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DIPLOMARBEIT

GUIDE VANE BEARINGS AND GUIDE VANE SEALS OF PUMP TURBINE

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Preface

Das Layout dieses Projekts wurde von der Verbund Hydro-Power AG vorgeschlagen, dem größten Stromerzeuger aus Wasserkraft in Österreich und einer der führenden Hersteller von Wasserkraftwerken in Europa.

Die kurze Lebensdauer der Leitschaufellager in Francis Pumpturbinen ist ein häufiges Problem für die meisten operativen Betreiber in Pumpspeicherkraftwerken, dass selbst die Lager-Hersteller herausgefordert hat. In der Tat entsteht dieses Problem aufgrund der Vibrationen und der dynamischen Belastungen während des Vorgangs der Maschine im Pumpenbetrieb.

Da die Technologie des Gebrauchs der selbstschmierenden Lager als Leitschaufellager in Francis Pumpturbinen eher neu ist, existieren wenig Literatur und wissenschaftliche Studien. Darüber hinaus sind auch die Erfahrungen der Betreiber und Hersteller von Leitschaufellager für die Lösung des Problems nicht ausreichend.

Eines der wichtigsten Ziele dieses Projekts war Informationen und Erfahrungen vom Betreiber und Hersteller zu sammeln, um einige Lösungen und eventuelle Ansätze für weitere Forschung zur Optimierung und Verbesserung der Erhöhung der Lebensdauer von Leitschaufellager präsentieren zu können.

Ein Betreiber sollte sich immer neuen Produkten und ihren Eigenschaften bewusst sein, um die beste Wahl treffen zu können. Der erste wichtige Faktor in diesem Zusammenhang ist, dass der Betreiber genug Erkenntnisse von der Funktionsweise des Systems hat. Der erste Schritt bei der Wahl des passenden Leitschaufellagers ist, umfassende Tests durchzuführen. Da diese Tests mit umfangreichen Kosten für den Betreiber verbunden sind, können diese meist nur auf

Testergebnisse der Hersteller oder der verschiedenen Forschungsinstituten vertrauen und zurückgreifen.

Die Herausforderung bei der Montage des Leitschaufellagers ist ein möglichst passendes Spiel im Leitschaufellager zu erreichen. Ein kontroverses Argument während der Installation des Leitschaufellagers ist, Dichtungen zu verwenden, um den Eintritt der externen Partikeln zwischen Welle und Lager zu verhindern und somit diesen negativen Einfluss auf die Lagerlebensdauer des Lagers zu verringern. Gleichzeitig steigt aber der Aufwand für die Wartung und Austausch der Dichtungen.

Statische und dynamische Reibungswerte sind wichtige Indikatoren und Hilfsmitteln zur Optimierung der Lebensdauer von Leitschaufellager. Das ausgewählte Leitschaufellager sollte fast ähnliche Höhe der statischen und dynamischen Reibungswerte haben, da aus dem großen Unterschied zwischen diesen Werten der Stick-Slip Effekt resultiert. Dabei müssen ebenfalls die Verschleißwerte berücksichtigt werden. Denn niedrige Reibungswerte bringen nicht automatisch niedrige Verschleißwerte hervor.

Die Betreiber benötigen umfangreiche Erkenntnisse der Arbeitsbedingungen, da die erzielten Testergebnisse zeigen, dass sogar minimale Unterschiede in den Arbeitsbedingungen von zwei ähnliche Leitschaufellager zu verschiedene Lebensdauer führen können.

The layout of this project was proposed by the Verbund - Austrian Hydro Power AG which is the largest power provider in Austria and one of the leading producers of hydroelectric power in Europe.

Short life of the guide vane bearings in Francis pump turbine is a common problem for most operating companies in pumped storage power plants, which has even challenged manufacturers themselves. Indeed, this problem is raised due to vibrations and dynamic loads during the operation of the machine as a pump system and the related calculations and characterization is a very complicated process.

Because the technology - using self lubricating bearings in guide vanes of pump turbines - is rather new, very little literature can be found on using them optimally. In addition to this, the experience of operators and manufacturers of bearings for solving the problem are very inconsiderable. One of the main objectives of this project was gathering details and experiences from operators and manufacturers to be able to present some solutions or at least ideas for their optimization and longer operating life.

Through agreements with Verbund Company - designer of the project – and cooperation of RWE Group and SEO Vianden Group it was possible to visit the Vianden hydro power plant in Luxembourg. Very valuable details were obtained from this power plant through the agreement with two above-mentioned companies. In addition, some more information and experiences were found from Rodundwerk II power plant by Vorarlberger Illwerke AG.

During the project, two personal visits were set with two bearing manufacturers, Trelleborg Group and Deva GmbH. Although they were not allowed to disposal information about their customers, the session held principally to be familiar with their products.

The first chapter briefly explains hydroelectric power plants and their tasks. The next chapter introduces the wicket gate and types of bearings. It also contains some of their advantages and disadvantages and also summarizes how to select the proper bearing based on the manufacturers' recommendations. The next chapter deals with some tests and analysis about bearings done in some institutes and the chapter concludes some experiences from manufacturers and operators in the field.

Content

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1. Pumped Storage Power Plants

1.1. Background

The power plants in network are divided generally into two major categories. Some power plants provide the base load while others provide the peak load of the network. The base load power plants have lower utilization costs but they are less flexible against changes in the network. During the day, when the demand for energy is high, most power plants generate electricity regardless of the production cost. On the other hand, at times of low electrical demand only base load power plants are located in network. This leads to price differences in energy production during low load and high load periods of network. On top of that, price of electricity in different seasons usually fluctuates. The storage power plant can reduce this difference in energy prices.

Pumped storage hydroelectricity is a method of the storage and the production of electricity to supply high peak demands for electricity in network. Generally the storage and the production of electricity are obtained by moving water between two reservoirs at different elevations. The water movement occurs during high and low demands for electricity in network. At times of low electrical demand, extra capacity of other power plants is used to pump water into the higher reservoir and during the time of high demand the storage energy is used to produce electricity. In this way the production and consumption of energy would be balanced. Moreover the voltage and frequency of network could be controlled. When there is higher demand in network, water is released back into the lower reservoir through a turbine to generate the hydroelectricity. Reversible turbine/generator assemblies act as pump and turbine. In short, the pumped storage hydroelectricity is a huge battery to save the electricity.

Frequency and voltage are two important parameters in controlling and utilization of network. Changes in production processes and energy consumption lead to changes in network voltage and frequency. In power grid when energy production is more than energy requirement, the network voltage and frequency increase and when the energy produced is less than energy requirements, these values are being reduced. Hard changes in voltage and frequency of network due to factors such as network disconnected lines, leaving the large manufacturing units form network, rapid changes in energy requirements, cause damage in plant equipments and costumer equipments, network imbalance and network collapse. Hence a fast response during disasters is very important to control and balance the frequency and voltage of network. The thermal power plants are very slow to response against changes in energy requirements and the produced energy must be consumed immediately and cannot be stored.

It should be noted that these pumped storage plants are built in the place where the nature prepared the condition for these plants and it is not economical to prepare the condition artificially for the plants. These power plants are very useful in countries that the large part of their energy will be supplied by nuclear power since removing nuclear fuel and nuclear power plants from cycle work and network is very difficult and costly.

For the first time, the pumped storage system was exploited in 1890 in Italy and Switzerland. However reversible turbines were used in 1930 for the first time. In the primary division, these power plants are divided into simple and combined plants. In simple pumped storage plants only a certain volume of water will be displaced between two reservoirs. On the other hand in combined pumped storage plants, in addition to pumping water, the natural flow of rivers is the source of energy production too. In addition to this classification, these power plants are divided based on upper and lower reservoirs. These reservoirs can be natural or artificial. Fig.1.1 shows schematic of pumped storage plant and generating plant.

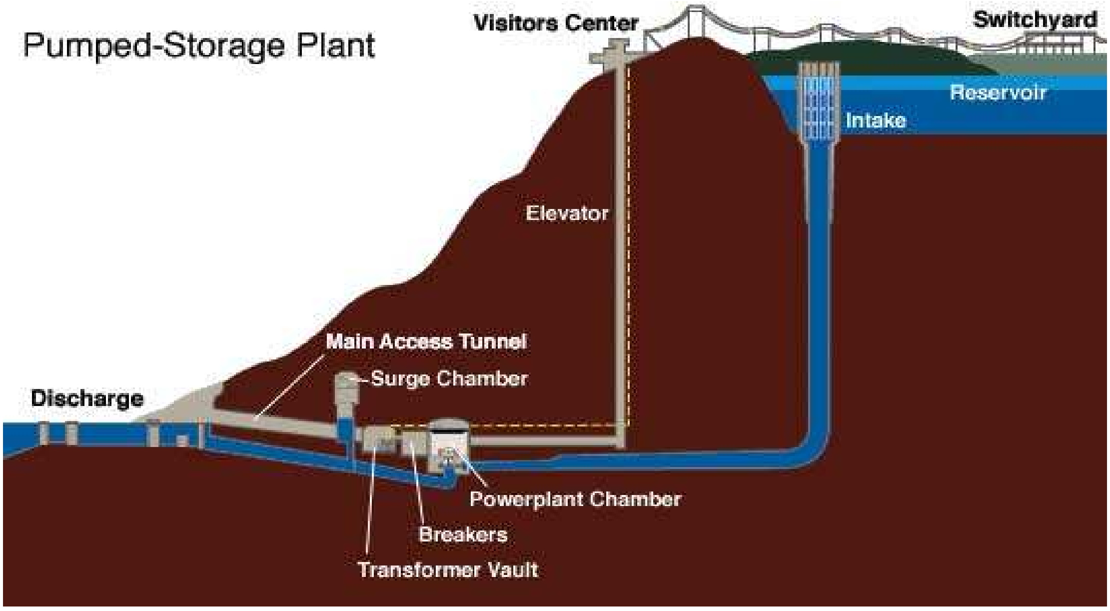

Fig.1. 1: Schematic of pumped storage plant [1].

The total efficiency of pumped storage plants, the ratio of energy output to energy input, will be obtained from the total pumping and generating cycle. In the new pumped storage plants, depending on design, equipments, head and how the plant operates, the efficiency can be more than 75 % [2]. Fig.1.2 shows schematic of estimated losses and efficiency values of a pumped storage plant.

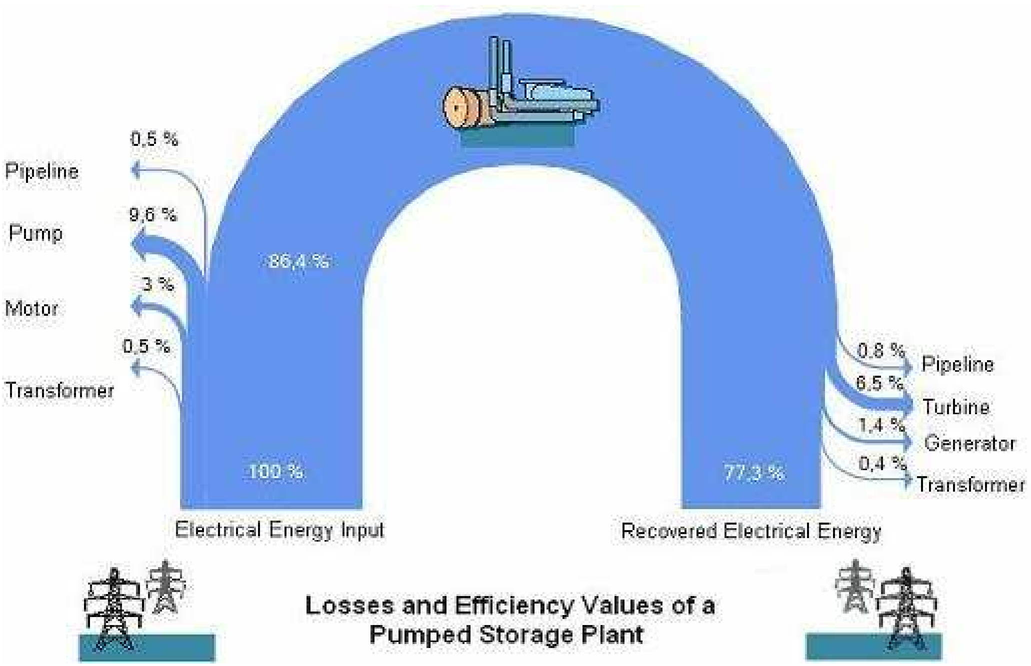

Fig.1. 2: Estimated losses and efficiency values of a pumped storage plant [3].

1.2. Upper and Lower Reservoirs

As mentioned, the pumped storage plants can be classified based on upper and lower reservoirs. The reservoirs are important criteria for choosing a power plant construction site. The height between the reservoirs and water transfer route are very important factors that must be considered in designing the power plant.

Fig.1. 3: Okinawa seawater pumped storage plant [4].

The reservoirs can be classified based on different criteria [2]:

- Inflow conditions: with natural or without natural inflow
- Their position in a river system or lake system
- Position and operation mode
- Storage cycle: daily, weekly, etc.

Fig.1.3 and fig.1.4 show the reservoirs of Okinawa seawater pumped storage plant and pumped storage power plant Goldisthal.

Generally the reservoirs can be natural or artificial. Constructing the power plant where natural reservoirs are available, can be very economical for project.

Fig.1. 4: Pumped storage power plant Goldisthal [5].

1.3. Causes and Benefits of Pumped Storage Usage

Several causes and benefits can be counted for these power plants that they can show the importance of these power plants along with other conventional power plants.

1. Use of the extra production in low demand.
2. Adjusting the electricity and avoiding voltage drop and frequency drop in network.
3. Cost saving in starting of thermal power plants.
4. The adjustment and the optimization of thermal power utilization which results in saving fuel thermal.
5. Increase in life time of thermal power plants due to reduction of thermal and mechanical stress effects which are occurred by sudden load changes in power plants.
6. Less fossil fuel consumption results in less pollution of environment.
7. Balance in the country's electricity network at high and low demand times.
8. Increase flexibility of network during sudden changes in consumption.
9. Management of energy at network.
10. The correction of load curve.

Although these power plants have many advantages, they have some disadvantages too. Construction and design costs of these power plants are expensive. Moreover, the cost of transmission lines is high. In dry seasons, when the available amount of water is reduced the energy production is affected. Furthermore, by creating artificial reservoirs, tangible changes in the ecology system in the area of the reservoirs would happen [6], [2].

1.4. Power Stations

Based on the location of plant equipment installation, the pumped storage plants are classified into aboveground and underground pumped storage plants. Underground power plants are called cavern. There are some factors that should be taken into consideration in the decisions related to whether the power plant must be aboveground or underground. Fig.1.5 shows the schematic of the Siah Bishe pumped storage in Iran. The Siah Bishe power plant is the first pump storage power plant in Iran. It’s anticipated that this national project will be put into operation in 2011.

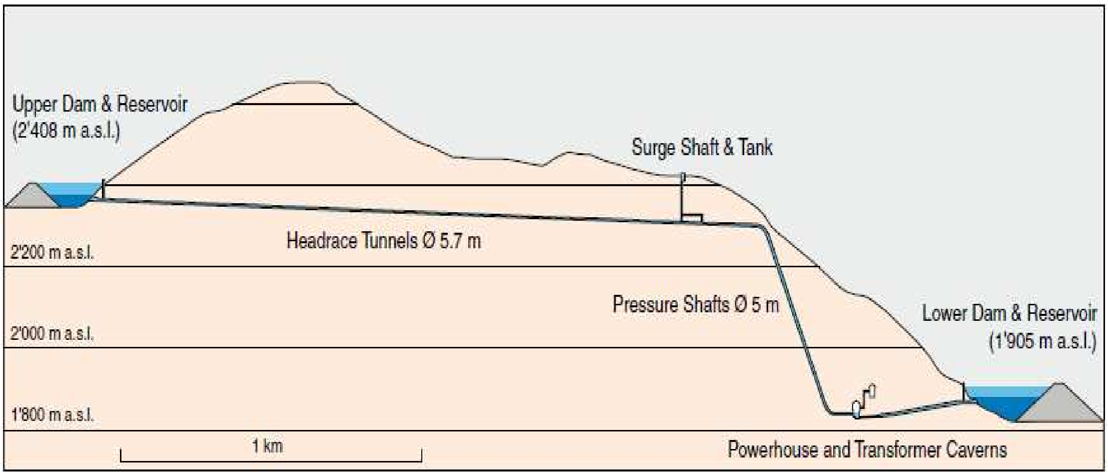

Fig.1. 5: The Siah Bishe pumped storage in Iran [6].

The space and the geological conditions are two influential factors for this decision. Although factors such as geology, soil conditions and earthquake rate are important for other power plants too, these factors are more important if the equipment is to be installed underground. Generally, the underground power plants need more exploration work and longer planning phase and also the control and securing expenses for these power plants are more than aboveground power plants' expenses. On the other hand the underground power plants have various advantages. From the perspective of the environmental organization these power plants can be better, since these plants has less impact on the environment and from the perspective of the engineers, the environment such as seasonal conditions, avalanches and landslides have less influence on the plant. Fig.1.6 shows the powerhouse of Siah Bishe power plant. In underground power plant, pipelines are shorter, have less costs and less pressure drop in system and also more favorable supply levels for machine to avoid the cavitation [2].

Fig.1. 6: The Siah Bishe pumped storage in Iran [6].

1.5. Mechanical Equipment

The pumped storage unit, which includes main equipment such as a motor generator, a pump and a turbine, can be installed vertically or horizontally. Fig.1.7 and fig.1.8 show schematic of horizontal design and vertical design. The equipments of the horizontal design can be assembled or removed easier than equipments of vertical design because of the accessibility of parts. Nevertheless depending on the local situation, their construction need more work. There can be some problems in this design for services such as the alternating loads of the shaft and cavitation risk [2].

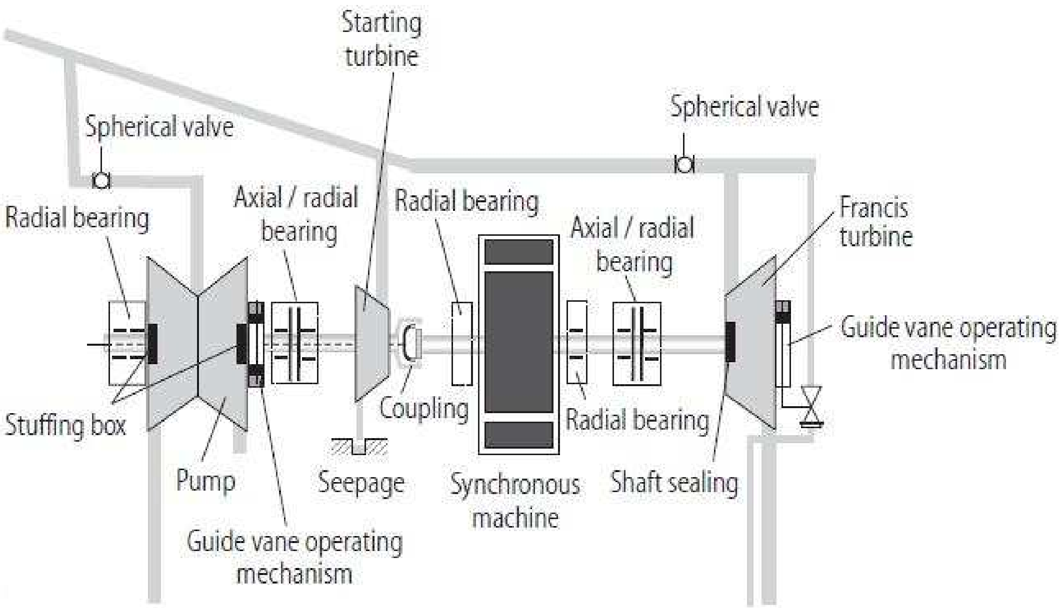

Fig.1. 7: Horizontal design, pumped storage power plant Niederwartha [2].

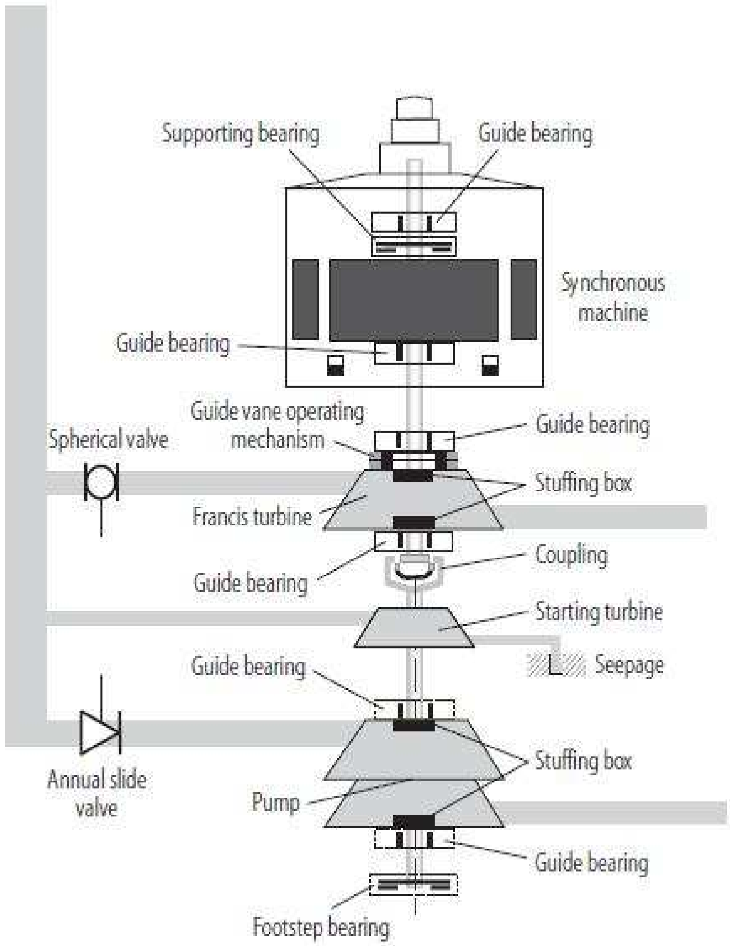

Fig.1. 8: Vertical design, pumped storage power plant Hohenwarte II [2].

With development of technology in hydro power plants, the reversible pump turbines are used depending on the conditions of pumped storage unit. The reversible pump turbine is a hydraulic machine with two modes of operations. The Francis turbine has the capability to be designed for this system, which may work as a generating unit or as a pumping unit by reversing the direction of the shaft rotation. Fig.1.9 shows a sectional view of Francis pump turbine. Because of the usage of one machine instead of two machines in these units, the evolution of reversible machines can be reduced the capital cost such as construction costs of pumped storage power plants. Nevertheless the cost of the machine increases due to the difficulty in designing. The main advantage of this unit is the elimination of shaft coupling. The total efficiency of the reversible machine especially with fixed speed is lower. When the operating mode is changed from generating to pumping or contrariwise, the change over time is longer than a power plant with two machines [2].

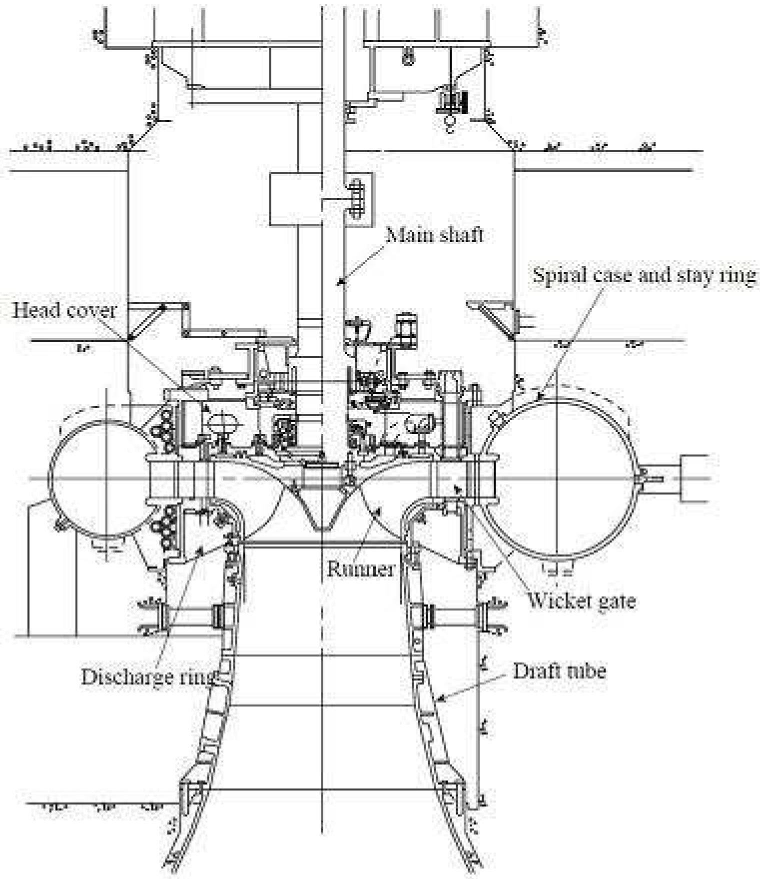

Fig.1. 9: Sectional view of pump turbine [2].

According to the shaft arrangement, the Francis turbines are divided into vertical shaft arrangement and horizontal shaft arrangement. Fig.1.10 shows a horizontal Francis turbine and fig.1.11 shows a vertical Francis turbine. Francis turbines generally include a main shaft, stay vanes, guide vanes, a runner, bearings and spiral case. The guide vane cascade encompasses full circumference of the runner. The adjustable vanes with equal shape and size regulate the flow direction during the discharge or before entering the runner.

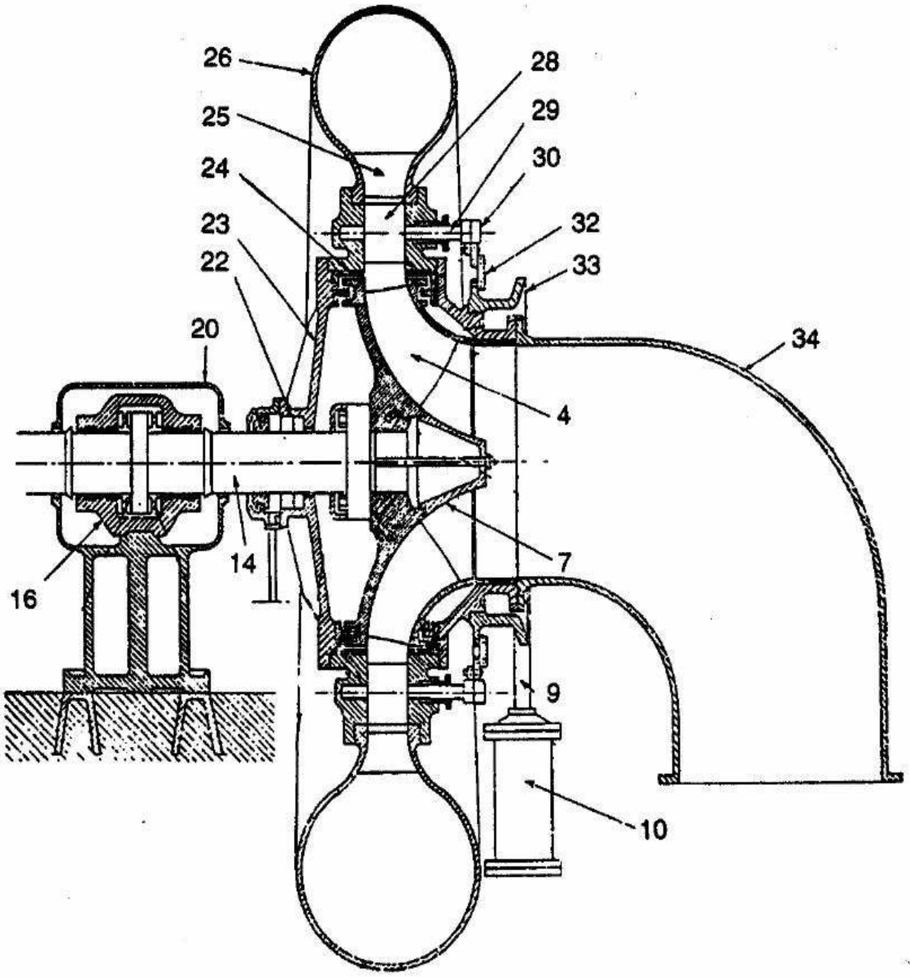

Fig.1. 10: Horizontal Francis Turbine; 4.Runner, 7.The runner cone, 9.Servomotor rod, 10.Servomotor, 14.Turbine shaft, 16.Bearing pad, 20.Bearing cover, 22.Shaft sealing box, 23.Turbine cover, 24.Runner seal ring, 25.Stay vane, 26.Scroll case, 28.Guide vane, 29.Guide vane stem, 30.Giude vane lever, 32.Link, 33.Regulating ring, 34.Draft tube [7].

A Francis pump turbine has to be adapted to both operations. During the frequent starting and change over processes, the static and dynamic loads and pressures on the components, particularly on the guide vanes bearings are heavier. The pump turbine needs a greater runner’s diameter than a Francis turbine. Since the flow controlling in the turbine runner is easier than in the pump runner, the pump turbine’ runner must be designed to reduce the cavitation risk in pumping operation. The flow should have time in runner to achieve the stability [2].

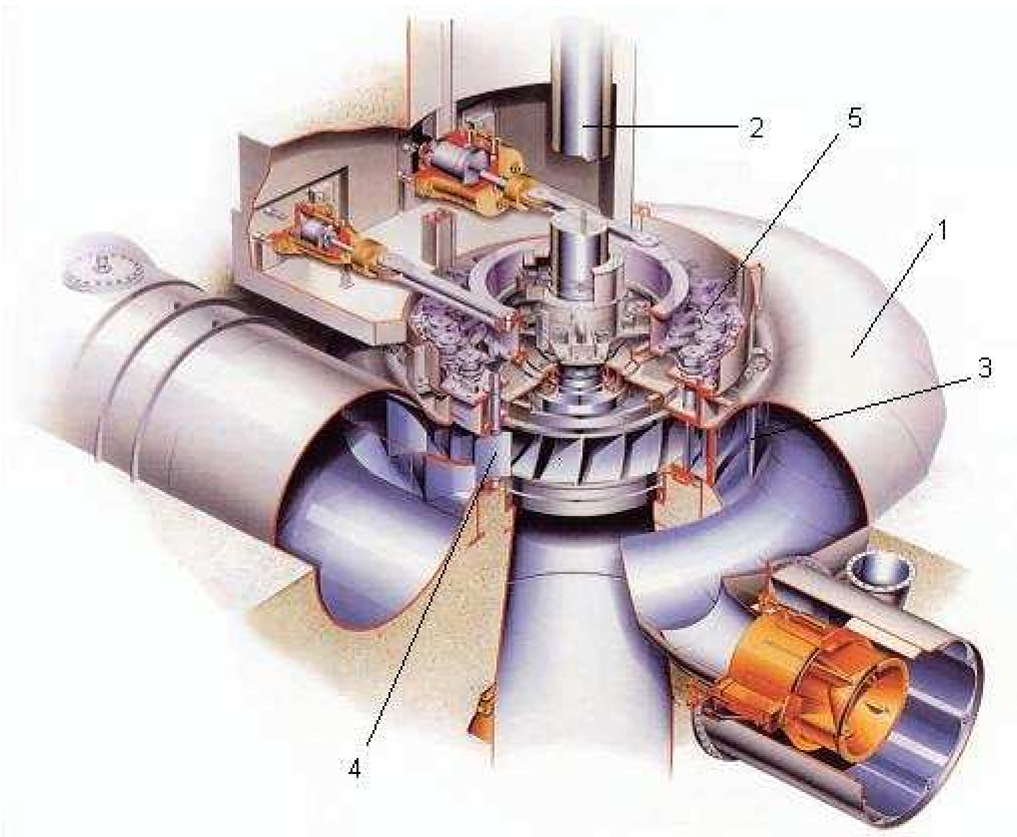

Fig.1. 11: Vertical Francis turbine: 1.Scroll case, 2.Shaft, 3.Stay vane, 4.Guide vane, 5.Wicket gates operating mechanism [7].

As previously noted, the total efficiency of the pump turbine is lower, especially the pump turbine with fixed speed. The efficiency can be improved with using variable speed. The modern technology of variable speed requires electronic power converters, control equipment with high capability and especial design of motor-generator and pump-turbine.

For example, in the pumped storage plant of Goldisthal with the adjustable speed from 300 to 346 rpm, in which synchronous speed is 333 rpm, the efficiency improvement of up to 10% at partial load is achieved [2]. Fig.1.12 compares the turbine efficiency curves at fixed/variable speed for the pumped storage power plant Goldisthal.

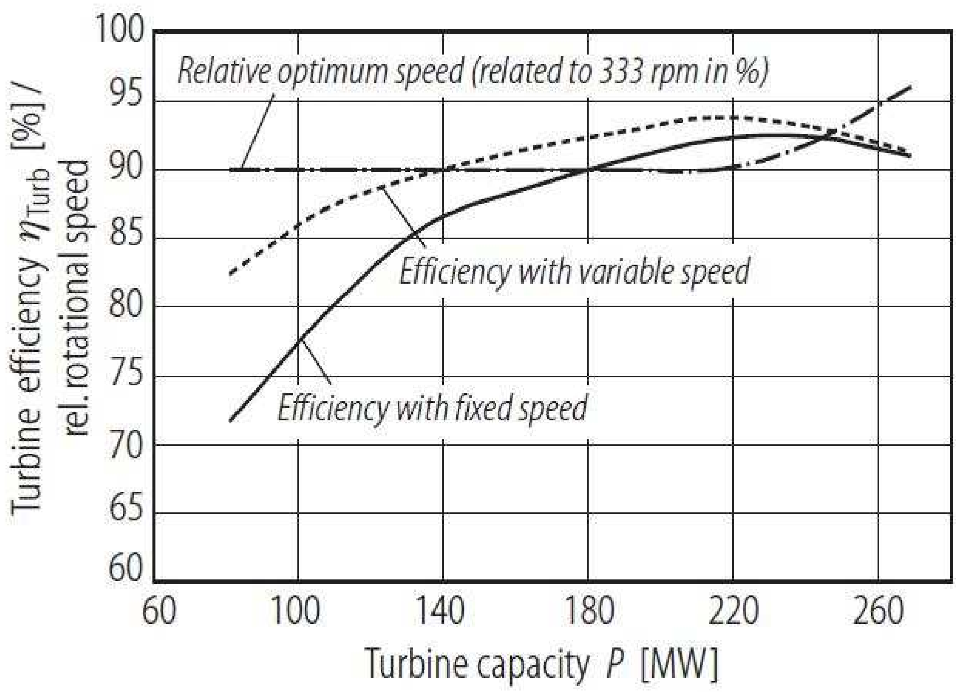

Fig.1. 12: Comparison of turbine efficiency curves at fixed/variable speed for the pumped storage power plant Goldisthal [2].

With the synchronous speed, the maximum efficiency just occurs in design point and depending on the head and the discharge, faraway from the design point, the efficiency is reduced. With adjustable speed, the efficiency can be kept on the optimum value. The pump input power can be changed by adjusting the rotational speed. This enables the frequency control of operation in pumping operation as well as in generating operation. For very rapid variation of the load in the network, adjustable speed units can follow the variation rapidly. Due to this process, the power output can be changed to stabilize the high frequency load variations or system noises. However, due to power electronic devices used for the low frequency alternative current excitation system, the adjustable speed system is costly. Fig.1.13 compares operation performance of a pump with fixed and variable speed, in other words, with synchronous and adjustable speed.

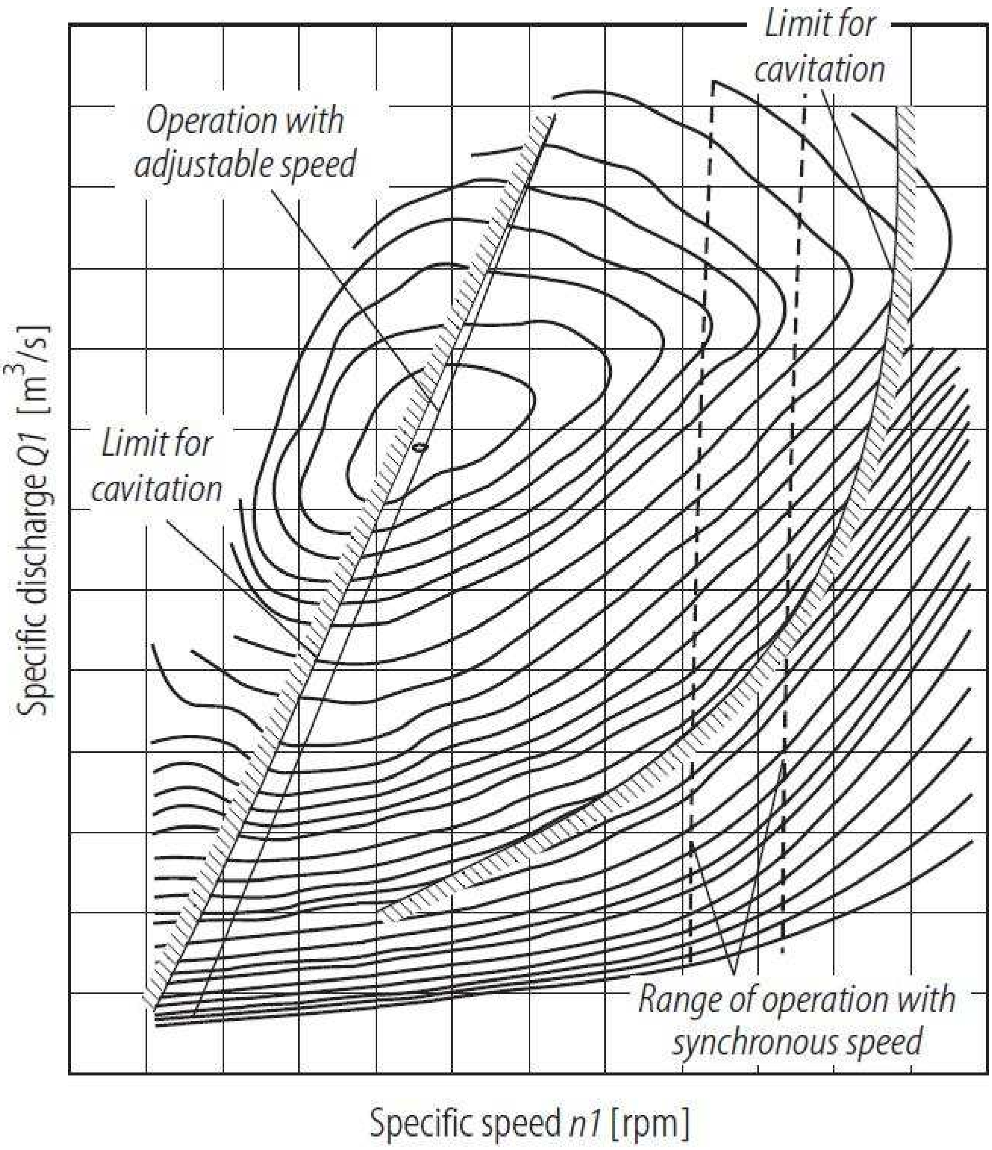

Fig.1. 13: Operation performance graph of a pump turbine with variable speed [2].

Another method to improve the efficiency is the usage of two stage reversible pump turbine. Commonly for the height operation below 600m and with regard to hydraulic conditions, the one stage pump turbine is used and for the above 800m the two or multi stage pump turbine is used. For the boundary conditions, namely the range between 600 to 800m, with regard to the economic and technical conditions, the one of them can be used. The usage of the two stage pump turbine in the upper range of boundary conditions has several advantages from the technical and economic perspective. The two stage pump turbine leads to the reduction of the vibrations, the oscillation on the wicket gates and the bearings, the stresses on the turbine components and the abrasion due to speed reduction and also the facility of the maintenance and the inspection. However, with increasing the stages, the waste increases between stages but total efficiency will be increased. With increasing of each stage in the multi stage pump turbine, the total equipment cost of the machine will be increased. This machine has the complex structure and complexity to control the operation.

2. Wicket Gate and Bearing

2.1 Introduction

In hydropower generation, hydrodynamic effects in turbines are very important in the design of these machines such as the draft tube rope at part load and the wicket gate runner blade interaction. Especially in high head Francis vibrations from the rotor-stator interaction between wicket gates and runner blades are an important factor to design the Francis turbine. In a water turbine, a wicket gate is a member to control and stop the water flow through the runner.

Design of the wicket gates must meet the requirements of both hydraulic and structural strength. The wicket gates are usually made of carbon steel castings for the low head, while stainless steel castings are suitable for the high load. The wicket gate consists of several guide vanes and each guide vane is to adjust the wicket gate openings. The openings of the guide vanes are adjustable by the regulating ring, the links and levers. The vanes have a smooth surface and they are shaped in accordance with hydraulic design specifications.

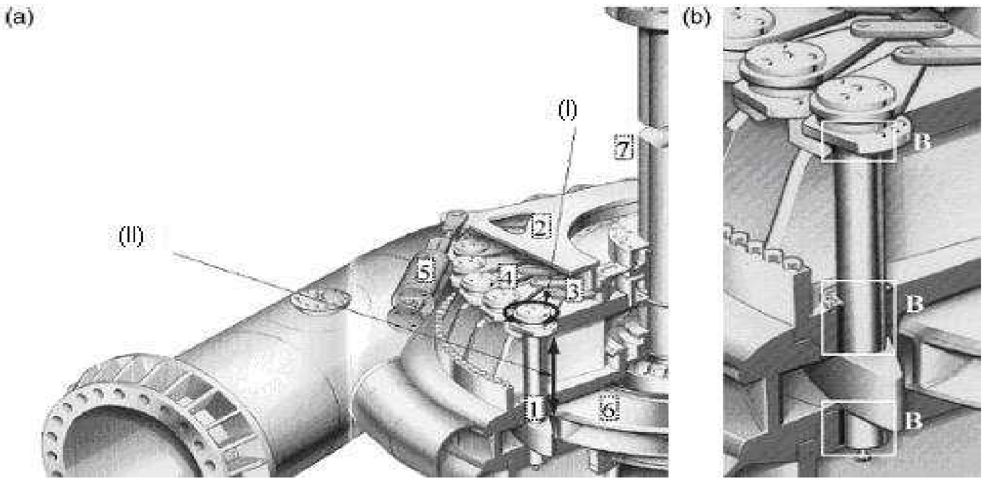

Fig.2. 1: Cross-section of (a) Francis water turbine with guide vane: 1.guide vanes; 2.central ring; 3.link; 4.lever; 5.hydraulic pistons; 6.turbine runner; 7.shaft; (b) guide vane support: B, bearings [8].

Guide vanes will be rotated to achieve the adjustment and the control of the water flow and also to start or stop the machine, see fig.2.1.a, the rotation (I). The movement (II) in fig.2.1a, during operation of the machine irregular water flow, generates rotating movements of guide and also axial movements of bushes in respect to guide vanes shafts [8].

The wicket gate operating mechanism, fig.2.2, is installed with eccentric pins between the gate operating ring and each wicket gate. Shear pins with carefully calculated size are provided with an operating mechanism. A pin will shear, should a wicket gate become blocked, and the remaining gates can be operated as required. In some stations, a friction device is installed, which prevents a free wicket gate from oscillation or irregular movement without constrictive normal operation of the remaining gates when a shear pin breaks. The guide vane mechanism provides the regulation of the turbine output [11].

The servomotor transfer the force through a rod to the regulating ring and the servomotor is controlled by the turbine governor. The guide vane and the governor have ability to provide a stable speed of the unit and also to maintain the frequency in the electrical distribution grid. The ring transfers the movement to the guide vanes through a rod, lever and link construction. The movement leads to angular displacement of the guide vanes to increase or decrease the water flow into the turbine runner fixed to the shaft.

The vane levers are mounted on the upper trunnion and fixed by a wedge, shear pins or pure friction joint. The guide vane lever and regulating ring are connected through links. These links are connected through self lubricated bearings on trunnions on the regulating ring and the lever respectively. The trunnions are positioned on the guide vanes for achieving minimal regulating forces from the hydraulic forces acting on them.

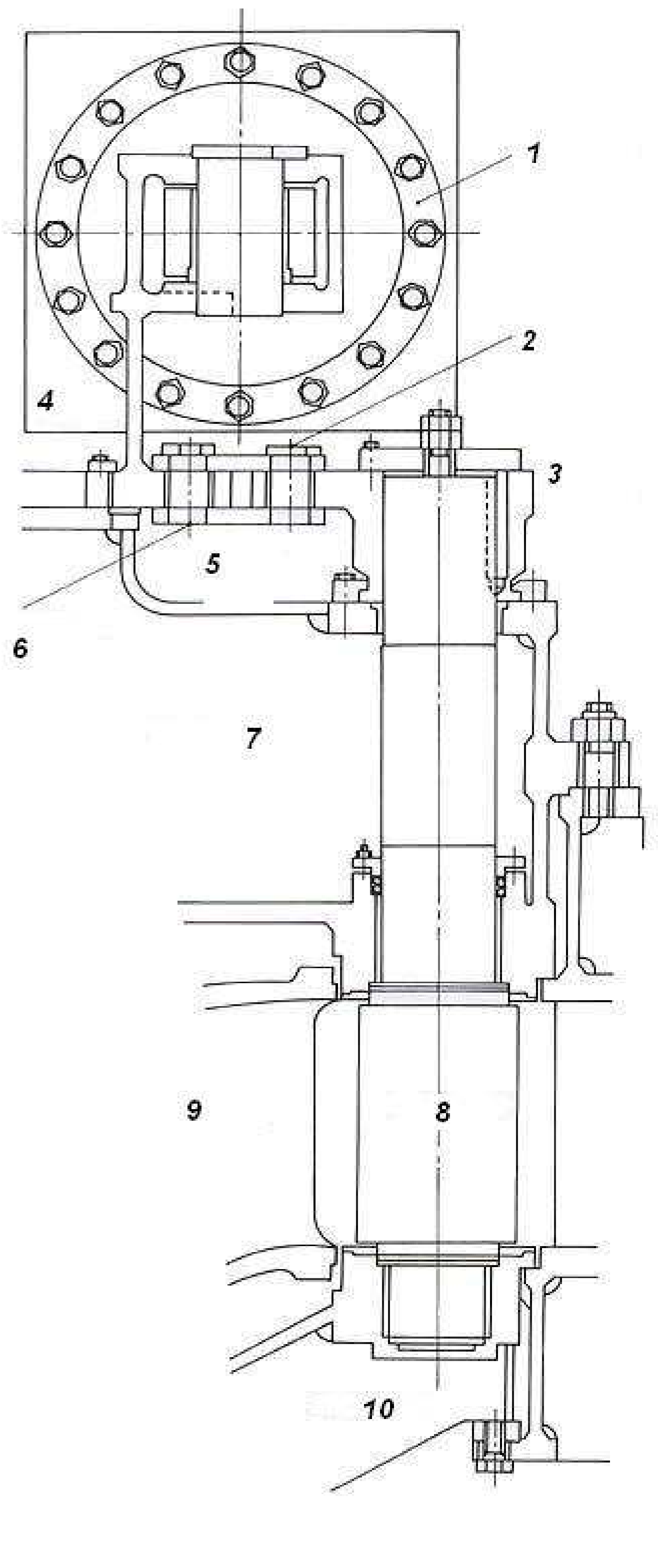

Fig.2. 2: Wicket gates operating mechanism: 1.servomoror, 2.shear pin, 3.gate arm, 4.gate ring, 5.gate link, 6.eccentric pin, 7.head cover, 8.wicket gate, 9.runner, 10.bottom ring [11].

The strong machine vibrations occur due to pressure waves that are appeared by the interaction between blades and wicket gates. The waves can be amplified to cause intense vibrations. The high velocity at the wicket gate outlet and the small radial gap between the blades rows lead to high rotor stator interaction of wicket gates in Francis turbines. The non uniform circumferentially of flow between wicket gates and runner creates the circumferential static pressure and velocity. For this reason, the accurate measurement of their effects on bearings is difficult. The small amplitude unintentional guide vane movements lead to the sliding distance greater than opening, closing and regulatory movements. It means that vibrations of the unit result from these unintentional movements and vice versa. The part of movements can occur without sliding due to the stick slip. The stick slip can change the real sliding distance.

A pump turbine in contrast to a pure turbine or a pure pump is not hydraulically optimal due to its double function. Because of the non-optimal exploitation of hydraulic conditions, the vibrations arise up to 100 Hz during the start-up and operation of these pumps turbines. The torsional vibrations on the bearings of the guide vane stems result in alternating stresses which can lead to some problems in the guide vane crank. These torsional vibrations produce increased abrasion on the guide vane bearing more than the bearing of a normal turbine.

In pump operation, pump turbines with fixed speed cannot be adjusted. Although wicket gates are provided on pump turbines primarily for controlling turbine load, wicket gates are beneficial for pump operation too. The position of the wicket gates can be regulated to obtain the best efficiency during the pump operation. With regulating the gate opening, the head in operation can be regulated. In pump operation at low head, the wicket gates have ability to decrease the power input, which result in the reduction in the capacity and consequently existence of cavitations. During the pump operation outside the design point, the reduction of efficiency for pump turbines with wicket gates is less than the efficiency reduction of pump turbines without wicket gates. For pump turbine with wicket gate, the shutoff heads are higher because the wicket gate limits the circulation of flow that leads to more efficiency conversion of velocity to pressure [10].

2.2. Grease Lubricants and Environmental Lubricants

Grease in the tailrace of power plants is an environmental concern. The greases must be evaluated on the basis of toxicity, mechanical properties and water quality to be environmentally safe.

Greased bushings have a continuous service capability of about 30 years. Although there are not any documentation or field experience that shows a long life period for self lubricated bushings, nowadays due to the environmental problems and continual maintenance of grease lubricated bushings, self lubricated bushings are preferred. Although the usages of greases increase the life time of bushings, many disadvantages can be seen in the use of grease lubricants.

The biggest problem is environmental, that greases, hydraulic fluids, and oil leaking from equipment may be carried into the waterway. The lubrication equipments require continuous maintenance and it lead to increasing costs and also continuous maintenance increase risk to maintenance personnel.

About the usage of greases or environmental lubricants, there is no clearly standard or law to define the attribute of an environmentally safe fluid and US federal regulations do not accept the term “environmentally acceptable lubricant” but manufactures and end users agree that lubricants should be biodegradable and non toxic.

There are some general agreements on the characteristics of an environmentally acceptable lubricant such as biodegradability and aquatic toxicity but they are not enough for environmental organizations:

a) Biodegradability: this is the chemical breakdown of materials by a physiological environment.
b) Aquatic Toxicity: this is a critical step when assessing whether a chemical is safe for living organisms.

The standard organizations always search to define various environmental criteria for these lubricants such as renewability content and restriction of hazardous substances.

2.3. Self Lubricated Bushings

2.3.1. Introduction

So far, many companies have provided the bearings of the guide vanes and the guide vane cranks with grease-lubricated bronze bearing. However, since the process water in pump turbine plants is pumped back and forth repeatedly, incessant lubrication produces unwanted contamination of water. To prevent this pollution, maintenance-free alternatives are being searched.

Because of unique tribological properties of grease lubricated bushings and possibility of greaseless lubrication and environmental pollution and maintenance costs of them in last decenniums, the interest to replace grease lubricated bearings with self lubricated in guide vanes of water turbines was increased. However, in some cases, the experiences of operators and manufacturers show that the greaseless materials did not meet durability requirements and were not specifically developed for high load, low speed oscillating operating conditions typical for power generation machinery.

A special testing device must be carried out to measure wear rate of self lubricating materials in conditions of high loads and small oscillatory movement. Most of the tests were carried out to examine long term performance, reliability, and environmental stewardship and evaluation of new materials for incorporation into hydroelectric. On the other hand no standard specifications or laboratory tests for such applications have yet been developed and widely accepted. The application of any bearing material requires accurate knowledge of the physical, chemical, absorptive, frictional, and wear characteristics of that material [8].

A large variety of greaseless bushing materials are available. However there are significant differences in material performance as well as discrepancies between manufacturers' performance claims and in-service performance. Some of the available greaseless bushing systems have higher coefficients of friction and/or higher wear rates than the manufacturers' published values due to corrosion of the shaft and sometimes swelling of the bearing material due to water absorption.

The key advantages of self-lubricating bearings include:

a) No requirements for ancillary lubrication systems
b) One time initial cost
c) Reduction in operation and maintenance costs
d) Elimination of environmental risk for lubricants being released into the water
e) Top of head cover and gate operating mechanism is clean which results in a reduced risk of accidents for maintenance personnel

These self-lubricated bushings often can be classified into the following two categories [13]:

1. Metal-based: this type of bushing consists of a bronze substance with a bronze-lubricant inner structure.

2. Plastic-based: this type of bushing consists of either a metal or plastic backing material with a Teflon or plastic lubricant inner structure.

Category	Type	Description	Example
Metal-based	Tin-bronze material	with graphite lubricant dispersed throughout	DEVA Metal Strip
	Solid type	with lubricant plugs	Lubron AQ30, DEVA Devaglide, Oiles, or Lubrite
Plastic-based	Teflon/Dacron fabric	glued to a metal or moulded phenolic backing	Fiberglide
	Teflon-Lead	overlay on a bronze inner structure with a steel backing	Garlock DU
	Teflon liner	Teflon liner mechanically held in a bronze bushing	Lubron AQ100
	Plastic – Synthetic	polymer alloy bushing, can be glued into a metal backing	Thordon or Delrin

Table.2. 1: The self lubricated bushings [13].

Various problems of self-lubricated wicket gate bushings stem from premature bushing failure and subsequent maintenance activities. Some types of plastic-based wicket gate bushings cannot be formed to compensate any lacks or damages caused during mounting. But metal-based bushings can be machined to compensate any imperfections. Since some plastic-based bushings are adhered to the bearing backing, glue failures have been reported. Certain types of solvents and oils can affect the bushing surface in some plastic-based type bushing materials.

The currently used types of greaseless bushings are Teflon containing types such as Fiberglide and sintered bronze alloys such as Deva Metal bearings, which contain graphite or other lubricants dispersed throughout the metal matrix.

2.3.2. General Experiences and Comments of Greaseless Bushings

The thin Teflon coating liner breaks apart on plastic type bushings such as the Lubron AQ100. Clearances can cause demolition of the bushing because of consecutively opening of the original design.

Problems sometimes occur in bushings with lubricant plugs. During the operation of metal-based solid bushings with lubricant plugs such as Lubron AQ30 or DEVA Devaglide, the lubricant plugs of these bushings become loose. Consequently, the wear rate increases and arises from metal to metal contact without the benefit of lubrication. Another problem is galvanic action and subsequent corrosion of the wicket gate stem mating material. This problem can occur because of the application of metal-based tin-bronze bushings impregnated with graphite.

Wicket gate bushings are usually in contact with water and external particles such as mud, silt and other external contaminants. External contaminants can influence the sliding performance and the endurance of the bushings. Although the wicket gate bushings can be equipped with seal to prevent the entrance of external particles, particles could pass the seal and enter the bushing gap. The metallic structure of the metal-based bushings such as DEVA Metal Strip does not allow external particles to enter in its sliding surface and bushing gap. And also, the metal-based bushings can be fabricated with self cleaning grooves. This property has more benefits against external substances in water.

Another important factor is the thickness of the sliding layer, which influence on the length of bushing life time. The thin sliding surfaces of plastic-based bushing are more likely to experience quick failure than those of metal-based bushings. Because of edge pressures on the bushing materials, the sliding layer may wear away, so the sufficient wear surface is so important, when the wicket gate deflection occur during operation.

Among the selecting criteria, friction coefficient is an important one. The load, velocity, and finishing of bearing and hardness of the mating material influence on friction coefficients. Friction coefficients of the plastic-based self-lubricating bushings are lower than the metal-based bushings. The wear rate criterion is important as much as friction coefficient criterion. Others important factor for selecting bushing material are hardness, corrosion resistance, and smoothness of finishing surface.

The use of corrosion resistant shafts or sleeves in contact with the bushing is importance. Various bushing manufacturers suggest a stainless steel mating material to protect against corrosion. For this application is recommended heat treated 17-4PH. 17-4PH is a precipitation-hardening martensitic stainless steel, which combines high strength and hardness (after heat treatment) with excellent corrosion. A hardness range of 271-301 BHN and the minimum surface finish on the mating material 16-32µin (0.4-0.8µm) is recommended [12].

In theory, almost all of these bushing can withstand high specific pressure. But because the effects of load and friction on wear and the dynamic hydraulic forces on the wicket gate bearing are not predictable, the actual load on the self-lubricated bushings should be meaningful lower that the specific pressure to ensure the long-life bushing service.

The effects of shaft deflection and edge loading directly influence the performance of any bearings material of wicket gates. These effects decrease the life time of bearings. The equipment that is misaligned because of machining errors, faulty assembly, or because of deflection or large bearing clearances, can cause edge loading of bushings, which may be several times the design load. The metal based bushings due to the metallic composition are much more resistant to edge loading than plastic based bushings with Teflon liners. The plastic bushings tend to deform under relatively low edge pressures. The effective edge load should not exceed the static compressive strength of the material.

Although the initial cost of a self lubricated bushing is higher than grease lubricated bushing, the continuous costs of lubricant and the maintenance costs will excess the price of self lubricated bushing.

For thick walled bushings L/D ratio of 1.0 to 2.0 is suggested and for thin walled bushings it ranges from 0.35 to 0.40 for diameters larger than 10 in (254 mm) and from 0.75 to 0.8 for bushing diameters up to 10 in (254 mm) [12].

Bushing manufacturers believe that their bushings need no seals to provide the entrance of water or other foreign matter and their negative effects, but researchers believe that seals are required where water or other contaminants may be present [12].

Sometimes friction coefficients in service are significantly higher than manufacturers' published values. Consideration of service condition is extremely importance. Sometimes the bearing materials have different friction coefficients in wet or dry conditions and also wear rates too.

The difference between static and dynamic coefficients of friction for a bearing material in a unit condition is important. This difference leads to stick-slip or stiction. The stiction can cause vibrations, noises and damages in the equipment.

During the installation, a very important factor to be considered is the bearing clearance, which is specified by manufacturer. It must be noted that some bushing material require greater operating clearance because of swell, which is due to absorption of fluid.

3. Material Analysis and Experiences

3.1. Test (Division of Machine Elements, Sweden)

3.1.1. Introduction

In institution of Division of Machine Elements in Sweden, material of four journal bearings, which are prevalent in trade, were examined in water lubricated conditions with an oscillating stainless steel shaft. Identical tests were performed using a bronze bushing with an environmentally adapted synthetic ester lubricant that has good oxidation stability. Tap water was used for water lubricated test. Finally the results were compared. All tests were conducted under boundary lubricated conditions. Although this test is simulated according to the Kaplan turbine, studying the results can help us get better understanding of the different materials and of their properties. Fig.3.1 shows the bearing materials which were examined in this test.

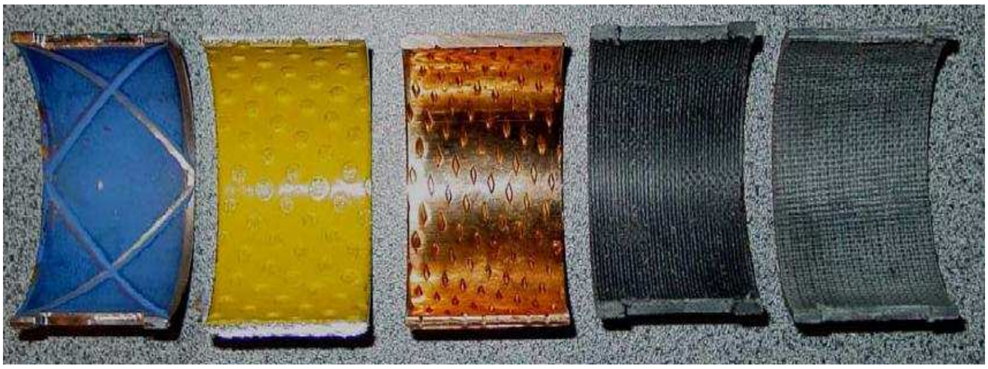

Fig.3. 1: From left: Deva, DEX, Tin-Bronze, Orkot and Tufcot [14].

The tests were carried out in a journal bearing friction and wear tester. The testing facility comprises two bearing halves pressed against a rotating shaft allowing for control and monitoring of findings such as friction, load, speed, temperature and decreasing clearance.

The shaft diameter was 40 mm and bearing length was 30mm. The shaft was made of stainless steel turned to surface finish Ra = 0.4 μm with hardness of 325 HV. The shaft diameter tolerance was f7 and the housing, where the bearing halves were clamped, had diameter of 44 mm with H7 tolerance.

Journal bearing	Description	Recommended lubricant	Thickness (mm)	Diametric clearance (μm)	Lubricant	Bearing pressure (MPa)
Deva BM018 CuSn8713/9PE	Bronze on steel backing with a run-in layer	Dry/Water	2	Min 25 Max 75	Water	19.8
DEX	POM lined steel backing	Grease/oil	2	Min 25 Max 75	Water	19.1
Tin-bronze	8 % Sn 92 % Cu	Grease/oil	2	Min 25 Max 75	EAL	19.4
Orkot TXM Marine	Composite with PTFE	Dry/Water	2	Min 25 Max 75	Water	19.5
Tufcot T100XM	Composite (undefined)	Dry/Water	2	Min 25 Max 75	Water	19.0

Table.3. 1: Tested journal bearing material [14].

The description of each material and the test conditions such as thickness, clearance, lubricant and bearing pressure are defined in table 3.1. The bearing pressure in table 3.1 is the average bearing pressure which was held around 20 MPa during the test.

The shaft motion pattern was a cycle consisting of a 60 degree back and forth motion with a 2 seconds pause before and after each direction change. After that, ten smaller ± 12 degree motions were performed with a 1 second pause between each sliding direction change. This pattern was repeated about 15000 times or until the test was stopped. This means that the total amount of motions was about 330000. The maximum sliding speed was 5.6 mm/s [14].

3.1.2. Results

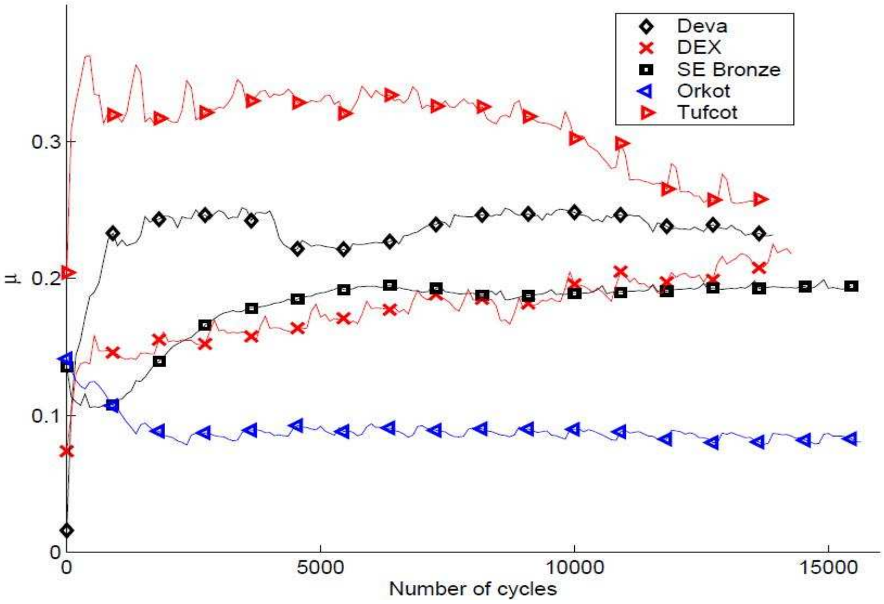

Fig.3. 2: Friction values during the test [14].

During the test, the Deva material had friction values approximately between μ=0.2 and μ=0.25. The DEX material introduced increasing friction values to above μ=0.2. The Tin Bronze material at first had friction values of approximately μ=0.1 but after a few thousand cycles, it increased to μ=0.2. The Orkot material showed the best friction values, which was about μ=0.1during the test. The Tufcot material showed the worst results in this test. It should be noted that friction values of materials can not be compared with other similar materials in other references. It should be mentioned that all test conditions are identical. Fig.3.2 presents the friction values during the test.

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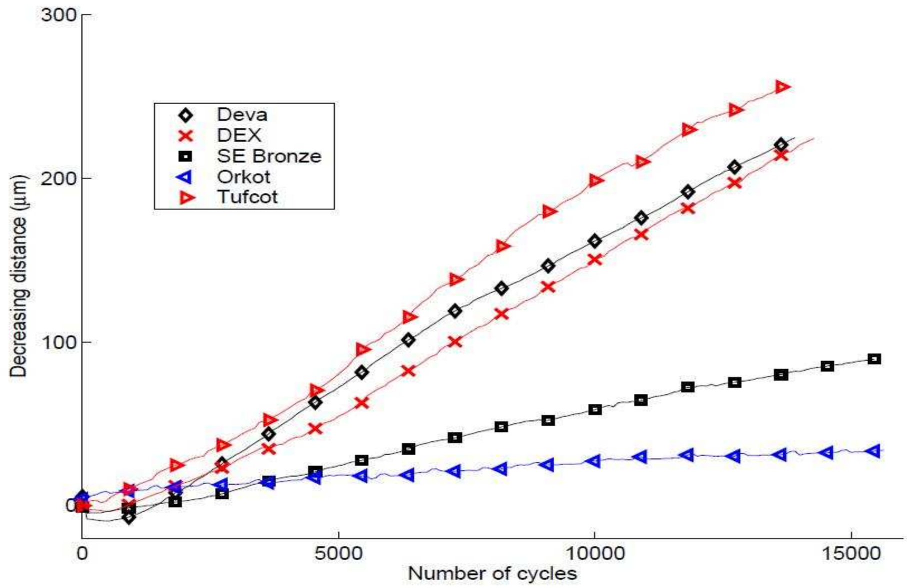

Fig.3. 3: Linear wear [14].

During the test, the distance between the bearing halves was measured to define the linear wear. Fig.3.3 presents the linear wear of materials during the test. At the beginning of some tests, an increasing distance has been observed. This is due to thermal expansion of the shaft. If the rate is lower than thermal expansion, the linear wear becomes negative. The Deva, DEX and Tufcot materials had similar curves. They showed relatively high wear rates. The best result was for Orkot material. During the test, the wear rate of Orkot material was almost constant and also it has the lowest wear rate. The maximum and minimum length (width) of the bearing materials was measured. Only the Tufcot material had a dimension change after the test. Fig.3.4 shows worn bearings after the test.

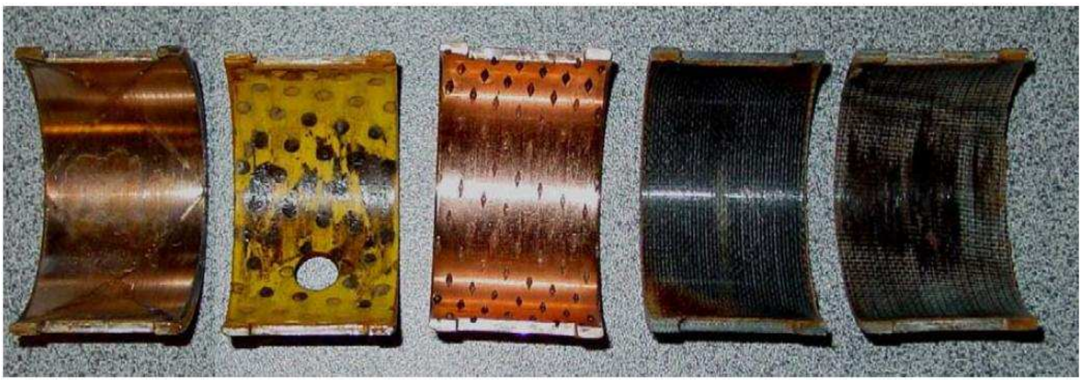

Fig.3. 4: Worn bearings, from left: Deva, DEX, Tin Bronze, Orkot and Tufcot [14].

3.2. Test (Gdansk University of Technology, Poland)

3.2.1. Introduction

This test was carried out at Gdansk University of Technology to measure wear rate of self-lubricating materials in conditions of high loads and small oscillatory movement, which turned out to be dominant component of movements in guide vane bearings. The testing facility (SOOG) was constructed to simulate sliding contact conditions of water turbine bearing in guide vane elements. Four dry bearing materials were examined in this test. The bearing materials included Glacier Devatex, Tenmat Feroform T814, OrkotTXMMarine and Kamatics KPD2724.

Fig.3.5 shows the testing facility in this test. The tests were carried out under following conditions:

- Contact pressure of about 29 MPa
- Temperature of almost $53^{\circ}C$
- Reciprocating movements with amplitude of $\varepsilon = 120\mu m$
- Surface finish of counter specimen with $R_a = 0.8 - 1.2\mu m$

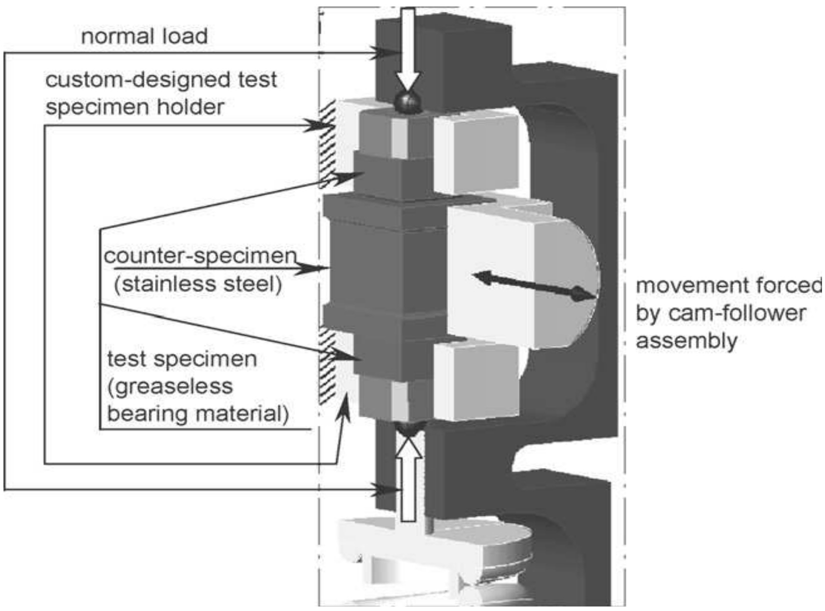

Fig.3. 5: Part of test of head for testing self-lubricating materials under oscillatory movements to simulate power plant conditions [8].

3.2.2. Results

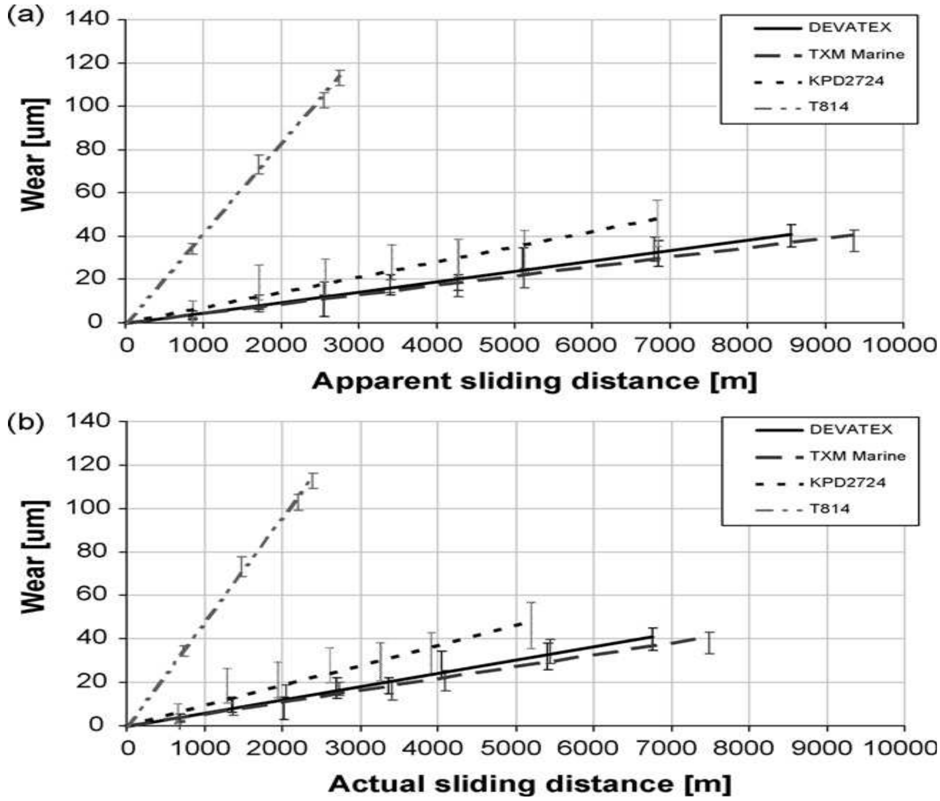

Fig.3. 6: Wear of materials with high resistance to contact pressure vs. sliding distance: (a) apparent; (b) actual [8].

The fig.3.6 shows the wear of materials against the apparent sliding distance and the actual sliding distance. In oscillatory movement for elastic materials, the actual and apparent sliding distance occurs, when stick slip is present. The error bars on the diagram define the difference between maximum and minimum wear obtained in the tests. In this test, the Tenmat T814 showed the worst result and other materials had approximately similar results, but after the test, according to experiences, the Devatex material was selected to use in a water turbine in Hydro Electro Power Plant Solina in Poland.

On the study it was mentioned that at the time of the paper preparation, the bearings have been in operation for 5 years without failure but it was not specified that the bearing was used for which part of guide vane (lower, middle or upper bearing) [8]. The water power plant in Solina (a water storage station) was put into operation in 1968. It contains two classic reversible pump turbines with the total power output of 136 MW. In 2003 the modernization of the plant was completed and its output was increased to 200 MW, while electric power production grew from 137 to 239 GWh and the efficiency of turbines was increased as well.

3.3. Test (Powertech Inc.)

3.3.1. Introduction

The other test that Powertech Inc. has published evaluated the following bushings on the basis of wear, coefficients of static and dynamic friction, and temperature rise: Deva metal, Oiles SP500, Lubron AQ100, Lubron AQ30, Fibreglide 64, and Thordon SXL.

The tests were carried out under the following conditions [15]:

- Constant radial bushing pressure: 3000 psi (20.7 MPa) radial over the projected area of the bushing.
- Minor oscillations: ± 2 degrees continuously at a 2 Hz frequency except during major swing.
- Major swing: ± 15 degrees once every 15 minutes.
- Length of test: 120 hours (approximately 2 years of normal operation of a typical unit).
- Tests were run under both wet and dry conditions.

3.3.2. Results

The Thordon SXL bushing had insignificant wear and had the second lowest coefficients of friction of the bushings tested. This bushing is a polyurethane based resin bonded to a bronze backing. Polyurethane falls under the category of elastomeric resins and would be much more resistant to fatigue and brittle failure than epoxies or phenolic resins. This would be more important in oscillating applications like the wicket gate than in rotating applications. The manufacturer claims that the Thordon SXL bushings function in an abrasive environment don't need install seals and can be machined easily to finished sizes using conventional metal working tools. This style of bushing appears to be suited for wicket gate application and moist environment with cyclic motion.

The Fibreglide bushing showed the best results during the tests. This bushing exhibited the lowest coefficients of friction and insignificant wear. However, the Thordon SXL bushing was recommended for wicket gate service. The Fiberglide bushing is sensible against wet conditions and contaminants in water. As a result, the bushing seal is required for this bushing in wet condition. It leads to additional cost in material, increase in maintenance cost and installing time for these bushing materials.

Other materials in this test did not show good results as well as the Fiberglide bushing and the Thordon SXL bushing. The Deva metal bushing had the highest friction coefficients and wear rate. The Lubron AQ100 had a stick slip problem that is due to high difference between static and dynamic friction coefficients.

3.4. Test (US Army Corps of Engineers, at Powertech Laboratories)

3.4.1. Introduction

There are different self lubricated bushing materials in market but usually deviations occur for bushings performance between manufacturers' claims and in-service operation. The U.S. Army Construction Engineering Research Laboratory thinks that a series of tests to standardization the bushings materials can be useful for engineers and designers, although, it is almost impossible to test all of materials in market. This department has chosen various materials to test according to commercial availability, current applications and overall field performance.

Corps experiences with self lubricated bushings have been negative. Corps experiences have shown that most bearings using lubricant plugs perform poorly in small movement applications and this is almost the only kind of motion that occurs in the targeted hydropower applications. Tested and classified materials help engineers to choose the material for special applications. To select a bushing for any application, it is required to know the physical, chemical, absorptive, frictional, and wear characteristics of the material. It must be noted that to compare the different materials, all bearings must be examined under exactly the same conditions and also these conditions must be closed to service conditions. However it is a big problem to simulate real conditions for a test. It leads to different material characteristics between operating and manufacturers' published values such as friction coefficients and wear rate and also service life of these bushings.

This department has examined the chosen materials for the following parameters:

a) Coefficients of friction, both wet and dry
b) Wear rates, both wet and dry
c) Swell in water and oil
d) Bearing and bearing bond resistance to vibration
e) Resistance to edge damage

The test condition consists of a heat treated stainless steel with an inside diameter of 5 in (127 mm). The static loading of the test bushing is upto 3300 psi (22.737 MPa) with a plus and minus load of 1000 psi (6.890 MPa). It produces a load range between 2300 psi (15.847 MPa) and 4300 psi (29.627 MPa). The oscillating loaded sleeve produces a rotation of 8 degrees per second. The tested bearings are 5 in (127 mm) in diameter by 3 in (76.2 mm) in length. Every 15 minutes the test sleeve is rotated through plus and minus 15 degrees. Standard time for tests was 144 hours, 24 hours for the set and creep test and 120 hours for the friction and wear rate test. The wear rates are formulated in mils per 100 test hours (0.025 mm per 100 test hours) to evaluate the life time of bearings [12].

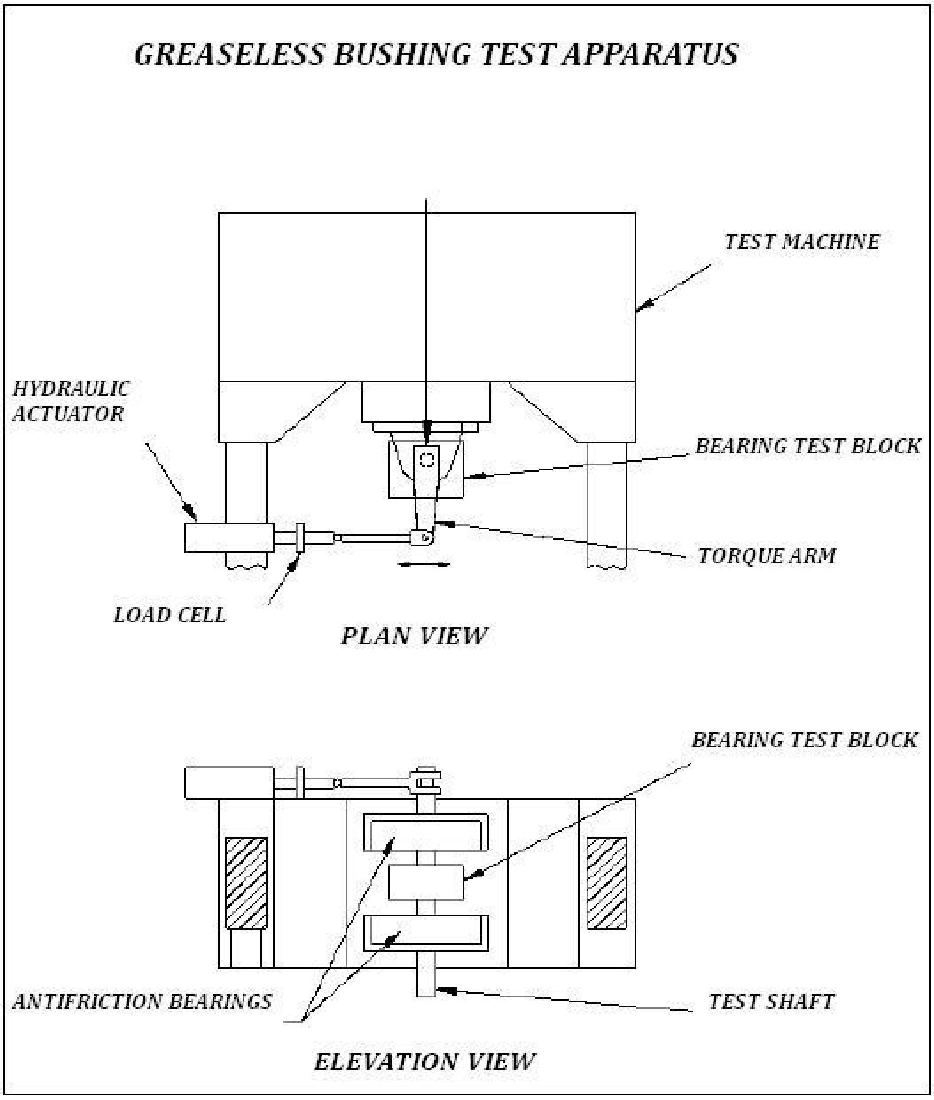

Fig.3. 7: Greaseless bushing test apparatus [12].

The test apparatus, in fig.3.7, consist of a shaft supported by two antifrictions, self aligning, and double roller bearings capable of sustaining the test radial load of 52000 lb (23556 kg) plus the superimposed load of 15760 lb (7139.28 kg). The bearing test is mounted between two antifriction bearings. With using a hydraulic cylinder, the radial load is applied. A torque arm provides the oscillation of shaft for the test. The hydraulic cylinder and the load cell control the torque arm. The superimposed variable load will be provided by an additional hydraulic cylinder. Cooling water shall be provided for the test sleeves to keep the sleeve and the bushing temperature below 35˚C during the test.

3.4.2. Results

Several bushing materials have acceptable friction coefficients for wicket gate application. However materials with least stiction are recommended by researchers. The stiction or stick slip occurs due to difference between static and dynamic coefficients of friction when a system starts to work. This test showed that friction coefficients of tested materials are often higher than manufacturers' published values. The values of friction coefficients are entirely dependent on the working conditions. This test showed that the bushing materials have different coefficients in wet and dry conditions, while wear rates are the same. It must be mentioned that a lower friction coefficient is not equivalent to lower wear rate. The use of stainless steel shafts and sleeves and also seals for bushings help to avoid the increasing friction coefficients. The wet conditions cause corrosion of any non stainless steel shaft or sleeve and leads to increasing friction coefficients.

In this test and in wet conditions, the least static coefficient of friction was for Fiberglide material and the least dynamic coefficient of friction and wear rate were for Tenamt T814 material and in dry conditions, the least static coefficient of friction was for Tenamt T814 material and the least dynamic coefficient of friction was for Karon V material and the least wear rate was for Fiberglide material. Fig.3.8 and fig.3.9 show the friction coefficients of materials in wet and dry conditions. And fig.3.10 shows the wear rates of materials in wet and dry conditions. As you can see in fig.3.10, the wear rate of Fiberglide bushing in dry condition, a wear rate can be negative, which means that the wear rate is less than the amount of expansion, depending on the swelling and thermal expansion.

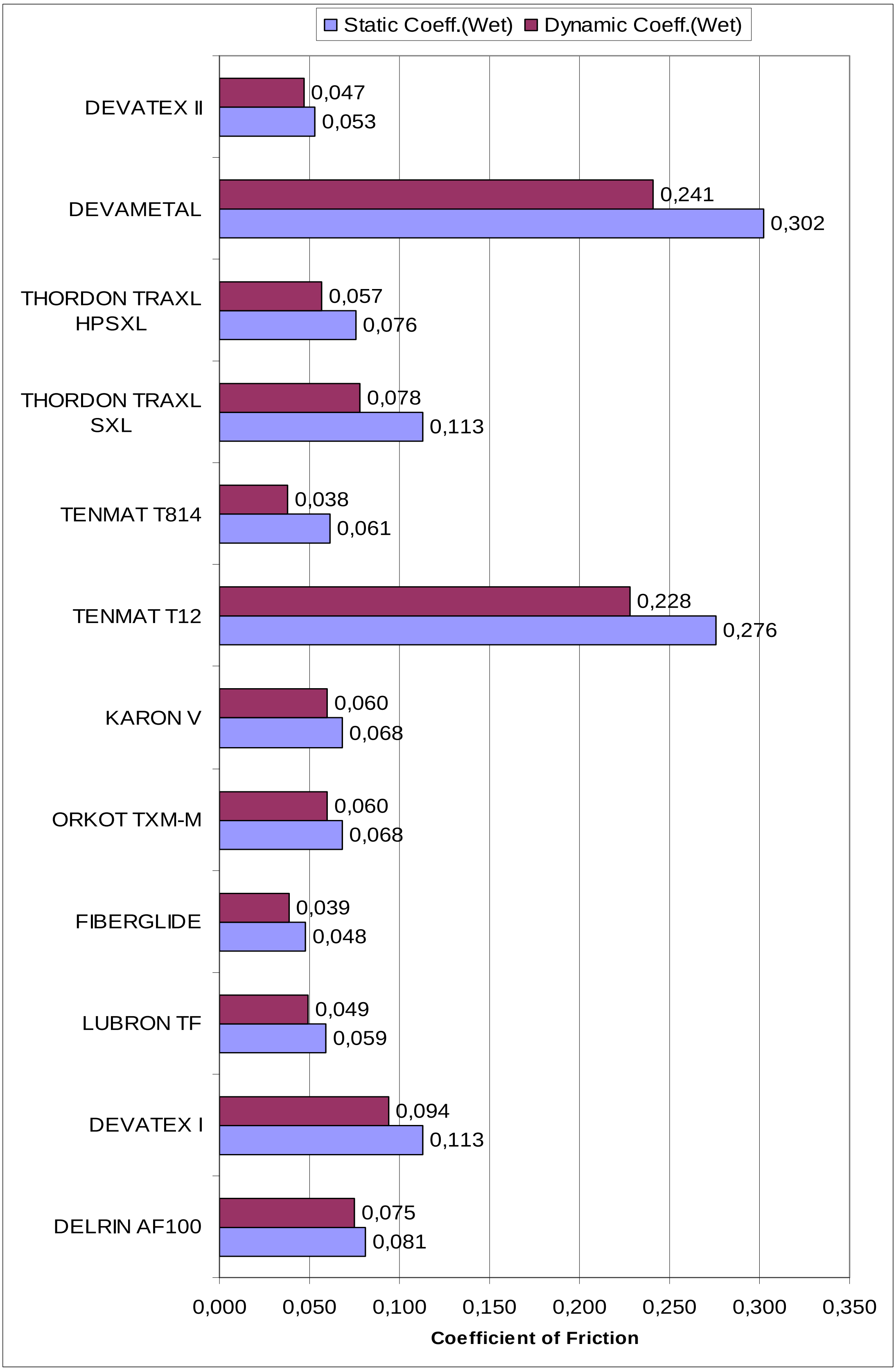

Fig.3. 8: Friction coefficients of tested materials in wet condition [12].

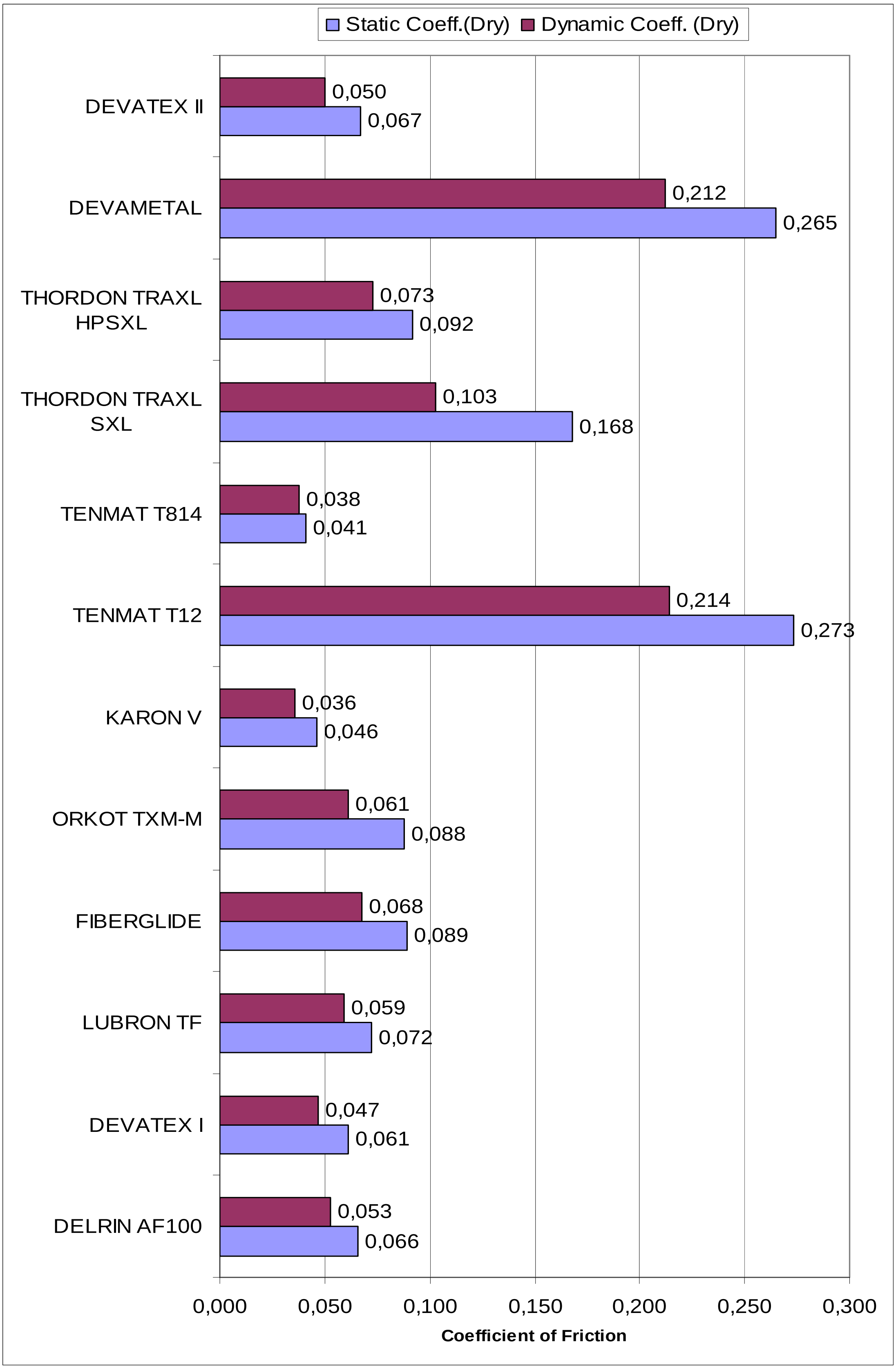

Fig.3. 9: Friction coefficients of tested materials in dry condition [12].

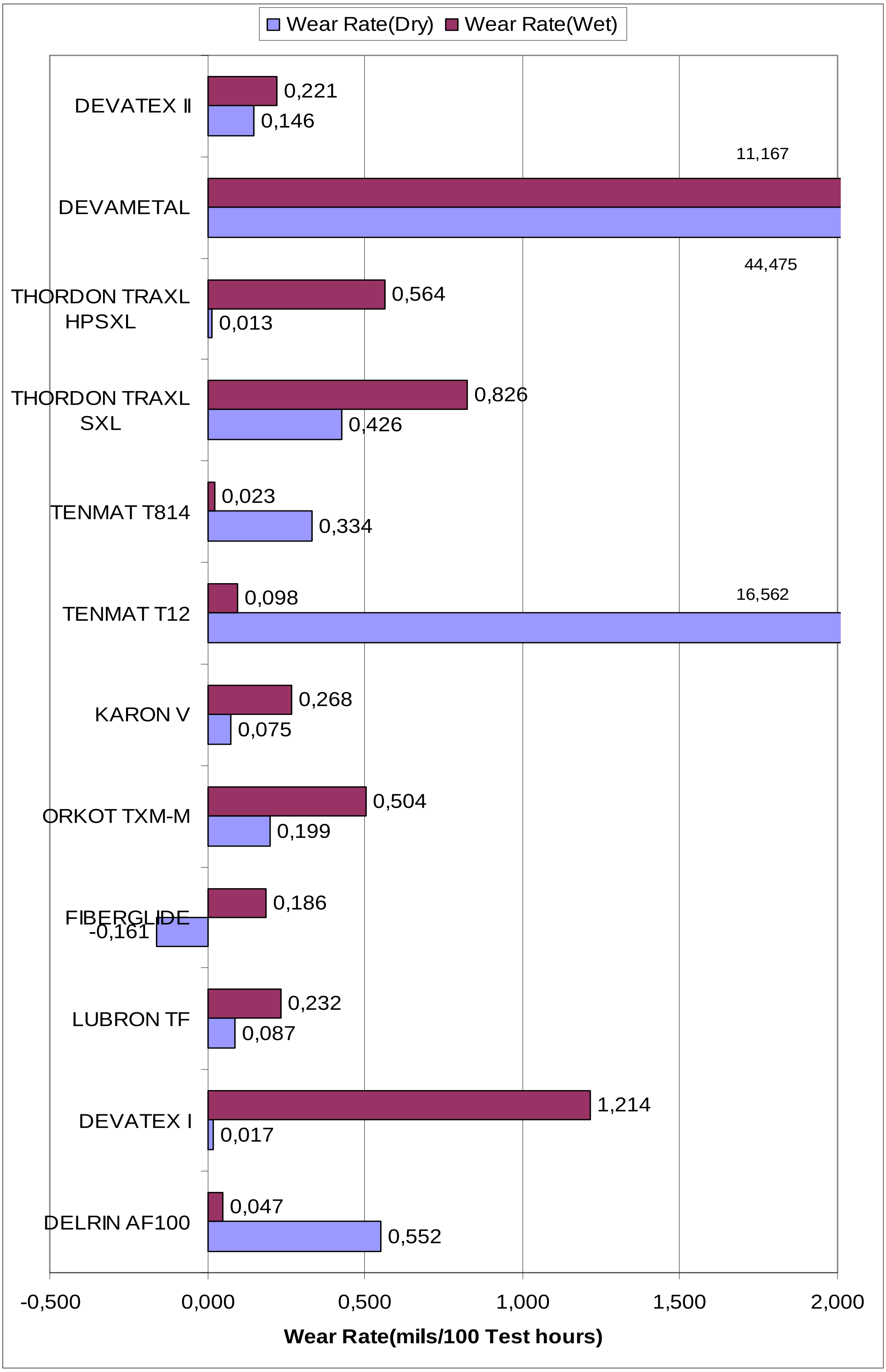

Fig.3. 10: Wear rates of tested materials [12].

Another test was carried out to determine the swell, change in thickness and length for each material. The test was carried out in water and oil conditions. Distilled water was used for water condition and R&O 68 TURBINE OIL for oil condition. The test showed that between 90 and 95 percent of full swell has occurred by the end of 8 months and full swell occur in less than 14 months. After the test in water condition, the Orkot TXMM material and the Tenmat T814 exhibited a decrease in volume. Almost all of materials in oil condition exhibited a decrease in volume. Fig.3.11 and fig.3.12 show the suggested diametrical allowance for swell in water and oil conditions. The materials need negative clearance to install, which shows a decrease in volume, respectively they need pre-load to install.

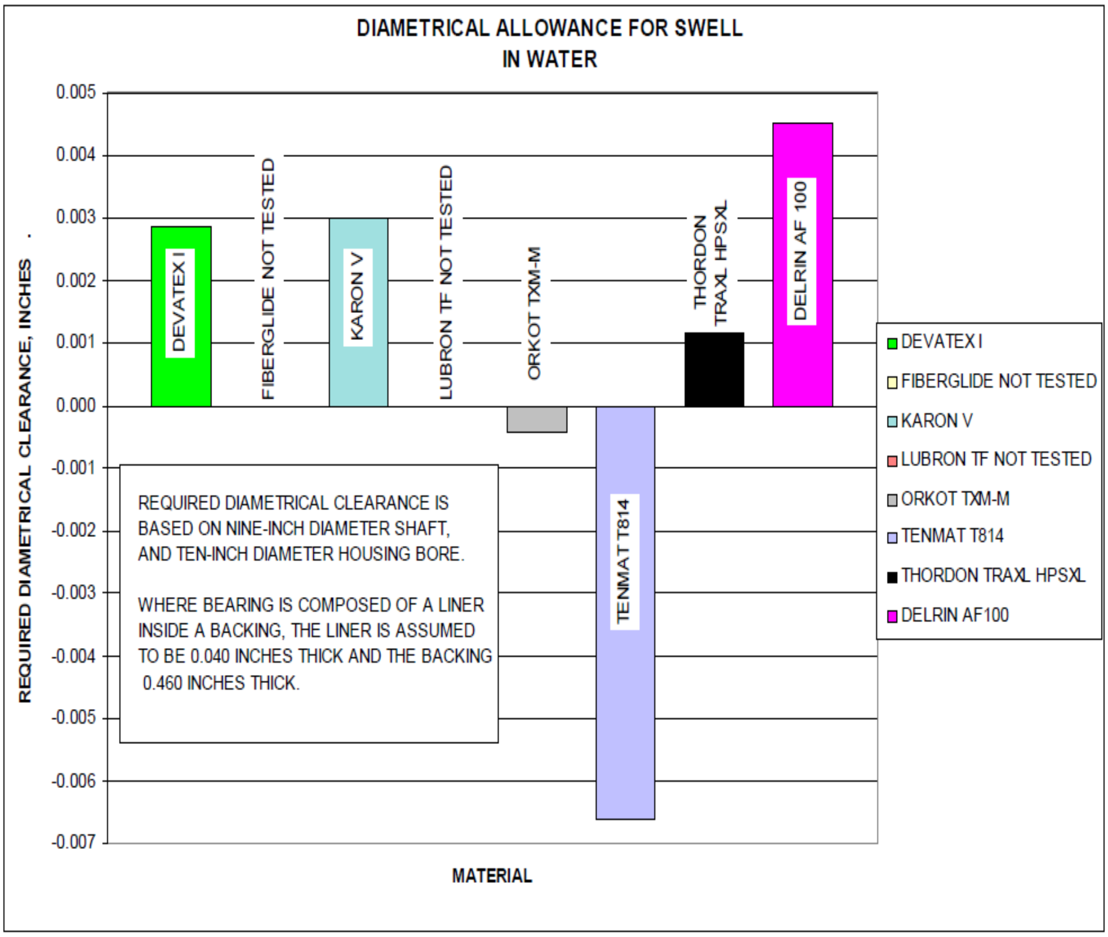

Fig.3. 11: Diametrical allowance for swell in water [12].

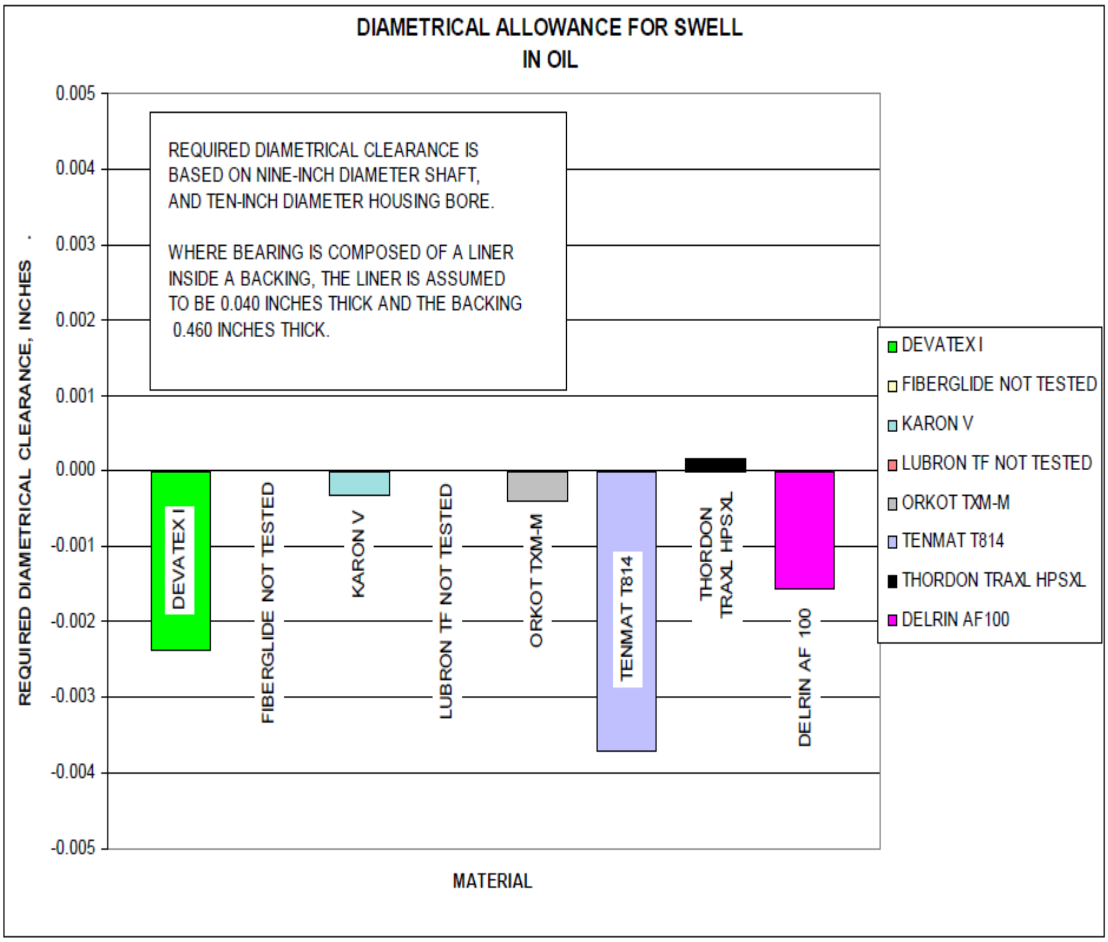

Fig.3. 12: Diametrical allowance for swell in oil [12].

Finally all of the results were accumulated and for each test a point was noted. For example in static and dynamic friction coefficient test, for greased Bronze 50 points were recorded. Only material that has same value gets 50 points, and for each 0.01 less than that value gets extra 1 point, and for each 0.01 more than that value gets negative 1 point, and also for other example in wear rate test, for greased Bronze put 100 points and only material that has same value gets 100 points, and for each 0.02 mil per 100 test hours less than that value gets extra 1 point, and for each 0.02 mil per 100 test hours more than that value gets negative 1 point**.** Also separately for various applications an extra rating system was considered. A rating system for upper, intermediate and lower wicket gates stem bushings applications was considered according to damage susceptibility relating to peelability. If lubricant layer peelable from substrate doesn't occur, no point is subtracted from material, if it is difficult, 50 points are subtracted for upper and intermediate bushings and 100 points for lower bushings, and if it isn’t difficult, 100 points are subtracted for upper and intermediate bushings and 200 points for lower bushings. Each material from each test has a point and the ranking of the materials for each application was showed on the diagram. These rankings are visible in fig.3.13, fig.3.14 and fig.3.15. From the chart it can be figured out that the greaseless bushings for use in hydropower applications vary widely in performance and the performance of most individual products is dependent on whether the operation is in wet or dry conditions, or is edge loaded.

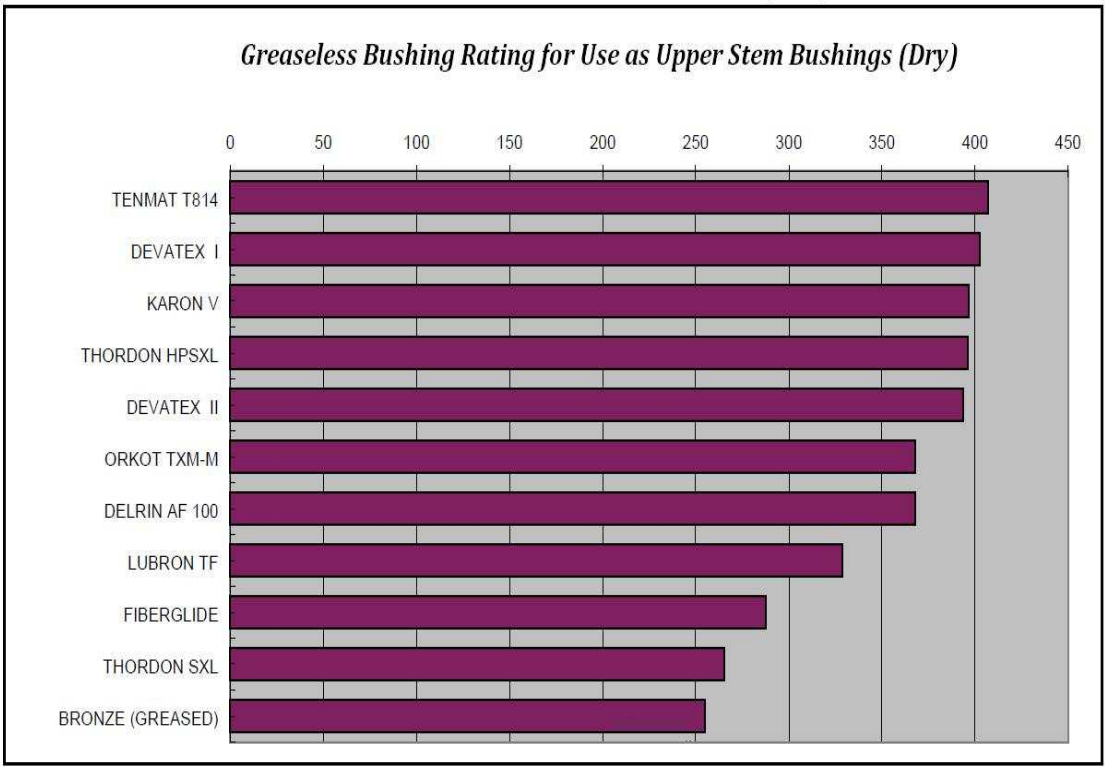

Fig.3. 13: Greaseless bushing rating for use as upper stem bushings (dry) [12].

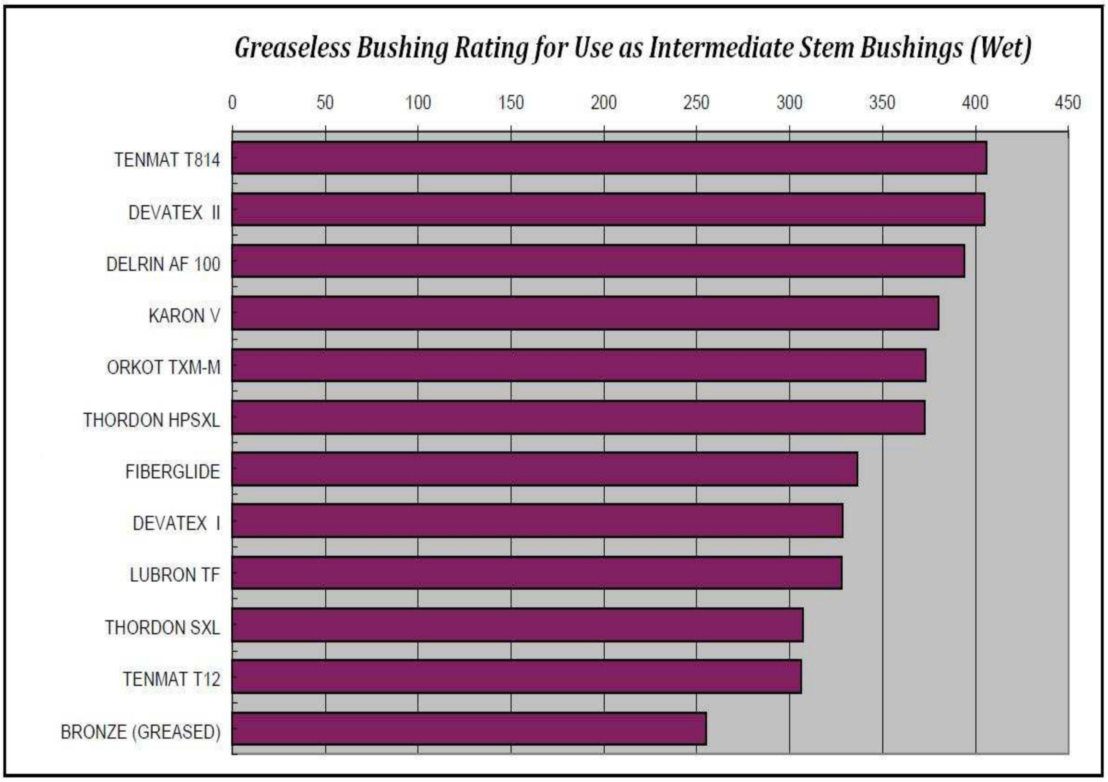

Fig.3. 14: Greaseless bushing rating for use as intermediate stem bushings (wet) [12].

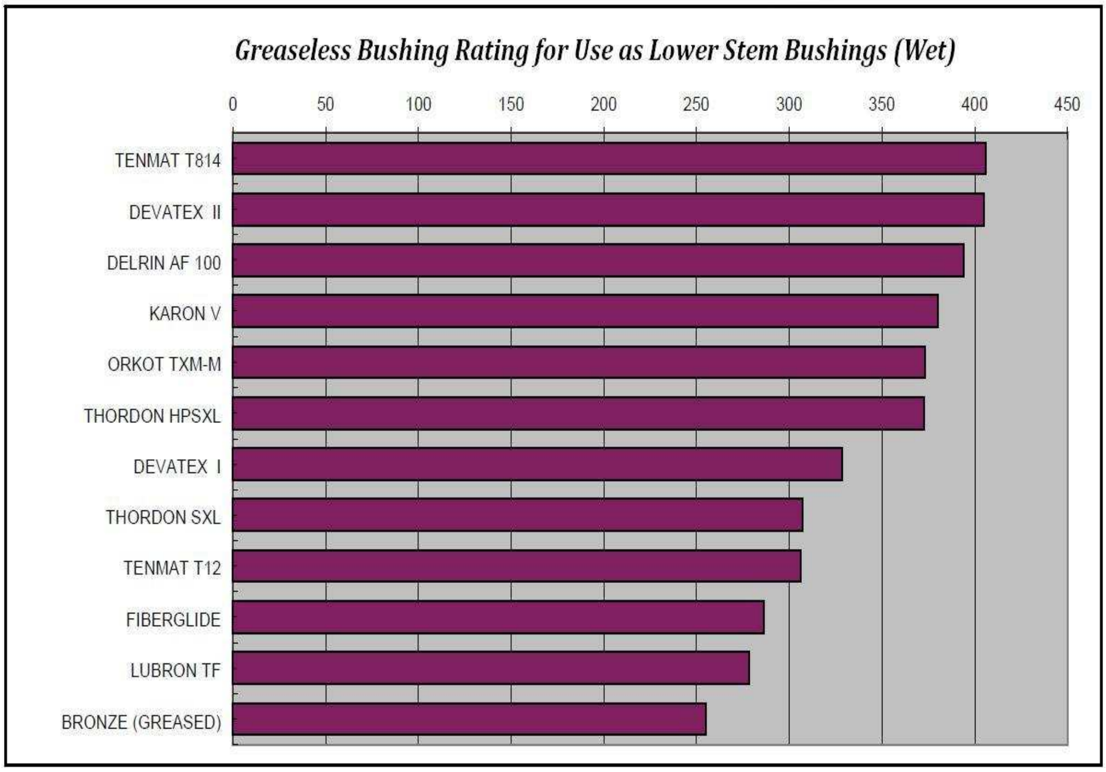

Fig.3. 15: Greaseless bushing rating for use as lower stem bushings (wet) [12].

3.5. Review and Comparison of Tests

Due to differences in test conditions, the accurate comparison of these tests is almost impossible but the tests can be compared relatively.

In test of *Division of Machine Elements, Sweden*, section 3.1, it can be seen that for example, the Orkot TXMM bushing material is recommended for both wet and dry condition from the manufacturer. However, test of *US Army Corps of Engineers, at Powertech Laboratories*, section 3.4, showed that this material like other materials has difference results in different conditions.

The Deva metal, Fiberglide and Thordon SXL material were tested in test of *Powertech Inc.*, section 3.3, and test section 3.4 These materials in both tests have relatively the same results. In test section 3.3 it is concluded that according to experience, the Thordon SLX is recommended for wicket gate service against Fiberglide. The ranking table in test section 3.4 confirms this assertion.

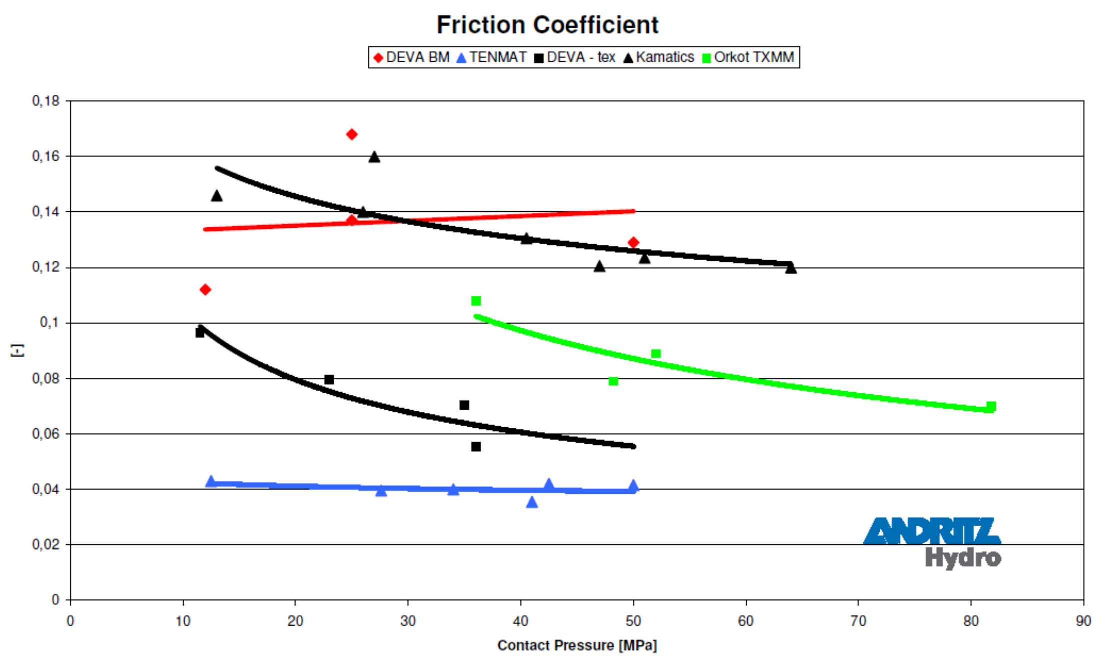

Fig.3. 16: Friction coefficients of materials from ANDRITZ HYDRO GmbH.

This diagram in fig.3.16 is a result of a test but doesn’t define the test condition. Both of this diagram and test no.1 confirms that the friction coefficient of Deva BM is higher than the friction coefficient of Orkot TXMM.

Another problem in diagram of Andritz is that the type of materials is not clear. In this diagram it is showed that the Tenmat material has the least friction coefficient. On the other hand the test no.4 shows that there is large difference between types of Tenmat materials, for example, Tenmat T814 and Tenmat T12.

There are three similar materials between the test of Andritz and test in section 3.4, Devatex, Tenmat and Orkot TXMM materials but it is impossible to compare these materials. The Devatex (II) and Devatex (I) are the old names and nowadays (I) and (II) are for Devatex indistinctive. Moreover the type of Devatex in diagram of Andritz is unclear. As previously noted, the type of Tenmat is unclear too.

3.6. Experiences - Generally

In following section some experiences of the operator and manufacturer are mentioned. Although some of the experiences don’t concern reversible pump turbine power plants, readers can have an overview of durability and performance of various conventional guide vane bearings.

The Allianz Insurance Group in Germany found out that the overstressing and lack of lubricant are the primary causes of damage in wicket gate bushings. Allianz has found that metal-based bronze bushings achieve a durability that is 0.5-2 times longer than that of bushings with Teflon Inserts. Because of a rigid metal-based self-lubricating layer of sufficient thickness is destruction of the bushing in the presence of extreme edge pressure or special operating conditions practically impossible and metal-based bushings are not suddenly destroyed.

The turbine manufacturer, Voith of Germany, has carried out field tests on the wicket gate stem bushings at the Alzwerke GmbH turbine unit in Germany. The manufacturer has installed different types of self-lubricated bushings and the test results have indicated that the metal-based Tin-Bronze graphite impregnated wicket gate bushings manufactured by DEVA were superior to other types. Until this report was published, these metal-based bushings have been in service for 18 years.

The Canadian utilities, Ontario Hydro and B.C. Hydro, have chosen Thordon plastic-based wicket gate stem bushings for their turbine units. Because of less susceptible to damage during Installation, they prefer Thordon over a material such as Fiberglide. These Thordon bushings have only been with no problems in service for 5 to 10 years.

The B.C. Hydro reports the failure of Lubron AQ100 wicket gate stem bushings in the units of Schrum Power plant. These Lubron AQ100 bushings failed when corrosion occurred on the carbon steel wicket gate stems which caused the bushing Teflon liner to flake off and compress into the bushing operating clearance. At this point the wicket gates could not be opened by the servomotors.

B.C. Hydro has carried out tests on several different types of self-lubricated bushings. They extracted that articulately plastic-based bushings such as Thordon and Fiberglide had a lower coefficient of friction than the metal-based DEVA Metal Strip.

The United States Bureau of Reclamation has three facilities that have self lubricated wicket gate stem bushings. The Grand Coulee power plant with a bushing consisting of a Teflon liner bonded to a filament wound composite backing in 1983, the Heart Mountain power plant with Lubron AQ100 bushings in 1991 and the Stampede power plant were retrofitted with Fiberglide bushings in 1987. The units at these installations are all less than 15 MW and no bushing problems have been reported (The report was published in July 1992) [13].

In 1986-87 the BC Hydro experienced self lubricating bushings (Thordon SXL) at the Gordon M. Shrum generating station but approximately after two years, shear pins were failing on the wicket gate shafts. This was traced to increased friction from the wicket gate shafts. The Mica generating station of BC Hydro has the experiments about self lubricating bushings in the wicket gates that it has four Francis generating units, each rated at 434 MW. Thordon SXL was used for unit 3 in 1990 and for unit 4 in 1992. On both machines, approximately after five years of service because of breaking the upper wicket gate bushings were changed. The intermediate wicket gate bushings were functioning correctly. For upper wicket gates were alternative materials tried. Tenmat T814 for unit 3 and Thordon HPSXL for unit 4 were chosen which have rated higher in the testing and rating system. After three years and one year respectively, the new replacements have been satisfactory [16].

In 1975 Escher Wyss equipped the turbine 10 of equipment Vianden in Luxembourg with Fiberglide bearings type 64. This system has worked correctly until (1980). In Austria the pump turbine Rodund, the next machine, was equipped with Fiberglide bearing type 64.

In Rodund complex (Voith) the lower bearings due to wear in autumn 1979 were replaced after a period of four years. Half of the installed bearings were with reinforcement type 34 and the rest of them were with reinforcement type 44.

Experiences of Rodund in February 1983 are as follows: in Rodund, Fiberglide bearings and Dacron bearings are installed. Because of the abrasion on the bearing, some places faced high stress. The danger began on the pressure points in which the blade stem suffers an abrasion and the shaft goes wavy.

By the installation of new bearings on these places huge surface pressure occurred that leads to a rapid destruction of the bearing. This waviness in Rodund existed on the blade stems and was discovered too late.

The previous wear damage only affects the bearing on the lower lid, while the double bearing in the upper lid was without any problem. In addition to high surface pressure, the cause of the damage is the intrusion of dirty water through O-ring-seal. After the exchange of O-ring by groove ring sleeve with self-lubrication features, a significant improvement was expected [17].

The experience in Rodund as well as in Vianden indicates an advantage of the Fiberglide bearing opposite to the Dacron bearing.

3.7. Experiences – Vianden – Machine 10

The pump turbine is equipped with 16 guide vanes. These vanes are all made of 17.4 CrNi. The guide vanes have expandable bearing seal housings. In initial assembly, bearing seal housing is aligned by a plastic pipe. A good alignment is very important for the radial preload of the seal-gasket washer of the guide vane bar. This solution has been widely used until today. The track guiding is formed on the bottom bearing, see fig.3.17. The guide vanes will be pressed on the thrust bearing with pressure relief under the lower blade plate.

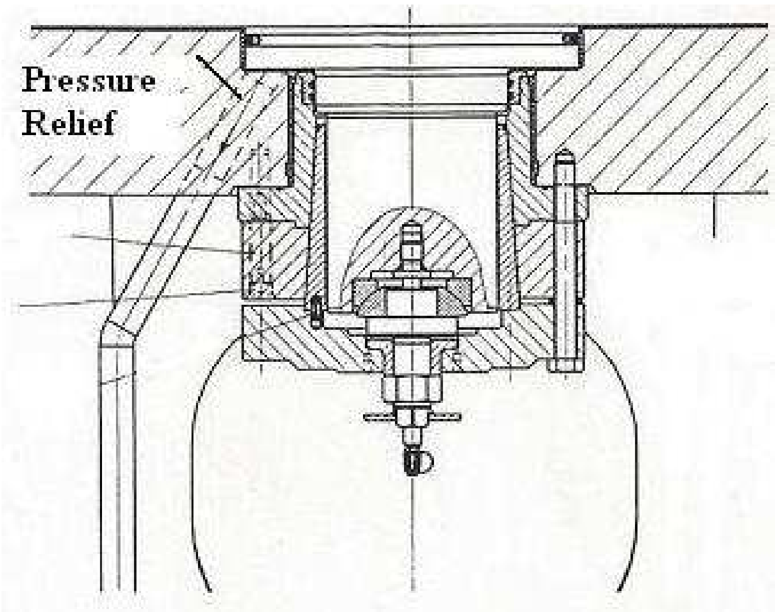

Fig.3. 17: The schematic of a wicket gate and track guiding on the lower bearing [17].

Originally the bronze bearing bushings were made of RG 7 and equipped with Teflon foil with 25% reinforced glass fibre as stock material. After first damage in 1976 it was decided to replace Teflon film with Fiberglide. But even this replacement Fiberglide and the improved Dacron couldn't solve the problem.

The lower bearings had to be replaced regularly at two year intervals. In August 1987, a series of tests was started with bearings made from different materials and even different preloads.

In order to measure and reduce the damages caused by bearing, a vibration sensor was located at the end of the shaft of guide vane. This sensor is shown in fig.3.18. By a modification to the pump starting-method it was possible to reduce the vibrations significantly.

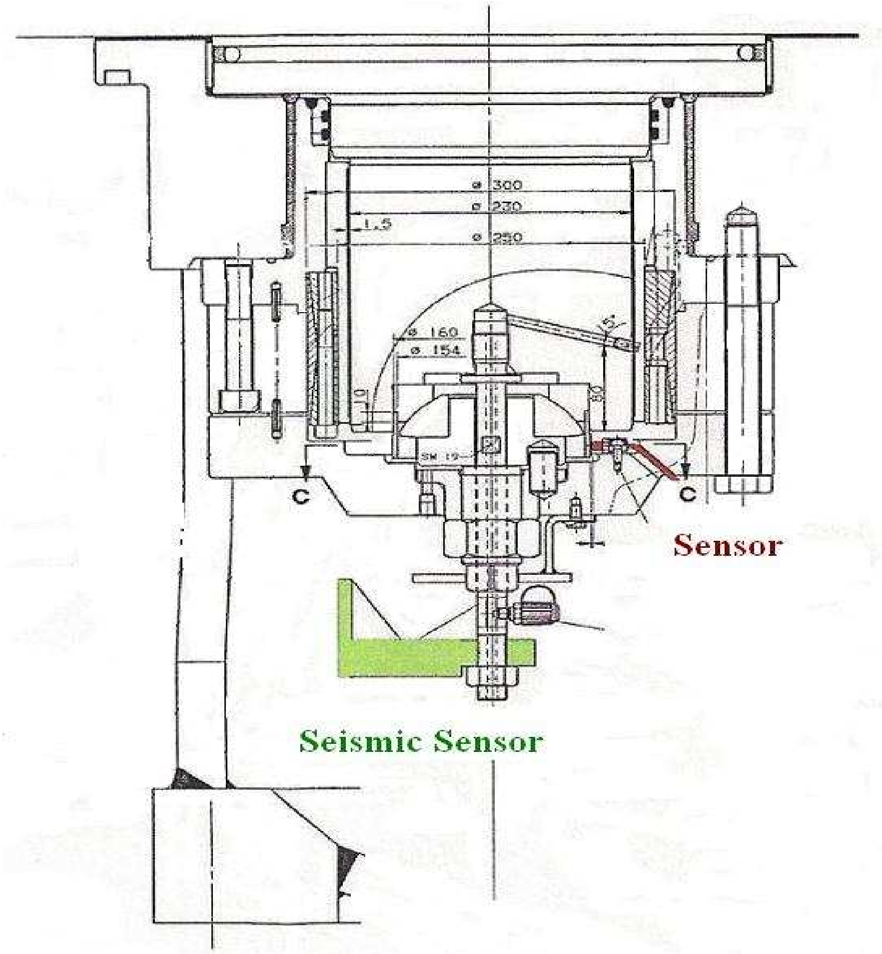

Fig.3. 18: The sensor [17].

A slight improvement has been achieved by optimizing the power switch at the pump stabling. Then relative vibration transducers were installed in the bearing seal housing in 2 axes. Here it was found that the vibrations were strongly absorbed with bearings that have a lower clearance. The torsion vibrations are cushioned as a result of the restraint bearing. Axial vibrations appear to play a secondary role in the development of damage.

It is therefore likely that the cause of damage primarily is due to a combination of strokes by bending vibration and torsion vibrations.

Two preloaded bearing types were used as prototypes, see fig.3.19:

1. Vulkollan pre-stressed bearing
2. Bi-Kon pre-stressed bearing

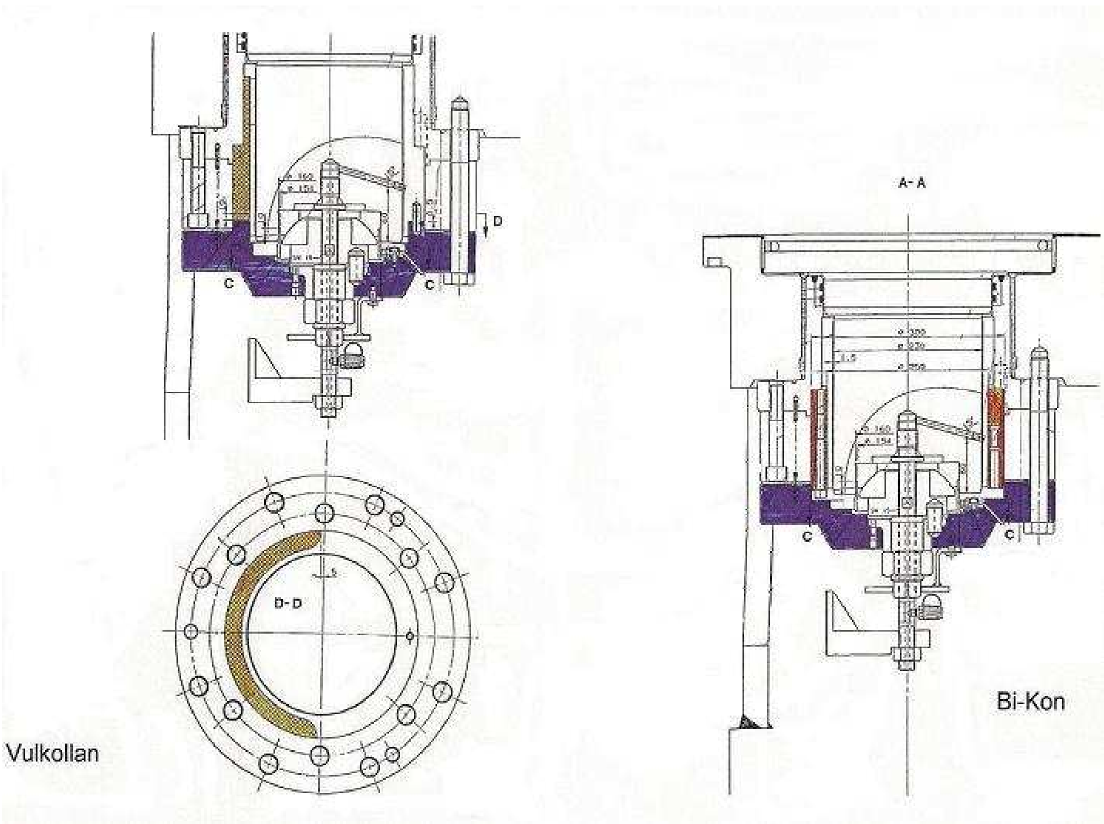

Fig.3. 19: Pre-stressed bearings by Vulkollan and Bi-Kon [17].

The Vulkollan pre-stressed bearing lost a part of its pre-stressing load. It was proven that the Bi-Kon preloaded bearing is better than the other one. A disadvantage of these bearings is the higher bending strain of guide vanes that is a result of the enlarged support space. All other non-pre-stressed bearings didn't lead to success.

In 1991 they decided to set up a conical pre-stressing bearing, which is made of RG7 with 25% glass fibre-filled Teflon film, in lower guide vane bearings, see fig.3.20. The conical bearing bushings were completed and preloaded with grooves flexibility in the direction of supporting. DU bushes were installed in the two positions of the upper bearing. These bearings have been installed with minimal clearance of 0.05-0.08 mm.

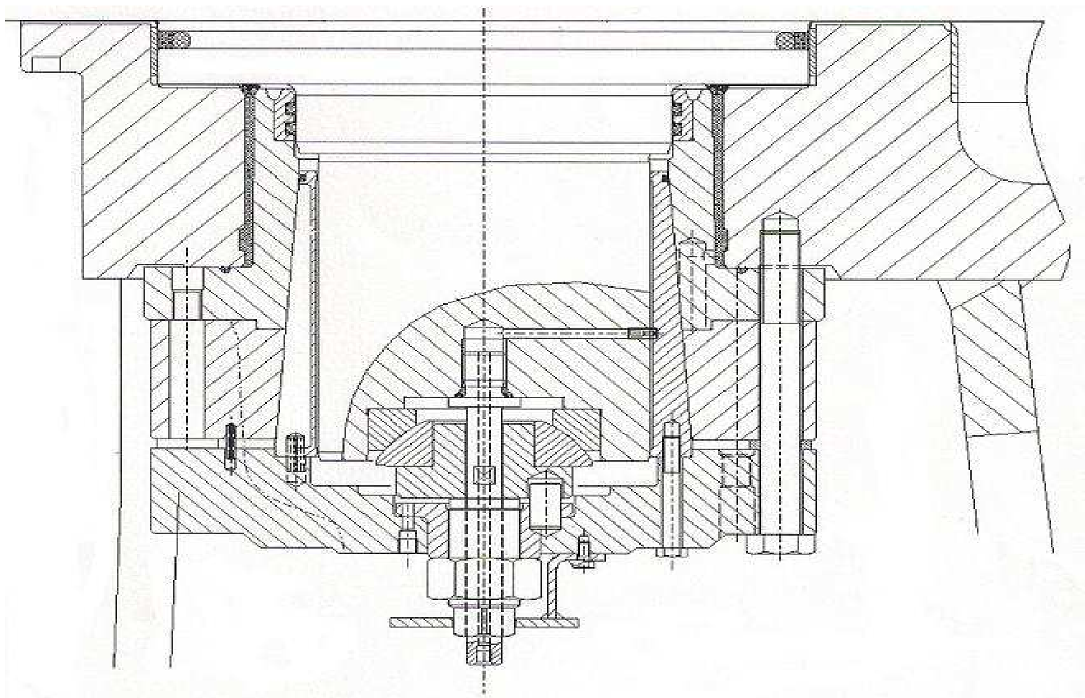

Fig.3. 20: Conical pre-stressing bearing [17].

From the revision of the 1993-1994, the bottom guide vane bushings had to be renewed again, because the guide vane stems were turned. The re-loaded bushing in the conception wasn't changed. Teflon film was selected for the material of the bearings that are on the generator side. The condition of bearings was good according to the checks during 1996-1998. They only discovered a deformation of the bronze bushings, known as flexibility grooving.

In 2001, after the check of a lower bearing a crack was found in the Teflon film. It was decided to remove all Teflon films and assemble new ones. In order to control the mass, they noticed that the bronze bushings are deformed. The deformations were consisting of clearance extension in the grooves (approx. 0.5-0.6 mm) and clearance reduction between the grooves (to about 0.3-0.4 mm). This deformation made the insertion of a new Teflon film without processing, impossible.

Fig.3. 21: Crack in the Teflon film of the lower bearing [17].

Fig.3. 22: The deformation of bronze bushing [17].

Because of the deformations during the pre-stressing, its adhesive responsibility couldn’t be guaranteed and as a result they waived sticking of a thicker Teflon film with subsequent processing, therefore the manufactured supporting device were the bronze bushings at the outer diameter reworked that again with a Teflon film 1.50 mm thick. The Preload of the bushings has been reduced from 150bar to 100-120bar.

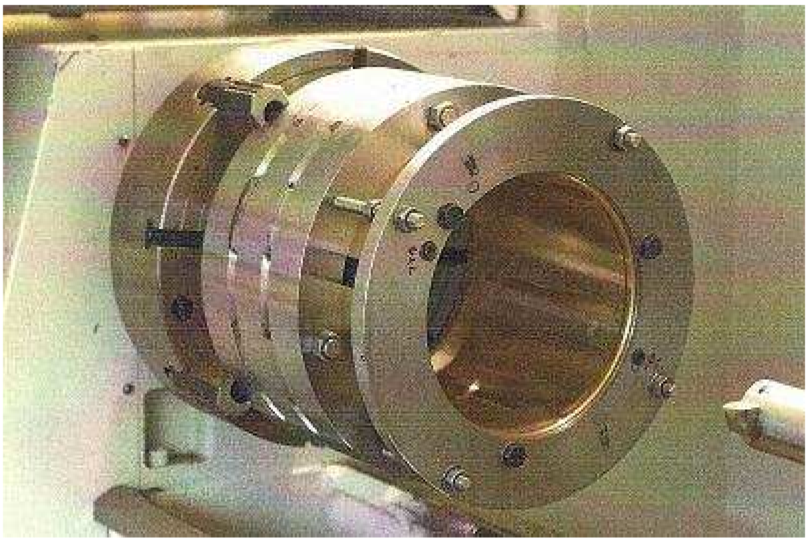

Fig.3. 23: Reworking of Teflon film on the bronze bushing [17].

The generator side bushes were generally in good condition. Wear was just appeared on the Teflon film under the two bushings. Here was a new Teflon film stick to the all of the bearings. The bearing was again installed with minimal clearance of 0.05-0.08 mm.

Fig.3. 24: Wear on the Teflon film [17].

The lower guide vane bearing was made without the flexibility grooving. An aluminium bronze with higher strength values was chosen for its material. Different forces are needed to adjust the pre-stressing and as a result a hydraulically-driven device must be used. They remain in the elastic range, under 0.4 mm diametrically, during deformation. Desired clearance for installation: 0.1-0.15 mm

In summary, the method of the pre-stressed bushing is primarily proved. The reduction of the vibrations by the preloaded bushing positively influenced the wear of the Teflon bearing material as well as the sealing edge wear of the closing edges at the guide vane. According to the fact that the bearings haven’t experienced any problems in operation since their start-up in 1993, their performance is primarily proven.

Advantages:

- The wear that is caused by relative motion is avoided by reduction of the vibrations, which also has a positive effect on the guide vane sealing edges.

- Gals filled Teflon foil can be used as a bearing material. As a result there would be no edge pressure and this will avoid any wear.
- A preload adjustment is possible anytime.
- As the preload bearing at the guide vane bearing is applied, where the step bearing is located, the vibration absorption affects the upper bearings.

Disadvantages:

- Bushings had been deformed, through the selection of RG7 and the flexibility grooving. As a result in order to insert a new Teflon film it was necessary to rotate the deformation. This disadvantage is eliminated with a choice of higher quality material and by the omitting of the grooves.

A research was carried out in Vianden (Machine 10) from August 1980 to July 1982. The bearings of all types indicated damage but the simple Fiberglide bearings were successful. Deva metal and Teflon bushings were totally or partially destroyed. The Dacron bushing was again installed, and received a new Dacron bushing. All other received new Fiberglide bushings in order to control the removed bearings.

The conditions of the bushings have been observed in July 1982 in Vianden:

1. Fiberglide: The wear seems to be low. In the upper bearing, the guide vane no.8 is of the Fiberglide lining of the bottom bushing (on the runner) was broken in several places.
2. Dacron: The abrasion in practically zero.
3. Deva metal: In the bushing groove, the large amounts of wear particles were discovered. The sliding surface was good and the wear was 0.02-0.07 mm.
4. Teflon film with glass filling: The film on the high pressure place was totally damaged. The Teflon had been squeezed and smeared in the rotating direction.
5. Teflon film with bronze filling: In the bearing, the guide vane no.7 had shown that the film during the assembling had been striped. In the bearing no.5 the bronze-Teflon had more damage and more severe breaks.

The conditions of the guide vane stems have been observed in July 1982 in Vianden:

1. Fiberglide: Despite superficial damage of Fiberglide - chuck the guide vane stems were still absolutely smooth. An abrasion of the chrome steel was not available.
2. Dacron: The surface of the guide vane stem was smooth but not quite as good as the Fiberglide. An abrasion was not available.
3. Deva metal: The guide vane stem was roughed and 0.05 worn. Here corrosion effect beside the abrasion existed and on the stem surface was a counter profile of bushing with groove.
4. Teflon film with glass filling: The guide vane stem was smooth despite the destroyed film, however was slightly worn out (0.02 - 0.03 mm).
5. Teflon film with bronze filling: Despite the damage of the Teflon on the blade no.5 und the completely destroyed film in bearing no.7, the stems were smooth and not worn.

After the improvements in reduction of pump-start-up duration and vibrations by SEO Vianden Group, by earlier opening of the ball valve, duration of increased vibrations should be reduced by:

- Choosing a larger pump opening
- Increasing the guide wheel- opening speed

The guide wheel opening strongly decreases the guide vane vibrations up to 50%. To achieve this time as quickly as possible, the opening was advanced more. The previous low vibration level during the de-aeration stage was prerequisite essential for this idea.

Adjustable guide vane bearing on the suction side in Vianden in July1986: It is found out empirically that the adjustments should be proportionate to diameters and abrasions. For this reason, SEO wants a larger adjustment range to facilitate the bearing mounting. This is possible only with a continuously slotted bushing. A larger adjustment capability would be useful because of the expected deformation and wear and tear of the Teflon. Escher Wyss explained operating mode of their bearing. Therefore, the Vulkollan was used as adjustment device like a liquid; the guide vane will be loaded unilaterally; thus, the Vulkollan will be squeezed on the loaded side and exhausted on the unloaded side. Statistical forces can not be carried over the adjustable body. Therefore, a slack clearance will be required.

Therefore a bearing liner was used that in upper area is thick-walled for the metal seat and in lower area is flexible for the adjustment capability by use of the Vulkollan. The bended guide vane connects unilaterally on the bearing liner in upper area and in adjustable area, slack clearance adapts to the Vulkollan of the stem inclination. It is expected that during the high frequency of load change, the flow effect of the high viscose compact body occurs with delay, thus a damping of radial vibration is achieved. An additional damping of torsional vibration occurs by the enlarged frictional force. The explained operating mode doesn’t allow a continuously slotting of bearing liner and the adjustment capability is too limited (0.3 mm).

It was realized that to improve the lifetime of the guide vane bearing selecting the best bearing type is not enough.

Therefore the following actions were taken:

1. investigating about the causes of damages and
2. reducing the effect of the causes if it's possible

It was discovered that a longer vibration phase occurs after the pump has reached the jump-pressure and during the ball valve opens. They decided to open the ball valve much sooner so that immediately after the adjustment of the jump-pressure the guiding device can be opened.

Consequently, the following results were achieved:

1. Reducing the vibration phase from 25 to 13 seconds
2. Reducing the start-up time at 44 seconds
3. Reducing the spiral pressure from 48 to 39 bar

Next they made experiments with various pumps - starting openings. Also here they achieved a negligible improvement.

They have also increased the opening speed of guiding system for the pumps starting. A change in the closing time during the pump shut-down, as well as the increasing of the adjustment speed during the other operating transitions, has not been possible with regard to the pressure changes in the pressure tunnel. Also it is possible to reduce the vibrations of pump during the shut-down stage by applying some modifications.

In a further step of the research, they tried to explore the types of different vibrations. For this purpose, relative sensors were installed in some of the lower bearing.

It was proved that bearings with normal clearance previously measured severe vibration mainly consist of bending vibrations of guide vanes. In bearings with a less clearance the bending vibrations were greatly damped, here also the static bending through the bearing-restraint was also greatly reduced. This bearing-restraint caused a cushioning of the torsional oscillations; however the amount of this cushioning was not too much to be the only reason of the damage. Obviously, the damage caused by high warming of the bearing materials.

The experiences about Vianden power plant - machine 10 - and the Langenprozelten are achieved from the Vianden hydro power plant in Luxembourg and also use of the information in their documents.

3.8. Experiences – The Langenprozelten Plant

In 1976, the Langenprozelten plant was equipped with Fiberglide bearing type 64. This complex exhibits particularly unfavourable vibrations because the dimensioning of both pump turbines was carried very close. The complex was operated by the German Federal Railways. A high switching frequency of pump turbines, which was up to ten times per day, was required by the train schedule.

The Fiberglide bearings types 64 were worn after two years because of the extremely unfavourable operating conditions. It was clear that the bottom guide vane bearings, which are the most critical ones in pump turbine, were used under conditions that must be necessarily avoided in future designs.

It is a question of the following items:

1. The lower guide vane stem was designed a little short. Thereby the bearing was assembled at first with outside diameter bearing into the housing. After that, the guide vane spigot due to short guide vane stem was inserted in inside diameter bearing. Since the guide vane spigot does not sit exactly in the middle of the drill hole, one side of the spigot against Fiberglide-fibre was necessarily butted during assembly. Thereby, a number of bearings were damaged unintentionally during the installation and the damaged bearings were not identified. In the future the lower guide vane stem will be designed so long that the bearing can be inserted with inside diameter and then can be engaged the outside diameter bearing into the turbine housing.

2. The mounting of the lower guide vane spigot was supplied with a 45 ° phase sharp-edged. This necessarily caused some installation problems. Therefore, the installation phases were basically carried out with approximately 15 ° phase and a rounding edge.

3. Both the sliding contact surface of Fiberglide bearing and the spigots were lubricated with grease. The high vibrations together with grease generated a softening effect. Until now, this effect was unknown. To prevent the softening effect and to ensure the proper conditions of wear in the bearing, the Fiberglide bearing must be used dry and only the seal must be lubricated with grease to achieve a better life.

The lubricated bronze bearings were used in the Langenprozelten complex after disassembly of the Fiberglide. This bronze bearing because of the high loads had a shorter life than the previously used Fiberglide.

The newly developed reinforcement type 34 (Teflon Dacron) and type 44 (Teflon Nomex) were some ways to test the critical lower bearing. After a period of one year in June 1978 the results showed only 0.025 mm wear with the reinforcement type 34. The test of type 44 could not be completed because the bearing was damaged during the test.

It is worthy to consider that following materials have been tested: Johnson (similar to Lubrite bearing), Metalloplast (Pampus), Deva (maintenance free Sinter bronze); According to the results, the Fiberglide reinforcement type 34 is the best. The Johnson bushings had a relatively high wear and led to fretting on the shaft. The Metalloplast exhibited a relatively high compression set as expected. The Deva bearing showed traces of fretting on the sliding layer.

Because of the good results with the reinforcement type 34, in January 1980 the Langenprozelten complex was completely reconverted with Fiberglide type 34. For experimental purposes, arrays of bearings were equipped with reinforcement type 44.

Each of the Vianden, Rodund and Langenprozelten complexes were designed with a specific bearing load of 160 bars. In the Conso complex, which was built by Escher Wyss, the Fiberglide bearings were loaded with a specific bearing load of 265 bars. The result of this high loading was the abrasion of the bearings (type 64).

In future constructions of pump turbines the load of 100 bars will be considered for Fiberglide bearing. This technique is necessary to achieve the expected life.

The Fiberglide bearing with reinforcement type 64 for the guide vane crank were used In the Bendeela complex in Australia in March 1977. Being under preload, bearing had been in use till 1980. The installation was done as follows:

1. The Fiberglide bearings are pressed in the guide vane crank and the inside diameter of pressed bearings are measured.
2. The bolts are manufactured and sorted with an oversize of 0.02 mm due the installed Fiberglide bearing.
3. The bolt must be exhibit a proper installation phase and it is pressed in the Fiberglide bearing.

By the pre-stressing of the Fiberglide reinforcement, no bearing clearance occurs during the pump turbine operation. Thereby the knocking will be prevented out of the bearing and at the same time a certain cushioning will be achieved. The bearings were installed with a specific bearing load from 120 bars to 160 bars. This specific bearing load is calculated without the consideration of the additional specific bearing load that comes from the pressing of the bolt.

After a year of operation, bearings of Bendeela complex were demounted and 24 hours after the dismounting of the bolt, the inside diameter of the bearing exhibit the same dimensions as those at the time of installation a year ago. This result proves that the Fiberglide bearings worked within the elastic limit of the Fiberglide reinforcement.

The availability of the new reinforcement types 34 and 44 must be tested for guide vane crank, because these materials are much harder and not yet certain whether an installation of bolts with oversize is possible.

Maintenance-free bearings are based generally on the fact that the lubrication occurs by an abrasion of the lubricating material. At the normal Fiberglide tissue type 64, the abrasion resistance is given only by pure Teflon fibres and the connecting phenolic resin. Dacron (Polyester) has a very high abrasion resistance and concurrently at the same time is relatively elastic .Nomex has a good abrasion resistance, extremely high load capacity and at the same time a high temperature resistance.

The tissue type 34 is suitable due to the elasticity of Dacron materials that are used primarily where vibrations occur. Results of a test have showed that the life period of this tissue is 2 to 3 times more than tissue type 64. The price of this material is about 30% higher consequently.

Tissue type 44 is suitable for highest strains and is in the same temperature range of use as the tissue type 64. The wear resistance is about two to three times more than reinforcement type 64. Type 44 is almost 30% more expensive than type 64 due to the difficulty of its manufacturing and application.

4. Conclusion

Hydropower owners and operators are continually investing in plants to increase the value of output, add capacity, improve reliability, reduce operating and maintenance expense, extend plant life, and comply with environmental and safety regulations or voluntarily-imposed standards. Many owners are paying particular attention to minimizing or eliminating their facilities’ negative environmental effects, including reducing the risk of discharge to receiving waters, by incorporating self lubricating materials into projects and using environmentally acceptable lubricants.

Researchers and operators should always review recent pertinent conference reports, publications, other literature and discussions about self lubricating materials and environmental lubricants to achieve the best solution.

To sustain hydro’s efficiency and competitiveness requires implementation of improved, more cost-effective maintenance and operating practices and the commitment to applying technological advancement. Significant investment is often needed to improve many hydro plants particularly older, conventional plants to restore or sustain efficiency and competitiveness, and to meet environmental objectives. However, economic justification of needed investments is often very difficult. Recent improvements in technology, particularly in the areas of hydro-turbine and component design and manufacture, control equipment and instrumentation, and improved life and maintenance management, have greatly enhanced the prospects for increasing production and economic efficiency, and extending the life, of existing hydro plants.

The most commonly reported environmental practice is reducing the use of lubricants by the installation of self-lubricating or greaseless bearings and bushings, and/or the use of oil-less or low-oil governors.

As this technology - using self lubricated bearings in guide vane - is still young, manufacturers have not succeeded yet to produce an ideal bearing with favourite life span, because it is not possible to simulate them adequately due to the dynamic forces. However, manufacturers gaining more experience will improve the quality of products continuously and they will use different materials in order to supply bearings. The result consists of various types of bearings available in the market which make some difficulties in choosing the proper type for operators. Although experiences show that none of them have reached an ideal condition.

An operator should always be aware of the new products and their characteristics to be able to select the best choice. The first factor in this regard is that the operator should have enough cognition from the conditions of system. As different bearings may show similar or close properties, some other factors such as the price or manufacturer's service can be effective on choosing a proper bearing. Although during the implementation of the project very little information and experience was obtained from the operators, their comparison reveals that it is very difficult to introduce a general solution for all operators because of their different working space. Therefore identifying the working space and concentrating on it by the operators can facilitate the job to reach the final goal which is longer bearing life.

The first step in choosing a bearing is doing tests, but since it is very expensive for the operator, one can just use test results from the commercial catalogues provided by bearing manufacturer or from different research institutes working on bearings. The fundamental problem in doing these tests is lack of a standard manual for running the tests. In addition, manufacturers sometimes mark the test results from an institute as unacceptable because they believe that the test conditions were not appropriate for their product. While the test conditions are tried to fit to the actual conditions, this claim may not logical. Although, sometimes these tests are more commercial rather than scientifically, which means that a manufacturer benefits from some advantages of the product in a test in order to show the results in favor of its product in comparison with other manufacturer. Here is the field in which operator's cognition can help to identify a proper test technique, so the adequate bearing will be recognized and chosen.

Generally, the main problem in running tests is related to the adequate simulation which causes the bearing life not last until the time expected by the manufacturer. Some of the most important factors in selecting a bearing are: coefficients of static and dynamic friction and also the wear rate. Many elements affect these factors whose adequate definition and efficacy are very intricate. Some of the elements are wearing in the shaft, swelling of the bearing, applied loads, temperature, speed of vibration, and hardness of the surface.

One important discussion for installation and mounting of the bearing is clearance which determines the amount of free space dedicated to the bearing. The amount of this clearance reported by the manufacturer and observed by the operator is of critical importance during the installation. Different test results show that some bearing materials in water have expansion or contraction in volume after some specific times, which includes changes in their diameter, thickness and length. Experience shows that the maximum amount of these changes occur 8 to 16 months after working in the system. Ignoring this crucial point while installing the bearing can have much influence on increasing both the coefficient of friction and the clearance of the bearing.

Another controversial argument during the installation of bearing is using seal which is provided in order to prevent external particles from entering the space between shaft and bearing and its negative influence on the bearing's life. Although some of the manufacturers believe using seals as unnecessary, most of the literature published by scientific institutes after implementation of tests on the bearings, seals are recommended apart from bearings' material. Use of seals is a disadvantage for bushings. It leads to extra need for maintenance service and increasing costs. As mentioned, bushing manufacturers offer their products without seals, but sometimes, experiences and tests show something else.

Techniques of installation play also a key role in the life of bearing. Inaccuracy in this step can lead to lower working lives by edge loading, because of the shaft deformation or bearing deformation and extra bearing clearance.

As mentioned before, coefficients of static and dynamic friction are important and effective factors on the bearings' life. It should be noted that the selected bearing must have close coefficients of static and dynamic friction, since a large difference between them may lead to stick-slip phenomenon which will cause vibration, noise and failure of the bearing.

The obtained test results show that even small differences in working conditions of two similar bearings can lead to different life times. As dry and wet environments can provide different conditions for the bearing, various proper bearings can be used up and down of the guide vane. Because the bearings have not gained the required ability to work long in completely severe conditions, work of the operators will find significant importance. Indeed, after selecting the appropriate choice, the operator should provide suitable working conditions for the bearing. An operator can control and decrease the dynamic loads exerted on the bearing by changing the turn on, turn off and function route of the pump.

By studying and scrutinizing the bearings, a proper type of it can be chosen for the system, while working adequately with them needs to have much experience and to be familiar with the system thoroughly. In order to reach higher life times for bearings, using the experience of many other operators could also be useful but unfortunately this was not possible in the present study.

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