Design and Construction of the Variable-Pitch Air Turbine for the Azores Wave Energy Plant

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ABSTRACT

A 400kW variable-pitch air-turbine will shortly be installed, alongside a fixed pitch Wells turbine of similar rating, in an oscillating water column wave-energy pilot plant in the Azores. It is hoped that the new turbine will demonstrate increased net energy productivity through a combination of low idling losses and better efficiencies in the high airflows produced by larger sea states. It may also permit attempts at optimising the wave energy plant through the use of reactive loading.

There are three critical design areas. The blade/spar radial thrust bearings should have low friction and acceptable life expectation. The blades must be light but strong and resistant to erosion and corrosion. The pitch-change actuator idling losses should be very low but it must be able to move the blades quickly and accurately.

The paper describes the background to the project, and outlines the final design of the turbine as built.

1. INTRODUCTION

A feature of wave energy has always been the number and variety of concepts proposed for encouraging the sea to make electricity. However, the oscillating water column (OWC) has remained popular for many years, whether as a shore-based, bottom-mounted or floating device. Furthermore virtually every OWC proposed or built in the last 20 years has one or more Wells turbines somewhere inside it.

The popularity of the OWC has a great deal to do with the convenience with which the Wells turbine converts bi-directional airflows between wave chamber and atmosphere into unidirectional bursts of torque in the coupling of the electrical generator. Moreover, during lulls in the sea or when the air velocity drops to zero during twice-per-wave flow reversal, the Wells turbine needs little power to stay rotating. However the range of wave, and therefore airflow, conditions over which the Wells turbine operates with reasonable efficiency is severely limited by blade stall.

Wells turbines use symmetrical profile blades with their chords in the plane of rotation. In 1993 Salter [1] presented a graphical analysis, based on published lift and drag data for a commonly used blade profile. He showed how a section of such a blade operating at fixed speed with zero pitch angle in steady unidirectional flow produces positive torque only for angles of incidence between 2 and 13 degrees.

Below 2 degrees, in the low air velocity operating area, the lift component is too small to produce positive torque and the rotor tends to lose speed. All turbines are exposed to this problem and no known type has lower idling drag than the Wells.

At angles of incidence above 13 degrees the blade section goes into stall. The rapidly increasing drag forces dominate the less rapidly declining lift forces. Efficiency collapses. The turbine may even have to be kept spinning by the generator running as a motor unless airflow velocity peaks are clipped by a fast-acting bypass valve.

If, however, the blade were able to change pitch so as to prevent the angle of incidence exceeding some maximum angle, for example 8 degrees, then it would produce positive torque at all angles of incidence above 2 degrees. Salter suggested that a variable-pitch turbine operating on this basis would be more productive than a fixed pitch turbine of comparable size and he presented an outline design for such a machine. As well as higher efficiencies, the variable-pitch turbine would also provide an opportunity to try out performance-enhancing reactive loading by using the generator and turbine to pump bursts of energy into the wave chamber.

As the very high centrifugal blade loads and the high frequency of pitching operations seemed to rule out the use of rolling bearings, the original turbine design was based around novel plano-convex hydrostatic thrust bearings. A prototype of such a bearing [2] demonstrated the feasibility of the concept and led to an invitation to build such a variable-pitch turbine as the ‘B’ machine for the Azores OWC pilot plant. [3].

2. THE PILOT PLANT MACHINES

The variable-pitch turbine is been built along with an associated high-speed stop-valve which is described elsewhere [4,5]. They will form an integrated unit when installed in the OWC pilot plant. The stop valve has various uses including implementation of 'latching control' on the OWC and rapidly deployable protection of the turbine against water ingestion.

The 'A' machine in the Azores plant is a Wells turbine. The wave chamber will be fitted with a fast-acting turbine bypass valve and the generators will connect to the utility via an electronic variable speed drive. The machine room will thus contain the components needed to test almost any suggested OWC control strategy.

3. THE PROJECT PARTNERS

The University of Edinburgh is responsible for the concept, co-ordination, design and making of the rotating components of the turbine.

The Instituto Superior Tecnico, Lisbon is responsible for the aerodynamic design of the blades, making of most of the non-rotating parts, system integration, commissioning and operation.

Applied Research & Technology Ltd., of Inverness have provided consultancy on the manufacturing of the blades.

4. GENERAL SPECIFICATION

The initial specification called for a variable-pitch turbine to drive a 400kW induction generator at a nominal speed of 1475rpm. This was fleshed out through many iterations of a fast 2-dimensional drawing package and a parametric analysis package (FastCad and MathCad respectively).

Rated electrical output	400 kW
Nominal Speed	1475 rpm
Tip diameter	1.7 m
Blade pitch range	-40° to +40°
Number of blades:	15
Blade tip speed	131 m/s
Hub ratio	0.71
Blade chord at root:	0.2 m
Blade length	0.25 m
Solidity	0.66
Idling loss:	6.7 kW
Design airflow rate	120 m^3s
Design pressure drop	5900 Pa
Damping coefficient	50 $Pa.m^{-3}.s$
Efficiency at full flow	58 %

Table 1: Key Design Parameters

Various factors affect the choice of tip diameter. The maximum diameter is limited by centrifugal blade loading, Mach effects, water droplet erosion and shipping considerations.

To reduce compressibility effects and thus maintain good lift to drag ratios over the full operating range of the turbine, the rotational speed at the blade tips should be well below the speed of sound (330m/s). Hovercraft propellers operating at 220m/s are successfully protected from erosion by polyurethane coatings and helicopter experience suggests that erosion is not a serious problem at speeds below 165m/s.

Water droplet erosion tests on aluminium alloy and on stainless steel, carried out as part of the turbine design programme, suggest that there is no clear defining speed below which erosion resistance ceases to be a consideration, though at 131m/s there is room for optimism.

For bi-directional pitching, the blades cannot be made with built-in twist as in aircraft propellers, so that the angles of incidence will always vary more than desired along the blade length. However the high hub ratio reduces this effect.

With regard to optimal loading of the OWC, a higher damping factor would be better. From this point of view, operation at up to 2000rpm, using the variable-speed drive, may be useful.

5. GENERAL DESIGN

Figure 1 shows the series configuration of turbine and stop-valve in the machine room of the OWC pilot plant. This figure also shows the modular'45-degree frame' support system. Profiled plates radiate at 45 degrees from the corners of these frames, running parallel to the airflow direction. They provide adaptable structural support to the machines and ducting with a minimum of drag.

Figure 2 shows the variable-pitch turbine in section and labels the principle parts.

Figure 3 shows the blade thrust bearing arrangement in more detail. The solid models of Figures 4 and 5 present a clearer image of the blade/spar component and their arrangement within the complete hub. Photographs 1 and 2 show the turbine before manufacture of the blades and during trials of the pitching actuator.

6. ENERGY & LIFE CONSIDERATIONS

It is hoped that variable pitching will convert more wave energy to electricity than is possible with fixed blades. However, to be economically realistic the turbine should use only a small fraction of the extra energy gained to power the pitching system.

In large waves the blades might be moving through seven or eight pitching cycles every minute, perhaps one million cycles per year. At 1.7m tip diameter and 1500rpm the centrifugal acceleration at the blades is over 1500g. Their loads must therefore be reacted by efficient thrust bearings. These should hold the promise of a reasonable mean time between failure (MTBF). Five years was set as a target MTBF for initial design studies.

The variable-pitch turbine will probably spend at least half of its working life at zero pitch. The pitch control system should draw an absolute minimum of power at these times whilst being instantly able to provide pitching torque when commanded.

7. BLADE THRUST BEARING

Rolling bearings were initially rejected for the project as being unable to deliver the 5-year MTBF.

The hydrostatic thrust bearing prototyped before the current project [2] demonstrated high load capability and very low actuation torque. All bearing surfaces are separated by a fluid film so wear is negligible. However in the context of a practical seashore-going turbine there were a number of daunting requirements. The first was the problem of constantly recovering oil from the rim of the turbine against the severe centrifugal gradient, whilst keeping the rotor in good balance. Secondly, each blade bearing had 26 hydrostatic pockets of which 24 were supplied with oil by precisely calibrated flow restrictors. 360 restrictors would have been needed for the complete turbine. It was difficult to feel optimistic regarding their long-term stability and reliability.

A 'tension-torsion strap' concept was next developed. This was based on an idea commonly used in helicopter main blade retention and most notably in the variable-pitch fans used in the NOTAR (no tail rotor) helicopter [6]. Tension-torsion straps are made by potting tows of uni-directional strong fibres within an elastomeric resin. Some kind of termination arrangement is provided for transferring the external loads to the fibres. The straps are very strong and stiff axially, but have low torsional stiffness. The NOTAR tension-torsion straps are made by looping many turns of Kevlar yarn around a pair of circular spools and then potting within a polyurethane elastomer. One spool is attached to the blade end and the other to the rotor. The strap transmits the centrifugal load to the hub but the blade can twist freely.

The tension-torsion strap idea looks promising for variable-pitch turbines but the development effort required is beyond the scope of the current project.

Finally, it was felt that the nominal 5-year MTBF requirement for the bearings was unduly harsh for a prototype machine and the rolling bearing option was reconsidered. There will be periods in the summer when generated power is so low that downtime to change bearings makes economic sense. The current design uses a series combination of a cylindrical roller thrust bearing and a dry spherical bearing to give a self-aligning action. The arrangement should be clear from Figures 1,3 and 5. The blade load is transferred from the inner end of the spar tubes, through the bearings to the back surface of mushroom-shaped high-tensile steel 'root spars'. The root spar ends are held in a threaded clamping arrangement at the hub, such that some radial length adjustment of the blades is possible. The bearings are packed for life with grease.

The failure probability calculation suggests a 10 percent chance of one of the fifteen cylindrical roller thrust bearings failing within 6 months of continuous use, giving a MTBF of 2½ years. Impending bearing failure should be detectable by monitoring the pitch actuator force.

8. BLADE CONCEPTS

Figures 2,4 and 5 show various aspects of the blades. Initially it was thought that sweeping the blades back would reduce unwanted compressibility effects, and help stabilise the blade in pitch by shifting the centre of pressure aft of the pitch axis. The effect of sweep back of the blades was studied by IST [7]. This revealed that sweep back was not in fact beneficial so the sweep was removed from the blade design. Instead the blades were stabilised in pitch by shifting the pitch axis forwards of the centre of pressure.

The blade profile has been specially developed by IST for the variable-pitch turbine to give good lift to drag ratios across the full operating range and deal with Mach effects. Moving from hub to tip, the point of maximum thickness moves forward toward the leading edge, to give a slightly bulbous aspect.

The high aerodynamic and centrifugal stresses on the blades and the need for the smallest possible weight to minimise the thrust load demand a very good strength to weight ratio. Resistance to water droplet erosion is also very important. The ideal material is titanium alloy. The best fabrication technique for titanium is super-plastic forming / diffusion bonding (SPDB) [8]. In this process, several thin sheets of titanium alloy are selectively fused together and then inflated within a mould at high temperature. This process can make seamless components with complex surfaces and multiple internal divisions. Unfortunately the price of the SPDB process proved to be beyond the budget of the current project. Setup and development costs dominate so that it should be a viable process in mass production. However for only fifteen blades a technology with lower initial costs was required.

The proposed construction for the blades is as a carbon/epoxy composite with an electroless nickel erosion protection coating. Ideally the blades and spars would be separate components such that blades could be removed and replaced without requiring a major strip down. However this would require three difficult composite to metallic transitions in each blade/spar set. As these increase blade mass and complicate manufacture it was decided to make the blade and spar as one component.

9. BLADE DESIGN

The proposed blade design is a hollow, partially foam-filled, thin-walled carbon-epoxy component with the carbon fibres laid up in a mixture of uni-axial and alternating +/-45-degree layers continuous from tip to attachment collar. Moving in from the root of the blade, the cross-section starts out as an airfoil, then goes through an oval-to-round blend section, a tubular spar section, and finally to a threaded metal attachment collar.

The tip and the hub ends of the blade are closed by spherical surfaces. Just below the oval-to-round blend, a collar locally thickens the spar tube for the outer plain bearing where the blade torque and thrust are transmitted to the rotor. Near the inner end of the spar, the pitch clamp couples the blade to the pitch actuator.

At the inner end of the spar tube the composite structure blends into a waisted aluminium-alloy attachment collar. To bind the composite and the metal into a structural whole, additional carbon fibres are wound around the spar tube at the waist.

The final lay up will be optimised through finite-element (FE) analysis of the whole component. The dry fibre lay-up will be made around a core cast from a low-melting temperature alloy. (A typical eutectic alloy of this sort is Wood's metal which melts at 73° C). The lay-up will then be closed inside an aluminium alloy mould and epoxy will be injected under pressure. After curing, the metal core will be melted out.

Initial development work will prove the tensile strength of the attachment collar termination.

10. GENERAL DETAILS

It will be evident from Figure 2 and Photograph 2 that the turbine rotor is cantilevered out from a steel support structure. This structure will itself be supported in the OWC machine hall by the 45-degree frames. The support structure houses two circular clamps which provide for precise axial and radial adjustment of the position of the 200mm diameter bearing housing tube. The turbine shaft is constrained inside the housing tube at the generator end by a pair of spherical thrust bearings and by a single spherical radial bearing at the rotor end. The shaft and housing tube is sealed and the bearings splash lubricated.

The rotor hub is attached to the shaft by a 30-degree (included angle) taper that is easily separated. The hub is in two parts. A ring part on the sea side clamps the root spars to the main part of the hub.

The rotor is made as a structural sandwich from a pair of 12.7mm thick, 1090mm diameter aluminium alloy plates separated by radial stiffening plates. It can be seen to the left in Photograph 2. Threaded holes in the stiffening plates are used for rotor balancing weights. Other plates near the outside, visible in Figure 5 but not fitted when the photograph was taken, hold the outer bearings.

A glass reinforced plastic (GRP) inner shroud and labyrinth seal, visible in Figure 1, will be fitted to reduce water ingestion to the turbine.

11. ACTUATION

As has already been stressed, the actuator should have very low idling losses. It should use as little power as possible when actively controlling pitch but instantly be able to supply whatever force is required to obey pitch command signals.

A novel pitch actuator has been developed in response to these requirements. This sub-system is described in detail in another paper [9] but a brief overview is given here.

The energy to drive the actuator is taken directly from the turbine shaft. Functionally, it is like an amplifier that uses a small amount of electrical energy to precisely control the transfer of shaft mechanical energy into the pitch control mechanism.

As shown in Figure 1, the blades are moved by individual push-rods attaching to pitch-arms on the spar tubes. The other ends of the push-rods are connected to the pitch push-plate assembly so that axial translation of that one component forces all fifteen blades to change pitch. The push-plate assembly spins coaxially with the shaft and is attached to the nut of a large recirculating ballscrew having a thread pitch of 10mm.

The ballscrew has an aluminium disc attached to it. A second disc is connected to the first disc by a differential gear. If disc ‘A’ is retarded by a braking torque, then disc ‘B’ speeds up in the opposite direction and the ball screw rotates in one direction relative to the rotating shaft. If disc ‘B’ is retarded by a braking torque, then the ballscrew rotates in the other direction.

Each disc spins between the poles of a set of stator coils. An amplifier drives current through one set of stator coils or the other. Eddy currents are induced in the corresponding disc and a braking torque is produced. The ballscrew then rotates relative to the rotating turbine. The ballscrew nut, the push-plate assembly and the pitch push-rods move axially and the blades change pitch. Energising the other set of stator coils changes pitch in the opposite direction.

Pitch position is measured by counting the passage of holes at the outer rims of the eddy-current discs past opto-sensors. The stator coils, amplifier and opto-sensors have been integrated into a stable pitch-angle command servo-system.

12. CONCLUSIONS

- The variable-pitch turbine should be able to demonstrate increased energy productivity compared with fixed-pitch machines whilst having comparably low idling losses.
- Carefully selected rolling element bearings give a MTBF of 2½ years. The tension-torsion strap idea is promising for future turbines.

- The best blades will be probably be made of super plastic formed / diffusion bonded titanium alloy though nickel plated carbon epoxy composite blades look promising.
- The eddy-current pitch actuation system should provide responsive and energy efficient pitching.

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The Commission of the European Communities JOULE III programme has supported the work described in this paper under contract JOR3CT950002 ‘Variable-Pitch Turbine and High-Speed Stop-Valve’. The authors would like to acknowledge the contribution of Laird Parker in the machining of turbine components.

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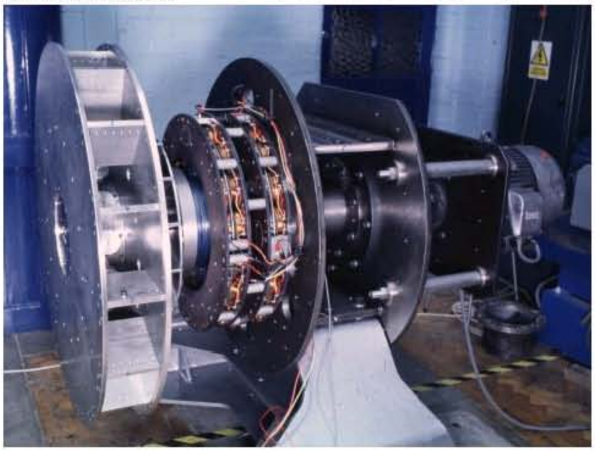
Photograph 1 (right)
Partly assembled turbine:
Pitching plate assembly on floor;
one set of coils has been removed to reveal aluminium eddy current disc.

Photograph 2 (below)
Rotating components and support structure on test stand during commissioning of pitch actuator;
18kW electric motor at right;
complete coil assemblies and ball-screw actuator left of centre.



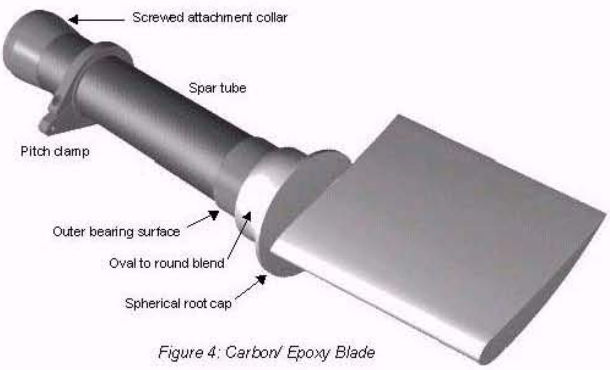


Figure 4: Carbon/ Epoxy Blade

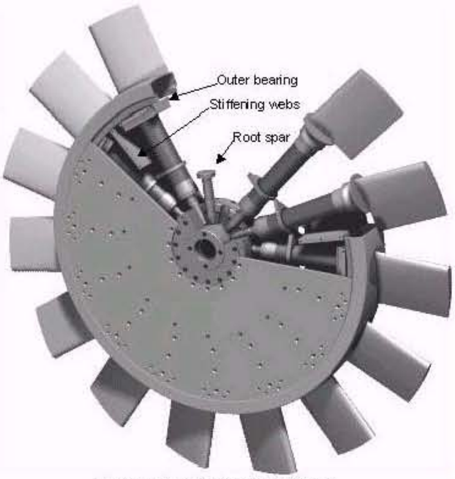


Figure 5: Partially Sectioned Rotor

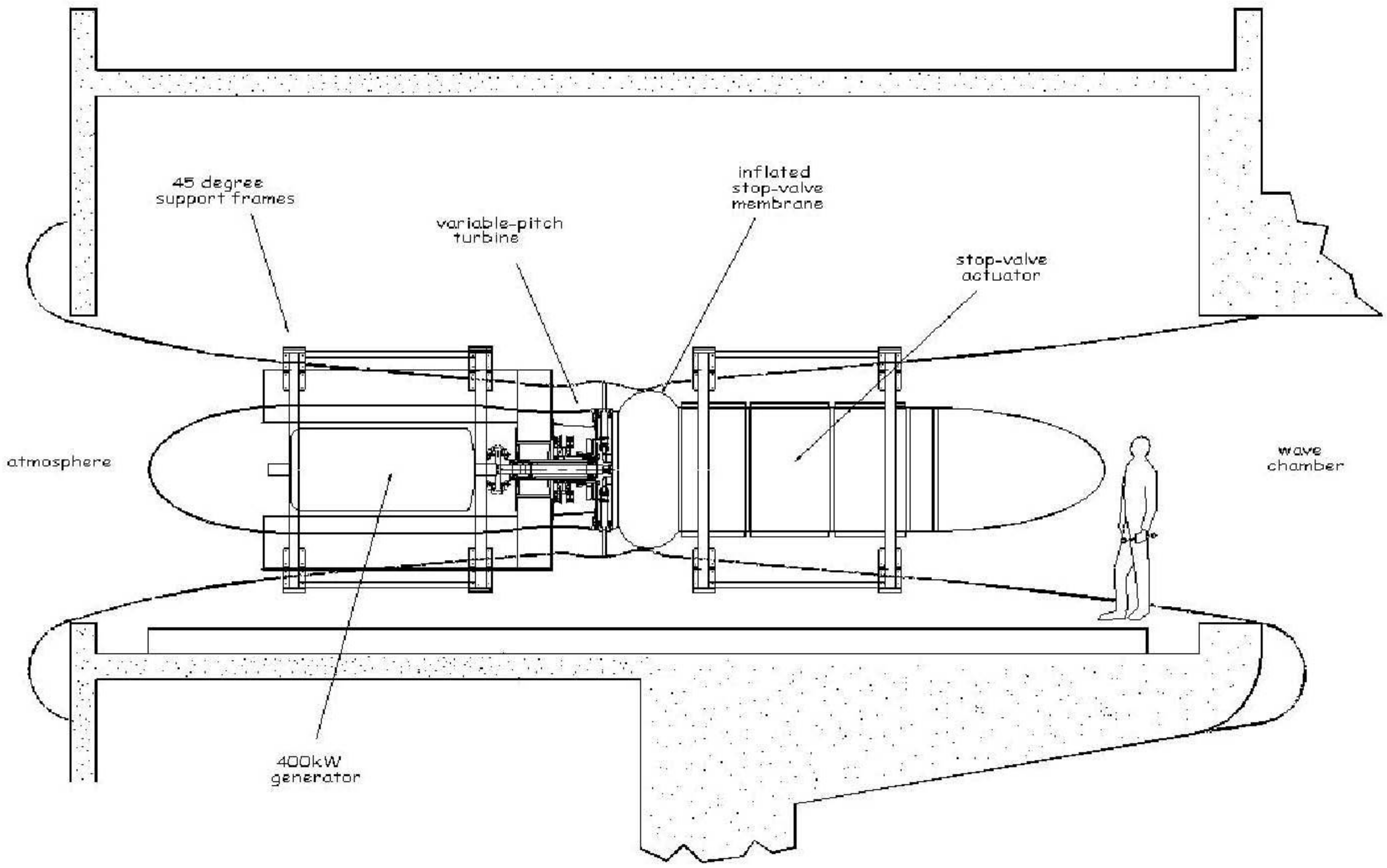


Figure 1: Variable-Pitch Turbine & High Speed Stop Valve in situ in Azores OWC pilot plant

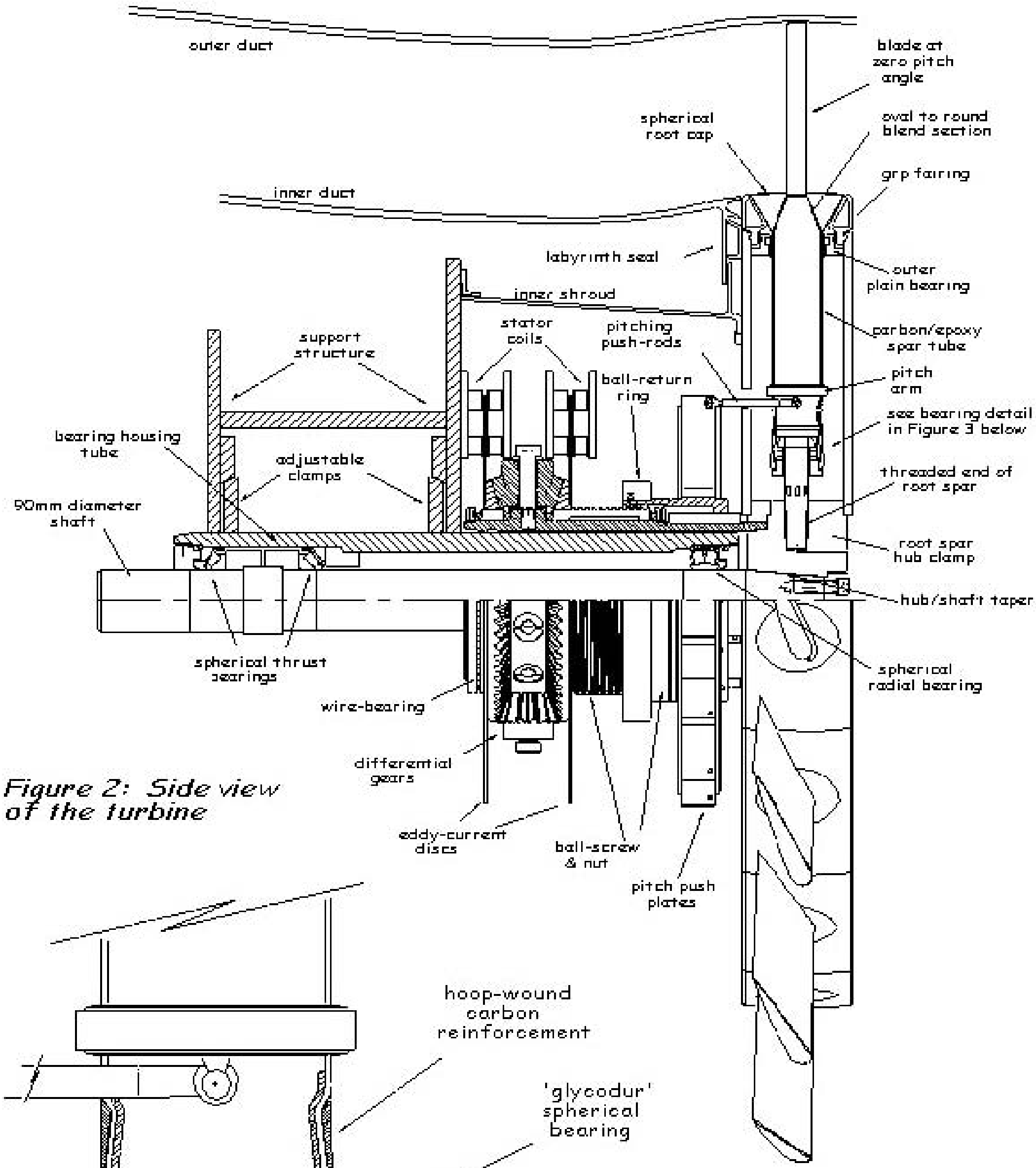


Figure 2: Side view of the turbine

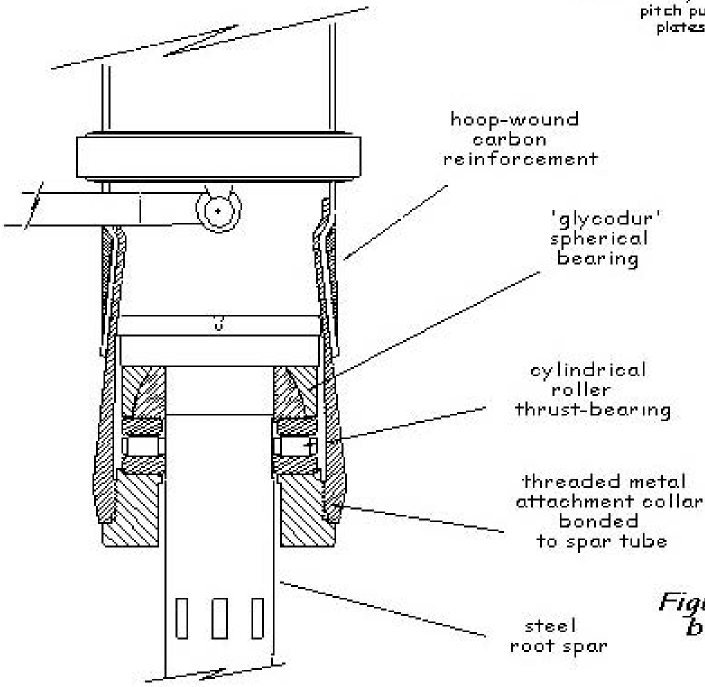


Figure 3: Detail showing blade thrust bearings