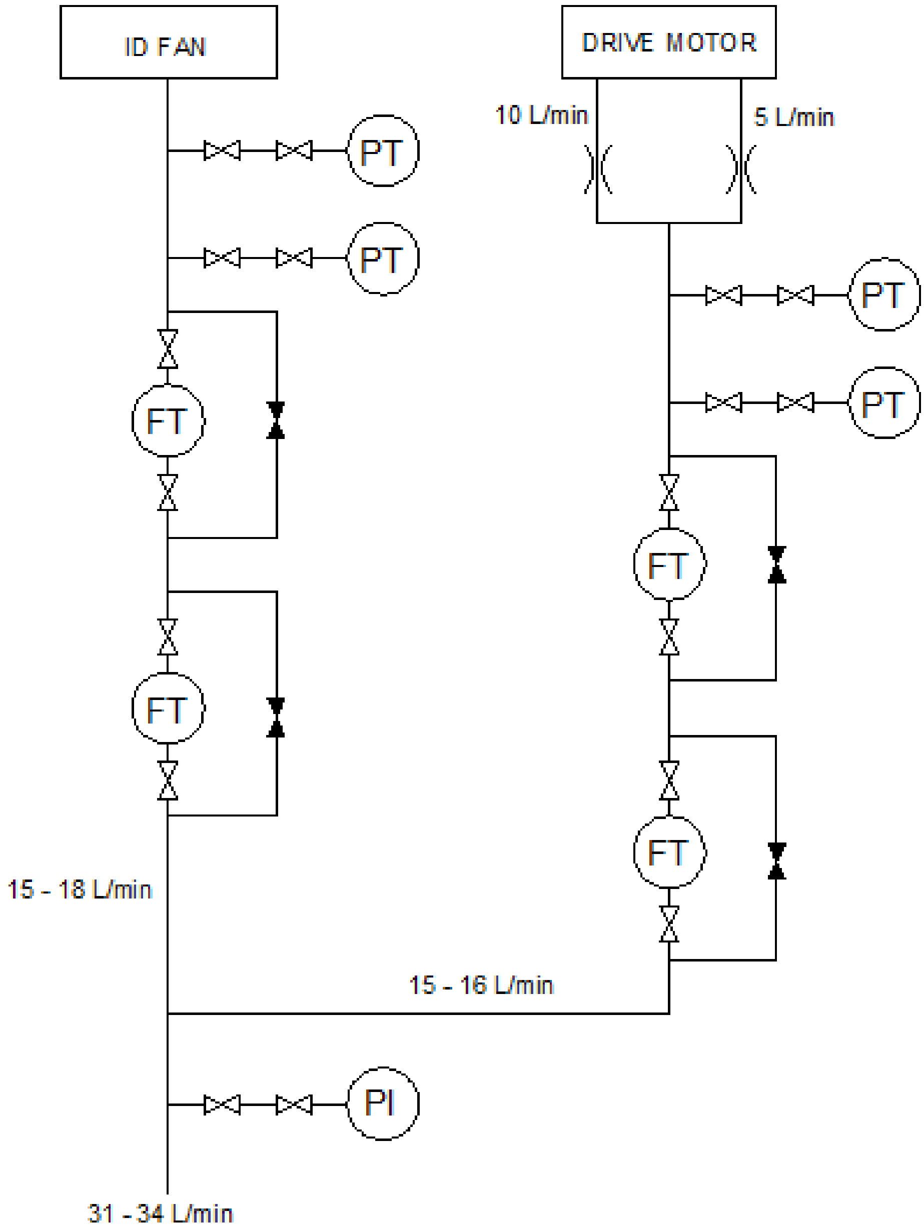


Figure 5.4: Modified lubrication system P&ID

5.1.5 Modified Pressure Alarm and Trip Values

To calculate new alarm and trip values for the ID fan lubrication system, head losses between the pressure transmitter and bearing inlet has to be determined for both induction motor and fan main shaft bearings. Calculations will be based on OEM flow rate conditions at maximum fluid temperature on 60℃. Table 5.3 illustrates the equipment protection conditions to the corresponding OEM flow rate conditions.

Table 5.3 – Alarm and trip conditions for head loss calculations

Equipment Protection Condition	**Flow Rate Condition**
Alarm	12 L/min → 0.0002 m^3/s
Start Condition	15 L/min → 0.00025 m^3/s
Trip	10 L/min → 0.0001667 m^3/s
Start stand-by pump	12 L/min → 0.0002 m^3/s

The fluid mechanic equations used to calculate head loss for each condition is from Fox, McDonald and Pritchard (2003).

Pipe cross-sectional area

$$A = \frac{\pi D^2}{4} = [m^2]$$

Fluid velocity

$$\overline{V} = \frac{\dot{V}}{A} = [m/s]$$

Reynolds number

$$\mathrm{Re} = \frac{\overline{V}D}{v}$$

Laminar Flow Reynolds Number: Re < 2300

Friction Factor for laminar flow

$$f = \frac{64}{\mathrm{Re}}$$

Major losses

$$h_1 = f\,\frac{L}{D}\,\frac{\overline{V}^2}{2} = [m]$$

Minor losses

$$h_{1m} = K\,\frac{\overline{V}^2}{2} = [m]$$

where standard radius elbow K factor = 0.6
Screwed socket connection K factor = 0.03

Potential head

$$h = [m]$$

Total head loss

$$H = h_1 + h_{1m} + h = [m]$$

Convert to pressure loss

$$p = \rho g h = [kPa]$$

The new alarm and trip values for each condition are determined using the above fluid mechanic equations with Matlab software. The Matlab script is shown in Appendix D.

Fan Main Shaft Bearings

Givens

Pipe length = 8 m

Pipe inside diameter = 15.8 mm → 0.0158 m

Standard radius elbow = 7

Screwed socket connection = 2

Kinematic viscosity @ 60℃ = 2 x 10^{-5} m^2/s (worst case scenario)

Table 5.4 – Fan main shaft bearing protection values

Description	**Alarm**	**Start Condition**	**Trip**	**Start stand-by pump**
Proposed values using fluid mechanic equations				
Fan Oil Pressure	x < 140 kPa	x1 > 150 kPa	x2 < 130 kPa	x < 140 kPa
2004 modification values				
Fan Oil Pressure	x < 200 kPa	x1 > 250 kPa	x2 < 150 kPa	x1 < 250 kPa

Induction Motor Bearings

Givens

Pipe length = 6 m

Pipe inside diameter = 15.8 mm →0.0158 m

Short radius elbow = 2

Screwed socket connections = 2

Kinematic viscosity @ 60°C = 2×10^{-5} m^2/s (worst case scenario)

Table 5.5 – Induction motor bearing protection values

Description	**Alarm**	**Start Condition**	**Trip**	**Start stand-by pump**
Proposed values using fluid mechanic equations				
Fan Oil Pressure	x < 150 kPa	x1 > 190 kPa	x2 < 130 kPa	x < 150 kPa
2004 modification values				
Fan Oil Pressure	x < 105 kPa	x1 > 130 kPa	x2 < 80 kPa	x1 <130 kPa

As clearly illustrated in Tables 5.4 and 5.5, the pressure protection values implemented in 2004 are vastly different from the calculated values. This highlights the importance of engineering input when selecting the correct pressure protection values. The proposed values also agree with the current operating pressures of the ID fan lubrication system. This is shown in Table 5.6 which compares the operating pressures against the proposed protection pressure values.

Table 5.6 – Comparison of pressure protection values

Condition	Fan Main Shaft Bearing Pressure (kPa)	Induction Motor Bearing Pressure (kPa)
Ambient oil temperature 25℃	x ≈ 400	x ≈ 450
Maximum oil temperature 60℃	x ≈ 230	x ≈ 250
Alarm	x < 140	x < 150
Start Condition	x1 > 150	x1 > 190
Trip	x2 < 130	x2 < 130
Start stand-by pump	x < 140	x < 150

5.2 Modification Design Overview

In summary, the proposed modification design for Callide C ID fan lubrication system consists of the five following components:

1. Change the existing filter fibre element from 25 to 10 µm with a beta efficiency ratio (β_{10}) = 100 (99%).
2. Modify the internal tank design by removing the existing two return lines and installing a de-aeration screen.
3. Simplify the lubrication piping system by removing the existing throttle valves and adjustable orifices. Then install a fixed-size orifice at the induction motor DE and NDE bearing.
4. Install two Macnaught M4 – ½" pulse and LC display flow meters on each pipeline with a by-pass line for each flowmeter and use the existing OEM flow rate protection values.
5. Amend the current pressure protection values in the ICMS with the values calculated using fluid mechanic equations.

Chapter 6 – Conclusion

6.1 Project Outcomes

The first part of this project was to conduct an engineering appraisal on the operational and reliability issues in the existing Callide C Power Station ID fan lubrication system. This appraisal highlighted the importance of proper bearing and lubrication selection in heavy rotating equipment. It also showed that the current ID fan bearing and lubrication selection is correct for this particular application. However, the design of the lubrication system is suffering from several failure modes which has plagued the ID fans from initial commissioning in 2001.

Using the Apollo RCA process, five primary causes and their associated failure modes were identified. Correct identification of root causes is the most crucial stage during an RCA because research into all possible design solutions can be then investigated. This project has highlighted the ability of RCA as a tool for solving the not-so-obvious re-occurring failures in the ID fan lubrication system.

The second part of this project was to research and evaluate effective design solutions. Once all solutions were identified, the advantages and disadvantages for each solution were researched thoroughly to ensure final design is both practical and cost effective for CS Energy. As discussed in Chapter 5, the proposed modification design consists of five components, all of which play a major role into preventing the current operational and reliability issues.

6.2 Further Work

The main aim of this project is the recommendation of a modification design for the ID fan lubrication system at Callide C Power Station. The next stage of implementation, which is outside the scope of this project, will require further financial justification, planning and installation.

To proceed forward with obtaining capital funds for this project, a "Project Concept Statement" must be prepared by the technical engineering department. Once the capital funds are approved, the next step will be to assign the project to a suitably skilled Project Manager, which will most likely be a Mechanical Engineer. The responsibility of the Project Manager will be to arrange procurement of necessary parts and then plan/schedule the installation using a combination of internal and external resources. Because the ID fans are required at times for maximum plant availability, the installation of the modification design can only occur during planned overhauls.

Once the installation and commissioning of the proposed modification design is complete, the final stage will be to revise and update the existing plant and operation manuals, and existing mechanical and electrical engineering drawings. Spare parts for the Macnaught flow transmitter will also need to be catalogued as stock items along with the development of preventive maintenance routines for future overhauls.

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Appendix A – Project Specification

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG4111/4112 Research Project
PROJECT SPECIFICATION

FOR: Stuart Baker

TOPIC: INDUCED DRAFT (ID) FAN LUBRICATION SYSTEM DESIGN REVIEW AND PROPOSED MODIFICATION UPGRADE AT CALLIDE C POWER STATION

SUPERVISOR: Mr Bob Fulcher

SPONSORSHIP: CS Energy, Callide Power Station, Biloela

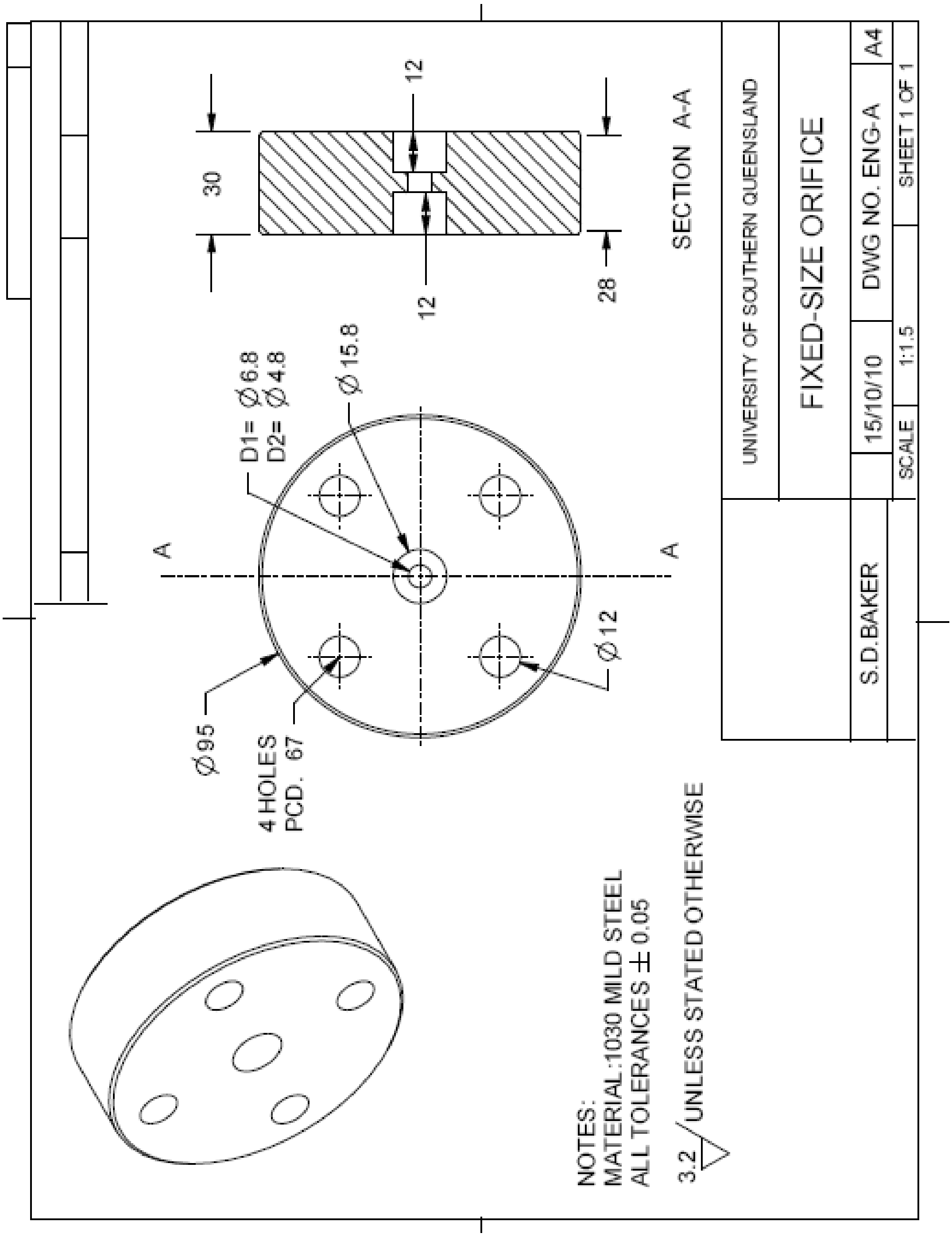
PROJECT AIM: To conduct an engineering appraisal on the operational and reliability issues in the existing ID fan lubrication system, and recommend a cost effective and reliable design for a future modification project.

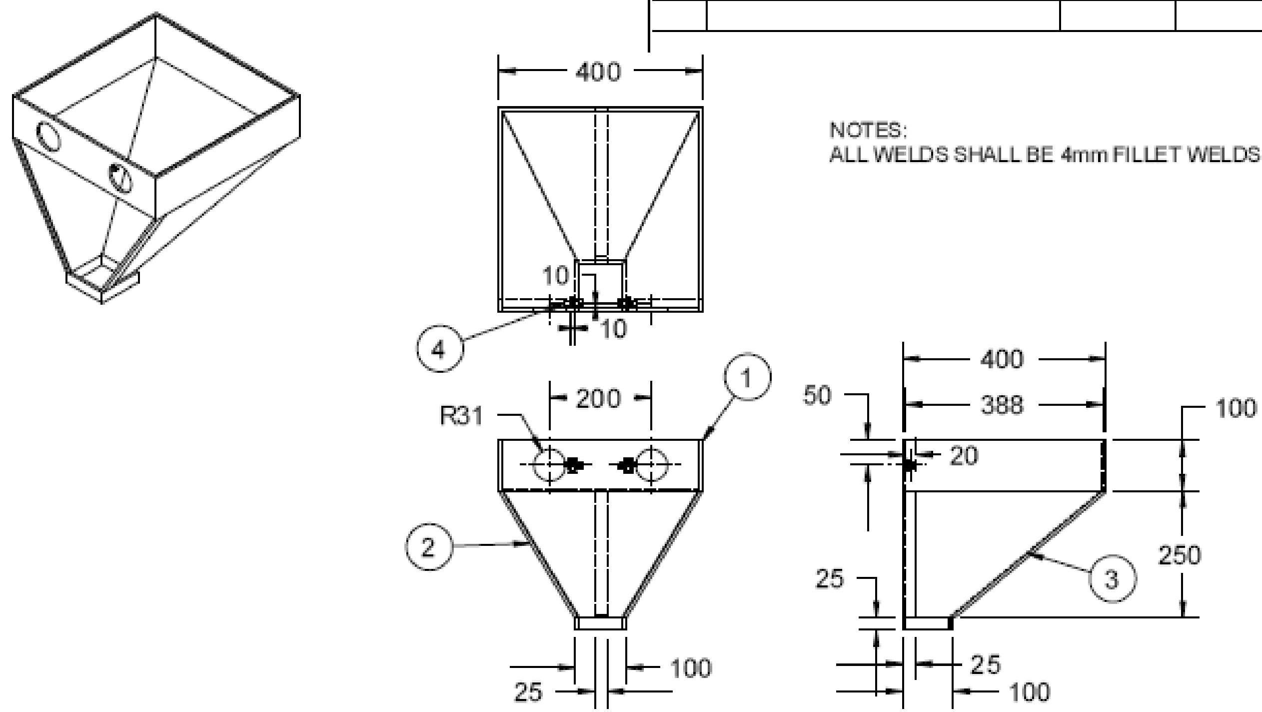
PROGRAMME: Version 1, 22nd March 2010

1. To describe the general operation of Callide C Power Station with special attention to boiler section pertaining to induced draft (ID) fan operations.
2. Research theory on lubrication and bearing systems in rotating equipment.
3. Investigate further information on the different types of bearings and lubrication used in heavy electric drives and axial fans.
4. Identify the operational and reliability failure modes in the existing ID fan lubrication system which is causing protection alarms and trips.
5. Research and evaluate effective design solutions to prevent reoccurring operational and reliability issues in the existing ID fan lubrication system.
6. Prepare a modification design recommendation that will make certain the lubrication system is designed for correct functionality and long-term reliability.
7. Submit an academic dissertation on the engineering research conducted and proposed design modification to the ID fan lubrication system.

AGREED S Baker (Student) R. Fulcher (Supervisor)
Date: 15 / 09 /2010 Date: 15 / 9 /2010

Appendix B – Engineering Design Drawings





NOTES:
ALL WELDS SHALL BE 4mm FILLET WELDS

PARTS LIST TABLE

ITEM	DESCRIPTION	MATERIAL
1	100x6 FLAT	316 SS
2	25x6 FLAT	316 SS
3	100 MESH SCREEN	316 SS
4	M8x30 BOLT	316 SS

UNIVERSITY OF SOUTHERN QUEENSLAND

DE-AERATION SCREEN

S.D.BAKER | 16/10/10 | DWG NO. ENG-B | A4

SCALE 1:10 | SHEET 1 OF 1

Appendix C – Flow Meter Specifications

M4 – 1/2" pulse and LC display meters

The M4 is a low to medium flow range model. It is a compact meter that has the ability to handle a wide range of fluid viscosities with exceptional levels of repeatability and durability.

Features

- Compact size.
- Two independent pulse units.
- Flexibility of installation options (e.g. can be mounted horizontally or vertically; no flow conditioning required).
- Low pressure drop allows for economical pump selection or gravity flow applications.
- Meter construction enables fast and easy on-site servicing without removal from application.
- Meter design minimises the number of wearable and replaceable parts and extends product life.
- Meter accuracy is verified by a factory calibration check after which an individual metrology report is issued.
- All LC Displays meet European CE directive for EMC.
- Display/Pulse Version has IP65/NEMA9 protection.
- Intrinsically safe LC Deluxe & Standard Displays. Certificate of conformity number PTB Nr. Ex-93.C.4033x & KEMA 05ATEX1168X

Specification:

Model	M4
Meter Type	Pulse / Standard LC Display / Deluxe LC Display/MR100 Display
Meter Body Material	Aluminium / 316 Stainless Steel / Bronze
Wetted Components:	
Rotor Material	PPS / 316 Stainless Steel
Shafts	316 Stainless Steel
O'ring	NBR (Nitrile)
Flow Rate Ranges (Litres Per Minute/US Gallons Per Minute)	
Above 5 cPs	2 To 30 / 0.5 To 8
Below 5 cPs	3 To 25 / 0.8 To 6.60
Accuracy- Within (Of Reading)	+/- 0.5%
Repeatability	0.03%
Maximum Viscosity (Of Standard Model)	1000 Centipoise (Optional Hi Viscosity Rotors)
Maximum Operating Pressure	5500kpa/ 800 Psi/ 55 Bar
Pulser Type	Hall Effect or Reed Switch or Combination HE / RS
Pulses Per Litre/Us Gallon	112/424
Max Operating Temperature	80°C / 176°F, High Temp Option 120°C / 248°F
Recommended Mesh Strainer Size	60 Mesh

*Note: For doubled pulse output or SAA flame proof certification use M5 series models (contact for more info)

M4ARP-X

Standard LC Display

Deluxe LC Display

MR100

Port Size

To order flowmeter you must replace 'X' with the relevant number. This number will determine the following specifications:

Port Size:	Calibrated In:	Electrical Connections
1 = 1/2" BSP (F) ports	Litres	20mm (F) Conduit Thread
2 = 1/2" NPT (F) ports	US Gallons	1/2" NPT

PPS = Polyphenylene Sulfide Resins

M4 – 1/2″ pulse and LC display meters

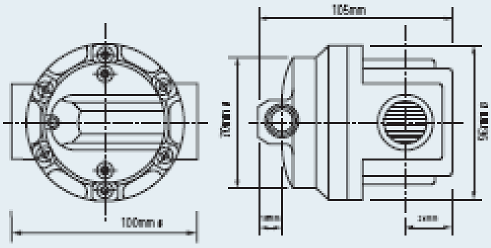
Options and accessories

M4 - 1/2" PULSE AND LC DISPLAY METERS	M4ARX-X	M4ASX-X	M4SRX-X	M4SSX-X	M4BRX-X
FKM O-Ring	○	○	○	○	○
FEP O-Ring	○	○	○	○	○
EPDM O-Ring	○	○	○	○	○
High Temp Rotors	○	●	○	●	○
High Viscosity Rotors	○	○	○	○	○
Hall Effect Sensor	○	○	○	○	○
Reed Switch	●	●	●	●	●
Solvent Kit	–	–	–	–	–
Hastalloy C Shafts	–	–	–	–	–
Protective Boot Deluxe LCD	–	+	+	+	○
Protective Boot Standard Display	–	+	+	+	○
Remote Mounted LC Display	+	+	+	+	+
Heating Jacket	–	–	–	–	–
4-20mA Module (Meter or Remote Mount)	+	+	+	+	+

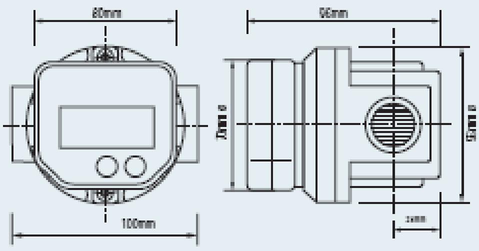
● Standard ○ Optional – Not Available + Accessory

Dimensions

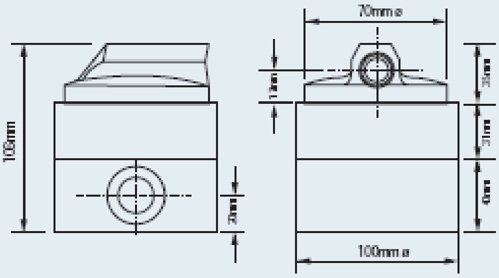
Standard display H 122mm x W 137mm.
Deluxe display H 122mm x W 137mm.



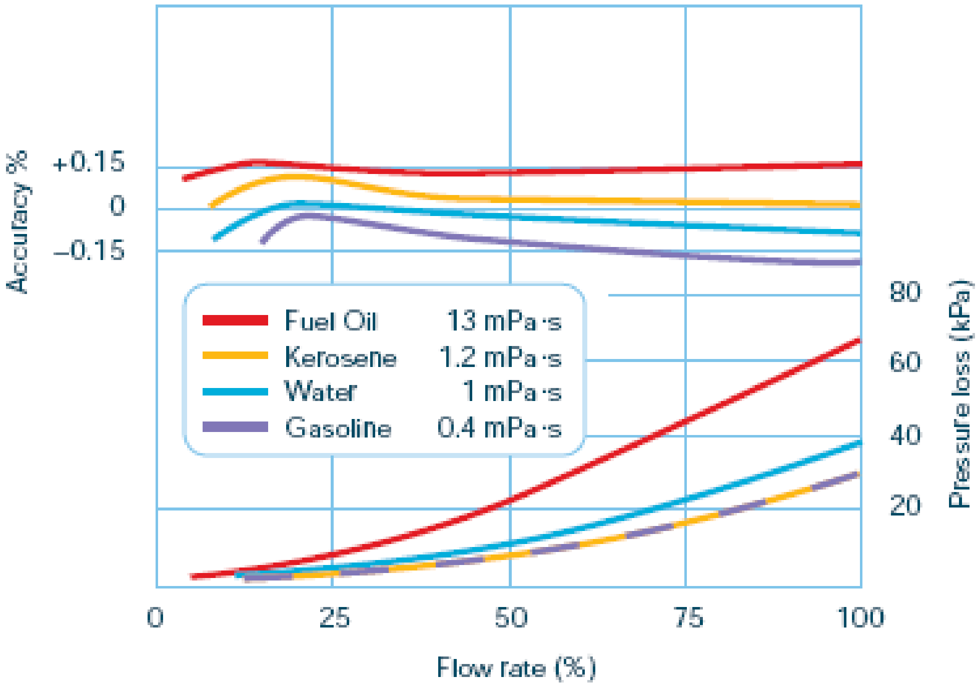
MR100 display H 70mm x W 80mm.



M4SSPH high pressure version



Meter sizes M4, M10 & M40



Appendix D – Matlab Programs

```
%Program: Fan_Protection_Values
%Written by Stuart Baker
%Last revision date 20/10/09

%This program has been developed to calculate pressure protection
%values for the ID fan main shaft bearings at three different
%flow rates (10, 12 and 15 L/min).
clc
clear
%Givens
Flow = [0.0001667,0.0002,0.00025];
ID = 0.0266;
L = 8;
K_Vis = 0.00002;
Elbows = 7;
Sockets = 2;
Elbow_K = 0.6;
Socket_K = 0.03;

%Pipe cross-sectional area
A = pi*ID^2/4;
%Fluid Velocity
V = Flow/A;
%Reynolds number
Re = V*ID/K_Vis;
%Determine if flow is laminar or turbulent
if (Re < 2300);
    fprintf('Flow is Laminar\n');
else
    fprintf('Flow is Turbulent\n');
end
%Friction Factor
f = 64./Re;
%Major Losses
h1 = f*(L/ID).*(V.^2/2);
%Minor Losses
h1m = (Elbows*Elbow_K + Sockets*Socket_K).*(V.^2/2);
%Potential head
h = 2;
%Total head
H = h1 + h1m + h;
%Convert to pressure loss (kPa) plus 100 kPa
p = (846*9.81*H./1000)+100;
%Results
fprintf('\n');
fprintf('Pressure loss at 10 L/min is: %3.1f kPa.\n',p(1));
fprintf('\n');
fprintf('Pressure loss at 12 L/min is: %3.1f kPa.\n',p(2));
fprintf('\n');
fprintf('Pressure loss at 15 L/min is: %3.1f kPa.\n',p(3));
%EOF
```

```
%Program: Motor_Protection_Values
%Written by Stuart Baker
%Last revision date 20/10/09

%This program has been developed to calculate pressure protection
%values for the ID fan induction motor bearings at three different
%flow rates (10, 12 and 15 L/min).
clc
clear
%Givens
Flow = [0.0001667,0.0002,0.00025];
ID = 0.0158;
L = 6;
K_Vis = 0.00002;
Elbows = 2;
Sockets = 2;
Elbow_K = 0.6;
Socket_K = 0.03;

%Pipe cross-sectional area
A = pi*ID^2/4;
%Fluid Velocity
V = Flow/A;
%Reynolds number
Re = V*ID/K_Vis;
%Determine if flow is laminar or turbulent
if (Re < 2300);
    fprintf('Flow is Laminar\n');
else
    fprintf('Flow is Turbulent\n');
end
%Friction Factor
f = 64./Re;
%Major Losses
h1 = f*(L/ID).*(V.^2/2);
%Minor Losses
h1m = (Elbows*Elbow_K + Sockets*Socket_K).*(V.^2/2);
%Potential head
h = 2;
%Total head
H = h1 + h1m + h;
%Convert to pressure loss (kPa)
p = (846*9.81*H./1000);
%Results
fprintf('\n');
fprintf('Pressure loss at 10 L/min is: %3.1f kPa.\n',p(1));
fprintf('\n');
fprintf('Pressure loss at 12 L/min is: %3.1f kPa.\n',p(2));
fprintf('\n');
fprintf('Pressure loss at 15 L/min is: %3.1f kPa.\n',p(3));
%EOF
```