**Sudan University of Science and Technology**
**College of Post Graduate Studies**
**Mechanical engineering department (power)**

# Effect of Roseires Dam Heightening on Thrust Bearing of the Power Generation Units

**أثر تعلية خزان الرصيرص على كراسي إسناد وحدات محطة توليد الكهربا**

**A Thesis for Submitted in Partial Fulfillment of Degree of M.Sc. in Mechanical Engineering (Power)**

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أَعُوذُ بِاللَّهِ مِنَ الشَّيْطَانِ الرَّجِيمِ

**بِسْمِ اللّهِ الرَّحْمَنِ الرَّحِيمِ**

**{اقْرَأْ بِاسْمِ رَبِّكَ الَّذِي خَلَقَ * خَلَقَ الإِنْسَانَ مِنْ عَلَقٍ * اقْرَأْ وَرَبُّكَ الأَكْرَمُ * الَّذِي عَلَّمَ بِالْقَلَمِ * عَلَّمَ الإِنْسَانَ مَا لَمْ يَعْلَمْ}**

[العلق: 1-5].

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Any work depends on the efforts of many different people and this research is no exception.

Firstly, Praise be to God, who made his grace deeds.

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# Dedications

To the most important persons in my life
My parents
My wife
And
My children

# Abstract:

After more than forty years without any apparent problems on their combined guide and thrust bearings, the seven 40 MW hydro-generators of ROSERIS Hydroelectric Power Plant were failing. The source of the failure (which happened after heightening the dam from 40 m to 50 m) was the over heat of the thrust bearing pads. This caused considerable losses of power with high costs. The solution proposed by the operation engineers was operating the thrust bearing high pressure pumps (In case of any problems with hydrodynamic lubrication- insufficient load carrying capacity- the hydrostatic jacking system been used permanently in order to increase reliability of the bearing). The direct cost was estimated to be SD 5,000,000,000 (Pumps operating and power losses).

# المستخلص:


بعد أكثر من أربعين عاما من العمل المتواصل بدون مشاكل ظاهرة في كراسي الاسناد والتحميل، في السبعة وحدات التي تولد 40 ميغاواط لمحطة توليد كهرباء الرصيرص، بدأت المشاكل في الظهور بعد الانتهاء من عمليات التعلية للسد من 40 مترا الى 50 مترا.

المشكلة الأساسية التي ظهرت كانت زيادة الحرارة في كرسي التحميل.

الحل الوحيد الذي توصل اليه مهندسو التشغيل كان هو تشغيل مضخات الضغط العالي الخاصة بكراسي التحميل لزيادة قدرة الكرسي على تحمل الاحمال الاضافية الناتجة من التعلية.

هذه الظروف الجديدة أدت الى فقد مقدر في الطاقة المنتجة مع فقد مادي كبير نتيجة لإنقاص انتاج الوحدات عند ارتفاع درجة الحرارة إضافة الى تكلفة تشغيل المضخات. التكلفة المادية بلغت حوالي خمسة مليار جنيه سوداني سنويا.

# Table of Contents

**Acknowledgement** ……………………………………………………ii
**Dedications** ……………………………………………………...iii
**Abstract**: ………………………………………………………………iv
المستخلص: ……………………………………………………………………v
**Table of Contents** …………………………………………………vi
**List of Figures** …………………………………………………..x
**List of Tables** ………………………………………………………xii
**Symbols and abbreviations** …………………………………………xiii
**Greek symbols** ……………………………………………………xiv
**Dimensionless Equations** ………………………………………..xv
1 **INTRODUCTION** ……………………………………………1
1.1 BACKGROUND ……………………………………………..1
1.1.1 Thrust Bearings ........................................................ 1
1.1.2 Hydropower Turbine................................................ 2
1.1.3 Friction, Lubrication and Their Impact in Hydropower Applications ……………………………………………………….2
1.2 IMPORTANCE OF THE STUDY.................................................. 3
1.3 PROBLEM STATEMENT ……………………………………3
1.4 OBJECTIVES OF THE PROJECT ................................................ 4
1.5 RESEARCH METHODOLOGY .................................................... 4
1.6 OUTLINES OF THE RESEARCH ................................................ 4
2 **LUBRICANTS** ……………………………………………………..5
2.1 INTRODUCTION ……………………………………………..5
2.2 LUBRICATION REGIMES ……………………………..5
2.2.1 Hydrodynamic Lubrication ....................................... 6
2.2.2 Elasto hydrodynamic Lubrication ............................. 6
2.2.3 Boundary Lubrication .............................................. 6
2.2.4 Partial Lubrication.................................................... 6
2.3 Stribeck Curve ……………………………………………………6
2.4 OIL VISCOSITY ………………………………………………...7
2.4.1 Dynamic Viscosity .................................................... 8

2.4.2 Kinematic Viscosity ................................................ 8
2.4.3 Viscosity Temperature Relationship ........................... 9
2.4.3.1 Viscosity-Temperature Equations ............. 9
2.4.3.2 Viscosity-Temperature Chart .................. 10
**3 HYDRODYNAMIC THRUST BEARINGS ……………..…….12**
3.1 INTRODUCTION ……………………………………………..12
3.2 THRUST BEARINGS ………………………………………..12
3.3 LARGE THRUST BEARING DESIGN ....................................... 13
3.3.1 Bearing Housing Arrangement ................................. 14
3.3.1.1 Support systems ........................................ 15
3.3.1.2 Plate spring design ..................................... 16
3.3.1.3 Spindle support design ............................. 17
3.3.1.4 Spring mattress design ............................. 17
3.3.1.5 Mattress of rubber springs 18
3.3.1.6 Membrane support .................................... 19
3.3.2 Auxiliaries .................................................................. 20
3.3.2.1 Hydrostatic jacking system 20
3.3.2.2 Cooling systems ........................................ 21
3.3.2.3 Oil tanks and control of the oil flow 23
3.3.2.4 Rotating hole ............................................ 24
3.3.2.5 Pumping plate ........................................... 25
3.3.2.6 Self-pumping pads .................................... 25
3.3.2.7 External pumps ......................................... 26
3.3.2.8 Filters and quality of the oil ........... 27
3.4 THRUSTBEARING DESIGN CALCULATIONS ......................... 28
3.4.1 Hydrodynamic lubrication ........................................ 28
3.4.2 Navier-Stokes Equation: ........................................... 28
3.4.3 Continuity Equation (For Incompressible Flow): ..... 29
3.4.4 Energy Equation: ...................................................... 29
3.4.5 Heat Transport Equation: .......................................... 29
3.4.6 State Equation: ......................................................... 29
3.4.7 Generalized Reynolds Equation ................................ 30
3.5 CURRENTLY MODELED EFFECTS .......................................... 31
3.5.1 Steady State Calculation ........................................... 31
3.5.2 Heat Transfer Effects ................................................ 32
3.5.3 Warm Oil Carryover Effect ....................................... 35
3.6 DEFORMATIONS OF PAD AND RUNNER ................................ 39

4 **DAM HEIGHTENING EFFECT ON STATION THRUST BEARING ……………………………………………………… 43**
4.1 INTRODUCTION……………………………………………….. 43
4.2 GENERAL THRUST BEARING THEORY. ...............................43
4.3 VISCOUS FLOW:………………………………………………44
4.4 PHYSICAL WEDGE TERM ..................................................45
4.5 ANALYTICAL SOLUTION: ………………………………… 46
4.5.1 Assumptions:.........................................................46
4.5.2 Shear Forces per Unit Width: .................................48
4.5.3 Static Equilibrium: ..................................................48
4.5.4 Power Loss for One Pad Is: ....................................48
4.6 FIXED–INCLINE SLIDER BEARING: .......................................49
4.6.1 Dimensionless Terms:............................................49
4.6.2 Pressure Distribution:.............................................49
4.6.3 Dimensional: .........................................................50
4.6.4 Normal Load Component: ......................................50
4.6.5 Shear Force Component:.........................................50
4.6.6 Friction Coefficient:...............................................51
4.6.7 Power Loss and Temperature Rise: .........................51
4.7 THRUSTBEARINGOVERHEATING IN ROSEIRIS POWER PLANT…………………………………………………………...53
4.7.1 Statement of The Problems ......................................53
4.7.2 ABB Design Calculations ........................................53
4.7.3 Case Data: ..............................................................57
4.7.4 Machine Used Oil Specifications: ............................60
4.8 COMPUTATIONAL DATA:.. ....................................................61
4.8.1 Calculations............................................................62
4.8.1.1 Before Heightening ................................62
4.8.1.2 After Heightening...................................62
5 **RESULTS AND DISCUSSION ……………………………………**63
5.1 RESULTS:…………………………………………………...63
5.2 DISCUSSION……………………………………………………..64
6 **CONCLUSIONAND RECOMMENDATIONS…………………**67
6.1.1 CONCLUSION:.....................................................67
6.1.2 Recommendations:.................................................68

**REFERENCES…………………………………………………………...**69

# List of Figures

Figure 2-1 Hersey Number ........................................................................ 7
Figure 2-2Viscosity-temperature characteristics 'ASTM slope'..................11
Figure 3-1Thrust bearing of a large hydro generator unit (ALSTOM Hydro) ..............................................................................................................13
Figure 3-2Typical combined thrust-guide bearing housing of a large hydro generator (ALSTOM Hydro) ......................................................................14
Figure 3-3Plate spring (ALSTOM Hydro) ..................................................16
Figure 3-4Spindle support (ALSTOM Hydro)............................................17
Figure 3-5spring bed support system..........................................................18
Figure 3-6Mattress of rubber springs support system ................................18
Figure 3-7Membrane support, principle of operation (ALSTOM Hydro)..19
Figure 3-8Membrane support (ALSTOM Hydro).......................................19
Figure 3-9Hydrostatic jacking system operation (ALSTOM Hydro) .........20
Figure 3-10Internal water cooler in the bearing housing (ALSTOM Hydro) ..............................................................................................................21
Figure 3-11External water cooler ...............................................................22
Figure 3-12External oil tank and coolers ....................................................24
Figure 3-13Rotating hole design .................................................................24
Figure 3-14Pump plate between thrust pads design and flow (ALSTOM Hydro) ........................................................................................................25
Figure 3-15Self-pumping pad principle (ALSTOM Hydro).......................26
Figure 3-16Hydrodynamic oil film coordinates system..............................28
Figure 3-17Velocity boundary layer thickness............................................33
Figure 3-18Turbulent velocity boundary layer for different runner radii ...34
Figure 3-19Warm oil carry over phenomena ..............................................35
Figure 3-20Warm oil carryover factor.........................................................36
Figure 3-21Oil injecting on the runner sliding surface, CFD simulation....37
Figure 3-22the oil flow (330 and 660 l/min) on the oil film inlet temperature distribution ...........................................................................37
Figure 3-23Graphical representation of the possible deformations of the bearing elements..........................................................................................39
Figure 3-24Scale factor influence on the bearing load carrying capacity...40

Figure 3-25Influence of the relative pad deformations on the load carrying capacity of the centrally pivoted pad ........................................................41
Figure 3-26Damaged sliding surface due to overload and loss of the load carrying capacity (wear).........................................................................42
Figure 4-1 physical wedge mechanism...................................................45
Figure 4-2thrust bearing geometry. ..........................................................47
Figure 4-3force component and oil film geometry in hydro-dynamically lubricated thrust bearing sector. ................................................................47
Figure 4-4fixed-incline slider bearing. .....................................................49
Figure 4-5 pressure distribution...............................................................49
Figure 4-6effect of film thickness ratio on normal load carrying capacity. 50
Figure 4-7effect of film thickness ratio on force component. .....................50
Figure 4-8effect of film thickness ratio on friction coefficient parameter. .51
Figure 4-9effect of film thickness ratio on non-dimensional adiabatic temperature rise. .........................................................................................51
Figure 4-10-3Chart for determining minimum film thickness corresponding to maximum load or minimum power loss for various pad proportions - fixed-incline-pad bearings. [From Raimondi and Boyd (1955)..................52
Figure 4-11chart of determining dimensionless temperature rise due to viscous shear heating.....................................................................................52
Figure 4-12Bearing Pad ...........................................................................54
Figure 4-13kf and Kp to find hmin Figure 4-14Temp rise in the oil film ..........................................................................................................55
Figure 4-15kf and Kp as a function in k= (hmax- hmin)/ hmin ..................55
Figure 4-16Thrust bearing frame with bearing pads and spring in position ..........................................................................................................56
Figure 4-17Full Capacity Load comparison ...............................................58
Figure 4-18Pressure Before Turbine ........................................................58
Figure 4-19Comparing max N.H ..............................................................59
Figure 4-20Temperature in Thrust Bearing................................................59
Figure 5-1Head and load capacity increase effect on Temperature of thrust bearing..........................................................................................................64
Figure 5-2Calculation results....................................................................65

# List of Tables

Table 2-1viscosity-temperature equations ................................................10
Table 4-1the simplifying assumptions necessary for derivation the Reynolds Equation ..........................................................................................46
Table 4-22011 and 2012 Data....................................................................57
Table 4-32013 and 2014 Data....................................................................57
Table 4-4Total oil specifications ...............................................................60
Table 4-5Calculation results.......................................................................62
Table 5-1temperature of bearing, load capacity and head on the years 2011 and 2012.......................................................................................................63

# Symbols and abbreviations

| | |
|---|---|
| A | penstock area, [m$^2$] |
| a, b, c, d | are constants |
| B | pad width, [m] |
| Cp | specific heat, [j/kg k] |
| F | friction force, [N] |
| $F$ | generator, shaft and turbine runner load, [$N$] |
| $F_h$ | downward hydraulic thrust load, [$N$] |
| $F_{tot}$ | total load, N |
| $h$ | oil film thickness, [m] |
| $h_{max}$ | thickness of oil film at entering edge, m |
| $h_{min}$ | thickness of oil film at trailing edge, m |
| $k$ | $(h_{max} - h_{min})/h_{min}$ |
| $k_f$ | oil film thickness coefficient |
| $k_p$ | frictional losses coefficient |
| L | pad length, [m] |
| P | Pressure, [Pa] |
| $P_f$ | frictional power losses, [W] |
| $P_f$ | frictional losses, W |
| $\dot{q}$ | heat flux density, [W/m$^2$] |
| $r_i$ | Pad inner diameter, [$m$] |
| $r_m$ | Pad mean diameter, [$m$] |
| $r_o$ | Pad outer diameter, [$m$] |
| T | absolute temperature, [k] |
| U | oil velocity, [m/s] |
| $u, v, w$ | velocity components in $x, y, z$ directions, [m/s] |
| v | mean velocity, m/s |
| W | normal applied load |
| $x$ | $L/B$ |

# Greek symbols

| | |
|---|---|
| α | convection coefficient, [W/$m^2$ k] |
| η | absolute viscosity (dynamic), [ Pas] |
| λ | heat conductivity, [W/m k] |
| μ | coefficient of friction |
| ν | kinematic viscosity, [$m^2$/s] |
| ρ | Fluid density, [kg/$m^3$] |
| τ | the shear stress acting on the fluid, [Pa] |
| Φ | function of dissipation, |
| Ω | Pad angle, [degree] |
| ΔT | temperature increase, [k] |
| W | rotational speed, [rpm] |

# Dimensionless Equations

| | |
|---|---|
| ***Nu*** | ***Nusselt number*** |
| ***Pr*** | ***Prandtl number*** |
| ***Re*** | ***Reynolds number*** |
| ***St*** | ***Stribeck number*** |
| $\mathbf{H_s}$ | ***Hersey number*** |

# 1 INTRODUCTION

## 1.1 BACKGROUND

In the late 1880's, experiments conducted on the lubrication of bearing surfaces. The idea of "*floating*" a load on a film of oil grew from the experiments of *Beauchamp* tower and the theoretical work of *Osborne Reynolds*.

### 1.1.1 Thrust Bearings

In 1896, inspired by the work of *Osborne Reynolds, Albert Kingsbury* conceived and tested a pivoted shoe thrust bearing.

*A.G.M. Michel* independently invented a bearing based on the same hydrodynamic principles.

$1^{ST}$ HYDRO APPLICATION - In 1912, Albert Kingsbury was contracted by the Pennsylvania Water and Power Company to apply his design in their hydroelectric plant at Holtwood. The existing roller bearings were causing extensive down times (several outages a year) for inspections, repair and replacement. The first hydrodynamic pivoted shoe thrust bearing was installed in Unit 5 on June 22, 1912. At startup of the 12,000 KW units, the bearing wiped! However, after properly finishing the runner and fitting the bearing, the unit ran with continued good operation. ASME as the 23rd International Historic Mechanical Engineering Landmark designated this bearing, owing to its merit of running 75 years with negligible wear under a load of 220 tons, on June 27, 1987.

EARLY SHIPBOARD APPLICATION - Prior to the development of the pivoted shoe thrust bearing, marine propulsion relied on a "horseshoe" bearing which consisted of several equally spaced collars to share the load, each on a sector of a thrust plate. The parallel surfaces rubbed, wore, and produced considerable friction. Design unit loads were about 40 psi. Comparison tests against a pivoted shoe thrust bearing of equal capacity showed: The pivoted shoe thrust bearing being only 1/4 the size, had 1/7 the area but operated successfully with only 1/10 the frictional drag of the horseshoe bearing.

Thus, the hydrodynamic pivoted shoe thrust bearings provided considerable benefits. They were smaller, less expensive, required less maintenance, lasted longer, and were more efficient. Indeed, the invention made it possible to build the high-tech machines and ships of today

.

### 1.1.2 Hydropower Turbine

Hydro Turbine Generators with vertical shafts have significant axial loads due to the large mass of the rotating machine components (*generator rotor, shaft, and turbine runner*), as well as any downward hydraulic thrust produced by the turbine. The axial loads are supported by very heavy-duty thrust bearings. Machines generating more than about 1 MW usually use fluid type thrust bearings where the axial thrust is supported on a thin layer of pressurized oil. While the machine is running, the faces of the bearing pads separated from the shaft collar by a thin oil-film, which provides a hydrodynamic buffer between the two surfaces to prevent metal-to-metal contact. The allowable hydraulic thrust load that accommodate by standard machine design is satisfactory for *Francis* or *Pelton* runners. However, *Kaplan* ("*variable-pitch propeller*") runners require provisions for higher than normal thrust loads due to the added downward loads produced by this type of runner.

### 1.1.3 Friction, Lubrication and Their Impact in Hydropower Applications

As with any type of machinery, hydro-electric generating plant is continuously undergoing development with the aim of improving efficiency and performance and maximizing generating capacity from ever smaller machines. Friction between moving parts is the number one enemy of the machine designer trying to improve efficiency. Friction can induce extreme localized rises in temperature and stress, which consequently contribute to material damage, increased rates and volumes of wear and a general reduction in system operating efficiency. All these factors play a role in shortening working life for the machinery and the incurrence of associated additional operating and maintenance costs. It is therefore desirable to minimize the influence of friction.

Any mechanical system having components moving in close proximity to one another requires some form of lubrication to reduce to a minimum friction between such surfaces. This in turn can achieve an extension of component life through the limitation of material damage.

Due to its "on-demand" availability, hydropower increasingly used as the generation method of choice when extra load placed on the electricity system (*i.e. at times of peak demand*). It takes only a matter of minutes (*seconds in certain cases*) to start-up a hydroelectric turbine as compared to hours or even days for fossil-fueled power stations.

## 1.2 IMPORTANCE OF THE STUDY

Thrust bearings frequently used in large energy converting machinery (*electrical engines, power plants etc*.). In hydro power plants, thrust bearings used for carrying the weight of the generators, turbines and hydro downward thrust. The friction in the thrust bearings is responsible for a large amount of the energy losses in hydro power plants, which may convert into a large amount of heat energy, (*like in our case*).

Operated correctly Thrust bearings have an almost infinite lifetime as wear can be completely avoided.

Due to the rising energy prices, the possibilities of reducing the friction in hydro power plants are receiving increased attention. The energy savings due to reduced friction can often fully depreciate the cost of operation and maintenance. Research discussing the causes of Thrust bearings over heat occurred in *ROSEIRS* power plant six 40 MW Units (*they are seven units but unit 6 is not available due to a big problem*) and how to reduce this temperature.

## 1.3 PROBLEM STATEMENT

Problem statement of this research lies on the Effect of *ROSEIRIS* dam heightening on the thrust bearing of the power station units. That after heightening affairs finished and lake filled power-generating head increased to be 50m instead of 40m.

A problem of thrust bearing over heating appeared in four units 40MW.

Units (*1, 2, 3and 4*). Until now, the only one solution that was done is operating the thrust bearing high-pressure pumps, so temperature decreases after few minutes.

## 1.4 OBJECTIVES OF THE PROJECT

The motivation of this project mentioned as follows:

- Improve performance of thrust bearing lubricating and cooling.
- Maintain increase of temperature.
- Reduce time of maintenance.
- Save money (*economic side*).

## 1.5 RESEARCH METHODOLOGY

The procedure to follow in this research is as follow:

- Data about this case will collect and study carefully.
- Data will analyze to find most effective causes.
- We will present all available solutions and will apply them.

## 1.6 OUTLINES OF THE RESEARCH

The outline of the thesis is as follows:

**Chapter 2:** discuss lubrication and wear mechanisms, what are lubricants made of, and their properties, oils different from greases, mineral oils and their uses, oils that are most suitable for application to gears bearings, the oil viscosity, the thermal properties and temperature characteristics of lubricants.

**Chapter 3:** Presents hydrodynamically lubricated bearing, Large thrust bearing and their design, bearing housing arrangement, Auxiliaries*, (cooling, oil pumping systems, Filters and quality of the oil)*, Thrust bearing calculations, description and derives the mathematical models which are used to study thrust bearings. The models based in the Reynolds equation and employ 2- or 3-dimensionaltreatment of the temperature distribution in the oil film and in the pad. Deformations of the pad and runner

**Chapter 4:** This chapter studies effect of *ROSARIES* reservoir dam heightening on thrust bearing of power plant turbines, collect data, drives mathematical formulas, which will be used to study the case, study and calculate the case situation before and after heightening, study the effect of additional downward thrust created, by water.

**Chapter 5:** presents the results and their discussion.

**Chapter 6:** Conclusion and Recommendations.

# 2 LUBRICANTS

## 2.1 INTRODUCTION

Before discussing lubrication and wear mechanisms, some information about lubricants is necessary.

What are lubricants made of, and what are their properties? Are oils different from greases? Can mineral oils use in high performance engines? Which oils are the most suitable for application to gears, bearings, etc.? What criteria should they meet? What is the oil viscosity, what are the thermal properties and temperature characteristics of lubricants? An engineer should know the answers to all these questions.

In simple terms, the function of a lubricant is to control friction and wear in a given system or a lubricant is any substance that reduces friction and wear and provides smooth running and a satisfactory life for machine elements.

The basic requirements therefore relate to the performance of the lubricant, i.e., its influence upon friction and wear characteristics of a system. Another important aspect is the lubricant quality, which reflects its resistance to degradation in service. Most lubricants are liquids (such as mineral oils, synthetic esters, silicone fluids, and water), but they may be solids (*such as poly-tetra-fluoro-ethylene, or PTFE*) for use in dry bearings, greases for use in rolling-element bearings, or gases (*such as air*) for use in gas bearings. The physical and chemical interactions between the lubricant and the lubricating surfaces must understand in order to provide the machine elements with satisfactory life. Present day lubricant research dedicated to the study, prevention and monitoring of oil degradation, since the lifetime of oil is as important as its initial level of performance. Apart from suffering degradation in service, which may cause damage to the operating machinery, Oil may cause corrosion of contacting surfaces. The oil quality, however, is not the only consideration. Economic considerations are also important. For example, in large machinery holding several thousand liters of lubricating oil, the cost of the oil can be very high.

## 2.2 LUBRICATION REGIMES

To understand the features that distinguishes the four lubrication regimes from one another. Perspective given, followed by a description of each regime.

### 2.2.1 Hydrodynamic Lubrication

Hydrodynamic lubrication (*HL*) generally characterize by conformal surfaces. A positive pressure develops in a hydro-dynamically lubricated journal or thrust bearing because the bearing surfaces converge and the relative motion and the viscosity of the fluid separate the surfaces. In hydrodynamic lubrication, the films are generally thick so that opposing solid surfaces prevented from coming into contact. This condition is often referred to as "the ideal form of lubrication," since it provides low friction and high resistance to wear.

### 2.2.2 Elasto hydrodynamic Lubrication

Elasto-hydrodynamic lubrication (*EHL*) is a form of hydrodynamic lubrication where elastic deformation of the lubricated surfaces becomes significant.

### 2.2.3 Boundary Lubrication

Because in boundary lubrication the solids do not separate by the lubricant, fluid effects are negligible and there is considerable asperity contact.

### 2.2.4 Partial Lubrication

If the pressures in Elasto hydro-dynamically lubricated machine elements are too high or the running speeds are too low, the lubricant film will penetrated. Some contact will take place between the asperities, and partial lubrication (*sometimes referred to as "mixed lubrication"*) will occur. The behavior of the conjunction in a partial lubrication regime governed by a combination of boundary and fluid film effects.

## 2.3 Stribeck Curve

A useful concept for the understanding of the role of different regimes of lubrication is the *Stribeck* curve as shown in Figure2.1 (*Stribeck, 1902*). The ordinate in Figure2.1 is the coefficient of friction under steady state conditions. The abscissa in Figure2.1 is a dimensionless number, sometimes referred to as the *Hersey* number, and is given as

$$H_s = \frac{\eta\omega}{p}$$

where

$\eta$ = absolute viscosity, Pa·s

$\omega$ = rotational speed, rps

$p$ = pressure, Pa 2.1

A high Hersey number usually means a relatively thick lubricant film, whereas a small number results in a very thin film. The regimes of lubrication indicated on Figure 2.1 and inferred from the friction behavior of the bearing system. At an extremely low Hersey number, no real lubricant film can develop and there is significant asperity contact, resulting in high friction. This high friction value is persistent with increasing *Hersey* number until a first threshold value reached. This represents the dominance of boundary lubrication in determining load transfer and friction between surfaces. As the Hersey number increases, a noticeable and rapid decrease in friction values observed.

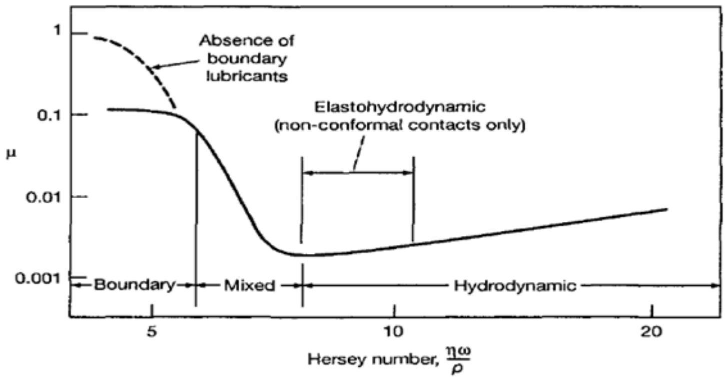



*Figure 2-1 Hersey Number*

## 2.4 OIL VISCOSITY

The parameter that plays a fundamental role in lubrication is oil viscosity. Different oils exhibit different viscosities. In addition, oil viscosity changes with temperature, shear rate and pressure, and the thickness of the generated oil film is usually proportional to it. Therefore, at first glance, it appears that the more viscous oils would give better performance, since the generated films would be thicker and a better separation of the two surfaces in contact achieved. This unfortunately is not always the case since oils that are more viscous require more power to shear. Consequently, the power losses are higher and more heat generated resulting in a substantial increase in the temperature of the contacting surfaces, which may lead to the failure of the component. For engineering applications, the oil viscosity usually chose to give optimum performance at the required temperature. Knowing the temperature at which the oil expected to operate is critical, as oil viscosity is extremely temperature dependent. The viscosity of different oils varies at different rates with temperature. The velocities of the operating surfaces (shear rates) can also affect it. The knowledge of the viscosity characteristics of a lubricant is therefore very important in the design and in the prediction of the behavior of a lubricated mechanical system.

In this part, a simplified concept of viscosity, sufficient for most engineering applications, is considered. Refinements to this model incorporating, for example, transfer of momentum between the adjacent layers of lubricant and transient visco-elastic effects, can be found in more specialized literature.

### 2.4.1 Dynamic Viscosity

$\eta = (F/A) / (u/h)$ or $\eta = \tau / (u/h)$ 2.2

Where:

$\eta$ is the dynamic viscosity [Pas];

$\tau$ = the shear stress acting on the fluid [Pa];

U/h is the shear rate, i.e., velocity gradient normal to the shear stress [s-l].

1 [P] = 100 [cP] ≈ 0. 1[Pas]

### 2.4.2 Kinematic Viscosity

$\nu = \eta / \rho$ 2.3

Where:

$\nu$ is the kinematic viscosity [m2/s]

$\eta$ is the dynamic viscosity [Pas];

$\rho$ is the fluid density [kg/m3].

1 [S] = 100 [cS] = 0.0001 [m2/s]

The densities of lubricating oils are usually in the range between 700 and 1200 [kg/m 3] (0.7&1.2 [g/cm3]). The typical density of mineral oil is 850 [kg/m 3] (0.85 [g/cm 3]). To find the dynamic viscosity of any oil in [cP] or [Pas] the viscosity of this oil in [cS] multiplied by its density in [g/cm$^3$], hence for a typical mineral oil:

Viscosity in [cP] = viscosity in [cS] x 0.85 [g/cm $^3$]

or

Viscosity in [Pas] = viscosity in [cS] x 0.85 [g/cm $^{3}$] x$10^{3}$.

## 2.4.3 Viscosity Temperature Relationship

The viscosity of lubricating oils is extremely sensitive to the operating temperature. With increasing temperature, the viscosity of oils falls quite rapidly. In some cases, the viscosity of oil can fall by about 80% with a temperature increase of 25°C. From the engineering viewpoint, it is important to know the viscosity at the operating temperature since it influences the lubricant film thickness separating two surfaces. The oil viscosity at a specific temperature either calculated from the viscosity-temperature equation or obtained from the viscosity-temperature ASTM (*American Society for Testing Materials*) chart.

### *2.4.3.1 Viscosity-Temperature Equations*

There are several viscosity-temperature equations available. Some of them are purely empirical while others derived from theoretical models. The most commonly used equations summarized in Table [2.1]. The most accurate of these is the Vogel equation.

Three viscosity measurements at different temperatures for specific oil needed in order to determine the three constants in this equation. The oil viscosity can then calculated at the required temperature, or the operating temperature calculated if the viscosity known. Apart from being very accurate, the *Vogel* equation is useful in numerical analysis.

A computer program 'VISCOSITY' for the *Vogel* equation is there. Based on the three temperature-viscosity measurements the program calculates the viscosity at the required temperature. It also includes the option for the calculation of temperature corresponding to a given viscosity.

*Table 2-1viscosity-temperature equations*

| Name | Equation | Comments |
|---|---|---|
| Reynolds | $\eta = \mathbf{b}e^{-\mathbf{a}T}$ | Early equation; accurate only for a very limited temperature range |
| Slotte | $\eta = \mathbf{a}/(\mathbf{b} + \mathbf{T})^{\mathbf{c}}$ | Reasonable; useful in numerical analysis |
| Walther | $(\upsilon + \mathbf{a}) = \mathbf{b}\mathbf{d}^{1/T^c}$ | Forms the basis of the ASTM viscosity-temperature chart |
| Vogel | $\eta = \mathbf{a}e^{\mathbf{b}/(\mathbf{T}-\mathbf{c})}$ | Most accurate; very useful in engineering calculations |

where:

**a**, **b**, **c**, **d** are constants;

$\upsilon$ is the kinematic viscosity $[m^2/s]$;

**T** is the absolute temperature [K].

### *2.4.3.2 Viscosity-Temperature Chart*

The most widely used chart is the ASTM Viscosity-Temperature chart (*ASTM D341*), which is entirely empirical and based on Walther's equation Table [2.1]

$(\nu + a) = b\, d^{\,1/Tc}$ (2.4)

In deriving the bases for the ASTM chart, logs taken from Walther's equation and (d) assumed to equal 10. The equation then written in the form:

$\log_{10}(\nu + a) = \log_{10} b + I / T^c$ (2.5)

It found that if ' ν ' is in [cS] then 'a' is approximately equal to 0.6. After substituting this into the equation, the logs taken again in the manner shown below:

$\log_{10}\log_{10}(\nu_{cs} + 0.6) = a' - c\log_{10}T$(2.6)

Where:

a', c are constants.

Although equation (2.6) forms successful bases for the ASTM viscosity-temperature chart, where the ordinate is $\log_{10}\log_{10}(\nu_{cs} + 0.6)$ and the abscissa is log10 T, from the mathematical viewpoint the above derivation is incorrect. This is because when taking logs, equation (2.6) should be in the form:

$\log_{10}\log_{10}(\nu_{cS} + 0.6) = \log_{10}(\log_{10}b + 1/T^{c})$

but

$a' - c\log_{10}T \neq \log_{10}(\log_{10}b + 1/T^{c})$

Despite this, the ASTM chart is quite successful and works very well for mineral and synthetic oils under normal conditions. Therefore, it well standardized that the viscosity-temperature characteristics sometimes specified as 'ASTM slope'.

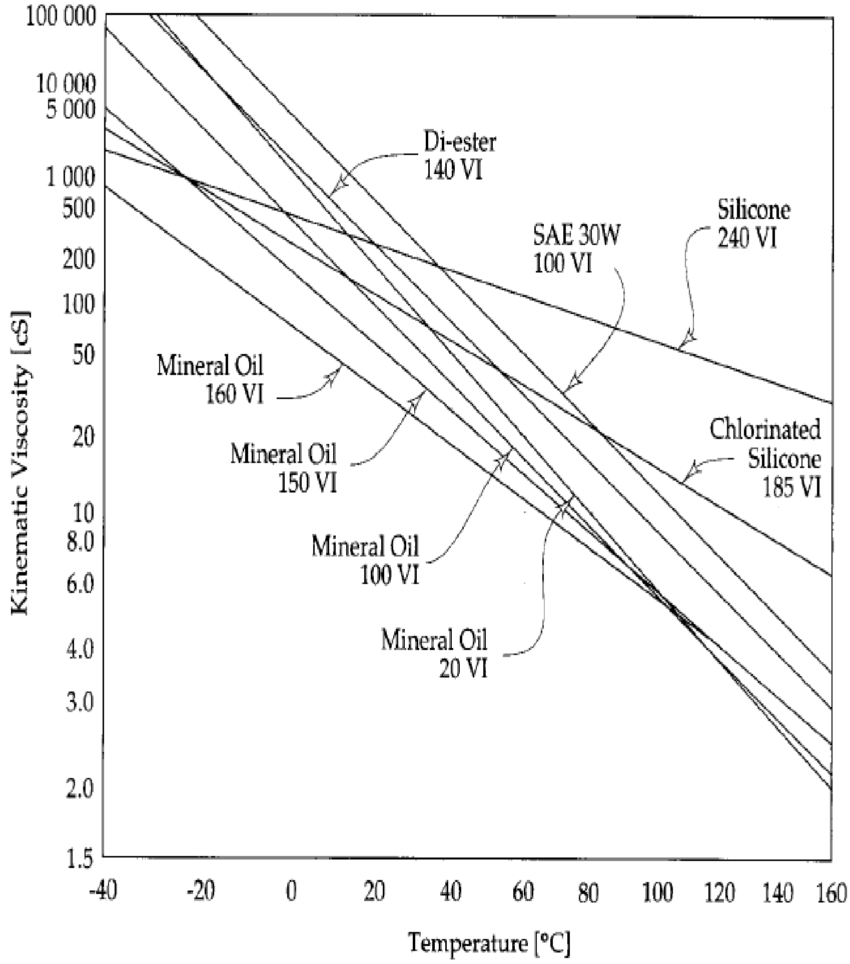



*Figure 2-2Viscosity-temperature characteristics 'ASTM slope'*

# 3 HYDRODYNAMIC THRUST BEARINGS

## 3.1 INTRODUCTION

A hydro dynamically lubricated bearing is a bearing that develops load carrying capacity by virtue of the relative motion of two surfaces separated by a fluid film.

Hydro Turbine Generators with vertical shafts have significant axial loads due to the large mass of the rotating machine components (*generator rotor, shaft, and turbine runner*), as well as any downward hydraulic thrust produced by the turbine. The axial loads supported by very heavy-duty thrust bearings. Machines generating more than about 1 MW usually use fluid type thrust bearings where the axial thrust is supported on a thin layer of pressurized oil. While the machine is running, the faces of the bearing pads separated from the shaft collar by a thin oil-film, which provides a hydrodynamic buffer between the two surfaces to prevent metal-to-metal contact. The allowable hydraulic thrust load that can accommodate by standard machine design is satisfactory for *Francis* or *Pelton* runners. However, *Kaplan* ("*variable-pitch propeller*") runners require provisions for higher than normal thrust loads due to the added downward loads produced by this type of runner.

## 3.2 THRUST BEARINGS

Large hydro generators have usually vertical axes of rotation in order to minimize deflections due to the weight, assembly inaccuracies and vibrations. Thus, the hydrodynamic thrust bearing carries the main load. It arranged in form of several tilting pads around the shaft. The load carried from the shaft by the means of the thrust collar and runner. Then through the oil, film to the bearing pads and further through the supporting system to the foundation.

A large hydrodynamic thrust bearing of a vertical hydro generator is Consists of following components: thrust and guide bearing pads and thrust collar attached to the shaft, bearing support system, the housing that transfers the load to the ground and at the same time has the function of the oil tank, lubrication- cooling system and other devices. The bearing housing ensures that both loads (*axial and radial*) transferred to the supporting foundation. It also has to ensure that the load carried independently of the thermal state of the system. The thrust bearing operates close to its operational limits in order to make its design on one hand side efficient and on the other very reliable.

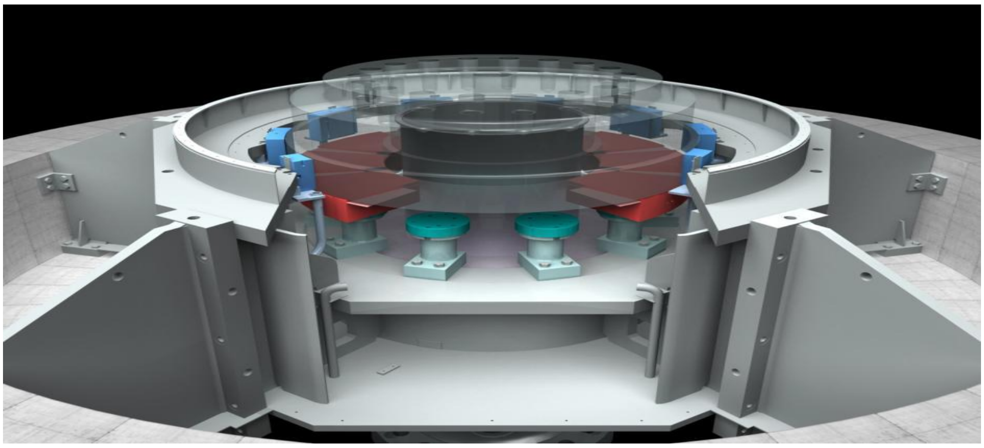


*Figure 3-1Thrust bearing of a large hydro generator unit (ALSTOM Hydro)*

## 3.3 LARGE THRUST BEARING DESIGN

Large hydro generators have usually vertical axes of rotation since such an arrangement makes the overall design more compact and robust. Deflections of the structure are smaller. This approach leads though to a thrust bearing high load, which is also loaded, in still stand and during startup and shut down – circumstances where the hydrodynamic action of the oil film is strongly limited. Due to variable axial deformations of the whole unit (*elastic and thermal*) there is always only one thrust bearing. The axial load consists of two components. First of them, the constant one, is the weight of the rotational part of the hydro generator. Generally, it can say that the weight of the hydro generator is about *80 %* of the whole unit and the rest is the turbine. Sometimes, it called dead weight. The second component of the axial load comes from the hydraulic thrust, which generated by the turbine. The power (*torque*) generated by the tangential component of the force; however, there occurs an additional axial component of the force. This part subjected to variations as a function of the actual power delivered to the network. It may change rapidly in transient (*emergency*) cases (*e.g. power rejection*). These variations occur due to changes of the water pressure, speed or even flow direction. They may have both directions (*downwards and upwards*) and are difficult to predict.

There may also be an additional thrust load from the magnetic thrust created by the generator. This component subjected to the assembly inaccuracies and usually small. It may also have both directions – it may create additional load but it may also reduce the overall load of the thrust bearing.

### 3.3.1 Bearing Housing Arrangement

The thrust bearing can be either single or combined with the guide bearing in order to save place and to use common auxiliary devices for both of them. In case of vertical hydro generators, bearing usually fully immersed in oil. In this case, the housing has an additional function of the oil tank. The space then divided in three separate chambers

- Cold oil, where fresh oil delivers and directs to the bearing pads.
- Warm oil, where the warm oil store – outlet to the cooling system.
- Air chamber, space for possible oil expansion due to temperature increase and air in oil.

The radial ribs in the housing prevent from the circumferential flow of the oil and allow for air separation.

Guide bearing combined with the thrust one in the same housing. Sometimes the guide-bearing pad used to pump the oil through the coolers instead of installing the external pump. In such case, the cooler has to be able to dissipate the power losses created by the both bearings together. Such approach quite often used during design of hydro generators. It reduces number of bearing brackets and auxiliary systems. On the other hand, design of both bearings has to fit to each other. It can for example lead to over dimensioning of the guide-bearing diameter.

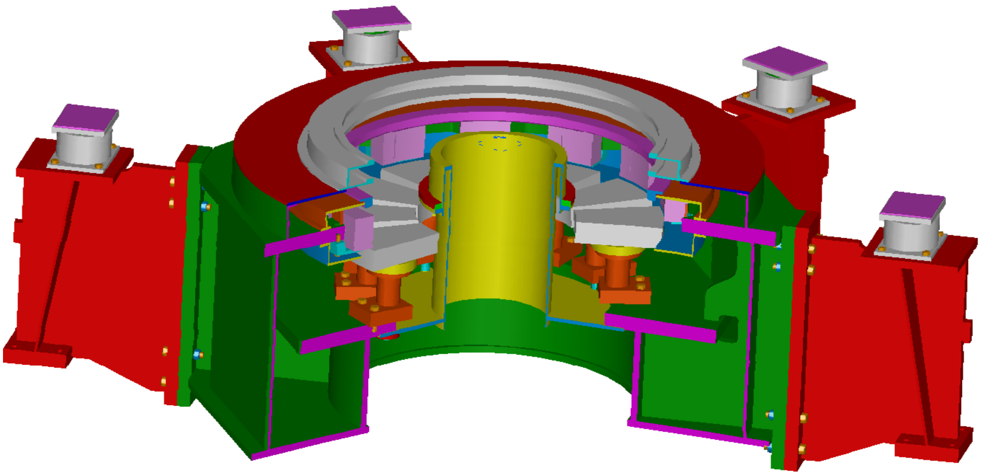


*Figure 3-2Typical combined thrust-guide bearing housing of a large hydro generator (ALSTOM Hydro)*

Some bearing manufactures make use of special devices that deliver the fresh oil close to the rotating shaft surface. Among them, one can distinguish different oil spray systems or LEG (*Leading Edge Groove*) technology offered by *Kingsbury* Company.

#### *3.3.1.1 Support systems*

The overall thrust bearing design is similar in most cases and the main differences may occur in design details. One of them is the support system of the bearing pads. Its main function except the most obvious (*load transfer from the bearing pad to the ground*) is the equalizing of the load among all the bearing pads. Different manufactures have developed their different designs that assure appropriate load distribution. Among them one can distinguish the most popular:

- Without any load equalizing.
- Mechanical load equalizing.
- Additional elasticity in the load chain.
- Hydraulic support.

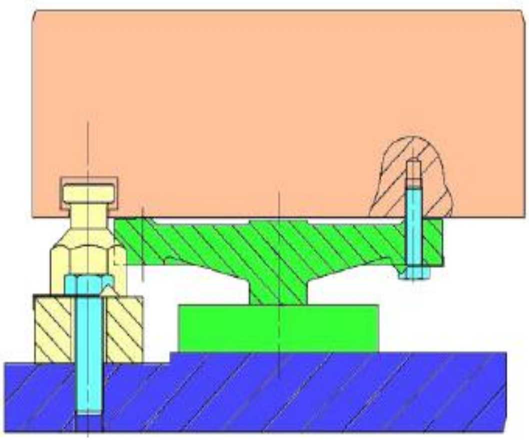
In the literature, one can find information about problems with appropriate load equalizing between the pads. For example, in some problems with hydro generator thrust bearing with the outer diameter equal to 1.8 m. The presented measurement data indicated that the temperatures between the pads varied within the range of18 °C. This suggested that some pads carried much higher load than others did. The careful analysis indicated that the warmest pad carried approximately doubled force. This example shows how important it is to ensure precise load distribution. Otherwise, the most loaded pad may initiate the process of the seizure of the whole bearing. This happens due to carry over of the melted Babbitt to the following pads and its aggregation on their leading edges. In this way, the oil film is broken away and the hydrodynamic lubrication of the next pad interrupted. In case of large thrust bearings, it is important to assure that pads are free to tilt in both directions. The tangential tilting is important because of the inclination of the pad and the hydrodynamic pressure generation and the radial tilting compensates thermal and elastic deformations of bearing components (*runner and bracket*). Pad and runner sliding surfaces remain almost parallel in the radial direction. For this reason, in the hydro generators almost only point supported thrust bearings are used. Kingsbury in the USA has patented this solution in

1912. Since that time, many different supporting systems developed but the general function of such bearing remains the same. No load equalizing system proposed in case of a small bearing where the manufacturing tolerances and elastic and thermal deformations are within a small range. In case of large-scale units, bearing equalizing system is a necessary solution. Since the oil, film layer is around several tens of microns thick it is essential to ensure that all the sliding surfaces of the segments lay on the same level or within a very tight range. Otherwise, some of the pads will be loaded with much higher forces while others will not carry the load at all.

A different approach of the load equal distribution is an additional elasticity in the load chain. Increased elasticity of a support system can give an ability of compensation of the assembly and manufacturing inaccuracies. Below some of the most common supporting systems that make use of the increased elasticity presented.

### *3.3.1.2 Plate spring design*

Plate spring is a solution for small hydro generators with the outer bearing diameter up to *1.2 m*. It is easy to manufacture and assembly. Its additional advantage is that it does not require much space in axial direction. In a design phase, the geometry adjusted in order to obtain the required axial stiffness. It should be as small as possible in order to ensure a wide range of axial deflection under the load but on the other hand is has to be large enough to keep the critical axial frequency of vibrations on a relatively high level. In the middle of the spring, one can see a sort of bumper. Its role is to limit the deformations and stresses in the spring in case of extremely high loads (*e.g. emergency or failure*).



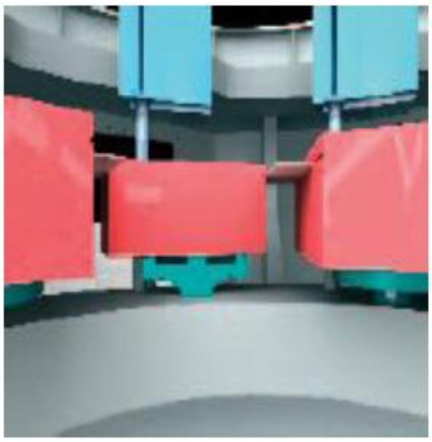


*Figure 3-3Plate spring (ALSTOM Hydro)*

#### 3.3.1.3 Spindle support design

Spindle support is a solution designed for medium and large sized generators. It allows for the adjustment of each of the pads separately during the final assembly of the generator. The lower end connected with the ground via a thread connection. Its relatively small cross section ensures the required elasticity, which adjusted during the design process. Inside there is a measurement pin attached to the top of the spindle so the axial deformation measured during assembly of the bearing. The deformations of all the spindles compared with each other and necessary adjustments made in order to equalize the load. The final confirmation of the appropriate load distribution among the pads made during first mechanical tests of the unit. If the pads are loaded equally the temperatures among the pads should also be equal. Normally there is certain range of tolerance for the measured temperatures. All values should be within the range of*2 – 3* °C.

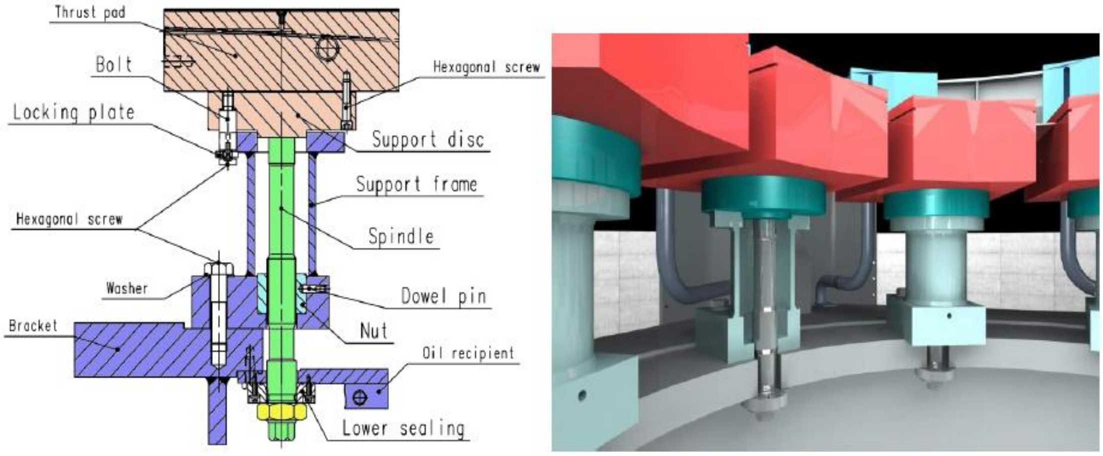



*Figure 3-4Spindle support (ALSTOM Hydro)*

#### 3.3.1.4 Spring mattress design

H. G. Riest at General Electric Company has invented spring mattress design around *1915*. At that time, it designed in form of a thin, washer type ring with one radial gap. Later this design was adapted to segmented thrust bearings. They used in order to obtain concave elastic bending of the pad and the compensation of the pad thermal crowning. However, in early designs the pads supported by springs from edge to edge, distribution and stiffness of the springs adjusted within a wide range during design in order to optimize elastic deformation of the pad.

Thickness of the pad should be relatively small in comparison with other dimensions in order to increase its elasticity. Otherwise, the elastic effects will not be sufficient. For bidirectional bearings, spring support has the disadvantage of an additional reaction moment of the support that reduces the tilt angle. The arrangement of the springs has to be symmetrical in order to ensure equal behavior in both directions (*center of effort*). Since the bidirectional tilt angle is already very small such effect can have a negative influence on the bearing temperature and minimum oil film thickness since the inlet oil gap is smaller and there will be less cold oil entering the oil film. This influence investigated further in this work.

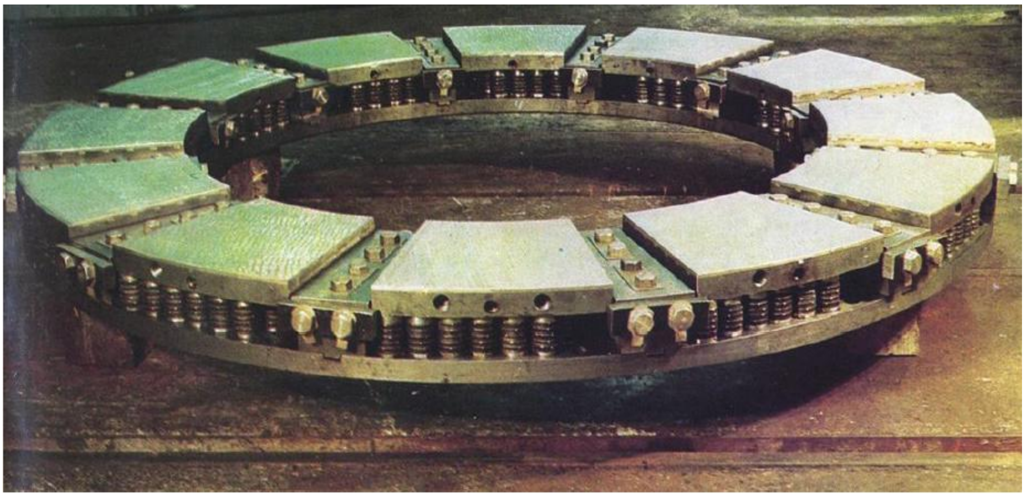


*Figure 3-5spring bed support system*

#### *3.3.1.5 Mattress of rubber springs*

Mattress of rubber springs is a similar solution to the spring bed one but rubber discs are used instead of helical springs (*see Figure 3.6*), although the problems with reaction moments remain still

*Figure 3-6Mattress of rubber springs support system*

### *3.3.1.6 Membrane support*

Membrane support makes use of a different approach in order to equalize the load among the pads. The hydraulic connection made between the pad supports in order to obtain equal reaction on the ground (*see Figure 3.7*). The connection realized by the system of bores in the base plate under the membranes. The same reaction forces among the pads means automatically equal load distribution according to the Newton Third Rule.

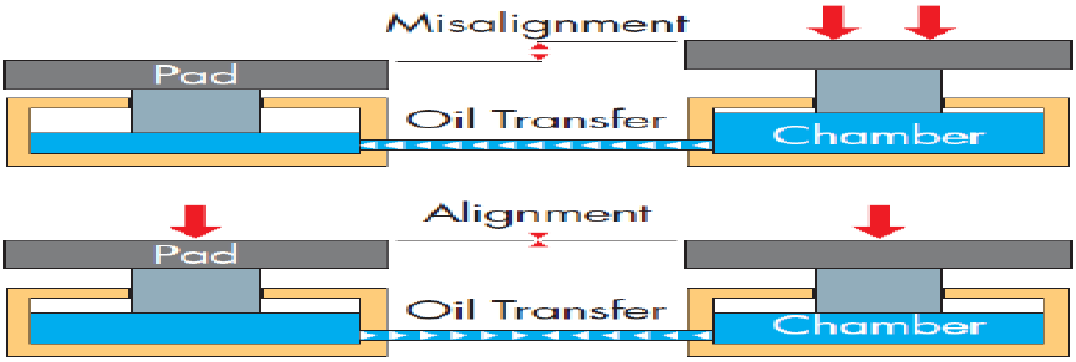
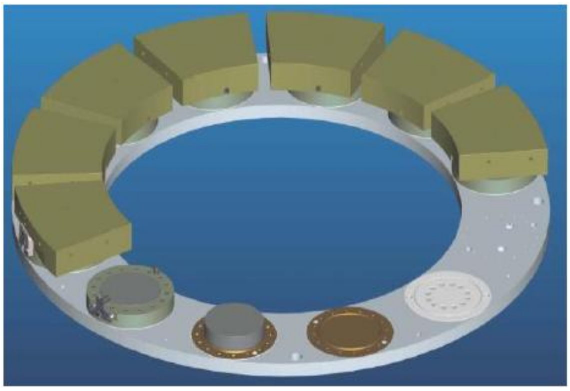



*Figure 3-7Membrane support, principle of operation (ALSTOM Hydro)*



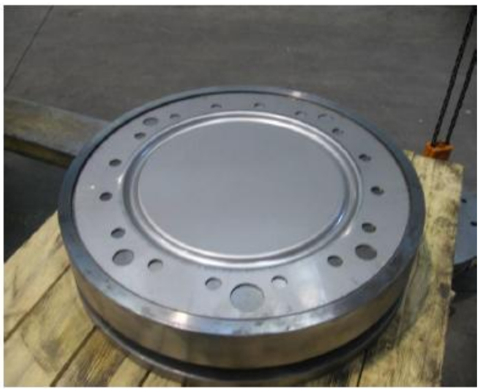


*Figure 3-8Membrane support (ALSTOM Hydro)*

Additional advantage of this system, used for large bearings, is that it does need neither assembly nor operational adjustments. Any possible deformation (*thermal or elastic*) of the bearing bracket or any other element compensated, within a certain range, by the movement of the membrane. This solution has been used for a long time but recently there has been a new, simplified design developed by *ALSTOM* Hydro company that made the membrane solution more compact and reliable (*see Figure 3.8*). The recent development made it compact, reliable and easy to assembly. On the other hand, the special tools needed in order to manufacture the membrane plate.

## 3.3.2 Auxiliaries

The bearing is equipped with many additional devices that support its main function. The most important ones described below.

### *3.3.2.1 Hydrostatic jacking system*

Since the hydrodynamic effect depends on the relative speed between the pad and the thrust collar in order to prevent wear of the sliding surfaces usually there is additional hydrostatic equipment that supports the bearing during startup and shut down. The additional oil pumped to the oil film assures the lubrication at low rotational speeds and thus the sliding surfaces always separated.

In case of small bearings, low specific pressures and in case of polymer coated bearing pads there is no need of the hydrostatic assistance. It is enough when before start up the shaft raised with the use of hydraulic brakes and the oil comes between the collar and the pads. Within the short time, the squeeze effect of the oil film can give sufficient lubrication at low speed. In this way, design and reliability of the thrust bearing and its design simplicity remarkably increased.

In case of any problems with hydrodynamic lubrication (*insufficient load carrying capacity*), the hydrostatic jacking system may be used permanently in order to increase reliability of the bearing. In such a situation, the lubrication system must be equipped with two or three high-pressure pumps in order to assure appropriate operation either in case of failure of one of the pumps, lack of power supply or any other reason. Energetic efficiency of the completely hydro generator not affected in this way since power of the high-pressure pump is usually very low (*several kilowatts*). Additionally, it does not operate at rated conditions of the generator.

*Figure 3-9Hydrostatic jacking system operation (ALSTOM Hydro)*

The hydraulic circuit is equipped with all necessary additional elements like backpressure valves and orifices, so the functionality of the jacking system remains uninfluenced also in case of failure of one (*or even more*) of the connections. The oil usually filtered up to *10 µm*.

### *3.3.2.2 Cooling systems*

In general, the cooling systems divided into the following groups:

- Air-cooling.
- Water cooling with internal cooler.
- Water cooling with external cooler.

For the bearing with relatively small losses and or allowable higher operational temperature ***air-cooled*** bearing housing proposed, especially in case of small hydro power plants. It makes the whole system compact, independent and robust. The air-cooled bearing housing can be realized either with natural or forced convection. In addition, large hydro generators or pumps with small rotational speeds (*power losses*) may profit from such simplified cooling methods. There has to be just enough oil in the bearing housing to disperse the energy delivered to the system due to the frictional losses.

***Internal water-cooling systems*** use the water pipes located in the bearing housing to exchange the heat. It is difficult to predict and control the efficiency of such a system. It can be influenced only by the amount of the cooling water and its temperature (*usually cooling water temperature cannot be controlled and additionally it is subjected to variations within a wide range depending on the season*). The convection coefficient between oil and the cooling pipes depends strongly on the oil flow in the bearing housing. Therefore, it varies with the rotational speed. There is also a danger of water leakage, which may require disassembly of the whole bearing.

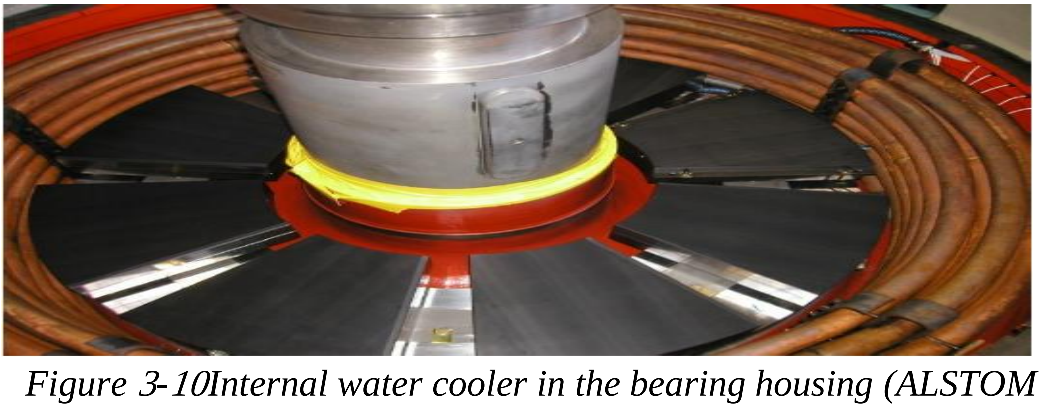


*Figure 3-10Internal water cooler in the bearing housing (ALSTOM Hydro)*

The most up to date solution is ***external water-cooling*** of the oil. The oil is pumped out from the bearing housing and then through the coolers. The oil can be pumped either with the use of external pumps supplied from the network or with the use of self-pumping devices. Self-pumping, a technology widely used; uses of viscous pump effects in the bearing housing. Rotation of the shaft used to pump the oil through the cooling system. This design allows for better control of the dispersed power since both the oil and water flows adjusted. In this way, constant cold oil temperature assured. Heat exchange in the coolers is also much better than in case of internal ones due to higher area and better convection coefficients.

Additional advantage of the external water-cooling is that the bearing may be equipped with the external oil tank where most of the oil is stored. Thus, the bearing housing may have compact dimensions, which might be important when the large amount of oil needed in the lubrication system. In contractual requirements, it usually said that, in case of emergency, hydro generator should run without water-cooling for *15 min*. For this case, the whole power loss created by the bearing has to accumulated in the oil and its temperature should not rise more than for safety reasons. Otherwise the oil viscosity might be too low and the temperature too high in order to maintain oil film thickness within the safe range.

*Figure 3-11External water cooler*

***Fouling:*** The deposition of any undesired material on oil coolers and any heat transfer surfaces called fouling. Fouling may significantly affect the thermal and mechanical performance of heat exchangers. It is a dynamic phenomenon, which changes with time increases the overall thermal resistance and lowers the overall heat transfer coefficient of heat exchangers. Also impedes fluid flow, accelerates corrosion and increases pressure drop across heat exchangers. Different types of fouling mechanisms identified. They can occur individually but often occur simultaneously. Descriptions of the most common fouling mechanisms provided below:

***Scaling/Crystallization Fouling***: Scaling is the most common type of fouling and is commonly associated with inverse solubility salts such as calcium carbonate (CaCO3) found in water.

***Particulate/Sedimentation Fouling***: Sedimentation occurs when particles (e.g. dirt, sand or rust) in the solution settle and deposit on the heat transfer surface. Like scale, these deposits may be difficult to remove mechanically depending on their nature.

***Corrosion Fouling***: Results from a chemical reaction, which involves the heat exchanger, surface material.

***Chemical Fouling***: Fouling due chemical reactions in the fluid stream, which result in the deposition of material on the heat exchanger surface.

***Freezing Fouling***: Occurs when a portion of the hot stream cooled to near the freezing point of one of its components.

***Biological Fouling***: Occurs when biological organisms grow on heat transfer surfaces.

### *3.3.2.3 Oil tanks and control of the oil flow*

Either oil for the bearing lubrication can be stored in the bearing housing or there may be an additional external tank in the lubrication system. The amount of oil can sometimes be very large in order to ensure an appropriate bearing cooling in case of lack of the cooling water. In such case, the bearing has to operate for a certain time (*15 min for example*) and the whole power loss has to dissipate in the oil volume. The time needed for the safe shut down of the unit and the temperature of the oil must not exceed the assumed value. Another reason for large amount of oil is the oil circulation in the cooling system. The time within which the whole oil exchanged should not be too short in order to allow the separation of the air that mixed with oil. Air usually mixed with oil in the areas of high speeds and where the separation between them is not well defined.

*Figure 3-12External oil tank and coolers*

The exchange of the warm oil usually forced in the bearing housing. It is important to ensure that fresh cold oil delivered to the oil-film inlet area. The bearing housing arranged a way that rotation of the runner usually used to enforce the oil flow. This is the so called viscous pump effect. This effect obtained by numerous designs. Some of them described below.

### *3.3.2.4* Rotating hole

One of the pumping effects used in order to force the oil flow in the bearing housing is the radial hole made in the runner (*see item 22 in Figure 3.13*). Due to centrifugal and viscous effects, the oil pumped in radial direction. The efficiency of such a solution depends on the geometry (*number and diameter of the holes, diameter of the thrust block, etc.)* and rotational speed of the shaft. In this figure, one can also notice that the relatively thin pad lies on the spring mattress. This particular example took from hydro generators catalogue from the ABB Company.

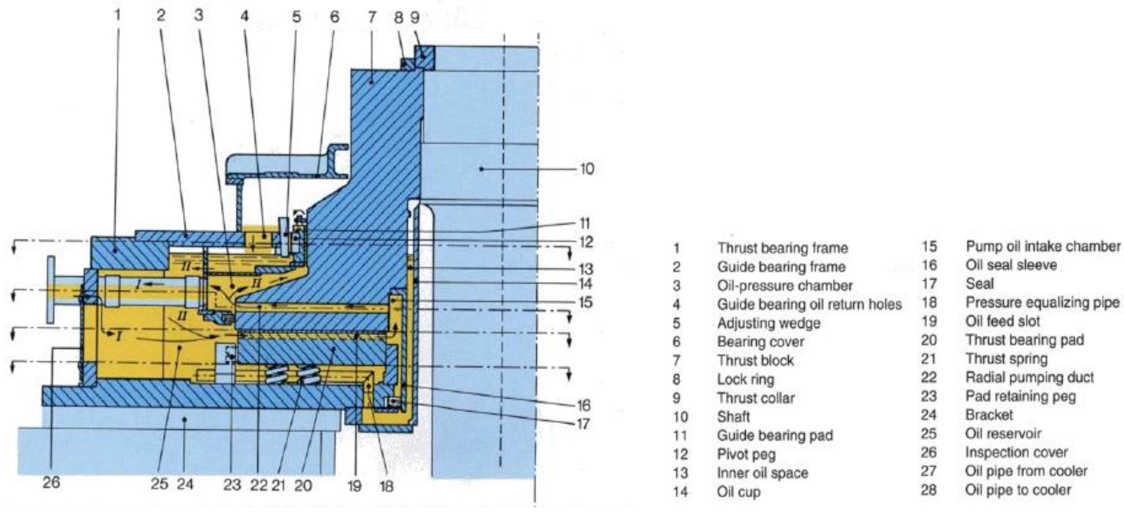



*Figure 3-13Rotating hole design*

### *3.3.2.5 Pumping plate*

Pumping plate between thrust pads used in order to separate warm and cold oil in the bearing housing and to deliver cold oil close to the oil film inlet. Additional function is to pump the oil in radial direction. In this way oil circulation in the bearing housing can be controlled. Such cross flow of the oil can cause better oil mixing between the pads and lower warm oil carry over effect.

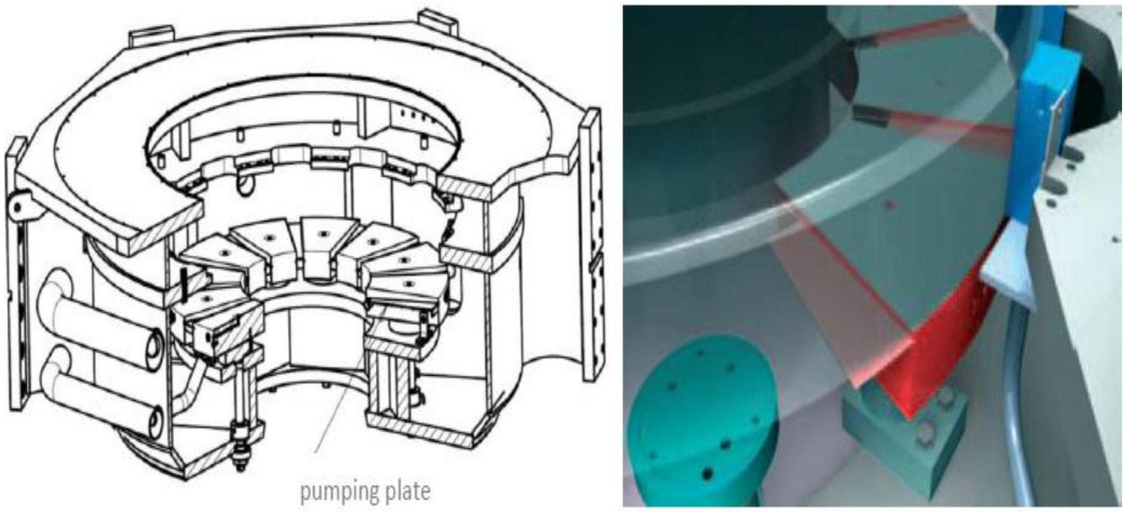



*Figure 3-14Pump plate between thrust pads design and flow (ALSTOM Hydro)*

### *3.3.2.6 Self-pumping pads*

*ALSTOM Hydro Ltd* (*formerly BBC*) company developed in 1980 is a self-pumping pads technology. At this moment, this is a highlight bearing technology of the *ALSTOM Hydro* Company and used in most of the current projects in case of the combined thrust – guide bearings. This design allows pumping oil through the cooling system with an external cooler and without any external pump. Specially designed guide-bearing pads used instead. Each of the pads has in the inlet area additional pocket so called pumping pockets where the pressure generated and used for pumping of the oil through the cooling system (*see Figure 3.15*). Pumping flows from all the pads then gathered in a common warm oil chamber or ring pipe. Pressure generated in this chamber causes the oil flow through the cooling system. Due to viscous character of such a pump, the resistance pressure has to keep on a relatively low level. For this reason, all pipelines and coolers must have cross sections large enough that assure low hydraulic losses. It also not allowed installing oil filter in such circuit. Main advantages of this technology are lack of maintenance, automatic operation and increase of the pumping flow as a function of the rotational speed of the shaft. In case of bidirectional bearing, self-pumping pockets need to manufacture on both sides of the pad.

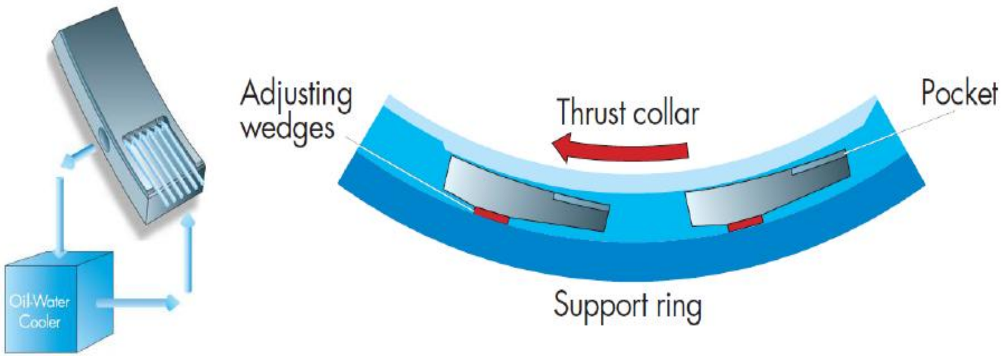



*Figure 3-15Self-pumping pad principle (ALSTOM Hydro)*

Theoretically, it is possible to use similar pumping pockets in the thrust bearing pads but this would reduce the valuable area of the bearing and increase the specific pressure.

### *3.3.2.7 External pumps*

In case of bearings with high power, losses it is possible to install an external pump that ensures circulation of the oil between the bearing housing and the external cooler (*see Figure 3.11*). Main advantage of such a solution is independence of the oil flow from the rotational speed of the bearing. Hydraulic losses in the cooling system can be also higher so there is no need of using large cross-sections in the cooling system. On the other hand, such a system is less reliable so very often the pumps and coolers have to double in order to ensure redundancy of the system. In order to ensure an appropriate function of the cooling system in case of emergency and lack of external power supply an additional sources of power have to assured. It can be a DC battery or an additional diesel generator. Maintenance of such system is more complicated and expensive. It requires also additional measurement and control devices that monitor its parameters.

### *3.3.2.8 Filters and quality of the oil*

Oil filtering in case of hydrodynamic bearings is very important since the operational oil film thickness is usually not larger than *20 – 40 μm*(*depending on the operational conditions*). It always ensured that both sliding surfaces separated, either due to hydrodynamic or hydrostatic film action. Thus, the wear can occur only if particles larger than oil film thickness are between the bearing parts. Especially bidirectional bearings have low load carrying capacity that results in low oil film thickness. In order to prevent scratching of the sliding surface it is necessary to assure that particles in the oil have smaller dimensions than the minimum oil film thickness. Because of these reasons lubricating oil has to fulfill the following classes of the contamination by the solid particles: SAE class 4 or NAS 1638 class 7 or ISO class --/16/13.

Remark: Additionally, in all classes, particles larger than $150\ \mu m$ are not allowed and water amount cannot exceed $200\ ppm$ ($2\ cm^3\ H_2O\ /\ 1m^3 oil$).

Water and solid particles accelerate the ageing of the oil since they cause additional wear and corrosion. A properly designed oil filtering system has to fulfill the following tasks:

- Filter solid particles of oil.
- Avoid interruptions between maintenance services.
- Assure high operational reliability.
- Increase utilization time of the oil.
- Maintain parameters of the oil.
- Avoid ageing of the oil.
- Enable easy maintenance.
- Increase durability of the hydraulic system components (*decrease of wear*).

## 3.4 THRUSTBEARING DESIGN CALCULATIONS

### 3.4.1 Hydrodynamic lubrication

The load carrying capacity of the hydrodynamic bearing is a function of the hydrodynamic pressure distribution in the oil film.

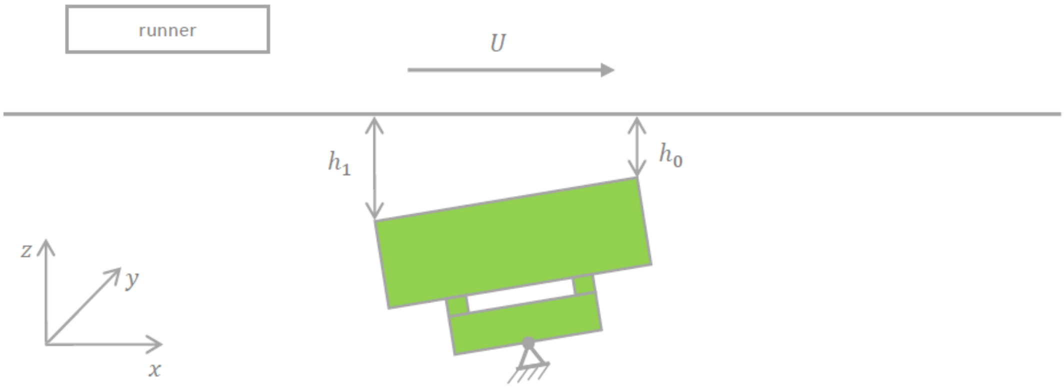



*Figure 3-16Hydrodynamic oil film coordinates system*

In general, case the oil flow described with the following equations:

### 3.4.2 Navier-Stokes Equation:

$$\rho\ \frac{d\boldsymbol{v}}{dt} = -grad(p) + \eta\ \Delta\boldsymbol{v} + S$$

This is a vector equation and written in form of three scalar equations for each spatial direction:

*x direction:*

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) =$$

$$= -\frac{\partial p}{\partial x} + \eta\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) + 2\frac{\partial \eta}{\partial x}\cdot\frac{\partial u}{\partial x} + \frac{\partial \eta}{\partial y}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) + \frac{\partial \eta}{\partial z}\left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}\right)$$

(3.1)

*y direction:*

$$\rho\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right) =$$

$$= -\frac{\partial p}{\partial y} + \eta\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + \frac{\partial \eta}{\partial x}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) + 1\frac{\partial \eta}{\partial y}\cdot\frac{\partial v}{\partial y} + \frac{\partial \eta}{\partial z}\left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)$$

(3.2)

*z direction:*

$$\rho\left(\frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) =$$

$$= -\frac{\partial p}{\partial z} + \eta\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) + \frac{\partial \eta}{\partial x}\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right) + \frac{\partial \eta}{\partial y}\left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right) + 2\frac{\partial \eta}{\partial z}\cdot\frac{\partial w}{\partial z}$$

(3.3)

### 3.4.3 Continuity Equation (For Incompressible Flow):

$$div(\boldsymbol{v}) = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

### 3.4.4 Energy Equation:

$$\rho\ c\ \frac{dT}{dt} = \lambda\ \Delta T\ + \eta\ \Phi \tag{3.4}$$

$$\Phi = 2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}\right)^2 \tag{3.5}$$

### 3.4.5 Heat Transport Equation:

$$\dot{q} = -\int_{(A)} \lambda\ \frac{\partial T}{\partial n}\ dA \tag{3.6}$$

### 3.4.6 State Equation:

$$\eta = \eta(T, p) \tag{3.7}$$

These 7 equations describe the problem of the hydrodynamic lubrication for 7 unknowns: $u$, $v$, $w$, $p$, $\eta$,$\dot{q}$ and $T$ thus the mathematical description of the bearing operating parameters in four dimensions ($x$, $y$, $z$, $t$) is possible.

Additionally, the simulation model used in order to model turbulent flow around the bearing pad. Turbulence models divided into following groups:

- Zero equation (*algebraic*) models.
- One equation models.
- Two equation models.
- Reynolds stress models.

Each of these models adds certain equations to the prior presented system of 7 equations and calculated additional parameters of the flow. Nowadays the most common in use are two equations turbulence models like($k$ - □) or ($k$ – $\omega$). These models add two separate transport equations to the previously defined 7. For example, in ($k$ –$\omega$)*turbulence* model first of the two additional unknowns is kinetic energy per unit mass of fluid arising from the turbulent fluctuations in velocity around the averaged velocity $k$. The second unknown is the frequency of large eddies $\omega$.

### 3.4.7 Generalized Reynolds Equation

Reynolds presented his theory of the hydrodynamic lubrication in 1886. This approach treats the hydrodynamic lubrication in a simplified way. The most important simplifications are:

- Laminar and incompressible flow assumption,
- Pressure gradient through the oil film thickness $\frac{dp}{dh} = 0$.
- Isothermal flow assumption,
- Calculation without influence of the pad and runner deformations.

With these assumptions Reynolds evaluated *2D* equation (*on x –y plane*) from the Navier-Stokes set of equations:

$$\frac{\partial}{\partial x}\left(\frac{\partial h^3}{\eta}\cdot\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial h^3}{\eta}\cdot\frac{\partial p}{\partial y}\right)=6\left[(U_1+U_2)\frac{\partial(\rho h)}{\partial x}+2\frac{\partial(\rho h)}{\partial t}\right] \tag{3.8}$$

*Reynolds* equation allows for the calculation of the hydrodynamic pressure profile for the infinite width of the bearing pad ($B \rightarrow \square$). So the lateral leakages neglected and overall load carrying capacity is overestimated.

One of the most significant disadvantages of the Reynolds equation solution is assumption of the constant temperature in the oil film. In a real bearing, there are temperature variations that have an influence on the viscosity of the oil.

Another disadvantage is the assumed oil film geometry. The bearing surfaces assumed flat. Therefore, there is no possibility to take into account deformations of the bearing elements.

Nowadays this solution does not have any practical importance but is very important from historical point of view. Reynolds equation solved numerically in order to take into account additional effects like more complicated oil gap geometry or variable viscosity in three dimensions.

## 3.5 CURRENTLY MODELED EFFECTS

### 3.5.1 Steady State Calculation

Steady state hydrodynamic calculation tools find the solution (*operational point*) for given constant operational parameters and are, until now, the most common approach to bearing calculations. Such an approach consequently been used for the bearing calculation since *1950 has* and is currently recognized as the standard calculation method. However, for large hydrodynamic bearings thermal inertia of the structure may be so huge that the bearing never reaches the equilibrium state. The operational parameters can change before the thermal balance occurs.

Currently used calculation methods for hydrodynamic lubrication divided into following groups:

- HD (Hydrodynamic).
- THD (Thermo-Hydro-Dynamic).
- TEHD (Thermo-Elasto-Hydro-Dynamic).

In order to simulate, in a realistic way, hydrodynamic bearing operational conditions it is necessary to take into account several physical phenomena. The most obvious part of the model is the hydrodynamic film, which most commonly calculated with the use of finite differences or finite elements methods that solve generalized *Reynolds* equation for the flow and energy equation for heat generation. This is *HD (Hydrodynamic)* solution with isothermal flow assumption. Currently used codes take into account oil heating and variable viscosity (*as a function of temperature*) and thus they commonly called *THD (Thermo-Hydro-Dynamic)*. In this group of codes, temperature field can be treated in either 2 or 3 dimensional ways. In the first case, temperature may vary only along the length and width of the oil film but remains constant through the oil film thickness. The second approach allows for variations of the temperature field in all three dimensions.

More advanced codes, so called *TEHD (Thermo-Elasto-Hydro-Dynamic)* use different methods to calculate thermal and elastic deformations of the bearing components (*both pad and runner or pad only*). Deformations calculated either analytically or numerically (*also with the use of commercial software*). Recent investigations showed that for large bearings runner deformations have to take into account due to large influence on the fluid film and consequently overall bearing.

### 3.5.2 Heat Transfer Effects

The easiest approach to the heat transfer analysis in the oil film is the assumption of the isothermal oil flow. The temperature value assumed either a priori or evaluated based on the overall heat balance in the whole bearing. This approach does not however take into account variations of the viscosity as a function of temperature. So-called effective viscosity is assumed once (*for each iteration*) for the whole oil film area. Constant viscosity assumption does not give good agreement with measurements since it has an influence on the tilt angle of the pad.

Assumption of the adiabatic oil flow means that both sliding surfaces are adiabatic boundaries and no heat transfer through them is possible. This assumption made in all bearing calculation programs that assume two-dimensional temperatures filed in the oil film. Constant oil film temperature means automatically no heat flow in this direction. The most advanced calculation models take into account thermal flow through the sliding surfaces of the pad and the runner.

Realistic analysis of the heat transfer in the bearing structure is a complex problem. Heat transported by the means of conduction and convection.

Conduction of heat described by the *Fourier* law, which for one-dimensional transient case written in form of the heat diffusion equation:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{a}\frac{\partial T}{\partial t} \tag{3.9}$$

Where *a* is thermal diffusivity and defined as:

$$a = \frac{\lambda}{\rho\ c} \tag{3.10}$$

Convectional heat transfer calculated by:

$$q = \alpha \cdot (T_{wall} - T_{bulk}) \tag{3.11}$$

In case of the convectional heat transfer rate the main difficulty is estimation of the convection coefficient $\alpha$. Just in one-dimensional case, it is a function of several parameters:

$$\alpha = f(\lambda, x, \rho, c, \mu, u_\infty)$$

Convection coefficient depends on the relation between thicknesses of the velocity and temperature boundary layers. The dimensionless Reynolds number can describe the velocity field of the fluid, the velocity boundary and character of the flow (either laminar or turbulent) layer

$$Re = \frac{\rho \cdot U \cdot L}{\mu} = \frac{inertial\ forces}{viscous\ forces} \tag{3.12}$$

Where the *Reynolds* number defined as:

$$Re = \frac{R^2 \cdot \omega}{\nu} \qquad (3.13)$$

According to the boundary layer theory, the velocity boundary layer may have significant thickness.

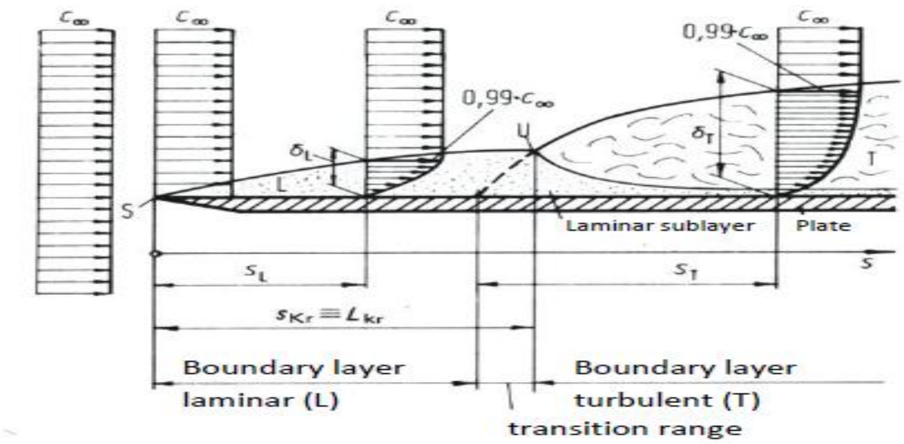



*Figure 3-17Velocity boundary layer thickness*

According to the one dimensional boundary layer theory its thickness is a function of the length and estimated according to the following formulas, respectively for laminar and turbulent flow regime:

$$\delta_L = 5 \cdot \sqrt{\frac{s_{cr} \cdot L_{cr}}{Re_{cr}}} \sim s_L^{0.5} \qquad (3.14)$$

$$\delta_T = 0.37 \cdot \sqrt[5]{\frac{s_{cr}^4 \cdot L_{cr}}{Re_{cr}}} \sim s_T^{0.8} \qquad (3.15)$$

The growth of the turbulent velocity boundary layer is much faster (□$_T \sim S_T^{0.8}$) than that of the laminar one (□$_L \sim S_L^{0.5}$).

For the disc with outer radius, which rotates in the oil the thickness of the turbulent velocity boundary layer, evaluated with a following form (*verified experimentally*):

$$\delta = \frac{0.526}{\sqrt[5]{Re}} \cdot R \qquad (3.16)$$

Critical Reynolds number (*laminar – turbulent transition*) is in this case:

$$Re_{cr} \approx 3 \cdot 10^5 \qquad (3.17)$$

For the thrust bearings used for hydro applications, this velocity boundary layer has thickness of several tens of millimeters. In the Figure (3.18), the boundary layer thickness shown for a given range of rotational speeds and radii. Curves are limited to the turbulent range only since only this flow regime seems to exist in the real bearing housings (*between the pads*). The development of the boundary velocity layer depends on the distance between the pads. If it is shorter than the defined sum of the lengths ($S_{cr}$ + $S_T$) in Figure [3.18] then its thickness does not reach the fully developed value.

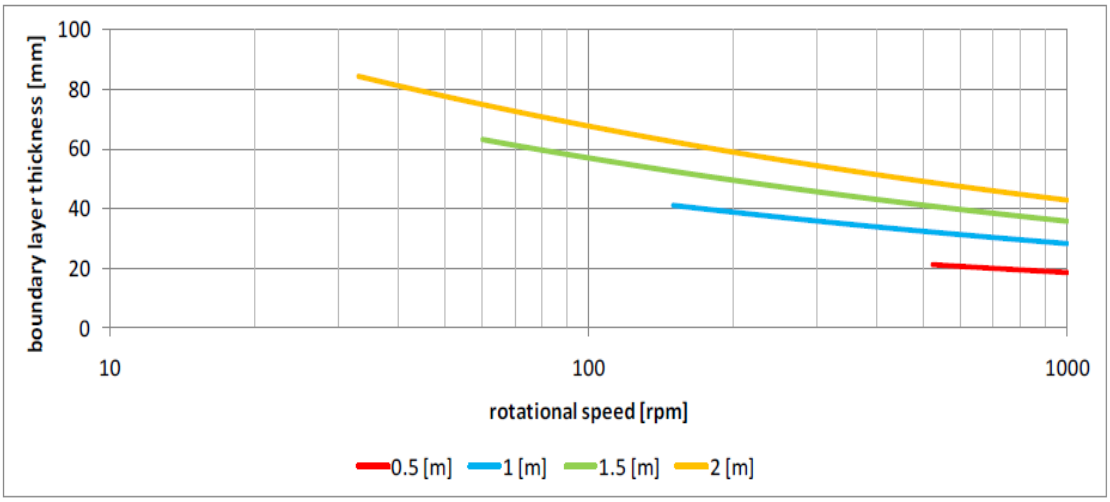



*Figure 3-18Turbulent velocity boundary layer for different runner radii*

According to the ratio between thicknesses of the velocity and temperature boundary layers idefined by dimensionless *Prandtl* number *Pr*.

$$\mathrm{Pr} = \frac{\nu}{a} = \frac{c \cdot \mu}{\lambda} = \frac{viscous\ diffusion\ rate}{thermal\ diffusion\ rate} \tag{3.18}$$

*Pr* is a physical property of the fluid and varies from $10^{-2}$for liquid metals up to$10^5$ for liquids with very complex molecular structure e.g. oils made of long-chain hydrocarbons.

For ISO VG46oil at the temperature 40 °C:

$$\mathrm{Pr}_{VG46} = \frac{2113.5 \cdot 0.0398}{0.123}$$

$$\mathrm{Pr}_{VG46} = 683.88$$

Finally, the dimensionless Nusseltnumber Nu, which is a function of the two previously described dimensionless numbers Re, and Pr. can define the convection.

$$Nu = f(Re, Pr)$$

$$Nu = \frac{\alpha \cdot L}{\lambda} = \frac{convective\ heat\ transfer}{conductive\ heat\ transfer} \qquad (3.19)$$

$L\ [m]$ is the characteristic length of the body, e.g. length of the plate, diameter of the cylinder, etc.

The *Nusselt* number is inversely proportional to the thickness of the thermal boundary layer:

$$Nu = \frac{L}{\delta_t} \qquad (3.20)$$

For example, for laminar incompressible case of the flow *Nu* is defined in the following way:

$$Nu = 0.332\, Re^{1/2} Pr^{1/3} \qquad \text{for} \qquad 0.6 \leq Pr \leq 50 \qquad (3.21)$$

$$Nu = \frac{0.3387\, Re^{1/2} Pr^{1/3}}{(1+(0.0468/Pr)^{2/3})^{1/4}} \qquad \text{for} \qquad Pr > 100 \qquad (3.22)$$

### 3.5.3 Warm Oil Carryover Effect

The bearing pads arranged in such way that the outlet of the preceding pad is located close to the inlet of the following one. The space between the pads is usually as small as possible in order to increase the area of the bearing. The surface covered by pads is approximately equal to 80 % of the runner area. Due to this reason, warm oil that comes out from the preceding pad goes directly into the inlet of the following one. This effect increases inlet temperature $T_{in}$ in reference to the cold oil temperature $T_{cold}$ (see Figure 3.19). Estimation of the oil-film inlet temperature has significant influence on the remaining part of the calculation since it is a sort of reference level for the oil film. Appropriate estimation of this temperature has influence on the overall accuracy of calculation.

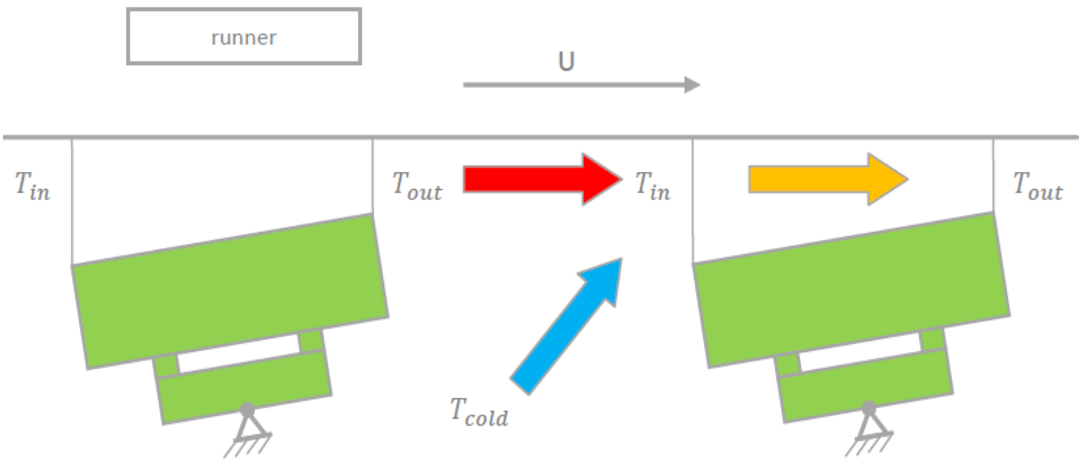



*Figure 3-19Warm oil carry over phenomena*

The ratio between $T_{out}$ and $T_{cold}$ depends on many factors. Among them we can distinguish the most important ones:

- Distance between the pads.
- Sliding velocity.
- Way of delivering the cold oil from the cooler LEG (*Leading Edge Groove*, *distributors*, etc.).
- Specific load of the bearing, tilt angle, pivot position.

*Ettles* introduced warm oil mixing factor *k* that allows estimating oil film-inlet temperature $T_{in}$ with the use of the iterative formula (3.23).

$$T_{in} = T_{cold} \cdot \left(\frac{1-k}{1-0.5\,k}\right) + T_{out} \cdot \left(\frac{0.5\,k}{1-0.5k}\right) \tag{3.23}$$

The mixing factor *k* estimated with the use of the diagram in the Figure (*3.20*) which based on measurements. In this analysis, the only parameters taken into account were sliding speed and distance between the pads. Oil-film inlet temperature $T_{in}$ has to be estimated in an iterative way since oil film outlet temperature $T_{out}$ also depends on it.

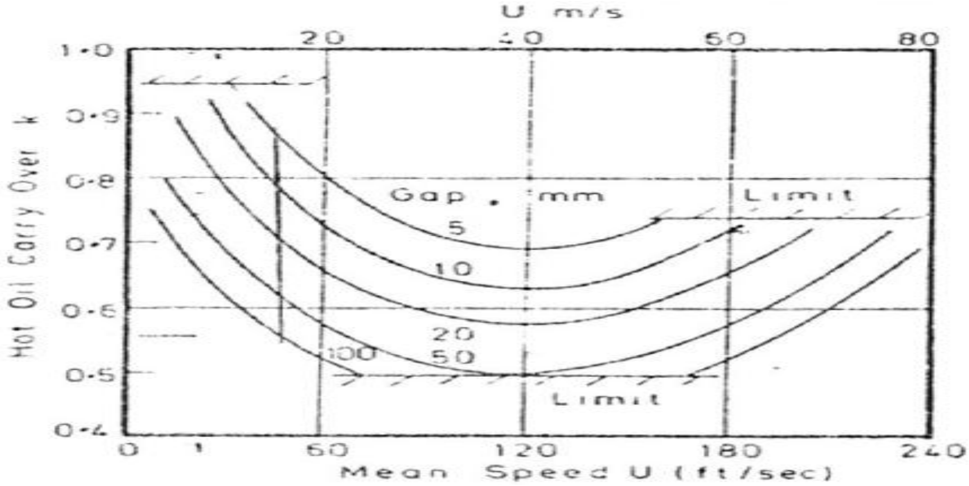



*Figure 3-20Warm oil carryover factor*

From this diagram (*Figure 3.20*) one can notice that the distance between the pads and the sliding speed are the key components for the warm oil carry over effects. Thus, the mixing has to consider for each bearing design separately.

There were attempts to decrease the hot oil carry over by mounting scrapers on the outlet of the pad that "*scratch*" the warm oil layer from the runner surface. The thermal boundary layer on the runner is very thin and does not exceed *0.5 mm* (*high Prandtl number for the oils*). These "*devices*" may work properly after mounting them in the bearing housing but wear of the sliding surface may reduce their efficiency after certain time of operation. They can also produce additional power loss and cause warming up the boundary layer on the sliding surface. The runner surface itself is also warm (*70 – 80 °C*) and due to rotation has a constant (*or almost constant*) temperature in the tangential direction so it causes warming up of the oil layer also between the pads and behind the scraper.

Warm oil carries over effect can be potentially reduced by directed lubrication devices mounted in the space between the pads but according to the literature and common knowledge these attempts do not allow influencing the operational parameters of the bearing in a significant way. The main reason of the lack of the influence is the oil bath in the bearing housing. Therefore, the oil flow from the injecting hole cannot "*reach*" the sliding surface of the runner and there is no influence on the mixing of the warm oil between the pads. Thrust bearings of large hydro generators immersed in oil bath in order to assure their safe operation and to fulfill contractual requirements.

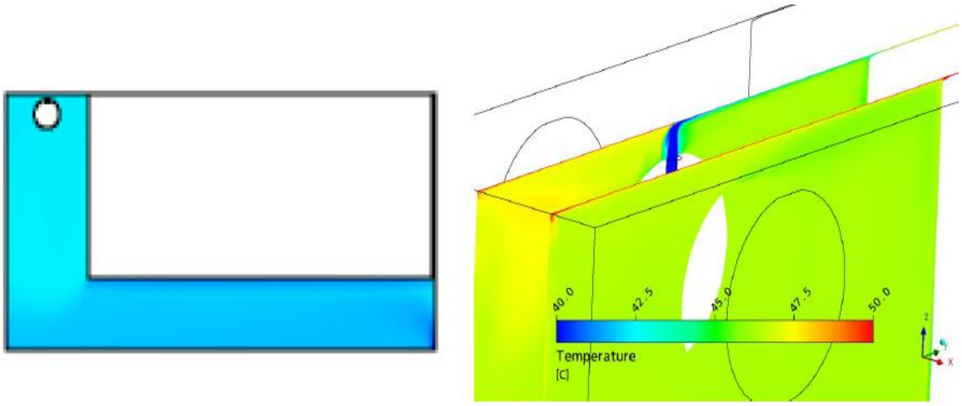



*Figure 3-21Oil injecting on the runner sliding surface, CFD simulation*

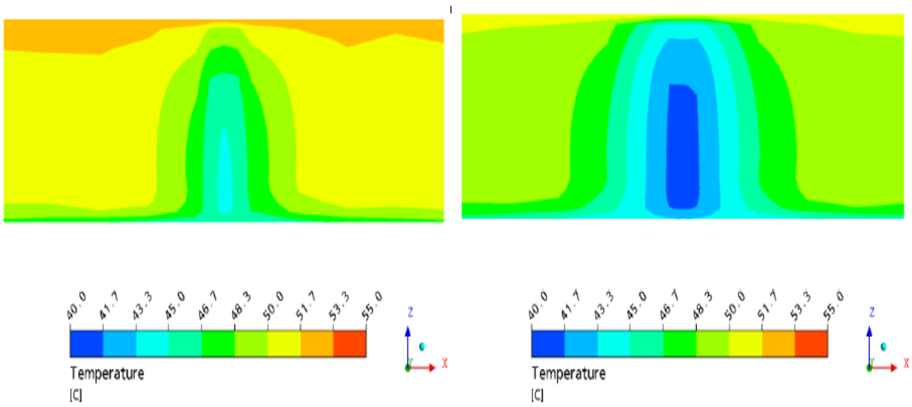



*Figure 3-22the oil flow (330 and 660 l/min) on the oil film inlet temperature distribution*

In (*Figure3.22*) one can see the influence of the injected oil flow on the oil-film inlet temperature. Increase of the injected oil flow from *330 to 660 l/min* results in change of the average oil-film inlet temperature from *49.4 to 46.2 °C*. There remains always a thin layer of the warm oil close to the runner-sliding surface. Also in the radial direction influence on the oil-film inlet temperature has only local character. Space between injection points remains almost not influenced at all.

The aim of all these attempts is to decrease the oil-film inlet temperature – to make it possibly close to the cold oil temperature. Thus, the best way to reduce warm oil carry over effects is "*dry*" bearing housing with oil presence only in the oil film. Together with directed lubrication methods, one can assure delivery of the cold oil directly to the oil film inlet without negative influence of any mixing effects.

The above results lead to an idea that oil supply realized not with the use of injection holes but with sort of an oil knife (*or several knives*). Such linear oil injection would deliver cold oil not only on several spots on the leading edge but on its radial length.

## 3.6 DEFORMATIONS OF PAD AND RUNNER

It well known that the pad elastic and thermal deformation has essential influence on the bearing performance. *Tanaka* indicated that in order to predict the load carrying capacity of the thrust hydrodynamic bearing the thermal and elastic deformation of the runner has to take into account. Otherwise, the oil-film thickness profile could be overestimated. For this reason, it seems necessary to include this effect in a modern calculation tool for the hydrodynamic thrust bearing. In (*Figure 3.23*) visual demonstration of the expected shape of the deformations and tilting of the pad shown (*for counterclockwise direction of rotation*).

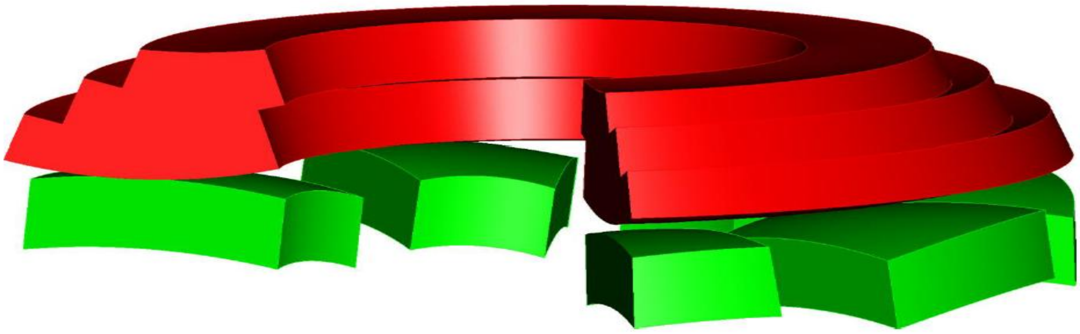


*Figure 3-23Graphical representation of the possible deformations of the bearing elements*

Both analytical and numerical (FEM) approaches used in order to estimate deformations of the bearing elements.

The analytical estimation of the deformations assumes their shape (*sphere for the pad and torus for the runner surface*). Since elastic deformation is very small, the deformation based mainly on distribution of temperature field in the bearing elements. The main unknown for such calculation are the thermal boundary conditions. Many researchers propose their own assumption of convection coefficients on the walls of the pad and runner that not connected with the oil film. The assumed values vary in a certain range and may have an influence on the obtained temperature field distribution and further on the deformation of the element. Until now, it is not clear what the real conditions on these surfaces are. According to these coefficients of free walls of the pad may vary within the range ($1\text{-}10\ KW / m^2\ k$) depending on the bearing arrangement. Additionally, it is very probable that these coefficients have different values at different walls of the bearing pad.

Runner boundary condition is also essential for the appropriate estimation of the thermal deformation of its sliding surface. This input parameter has a significant influence on the temperature distribution and further on the resulting deformation. The estimation of the temperatures on the cylindrical walls that rotate in oil and air is very difficult. It may vary for different bearings due to design and operational conditions like rotational speed or oil temperature in this area. The estimation of these temperatures (*boundary conditions*) most easily did with the use of the measurement data.

Until now, many researchers have not been taking deformation of the runner surface into account regardless the fact that it may have significant influence on the obtained results of the simulation. Deformations play a significant role for large sized bearings like the ones made for vertical hydro generators. *Ettles* showed that there is a significant influence of the scale factor on the bearing deformations. He demonstrated that Linear a increase of the bearing dimensions leads to more severe operating conditions at the same specific load level, which means that operational limits of the large hydrodynamic bearings are much lower than for smaller ones (*see Figure 3.24*)

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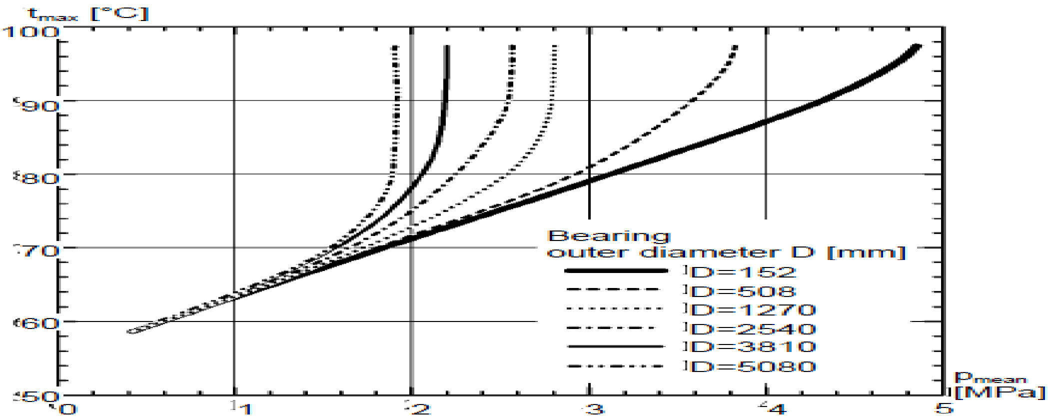



*Figure 3-24Scale factor influence on the bearing load carrying capacity*

These deformations are even more important for centrally pivoted pads, which shown by *Raimondi*. He showed that centrally pivoted pads need certain amount of thermal deformation in order to generate load carrying capacity.

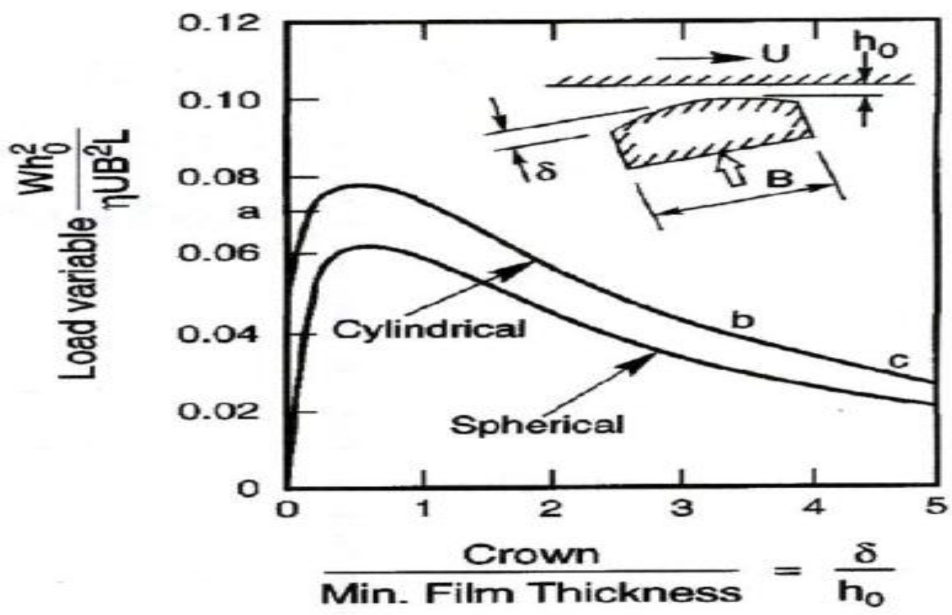



*Figure 3-25Influence of the relative pad deformations on the load carrying capacity of the centrally pivoted pad*

The thermal deformations of the pad depend on its size. According to the following formula, one can see that in order to obtain similar thermal deflection for larger bearings it is necessary to increase the pad thickness not linearly with the geometrical scale factor. For this reason, larger bearings have lower allowable operating specific pressures than the smaller bearings.

$$\delta_T = \frac{B^2 \cdot \lambda \cdot (T_{max} - T_{min})}{8 \cdot H} \qquad (3.24)$$

For optimum load carrying capacity (*marked by 'a'*) (*Figure 3.25*) of a bidirectional thrust bearing the $\square_T$ should be approximately equal to the minimum oil film thickness $h_0$. As the$h_0$ may vary for different operational conditions, it is necessary to assure during design of the bearing that this ratio remains close to unity. At the same time, one can notice a rapid decrease of the load carrying capacity of the pad for higher values of the deformations. This may occur for example during startup when the sliding surface reaches its operating temperature very quickly and the backing of the pad remains cold (*marked by 'b'*). The process may become also unstable since the decreased minimum oil film height leads to higher temperatures and higher thermal distortion of the pad (*marked by 'c'*). Finally, the seizure and plastic flow of lining material may occur as in the case shown in (*Figure 3.26*). The damaged area located in the middle of the pad indicates that the pad distorted heavily during operation.

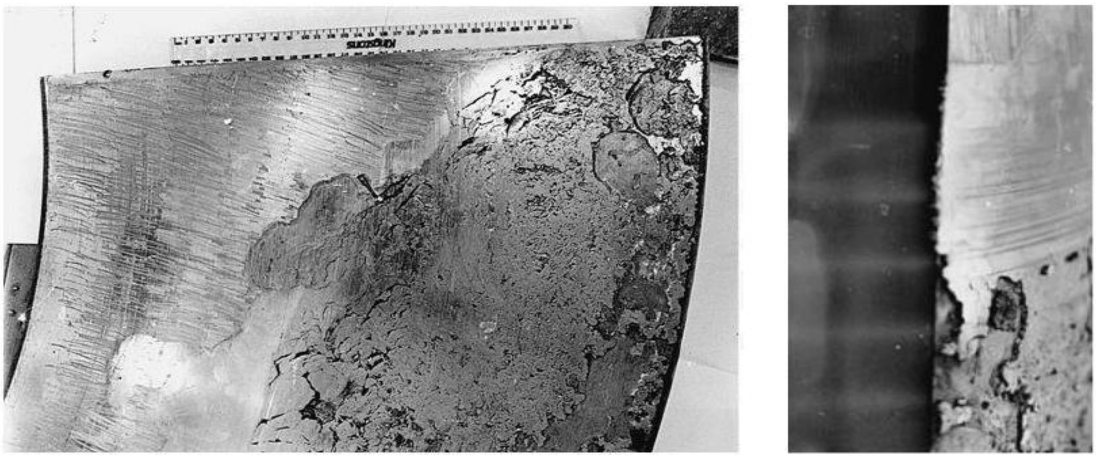


*Figure 3-26 Damaged sliding surface due to overload and loss of the load carrying capacity (wear)*

# 4 DAM HEIGHTENING EFFECT ON STATION THRUST BEARING

## 4.1 INTRODUCTION

A hydrodynamically lubricated bearing is a bearing that develops load-carrying capacity by virtue of the relative motion of two surfaces separated by a fluid film. The processes occurring in a bearing with fluid film lubrication better understood by considering qualitatively the development of oil pressure in such a bearing.

## 4.2 GENERAL THRUST BEARING THEORY

Solutions of the Reynolds equation for real bearing configurations usually obtained in approximate numerical form. Analytical solutions are possible only for the simplest problems. By restricting the Row to two dimensions, say the x-z plane, analytical solutions for many common bearing forms become available.

The quantitative value of these solutions is limited, since flow in the third dimension y, which known as "side leakage," plays an important part in fluid film bearing performance. The two-dimensional solutions have a definite value, since they provide a good deal of information about the general characteristics of bearings, information that leads to a clear physical picture of the performance of lubricating films. Besides neglecting side leakage, another simplification achieved by neglecting the pressure and temperature effects of the lubricant properties, namely, viscosity and density. The viscosity of common lubricants is particularly sensitive to temperature, and since the heat generated in hydrodynamic bearings is often considerable, the limitation imposed by this assumption is at once apparent.

Introducing variable viscosity and density into the analysis creates considerable complications, even in the case of two-dimensional flow. The temperature rise within the film calculated if it assumed that the lubricant (the adiabatic assumption) carries all the heat produced by the viscous action away. Several simplifying approximations have to make before a mathematical description of the fundamentals underlying mechanisms derived.

All the simplifying assumptions necessary for derivation the Reynolds equation summarized in [*Table 4.1*]

## 4.3 VISCOUS FLOW:

1. Making a fluid flow faster requires a greater force.
2. The flow involves irreversible change, and the work done appears as heat in the liquid.
3. A liquid becomes less viscous as its temperature raised.
4. The viscosity of a liquid in non-conformal conjunctions generally increases as the pressure increases.

$$\boldsymbol{\mu = \frac{\mathcal{F}}{w}} \qquad (4.1)$$

Where:

μ ≡ coefficient of friction

$\mathcal{F}$ ≡ Friction force (N)

W ≡ normal applied load

The differential equation governing the pressure distribution in fluid film known as *Reynolds* equation.

General Reynolds equation:

$$\mathbf{0} = \frac{\partial}{\partial x}\left(-\frac{\rho h^3}{12\mu}\frac{\delta p}{\delta x}\right) + \frac{\delta}{\delta y}\left(-\frac{\rho h^3}{12\mu}\frac{\delta p}{\delta x}\right) + \frac{\delta}{\delta x}\left(\rho h\frac{(u_a+u_b)}{2}\right) + \frac{\delta}{\delta y}\left(\rho h\frac{(v_a+v_b)}{2}\right) + \rho(w_a - w_b) - \rho u_a\frac{\delta h}{\delta x} - \rho v_a\frac{\delta h}{\delta y} + h\frac{\delta p}{\delta t} \qquad (4.2)$$

Derived from N.S and continuity equation.

- The first two terms (*Poiseuille*) describe the net flow rates due to pressure gradients.
- Third and fours (*Couette*) describe the net flow rates due to surface velocities.
- Fifth to seventh terms describe the net flow rates due to squeezing motion.
- Last term describes the net flow rate due to local expansion.

It can be seen that the *Couette* term leads to three distinct actions, but the important action in our case (*thrust bearing with several pads*) is the physical wedge pressure generating mechanism.

## 4.4 PHYSICAL WEDGE TERM $[\rho(\frac{u_a+u_b}{2})]\frac{\delta h}{\delta x}$:

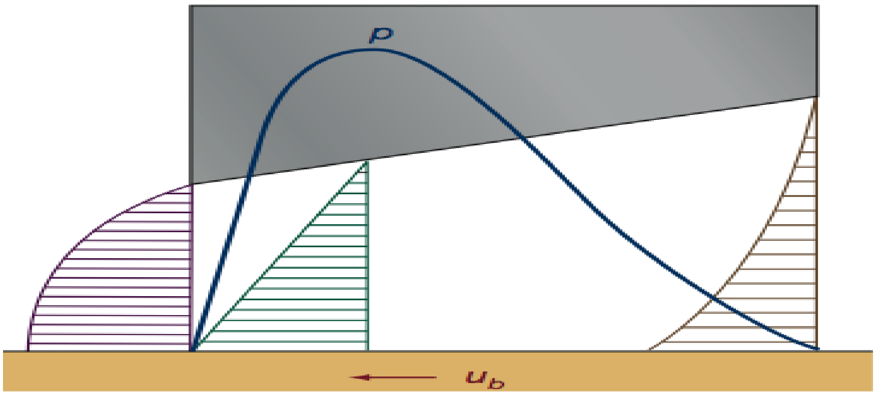



*Figure 4-1 physical wedge mechanism.*

The flow rate is proportional to the area of the triangle of height h and base u.

Flow continuity achieved only if a balancing pressure generated, and the thickness of lubricant film must decrease in the sliding direction.

-For only tangential motion, where:

$$w_a = u_a\frac{\delta h}{\delta x} + v_a\frac{\delta h}{\delta y} and w_b = 0$$

$$\frac{\delta}{\delta x}\left(\frac{\rho h^3}{N}\frac{\delta p}{\delta x}\right) + \frac{\delta}{\delta y}\left(\frac{\rho h^3}{N}\frac{\delta p}{\delta y}\right) = 12u_{\sim}\frac{\delta(\rho h)}{\delta x} + 12v_{\sim}\frac{\delta(\rho h)}{\delta y} \quad (4.3)$$

$$u_{\sim} = \frac{u_a + u_b}{2} = c$$

Equation above is applicable for Elasto-hydrodynamic bearings.

For hydrodynamic lubrication, the motion is pure sliding so ( $v = 0$), and if side leakage neglected (the pressure does not change in y direction):

$$\frac{\delta}{\delta x}\left(\frac{\rho h^3}{N}\frac{\delta p}{\delta x}\right) = 12u_{\sim}\frac{\delta(\rho h)}{\delta x} \quad (4.4)$$

Integrating this equation give:

$$\frac{1}{N}\frac{dp}{dx} = \frac{12u_{\sim}}{h^2} + \frac{C}{\rho h^3} \quad (4.5)$$

$$\frac{dp}{dx} = 0 when x = x_m \rho = \rho_m h = h_m$$

$$C = -12u_{\sim}\rho_m h_m$$

The integrated form of Reynolds equation with neglecting the variation of the density:

$$\frac{dp}{dx} = 12u_{\sim}\mu\frac{h-h_m}{2} \quad (4.6)$$

When considering various tangential and normal squeeze velocities (w), density is constant and neglected side leakage:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right) = \frac{u_a+u_b}{2}\frac{\partial h}{\partial x} + \frac{h}{2}\frac{\partial(u_a+u_b)}{\partial x} + w_a - w_b - u_a\frac{\partial h}{\partial x} \quad (4.7)$$

In Rosaries case:

$$u_a = w_a = 0 \quad , \qquad u_b = u_1 \quad , \quad w_b = 0$$

Then:

$$\frac{u_1}{2}\frac{dh}{dx} = \frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right) \quad (4.8)$$

Now analytically or numerically solution of the equation above for further analysis.

# 4.5 ANALYTICAL SOLUTION:

## 4.5.1 Assumptions:

*Table 4-1the simplifying assumptions necessary for derivation the Reynolds Equation*

| No | Assumption | Comment |
|---|---|---|
| 1 | Body forces are neglected | Always valid, since there are no extra outside fields of forces acting on fluid with an exception of magneto hydrodynamic fluids and their applications. |
| 2 | Pressure is constant through the film | Always valid, since the thickness of hydrodynamic is in the range of several micrometers. There might be some exceptions. |
| 3 | No slip at the boundaries | Always valid, since the velocity of the oil layer adjacent to the boundary is the same as that of the boundary. |
| 4 | Lubricant behaves as a Newtonian fluid | Usually valid with certain exceptions. |
| 5 | Flow laminar | Usually valid except large bearings. |
| 6 | Fluid inertia is neglected | Valid for low bearing speed or high loads . |
| 7 | Fluid density is constant | Usually valid for fluids when there is no much thermal exceptions. |
| 8 | Viscosity is constant through the generated fluid film. | Necessary to calculation, although is not true viscosity is not constant through the generated film. |

$\frac{dp}{dx} = 12u\mu\frac{h-h_m}{h^3}$(4.9)

The rotating shaft separated from the sector shaped bearing pads by a lubricant film. The load carrying capacity of bearing arises from the pressure generated by the geometry of the thrust plate over the bearing pads.

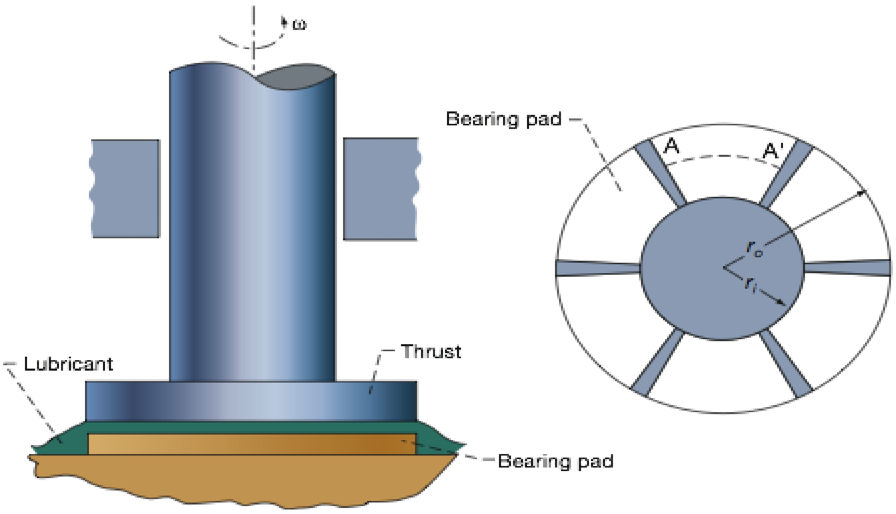



*Figure 4-2thrust bearing geometry.*

Total load carrying capacity:

$w_t = nw_z(r_o - r_i)$(4.10)

Where:

$n \equiv$ Pads number.

$w_z \equiv$ Normal load per unit width of one pad (N/m).

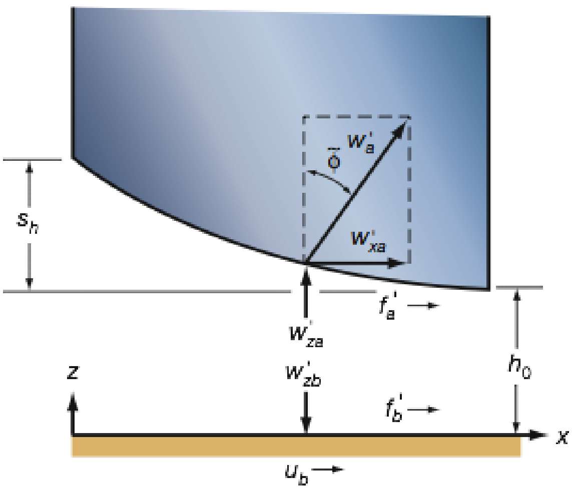



*Figure 4-3force component and oil film geometry in hydro-dynamically lubricated thrust bearing sector.*

The forces acting on the solids considered in two groups. The loads acting normal to the surfaces (*w*) and the viscous surface stress (*f in the x direction*).

Once the pressure obtained from Reynolds relations, the following are the force components acting on the solids:

$$\boldsymbol{w_{za} = w_{zb} = \int_0^l p dx} \quad (4.11)$$

$$\boldsymbol{w_{xb} = 0} \quad (4.12)$$

$$\boldsymbol{w_{xa} = \int_{h_o+s_h}^{h_o} p dh} \quad (4.13)$$

$$\boldsymbol{w_b = (w_{zb}^2 + w_{xb}^2)^{0.5}} \quad (4.14)$$

$$\boldsymbol{w_a = (w_{za}^2 + w_{xa}^2)^{0.5}} \quad (4.15)$$

$$\boldsymbol{\varphi = tan^{-1} \frac{w_{xa}}{w_{za}}} \quad (4.16)$$

## 4.5.2 Shear Forces per Unit Width:

$$\boldsymbol{f_b = \int_0^l (\tau_{zx})_{z=0} dx} \quad (4.17)$$

$$\boldsymbol{f_b = -\frac{w_{xa}}{2} - \int_0^l \frac{\mu u_b}{h} dx} \quad (4.18)$$

$$\boldsymbol{f_a = -\frac{w_{xa}}{2} + \int_0^l \frac{\mu u_b}{h} dx} \quad (4.19)$$

## 4.5.3 Static Equilibrium:

$$\boldsymbol{f_b + f_a + w_{xa} = 0} \quad (4.20)$$

$$\boldsymbol{w_{zb} - w_{za} = 0} \quad (4.21)$$

## 4.5.4 Power Loss for One Pad Is:

$$\boldsymbol{h_p = -f_b u_b = -f_b (r_o - r_i) u_b} \quad (4.22)$$

$$\boldsymbol{h_p = \rho q C_p (\Delta t_m)} \quad (4.23)$$

## 4.6 FIXED–INCLINE SLIDER BEARING:

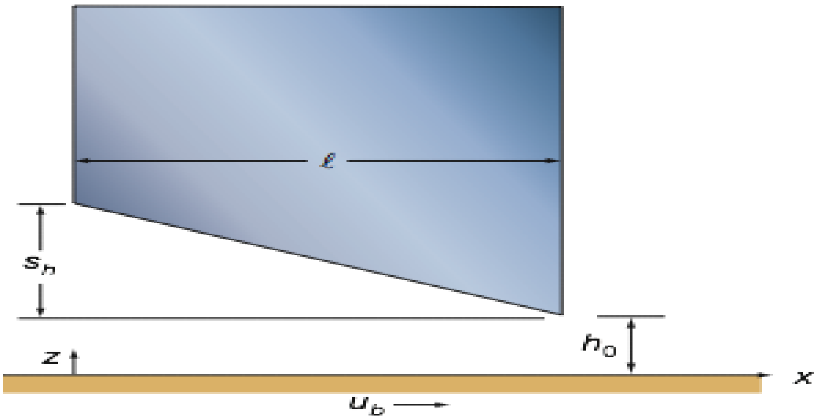



*Figure 4-4fixed-incline slider bearing.*

$$\frac{dp}{dx} = 6\mu u_b \frac{h-h_m}{h^3} \quad (4.24)$$

$$h = h_o + s_h\left(1 - \frac{x}{l}\right) \quad (4.25)$$

### 4.6.1 Dimensionless Terms:

$$P = \frac{ps_h^2}{\mu u_b l}, \qquad H = \frac{h}{s_h}, \qquad H_m = \frac{h_m}{s_h}, \qquad H_0 = \frac{h_0}{s_h}. \qquad X = \frac{x}{l}$$

### 4.6.2 Pressure Distribution:

$$P = \frac{6X(1-X)}{(H_0+1-X)^2(1+2H_0)} \quad (4.26)$$

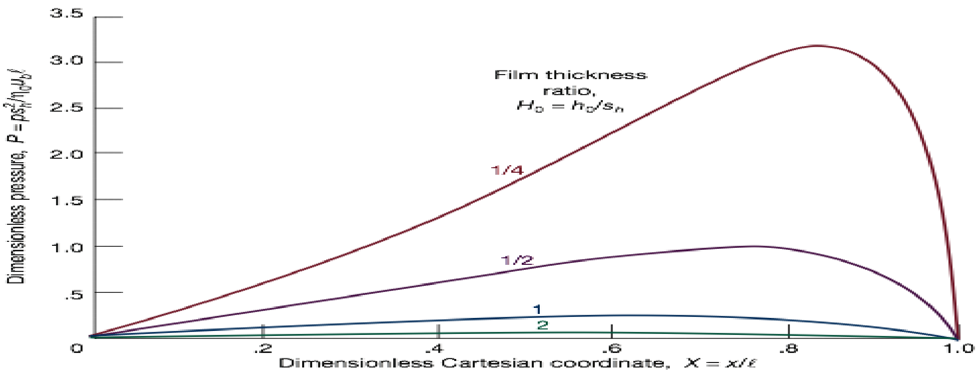



*Figure 4-5 pressure distribution*

$$P_m = \frac{3}{2H_0(1+H_0)(1+2H_0)} \quad (4.27)$$

### 4.6.3 Dimensional:

$$\boldsymbol{p_m} = \frac{\boldsymbol{3\mu u_b l s_h}}{\boldsymbol{2h_0(s_h+h_0)(s_h+2h_0)}}(4.28)$$

The shoulder height that produces Pm can be obtained by:

$$\frac{\boldsymbol{\partial p}}{\boldsymbol{\partial s_h}} = \boldsymbol{0}(4.29)$$

$$(\boldsymbol{s_h})_{\boldsymbol{opt.}} = \sqrt{\boldsymbol{2h_0}}(4.30)$$

### 4.6.4 Normal Load Component:

$$W_z = 6\ln\left(\frac{H_0+1}{H_0}\right) - \frac{121}{+2H_0}(4.31)$$

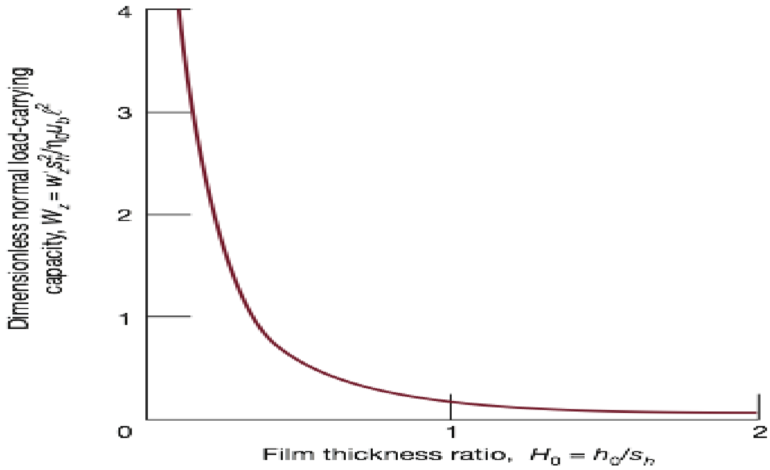



*Figure 4-6effect of film thickness ratio on normal load carrying capacity.*

### 4.6.5 Shear Force Component:

$$F_b = 4\ln\left(\frac{H_0}{H_0+1}\right) + \frac{61}{+2H_0}(4.32)$$

$$F_a = 2\ln(\frac{H_0}{H_0+1}) + \frac{61}{+2H_0}(4.33)$$

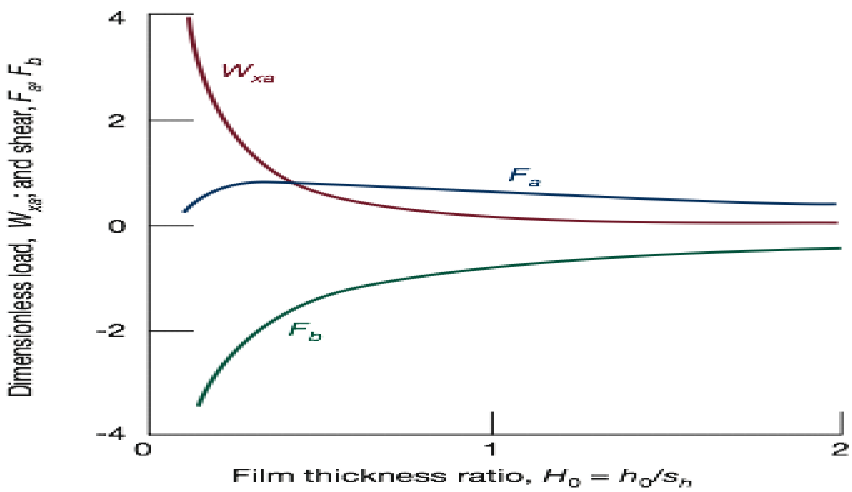



*Figure 4-7effect of film thickness ratio on force component.*

### 4.6.6 Friction Coefficient:

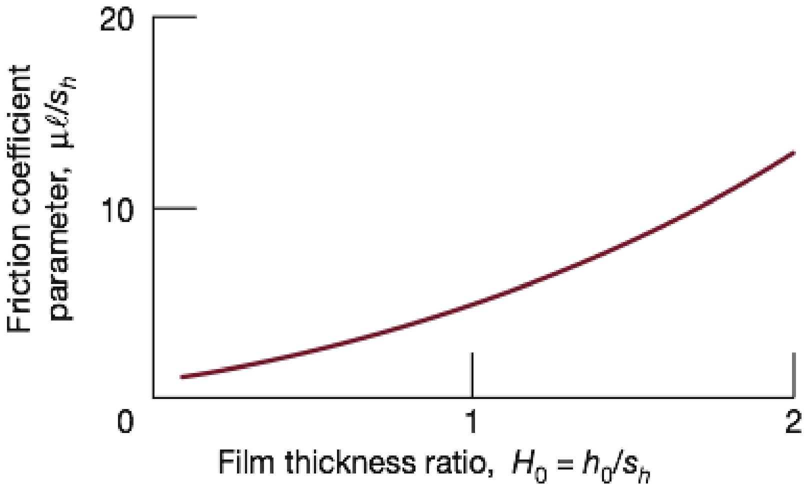



*Figure 4-8effect of film thickness ratio on friction coefficient parameter.*

### 4.6.7 Power Loss and Temperature Rise:

$$h_p = -f_b u_b = -f_b(r_o - r_i)u_b (4.34)$$

$$H_P = -4\ln\left(\frac{H_o}{H_0+1}\right) - \frac{61}{+2H_0}(4.35)$$

All the heat produced by viscous shearing assumed to carry away by the lubricant.

$$\Delta t_m = \frac{h_p}{\rho_0 q_x C_p} = \frac{2u_b l \mu_0}{\rho_0 C_p s_h^2}\frac{H_p}{Q}(4.36)$$

$\rho_0$ = constant lubricant density kg/m$^3$

$q_x$ =volume flow rate per unit width in sliding m$^3$/s

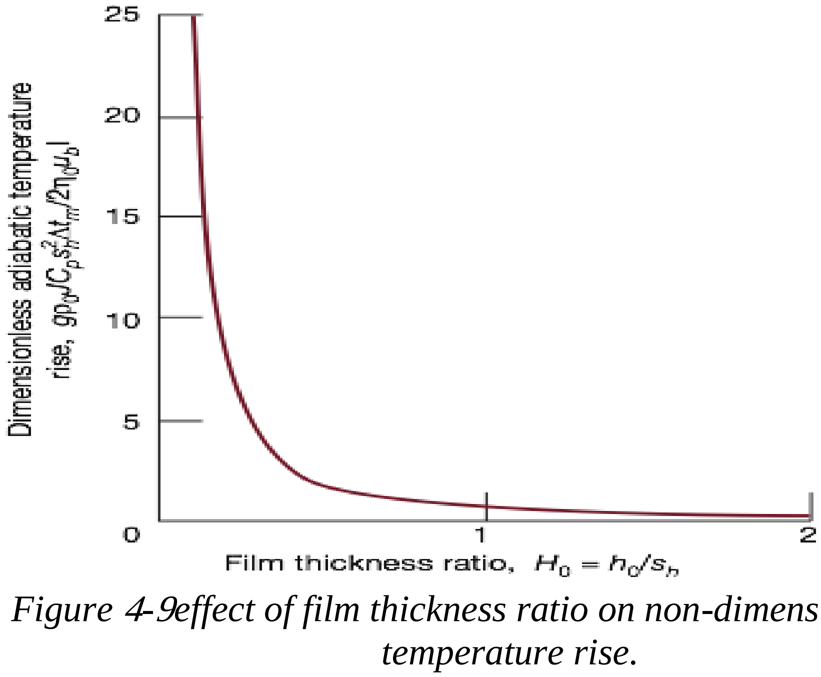



*Figure 4-9effect of film thickness ratio on non-dimensional adiabatic temperature rise.*

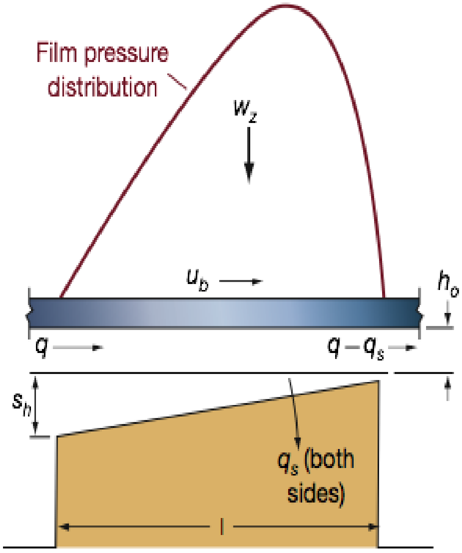



*Figure4.10.1:*

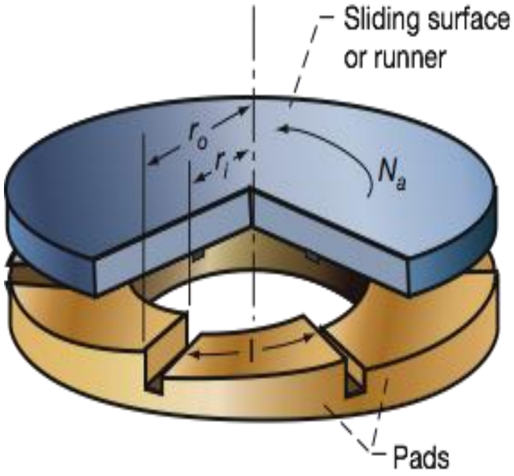



*Figure 4.10.2:*

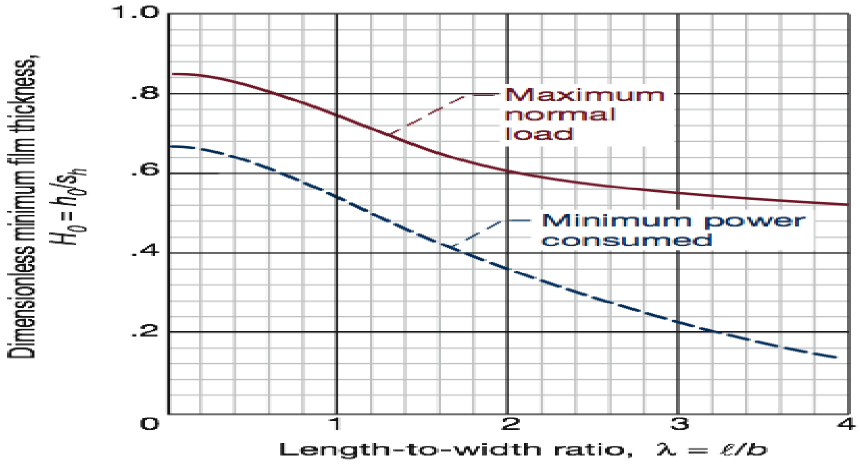



*Figure 4-10-3Chart for determining minimum film thickness corresponding to maximum load or minimum power loss for various pad proportions - fixed-incline-pad bearings. [From Raimondi and Boyd (1955)*

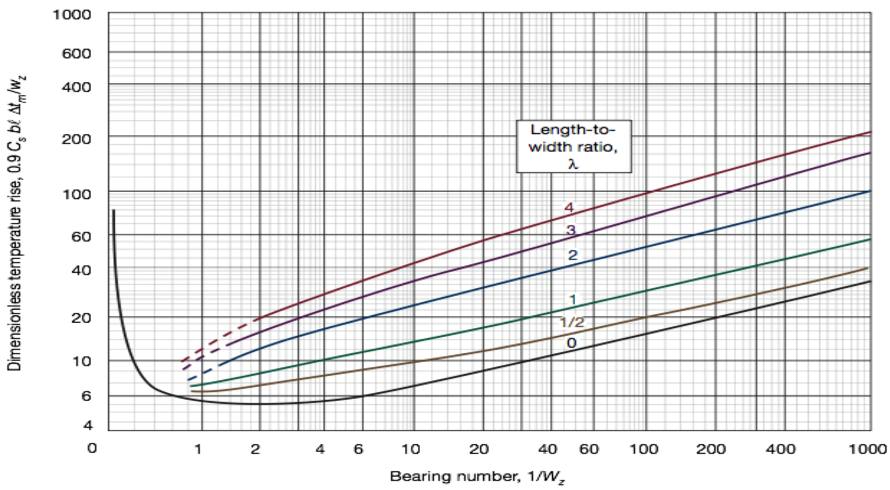



*Figure 4-11chart of determining dimensionless temperature rise due to viscous shear heating*

# 4.7 THRUSTBEARINGOVERHEATING IN ROSEIRIS POWER PLANT

## 4.7.1 Statement of The Problems

Failures were associated with high temperatures in the pads, temperature of cooling water raised also. A similar problem had occurred in the other two machines of the plant [*Unit 6 out of service*].

Sudanese Hydro Generation Company (SHGC) contacted the bearing manufacturer to evaluate and propose solutions for the issue. They arrived, did long tests, but said nothing yet.

Here some trials done to evaluate the problem and fined cause of temperature rising.

*ASEA BROWN BOVERI (ABB)* the manufacturer of thrust bearing developed some equations to calculate minimum oil film thickness and frictional power losses. Therefore, large long bearing approximation provide adequate estimate of load capacity and friction for the ratios of $L/B>3$. The bearing with ratio $1/3 < L/B < 3$ are called finite bearings. For these bearings all important parameters such as pressure, load capacity, friction force and lubricant flow are usually computed by numerical methods. In certain limited cases, however, it's possible to derive analytical expressions for finite bearings.

Here the ratio $x= L/B =1$. But *ABB* developed analytical expressions to calculate $h_{min}$ and $P_f$...

## 4.7.2 ABB Design Calculations

The bearing dimensions are determined from the following formulas and diagrams, which are solution to Reynolds equation for the hydrodynamic bearing theory.

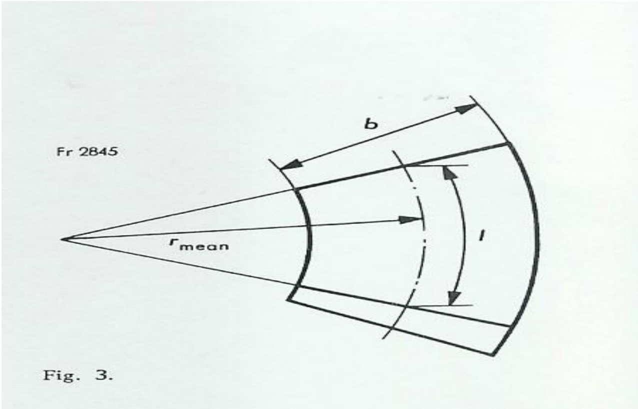



*Figure 4-12Bearing Pad*

The minimum oil film thickness at constant oil viscosity will be

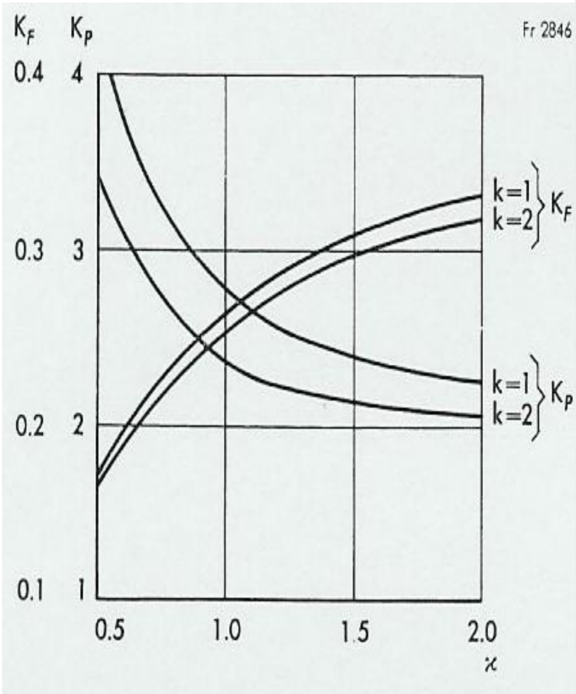
$$hmin = kf\sqrt{\frac{\eta v l}{p}} \quad (4.37)$$

And the frictional losses for the bearing face

$$Pf = kp \times Fv\sqrt{\frac{\eta v}{pl}} \quad (4.38)$$

Where

$\eta$ = viscosity of oil, $Ns/m^2$
$v$ = mean velocity, m/s
$p$ = mean pressure, $N/m^2$
$L$= mean length of pad, m
$B$= width of pad, m
$F$ = total load, N
$h_{min}$= thickness of oil film at trailing edge, m
$h_{max}$= thickness of oil film at entering edge, m
$P_f$ = frictional losses, W
$x = L/B$
$k = (h_{max} - h_{min})/h_{min}$



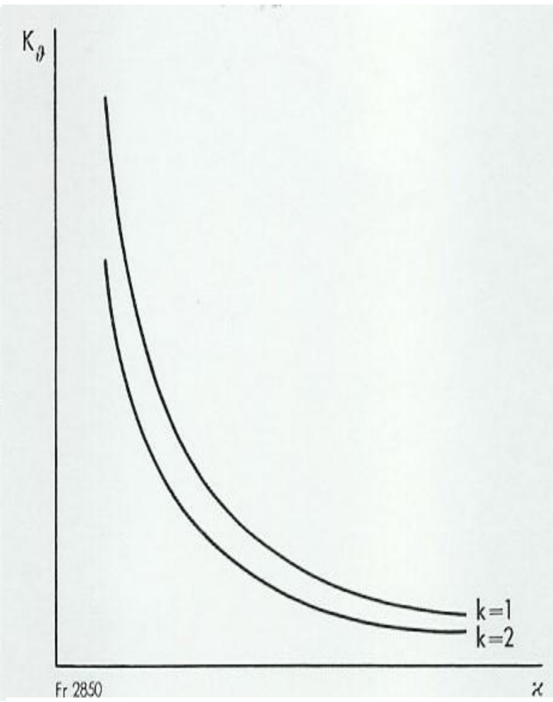




Figure 4-13kf and Kp to find hmin Figure 4-14Temp rise in the oil film

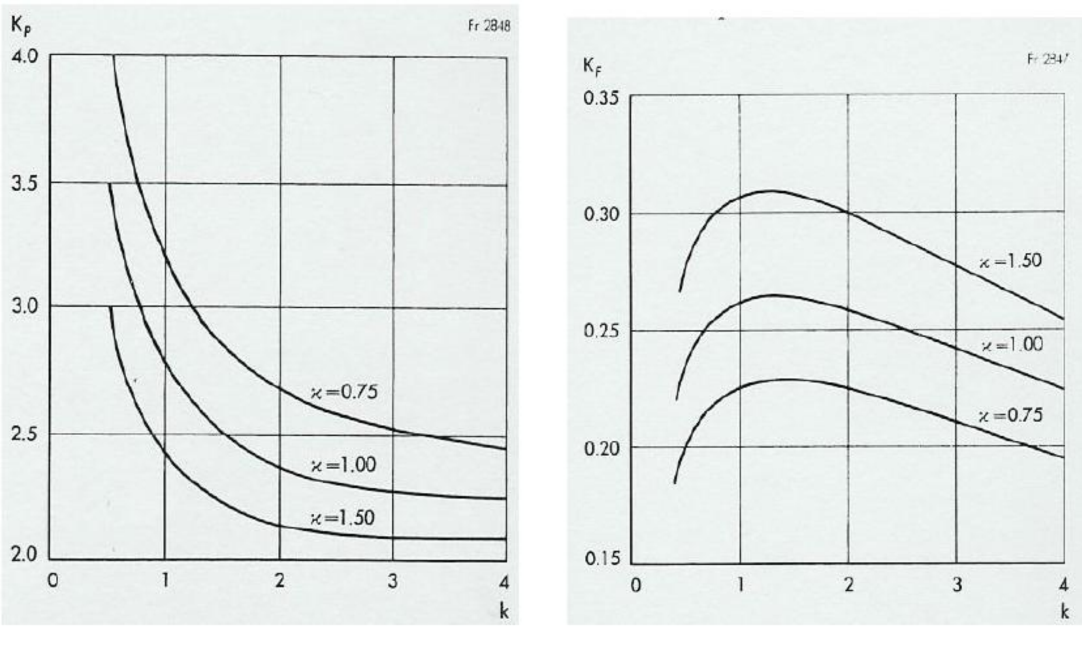



*Figure 4-15kf and Kp as a function in k= (hmax- hmin)/ hmin*

The dimensionless coefficient $k_f$ and $k_p$ are functions of the ratio $x$ and $k$. These are plotted in (figure4-13) for $k$ = 1 and 2.

An optimum bearing must have the largest possible load-carrying capacity and the minimum bearing losses.

To achieve this, the bearing must be designed for the correct values of $x$ and $k$ in (figure: 4-15) $k_f$ and $k_p$ have been plotted as function of k for $x$ = 0.75, 1 and 1.5 from the diagram it's apparent that the oil film thickness ($k_f$) will have its maximum value for a $k$-value slightly larger than unity. The frictional losses $k_p$ decreases with the increasing of $k$-value.

The thrust-bearing springs for the non-reversible machines are arranged so that the position of the resultant supporting force corresponding to $k$=2. The coefficient of temperature rise in the oil film has been plotted in (figure: 4.14) as a function of $x$. (*As can be seen, low value of x result in a substantial temperature rise in the oil film*). The temperature measured in the trailing edge may be high. For this reason, *ASEA*'s thrust-bearings are usually designed for $x$-value lying in the vicinity of unity. In certain cases, when the pad will be exceptionally large and difficult to handle, $x$-value of up to 1.2 are applied

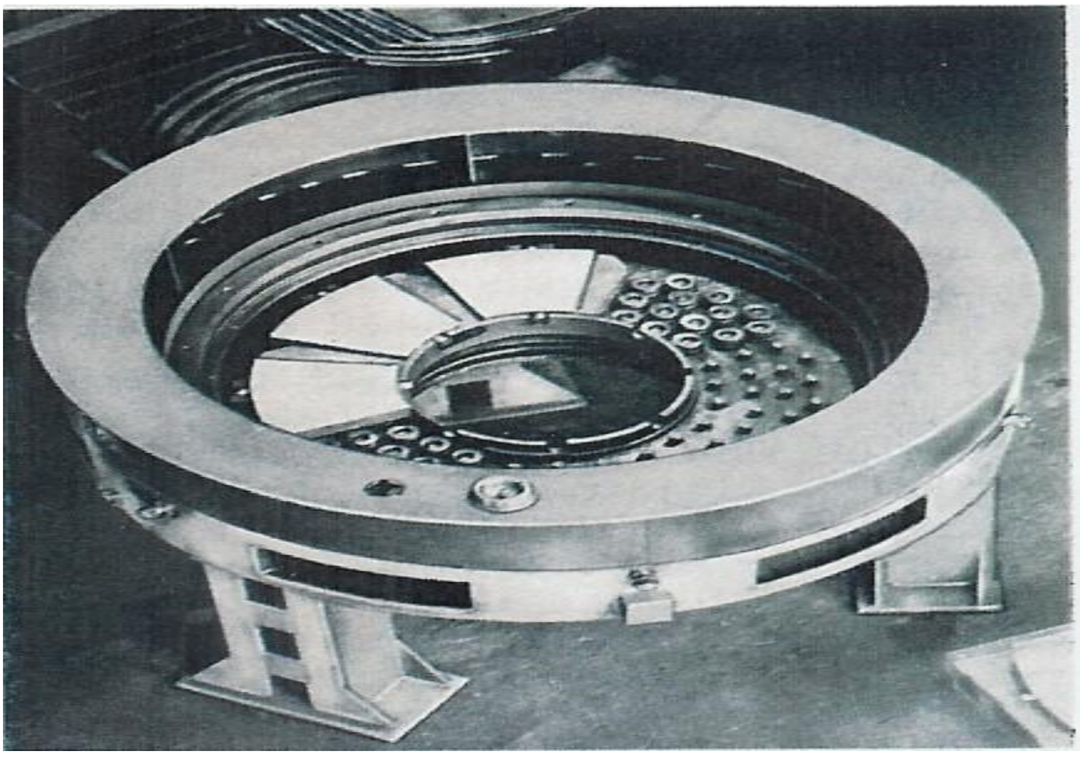


*Figure 4-16Thrust bearing frame with bearing pads and spring in position*

### 4.7.3 Case Data:

Unit *No 1* had been chosen as a sample, data collected during four years two years before heightening (*2011and 2012*) and two years after heightening (*2013and 2014*), data were arranged and summarized in tables and graphs were plotted to compare the differences.

*Table 4-22011 and 2012 Data*

| 2011 | | | | | | 2012 | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | MW | P.B.T | N.H | T.Th.B | | | MW | P.B.T | N.H | T.Th.B |
| jan | 27.00 | 3.60 | 35.80 | 68.00 | | jan | 37.00 | 3.50 | 35.00 | 68.00 |
| feb | 27.00 | 3.50 | 35.40 | 68.00 | | feb | 33.00 | 3.00 | 32.00 | 69.00 |
| mar | 30.00 | 3.17 | 32.40 | 69.00 | | mar | 25.00 | 2.70 | 31.00 | 68.00 |
| apr | 0.00 | 2.90 | 31.00 | 34.00 | | apr | 20.00 | 2.60 | 28.00 | 69.00 |
| may | 0.00 | 0.60 | 7.00 | 34.00 | | may | 22.00 | 2.50 | 27.00 | 70.00 |
| jun | 0.00 | 1.00 | 10.00 | 34.00 | | jun | 23.00 | 2.60 | 27.00 | 70.00 |
| jul | 0.00 | 0.10 | 0.50 | 34.00 | | jul | 22.00 | 2.60 | 25.00 | 71.00 |
| aug | 0.00 | 0.10 | 35.80 | 32.00 | | aug | 8.00 | 2.70 | 23.00 | 31.00 |
| sep | 25.00 | 2.60 | 26.00 | 68.00 | | sep | 25.00 | 3.50 | 40.00 | 70.00 |
| oct | 34.00 | 3.90 | 34.00 | 69.00 | | oct | 38.00 | 4.40 | 43.00 | 72.00 |
| nov | 37.00 | 3.70 | 34.00 | 70.00 | | nov | 36.00 | 4.40 | 43.00 | 72.00 |
| dec | 37.00 | 3.70 | 36.00 | 68.00 | | dec | 36.00 | 4.30 | 43.00 | 71.00 |

*Table 4-32013 and 2014 Data*

| 2013 | | | | | | 2014 | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | MW | P.B.T | N.H | T.Th.B | | | MW | P.B.T | N.H | T.Th.B |
| jan | 36.00 | 4.30 | 43.00 | 71.00 | | jan | 38.00 | 4.40 | 44.00 | 0.00 |
| feb | 36.00 | 4.30 | 43.00 | 71.00 | | feb | 38.00 | 4.30 | 43.00 | 72.00 |
| mar | 36.00 | 4.00 | 39.00 | 70.00 | | mar | 37.00 | 4.10 | 42.00 | 72.00 |
| apr | 36.00 | 3.50 | 35.00 | 71.00 | | apr | 37.00 | 3.90 | 39.00 | 72.00 |
| may | 36.00 | 3.30 | 32.00 | 72.00 | | may | 36.00 | 3.70 | 37.00 | 71.00 |
| jun | 31.00 | 3.00 | 30.00 | 71.00 | | jun | 32.00 | 3.20 | 30.00 | 70.00 |
| jul | 24.00 | 2.70 | 26.00 | 71.00 | | jul | 24.00 | 2.50 | 23.00 | 71.00 |
| aug | 23.00 | 2.70 | 22.00 | 71.00 | | aug | 22.00 | 2.40 | 22.00 | 71.00 |
| sep | 0.00 | 2.50 | 23.00 | 30.00 | | sep | 36.00 | 4.00 | 40.00 | 72.00 |
| oct | 37.00 | 4.50 | 44.00 | 30.00 | | oct | 39.00 | 4.40 | 42.00 | 70.00 |
| nov | 39.00 | 4.50 | 44.00 | 72.00 | | nov | 39.00 | 4.50 | 43.00 | 71.00 |
| dec | 39.00 | 4.50 | 45.00 | 72.00 | | dec | 39.00 | 4.50 | 43.00 | 71.00 |

Where:

MW = Generated load, *MW* P.B.T= Pressure before Turbine, *bar*

N.H = Net head, *m*T.Th.B = temperature in thrust bearing,◌̊*C*

These plots compare maximum values of Total load capacity, Pressure in penstock, Net head and Temperature in Thrust bearing during four years.

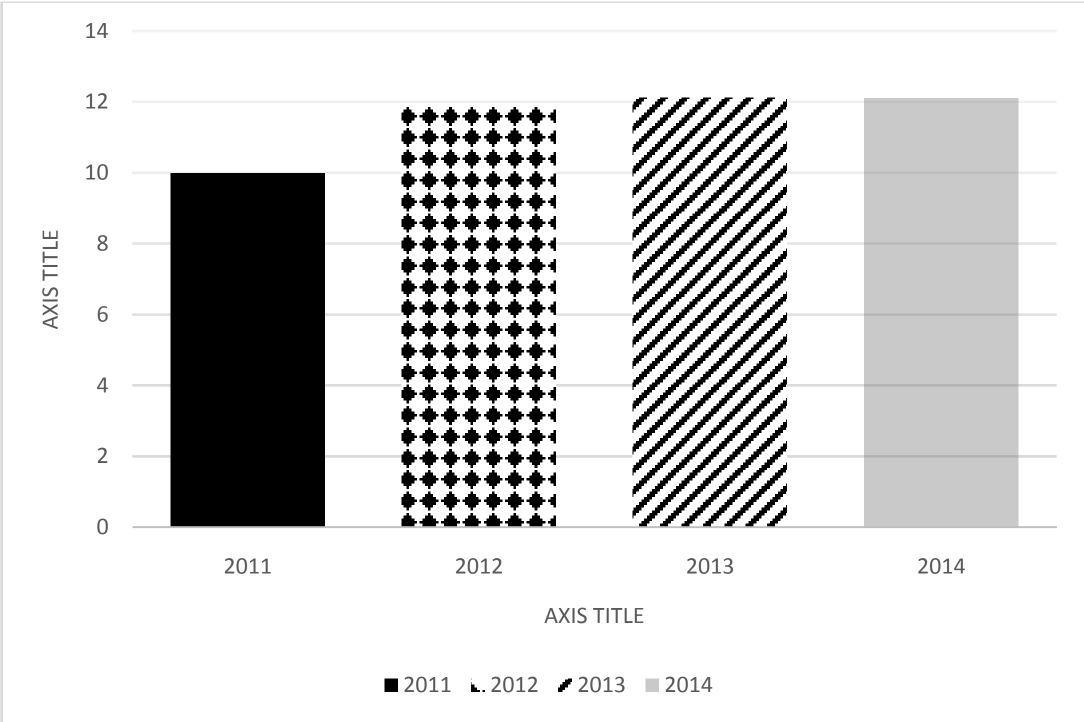



*Figure 4-17Full Capacity Load comparison*

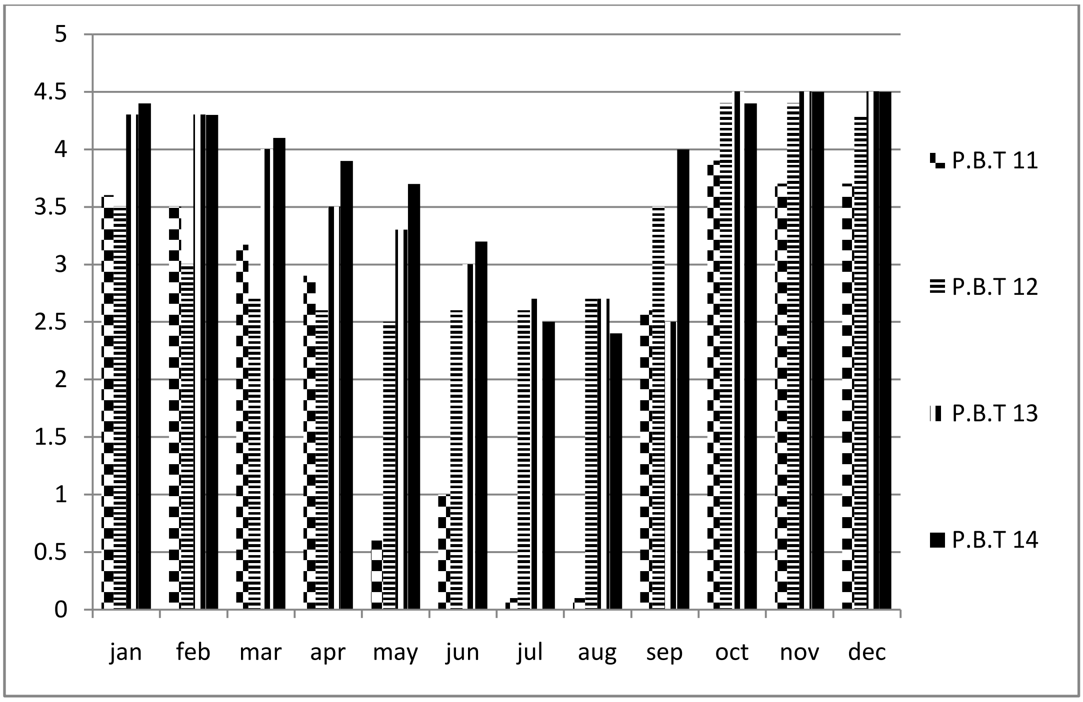



*Figure 4-18Pressure Before Turbine*

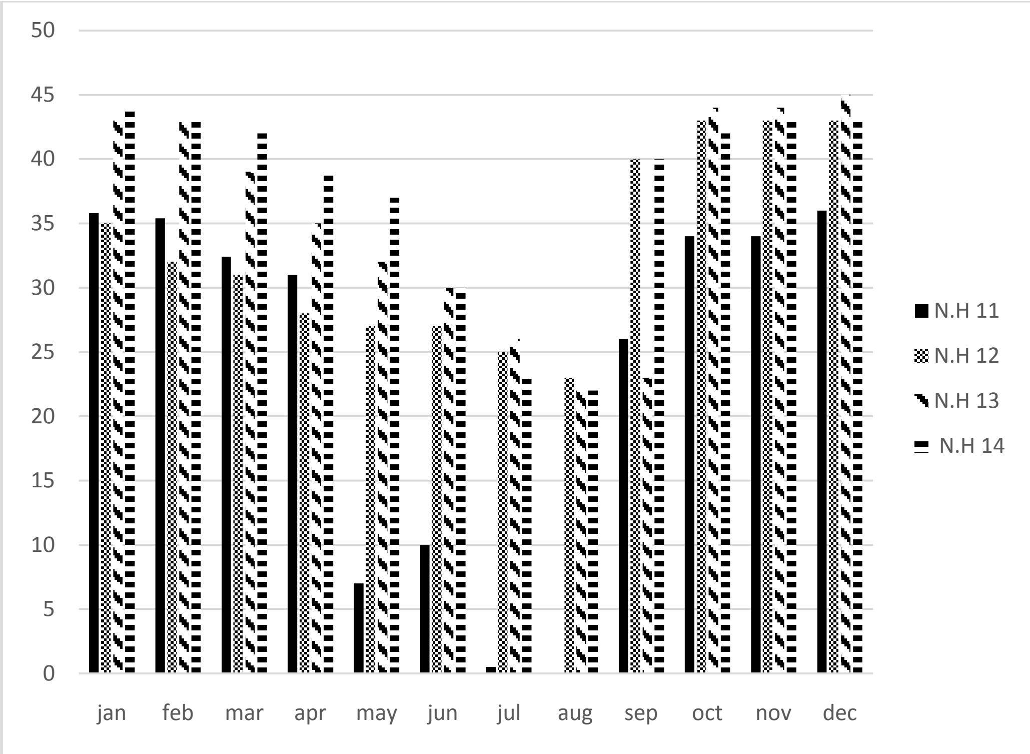



*Figure 4-19Comparing max N.H*

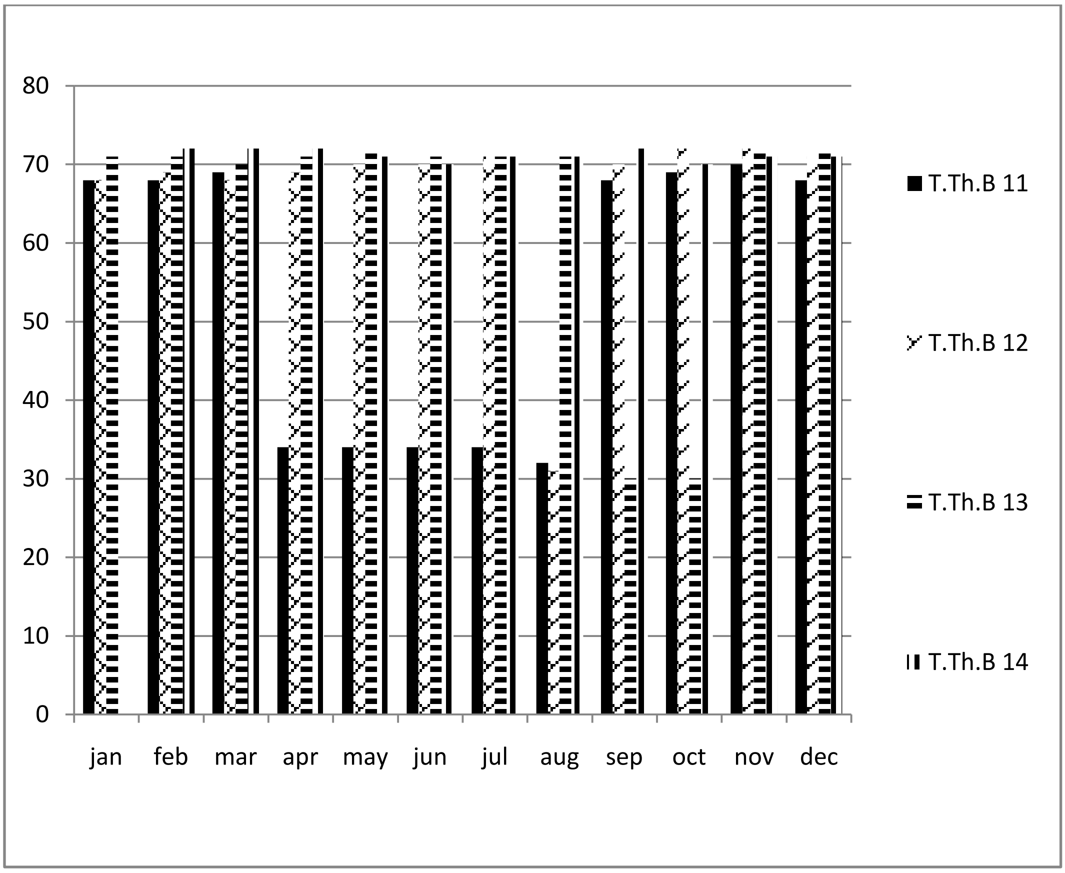



*Figure 4-20Temperature in Thrust Bearing*

### 4.7.4 Machine Used Oil Specifications:

*Table 9* provide the specifications lubricant oil used in *ROSEIRES* Hydro Power Station Machine units, *PRESLIA 68* made by *TOTAL* Company,

The lubricant oil characteristics are the following:

- High oxidation resistance, antifoam, air and water release performances.
- Reinforced antiwear and EP properties allowing the lubrication of the gearboxes driven by the turbine.
- High antirust and anticorrosion performances.
- Important « hydraulic properties » especially hydrolysis stability and filterability (with or without water).

*Table 4-4Total oil specifications*

| TYPICAL CHARACTERISTICS | METHODS | UNITS | |
|---|---|---|---|
| **PRESLIA 68** | | | 68 |
| **Density at 15 °C** | ISO 3675 | kg/m3 | 884 |
| **Viscosity at 40 °C** | ISO 3104 | mm2/s | 68 |
| **Viscosity at 100 °C** | ISO 3104 | mm2/s | 8,7 |
| **Viscosity index** | ISO 2909 | - | 100 |
| **Flash point** | ISO 2592 | °C | 240 |
| **Pour point** | ISO 3016 | °C | - 9 |
| **Demulsibility** | ISO 6614 | min | < 10 |

## 4.8 COMPUTATIONAL DATA:

- Number of pads = 8.
- Pad outer diameter, $r_o$ = <u>2</u>$m$
- Pad inner diameter, $r_i$ = <u>0.895</u>$m$
- Pad mean diameter, $r_m$ = <u>1.11</u>$m$
- Pad angle, $\Omega$ = 40°
- Generator rotor, shaft and turbine runner weight (dead weight) =<u>222350</u>$kg$
- Generator rotor, shaft and turbine runner load ($F$) =22350*9.81=<u>2.2</u>$MN$
- Downward hydraulic thrust pressure before heightening =<u>3.7</u>$bar$
- Downward hydraulic thrust pressure after heightening = <u>4.6</u>$bar$
- Penstock diameter = 5.8 $m$ Penstock Area A= $5.8^2*\pi/4$ =<u>26.42</u>$m^2$
- Downward hydraulic thrust load ($F_h$) before heightening= $3.5*10^5*26.42$ =<u>9.25</u>$MN$
- Downward hydraulic thrust load ($F_h$) after heightening = $4.7*10^5*26.42$ =<u>12.42</u>$MN$
- Total load ($F_{tot}$) before heightening = 9.25+2.2 = <u>11.45</u>$MN$
- Total load ($F_{tot}$) after heightening = 12.42+2.2 = <u>14.62</u>$MN$
- $b$ = <u>1.105</u>$m$, l = <u>1.01</u>$m$, $x$= $l/b$ = <u>1</u>, $k$ = <u>2</u>
- From plotted curves, $k_f$ = <u>0.26</u>, $kp$= <u>2.3</u>
- Normal temperature when machine is stopped = <u>34</u>°$C$
- Normal temperature when machine is running = <u>68</u>°$C$
- Temperature when machine is running after problem = <u>74</u>°$C$ (*Thrust bearing high pressure pumps been running to decrease temperature if no it will continue rising*)
- Oil viscosity, $\eta$ at 68 °$C$ = <u>$3.42*10^{-2}$</u>$Ns/m^2$
- Oil viscosity, $\eta$ at 74°$C$ = <u>$2.91*10^{-2}$</u>$Ns/m^2$
- Oil density, $\rho$ at 68 °$C$ = <u>849</u>$kg/m^3$
- Oil density, $\rho$ at 74°$C$ = <u>845</u>$kg/m^3$
- Oil heat capacity (specific heat), Cp at 68 °$C$= <u>2.048</u>$kj/kg\ k$
- Oil heat capacity (specific heat), Cp at 74°$C$= <u>2.069</u> $kj/kg\ k$
- Turbine rotating speed = <u>136.4</u>$rpm$
- Turbine linear speed at pad mean diameter, $v$ = <u>20.68</u>$m/s$

## 4.8.1 Calculations

### *4.8.1.1 Before Heightening*

Pressure due to total thrust load = F/area of pads(A)

$A_{total} = \pi*8(r_o^2 - r_i^2)\ \Omega\ /360 = \pi*8(2^2*0.895^2)*40/360 = \underline{8.96}m^2$

$P_{total} = F_{total} / A_{total} = 11.45*10^6 / 8.96 = \underline{1.28}MPa$

$l_{total} = 1.01*8 = \underline{8.08}$m

$$P_f = 2.3*1.43*10^6*20.68*\sqrt{\frac{0.0342*20.680}{.16*1000000*8.08}} = \underline{142.53}kW$$

Using equation (36)

$\Delta t = P_f / (\rho\ q\ Cp)$ so $q = P_f / \Delta t\ Cp\ \rho$

$\Delta t = 68 - 34 = 34$ ̊C = 307 k

$q = 142.53*10^3 / (307*2.048*849) = \underline{0.27}m^3/s$

$$h_{min} = 0.26*\sqrt{\frac{0.0342*20.68*1.011}{.28*1,000,000}} = \underline{0.00019}\text{ m}$$

### *4.8.1.2 After Heightening*

Assume constant flow rate although the load force was increased.

Pressure due to total thrust load = F/area of pads (A)

$A_{total} = \pi*8(r_o^2 - r_i^2)\ \Omega/360 = \pi*8(2^2*0.895^2)*40/360 = \underline{8.93}m^2$

$P_{total} = F_{total} / A_{total} = 14.62*10^6 / 8.96 = \underline{1.63}MPa$

$l = 1.01*8 = \underline{8.08}$m

$$P_f = 2.3*14.62*10^6*20.68*\sqrt{\frac{0.029*20.681}{.63*1000000*8.08}} = \underline{148.31}kW$$

$\Delta t = F / q\ Cp\ \rho = 148.31*10^3/(0.27*2.069*845) = \underline{317.70}$ k= $\underline{44.70}$ ̊C

$T = 44.70 + 34 = \underline{78.70}$ ̊C

Temperature will rise about $\underline{10.70}$ ̊C

$$h_{min} = 0.26*\sqrt{\frac{0.029*20.68*1.011}{.64*1,000,000}} = \underline{0.00016}\text{m}$$

*Table 4-5Calculation results*

| category | | before | after |
|---|---|---|---|
| **$P_{total}$ Mpa** | | 1.28 | 1.64 |
| **$F_{total}$ MN** | | 11.45 | 14.62 |
| **Pf kw** | | 142.53 | 148.31 |
| **q m3/s** | | 0.27 | 0.27 |
| **Δt** | | 34 | 44.70 |

| $h_{min}$ | | 0.00019 | 0.00016 |
|---|---|---|---|

# 5 RESULTS AND DISCUSSION

## 5.1 RESULTS:

The first heating alarm noted on 30 September 2012, so this date represents the first kick after dam heightening. Note that the heightening project completed in 2013(*as finishing works*) but first filling of dam reservoir was on 2012 the new head was 46 meters in December 2012, this high head of the new hydraulic load (*which plotted below*) formed an excessive load on the bearing, so bearing temperature started to rise until its alarm point of the control system (73 ͦC). Since this date 30 September 2012, the hydrostatic pumps operates continuously (*In case of any problems with hydrodynamic lubrication- insufficient load carrying capacity- the hydrostatic jacking system been used permanently in order to increase reliability of the bearing*) for assuring non-overheating of the bearing while it must stop working after arriving to 30%-35% of turbine rotational speed.

The next table and figure (5-1) shows temperature of bearing, load carrying capacity and head on the years 2011 and 2012:

*Table 5-1temperature of bearing, load capacity and head on the years 2011 and 2012*

| | Head (m) 2011 | Temp. (ͦC) 2011 | Head (m) 2012 | Temp. (ͦC) 2012 | Load (MN) 2011 | Load (MN) 2012 |
|---|---|---|---|---|---|---|
| jan | 35.8 | 68 | 35 | 68 | 11.7112 | 11.447 |
| feb | 35.4 | 68 | 32 | 69 | 11.447 | 10.126 |
| mar | 32.4 | 69 | 31 | 68 | 10.57514 | 9.3334 |
| apr | 31 | 34 | 28 | 69 | 9.8618 | 9.0692 |
| may | 7 | 34 | 27 | 70 | 3.7852 | 8.805 |
| jun | 10 | 34 | 27 | 70 | 4.842 | 9.0692 |
| jul | 0.5 | 34 | 25 | 71 | 2.4642 | 9.0692 |
| aug | 35.8 | 32 | 23 | 31 | 2.4642 | 9.3334 |
| sep | 26 | 68 | 43 | 74 | 9.0692 | 13.8248 |
| oct | 34 | 69 | 43 | 74 | 12.5038 | 13.8248 |
| nov | 34 | 70 | 44 | 74 | 11.9754 | 14.089 |
| dec | 36 | 68 | 46 | 74 | 11.9754 | 14.3532 |

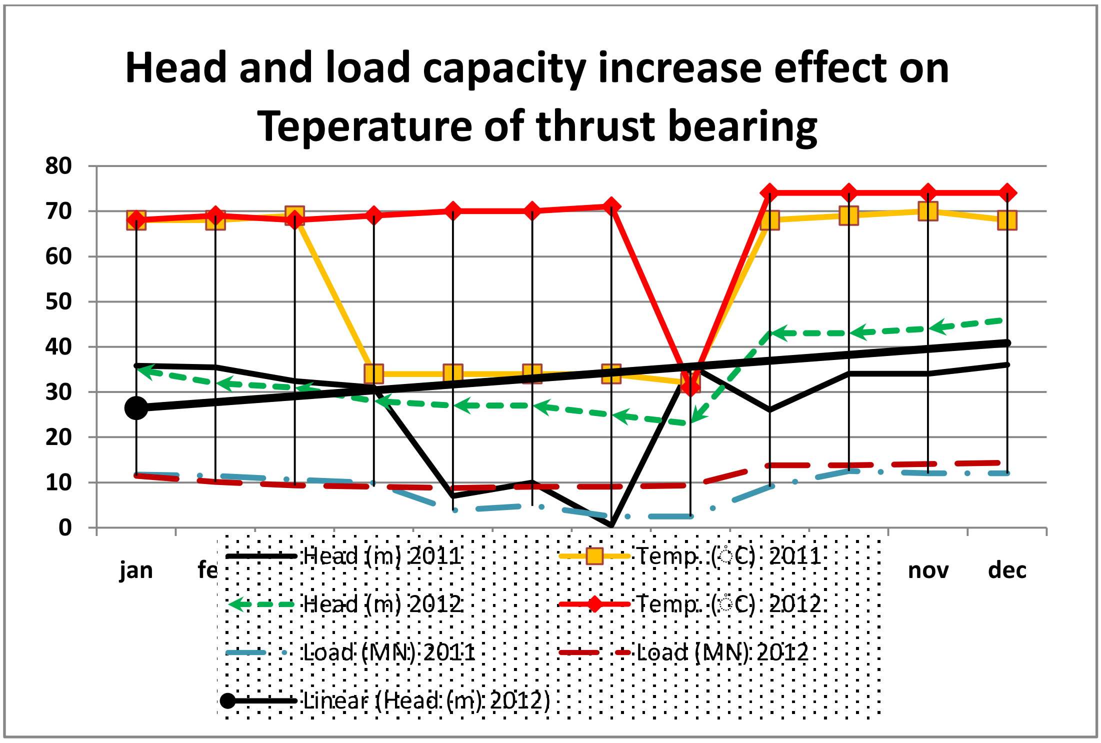



*Figure 5-1Head and load capacity increase effect on Temperature of thrust bearing*

*(Figure: 5-1*) shows clearly the temperature of thrust bearing is increasing proportional to net head value and load capacity which effect on lubricant thickness that leads to increase the value of power losses by friction, before heightening the maximum head in 2011 was around 36 m. and temperature was in range of (*68 to 70 ̊C figure: 5-1*), but after September 2012, the temperature exceeded the value of (*73 ̊C alarm point*) while the head exceeded 45*m*and load capacity been more than 14 MN.

## 5.2 DISCUSSION

*(Figure 5.1)* shows the results of the analytical *ABB* equations (37and 38), results presented here in the form of graphical correlations for the two cases of frictional losses in the bearing which created by the additional load of water due to heightening.

In*(Figure 5.11)* one notes that, with increasing of both the load capacity of hydrodynamic load*(F)*, and the maximum friction force in the layer (*Pf*), lubricant thickness (*h*) decreases (*decreasing of $h_{min}$*), that made a high frictional power converted into heat raised the ($\Delta t$) temperature to about ***44.70***◌̊C that means the temperature would arrive to ***78.70***◌̊C adding about ***10.70***◌̊C to the normal temperature when machine is running (*was about **68**◌̊C*). It has been assumed that the lubricant viscosity remains constant throughout the hydrodynamic film, (*practically, the bearing temperature is raised by frictional heat and the lubricant viscosity varies accordingly, Temperature rise as small as 25℃ can cause the lubricant viscosity to collapse to 20% of its original value*).

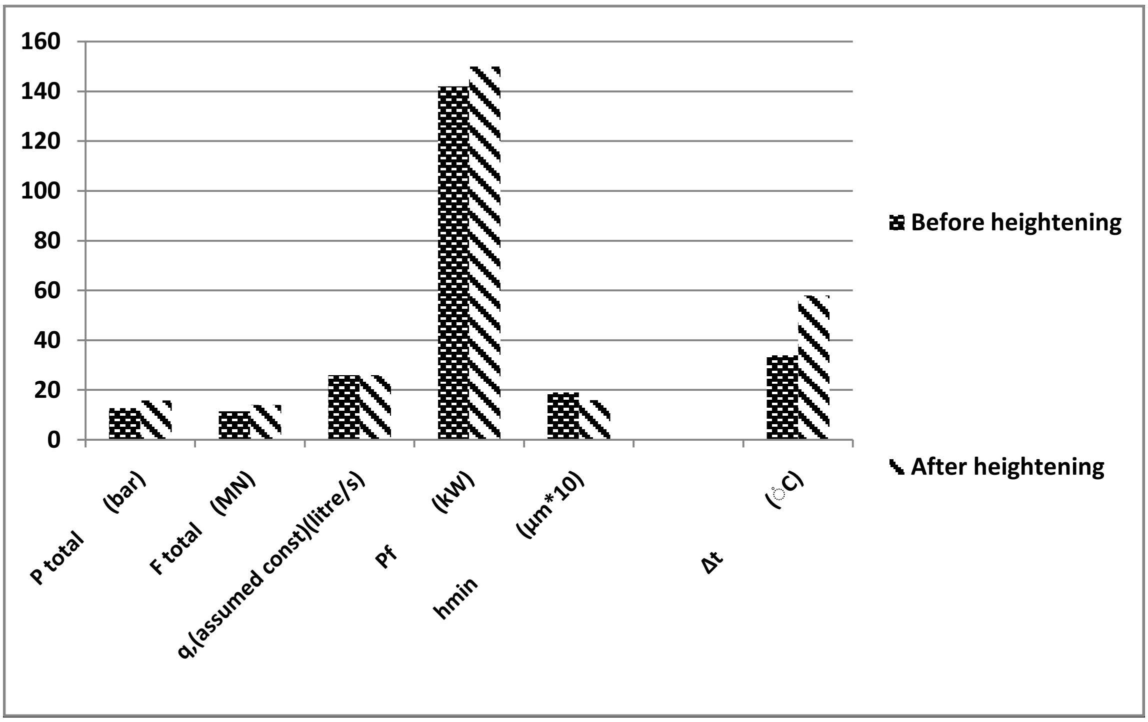



*Figure 5-2Calculation results*

The direct effects of heat in terms of lubricant hydrodynamic pressure, load capacity, friction and power losses can readily be catastrophic. More pernicious still are the indirect effects of thermal distortion on the bearing geometry, which can distort a film profile from the intended optimum to something far less satisfactory. It becomes clear that thermal effects play a major role in bearing operation and cannot ignore.

Lubricant flow rate is an important parameter since enough lubricant must supply to the bearing to fully separate the surfaces by a hydrodynamic film this effect can ever overweigh the direct bearing frictional power loss. Precise calculation of lubricant flow is necessary to prevent overheating of the bearing from either lack of lubricant or excessive churning.

# 6 CONCLUSIONAND RECOMMENDATIONS

## 6.1.1 CONCLUSION:

Rosaries dam heightened to maximum height of 50 m increasing its storage capacity to 7.4 milliard cubic meters, so a stable electricity supply had been available.

Overheating of thrust bearing that used to support the rotating parts noticed in September 2012, the temperature increased to 74 ̊C. It was necessary for the operation engineers in station to run the hydrostatic pumps to be sure of safe operating of the thrust bearing, fortunately temperature decreases to 68 ̊C.

As mentioned the additional head of water after heightening formed an additional thrust load exerted on thrust bearing, this additional load increases the frictional power loss. Loss in power appears as heat energy in the oil film, if assumption of adiabatic flow made.

To conclude we can say that the heightening of the dam represents an over load on the thrust bearing causing it to override the maximum allowable heat of 68 ̊C. This load coming from water column could not be the only reason behind overheating, there may be another causes behind this problem.

In a thrust bearing, alignment and load distribution are not perfect because of manufacturing tolerances in the housing, shaft and bearing elements, this misalignment results from thermal stresses and centrifugal forces from tension and flexibility of bolts and joints. Misalignment of the shaft with collar or the shaft with bearing could result in an unequal load distribution on the pads leading to overheating of particular pads that are loaded heavily.

Thrust bearing installed in Rosaries in 1971 and has been in service since then; long period of service could result in cracks, oil leakage may occur as a result of cracks causing reduction of the pressure generated in oil film, subsequently a reduction in the film thickness causing overheating of the bearing.

Beside fatigue and misalignment, elastic deformation of the pads could change the geometry of the oil film, also chemical corrosion and vibrations. Cooling system problems. All these failures could combine with water column additional load to cause overheating to thrust bearing. If these failures left untreated total bearing, failure may occur.

### 6.1.2 Recommendations:

- ❖ Oil Coolers must be check and clean from dirt and fouling. In addition, replace them by new ones if necessary.

- ❖ To determine whether replacing bearing is necessary or not, numerical solution must applied in order to calculate the total load carrying capacity of the bearing, taking into account the side leakage effect and change of oil properties due to temperature rise.

- ❖ Investigate the bearing physically to detect any crack or decomposed lubricant pipes and joints and to locate any broken bolt or joint.

- ❖ Disassembly of the bearing for further inspection to detect areas affected by fatigue, mechanical wear, chemical corrosion or thermal distortions…etc.

- ❖ Disassembly of the bearing may be costly, so in order to keep the unit in operation monitoring instruments used to provide data for operating characteristics during service. Analyzing data to determine fault location and reasons behind it.

- ❖ Keeping maintenance records to predict the failures and for easing the analyses of bearing performance. Observations noted during failure help in future problems.

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