

Eddy-Current Actuator for a Variable Pitch Air Turbine

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ABSTRACT

As part of the EU Joule 3 scheme, a variable pitch air turbine is being developed for an oscillating water column pilot plant in the Azores. The key requirements for the pitch actuation mechanism point to a new actuation technology based on the principle of the eddy current brake. Two eddy brake discs drive a custom ballscrew through a differential gear. The ball-nut pushes on pitching levers attached to the blades. A small electrical power input controls large amounts of mechanical power drawn directly from the turbine shaft. The paper describes the design and the development of the system.

1. Introduction

The project

Under the EU Joule 3 programme, an oscillating water column wave power device is being built on the island of Pico in the Azores. Two air turbines will be installed - a fixed pitch Wells turbine, and a variable pitch turbine with high speed stop valve.

The variable pitch turbine project involved collaboration between the University of Edinburgh, the Instituto Superior Técnico in Lisbon, and Advanced Research and Technology Ltd. in Inverness, Scotland. A separately presented paper gives a full overview [1]

A key component of the turbine is the pitching mechanism which allows the blades to change angle of attack as the duct airflow changes velocity and direction.

Key Specifications

Control over blade pitch allows many different control strategies to be tried. The simplest is pitching the blades to keep the angle of attack below a threshold to prevent stall. Over the operating cycle of a wave period the system should be able to pitch the blades up to 40degrees either side of the in plane (Wells turbine) position. For low absolute duct air velocity the blades will remain in the zero pitch position and the machine operates as a Wells turbine. Above a certain axial air velocity, the blades will change pitch to preserve a constant angle of attack, chosen to maximise power output. In a monochromatic sea the pitch demand signal would be a sine wave with crossover distortion dependent on duct velocity (figure 1). The maximum pitch rate of change assuming a maximum height 10 second wave is 32 degrees per second.

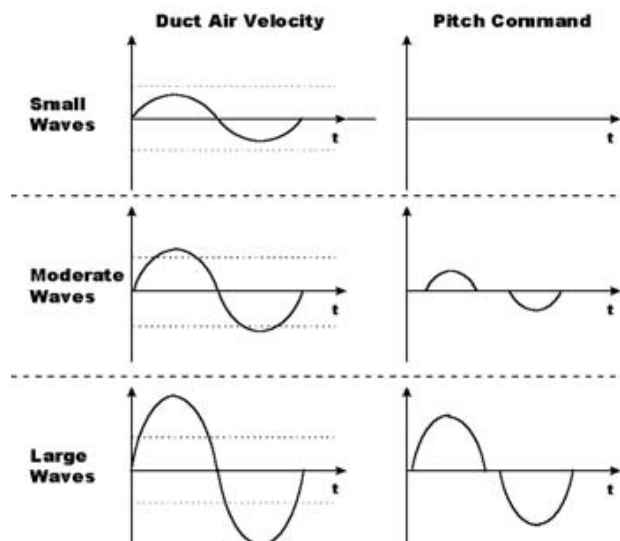


Figure 1: Simplified pitch command signals.

As the turbine will spend most of its time with zero pitch, it is crucial that the actuation system have low idle losses so as not to affect the overall efficiency. When working however it must be able to hold the blades at the demanded position regardless of aerodynamic loads. The system should be low maintenance and likely failure modes should not overstress the blades or jeopardise operating the turbine in fixed-pitch mode.

2. Electromechanical system

Design Concept

Figure 2 shows a cutaway solid model of the actuator. The 15 blades are attached to the hub with tubular spars. Roller thrust bearings allow the blades to change pitch while taking the centrifugal load of more than 15kN per blade. Rods push or pull levers clamped to the blade spars. The rods are attached to a sandwich construction pitching plate which is attached to the nut of a custom ballscrew. Rails constrain the nut so that it can only slide axially, and is locked in rotation to the hub. If the screw rotates faster than the turbine shaft, the rods pull the levers; if the screw rotates slower then the rods push the levers.

This speed variation is created by linking two eddy current brake disks in a differential configuration. Each disc is connected to a crown wheel linked by a pair of pinions, the axes of which rotate at the shaft speed. This maintains the average speed of the two discs to be equal to the shaft speed. If the disc nearest the blades is braked, then the screw rotates slower than the shaft, the nut screws out and the rods push the levers. If the disc furthest from the blades is braked then the screw speeds up by the differential action of the pinion. The nut screws in and the rods pull the levers.

A ring of electromagnetic coils ² creates an axial magnetic flux through each disc. The interaction of this flux with the tangential velocity induces radial in-plane eddy currents to flow under the poles. These currents interact with the applied flux to produce a braking torque. Dissipated energy heats the disc which rapidly loses heat to the surrounding air. Braking torque is controlled by varying the current flowing in the coil windings.

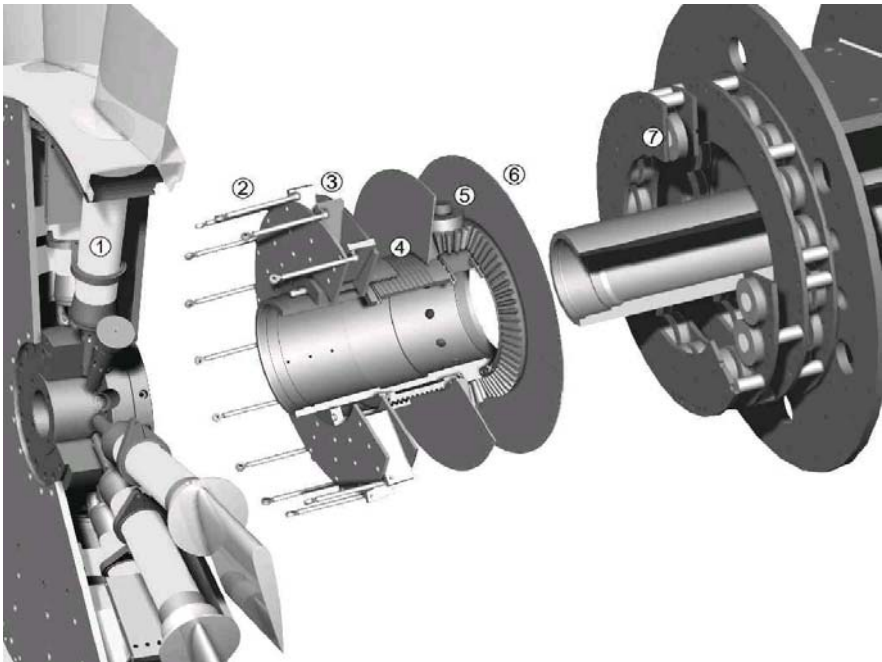


Figure 2: Exploded Solid Model of Pitch Actuator

- ① Blade spars & pitch levers
- ② Push rods
- ③ Pitch plate and ballnut
- ④ Ballscrew
- ⑤ Differential gears
- ⑥ Eddy current brake discs
- ⑦ Magnetic coils and poles

2.2 Custom Ballscrew

The loads on the ballscrew of maximum thrust 2.85kN are not severe and could comfortably be met by a single off-the-shelf 30mm diameter screw. However the axis of the turbine is occupied by the main shaft so multiple screws would be required at some radial distance from the main shaft. This would impose centrifugal loads on the screw radial bearings much higher than the thrust loads, as well as complicating the disc and gearing system.

The solution is to make a custom tubular ballscrew coaxial with the turbine shaft. As a tube is inherently balanced, the screw bearings need only take the thrust loads. The screw is turned with a form tool then finished with a form grinding wheel. The ogive section of the ball race reduces Hertzian stresses in the balls and on the contact surfaces. Steel balls might not have shared load effectively and would have required constant lubrication. DuPont Vespel polyimide balls have a compressive modulus of 2.4 Gpa compared to 200 for steel and can run un-lubricated without damage. 220 balls of 5/16" diameter are used, recirculating through a groove in a separate ring outside the nut.

2.3 Differential Gearing

The differential system is made from modified off-the-shelf gears. With steel pinions the gears are far stronger than they need be, they are noisy and centrifugal load on the pinion radial thrust bearing is large, causing undesirable friction. Mating steel gears require constant lubrication, which is difficult to arrange.

The solution to these problems is to use pinions made from Nylatron GSM, a molybdenum filled nylon 66. With a density a quarter that of steel the bearing friction is reduced accordingly, noise improved, and the gears are self-lubricating.

The steel crown wheels and the ballscrew run on angular contact ball bearings with ground wire races, which take up very little space.

3. Eddy current brakes

Theory for sparse poles

Axial magnetic flux at a radius from the axis of a rotating conductive disc will induce eddy currents to flow in the plane of the disc. Interaction of these currents with the applied magnetic field causes a retarding torque. At low speeds eddy current brakes can be modelled as perfect dampers, the torque rising linearly with speed.

The currents in the disc generate a magnetic flux which acts against the applied flux. At low speeds they are not significant but above "critical speed" they tend to cancel the applied flux. The torque asymptotes to the inverse of the speed and it becomes a constant power machine. The peak torque occurs at this speed and is termed the "critical torque". The equations that model this behaviour and a review of corroborating experimental work are presented in an invaluable paper by Wouterse [2].

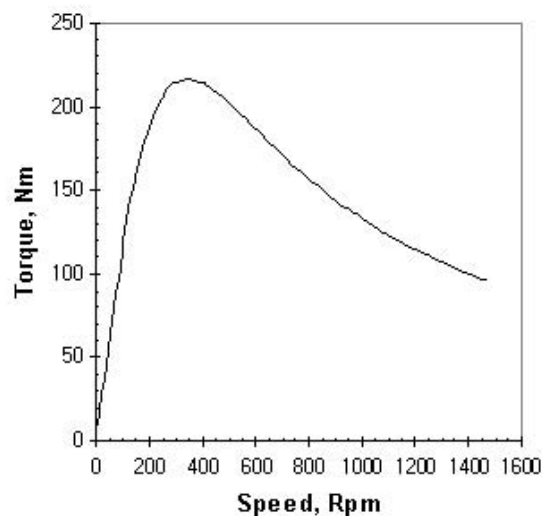


Figure 3: MathCad model torque/speed curve, final design, 10A coil current.

A numerical model based on these equations was constructed in MathCad (figure 3 above) and calibrated with experimental results. Published optimisation procedures for eddy current dampers eg. Mikulinsky & Shtrikman[3], Nagaya & Karube [4] are valid only in the sub-critical region and assume parameters are continuously variable. However design of the coils and disc is a "fuzzy" optimisation problem-some parameters e.g. pole pitch circle diameter are effectively continuous, other like disc thickness are quantised by available stock sizes, whilst others like disc conductivity come in only a few discrete values.

After many iterations the design has converged on a solution offering much more torque than should ever be needed, a reasonable safety factor on the strength of the discs and a power amplification gain of around 10 times, so 1kW of input electrical power directs 10kW of mechanical shaft power into the actuator.

Number of pole pairs	16
Pole diameter	40 mm
Pole circle diameter	520 mm
Number of coils	32 in series
Turns per coil	200
Wire diameter	1.4mm
Total resistance	17.2 W
Total inductance	270 mH
Disc resistivity	37 nW m
Disc diameter	600 mm
Thickness	3.0 mm
Air gap both sides	2.25 mm
Maximum flux density	0.80 T at 12A

Figure 4: Table of Important Parameters.

3.2 Effect of pole spacing and polarity

The Wouterse model assumes that poles are widely separated and do not interact. Practical designs pack as many poles as possible around the disc to maximise torque, so interactions between poles become important. Disappointing initial results with closely spaced poles of identical polarity prompted experimental study of the effect of pole spacing and polarity (figure 5 below).

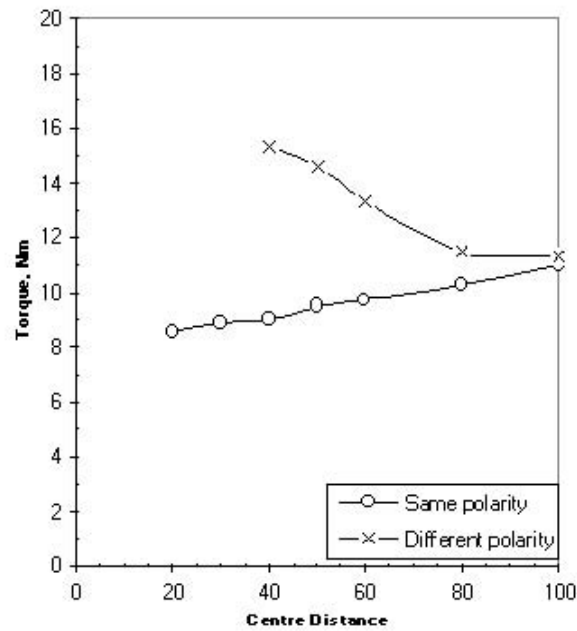


Figure 5: Interactions with 25mm poles.

A tentative explanation is as follows (see figure 6 below). With adjacent poles of the same polarity, current flowing under the poles returns in between the poles, so stray flux through the disc in between the poles acts to cancel some of the current, reducing torque. This stray flux increases as the poles are packed closer together. With an opposite poles configuration radial current flow occurs only directly under the poles. Stray flux between the poles tends to cancel by symmetry and anyway does not act against the eddy current flow. Indeed experimental results show that poles of opposite polarity actually work better when closer together.

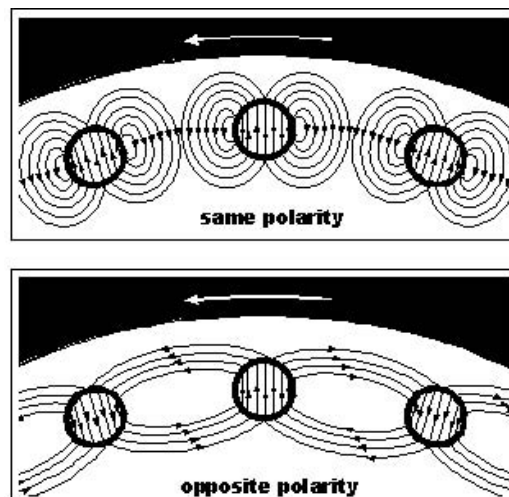


Figure 6: Suggested eddy current flow paths.

3.3 Eddy brake discs

Eddy brake designs intended for low speed operation as a damper typically use copper discs for highest damping coefficient. For our application copper is too conductive; large currents flow at low speeds giving a low critical speed. Copper also has a low strength to weight ratio leading to low safety factors for centrifugal stresses. The relatively low electrical conductivity of aluminium alloy H30 compared to copper raises the critical speed, while its better strength to weight ratio allows higher factors of safety. Keeping the critical speed below the operating speed ensures that as the disc conductivity increases with temperature the critical speed rises, so the available torque rises slightly, and the eddy actuator is not impaired.

Braking power heats the disc so heat transfer must be considered. A powerful radial airflow from the disk is encouraged by the large air gap between the poles. Experiments indicate that with an average of 5kW dissipated in each disc, the temperature of the disk will be under 80degrees above ambient. This does not pose a problem for performance as an eddy current brake, but initial trials showed a tendency for the disc to become warped due to differential thermal expansion of the rim (where the most of the heat is generated) relative to the hub. Large compressive hoop stresses buckled the relatively thin disc. The solution was to cut 4 radial slits 1.5mm wide in each disc, which close up as the rim thermally expands. A finite element study indicated that these slits would prevent buckling up to a relative temperature of 200degrees, well beyond that anticipated.

256 holes drilled in the rim of each disc gate a transmissive opto switch, which provides position feedback for the actuator. The discs are turned into incremental encoders giving pitch resolution down to 0.03° .

Results

A complete assembly for a single disc was mounted on the bed of a milling machine with the disc mounted on the spindle. Monitoring of spindle motor current allowed torque to be calculated from a previous calibration.

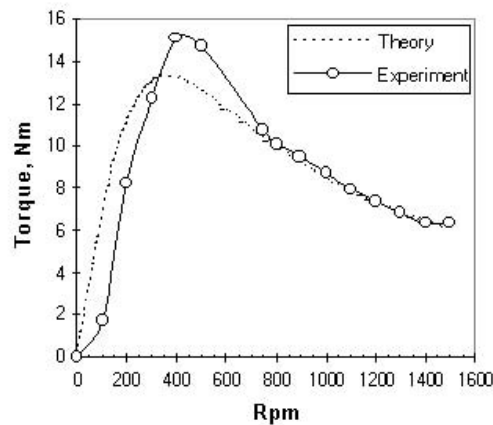


Figure 7: Torque vs. Speed.

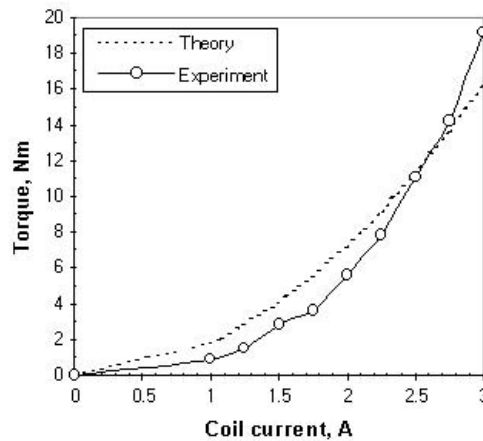


Figure 8: Torque vs. Current.

It was expected that the torque would be proportional to the square of the current. However the exponent measured was 2.8. This effect is welcome but so far unexplained.

4. Control system

4.1 Overview

As far as is known this is the first application of eddy current actuation to position control. A speed control algorithm has been simulated (Simeu & Georges [6]) but is valid only for the subcritical region.

The first system to be tried is an analogue PID controller with position feedback from the incremental encoders built into the disk. This system does not attempt to linearise the coil current/torque curve nor correct for torque variations with disc speed due to the critical speed effect, which would be necessary to implement a truly linear control system.

4.2 System Description

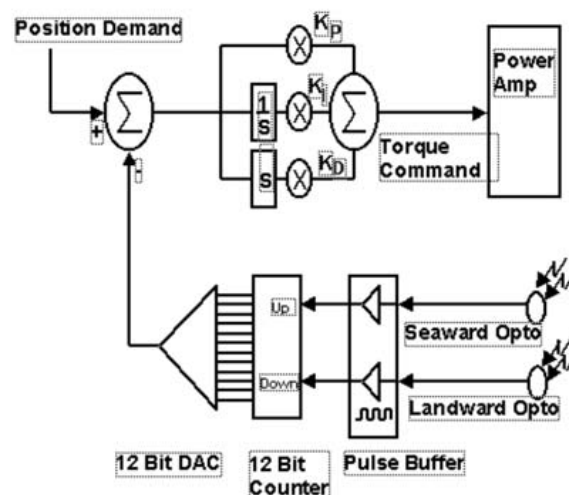


Figure 9: Control System.

The block diagram above shows key system components. Opto sensors on each disc generate pulses into a 12-bit up/down counter. Pulses from one disc decrement the counter, whilst those from the other disc increment it. When the actuator is at constant pitch, the rate of pulses from each disc is identical, so the position register stays static with a 1 bit dither. A clocked pulse buffer and synchroniser before the counter reduces this dither and eliminates the risk of a pulse being missed or counted twice. The output of the counter is fed to a 12-bit DAC to give an analogue position signal.

The first junction calculates the position error from the command and the feedback signals. The second junction sums the position error with its differential and integral, each of which has a gain control, to implement the PID function. Its output is effectively a torque demand of the actuator, positive and negative torque demands being directed to separate coil sets. Before feeding to the amplifier a pre-distorting correction circuit would compensate for the non-linear current/torque curve. A second speed dependent distortion would correct for the torque variation with speed due to the critical speed effect. However these have not been found necessary to implement a stable control loop.

4.3 Coil power

Supply of current to the coils must be from a current mode amplifier to eliminate the effect of the increase of coil

resistance with temperature. Only one set of coils should be energised at one time; simultaneous conduction will produce no net actuation torque but will waste shaft power on heating the discs.

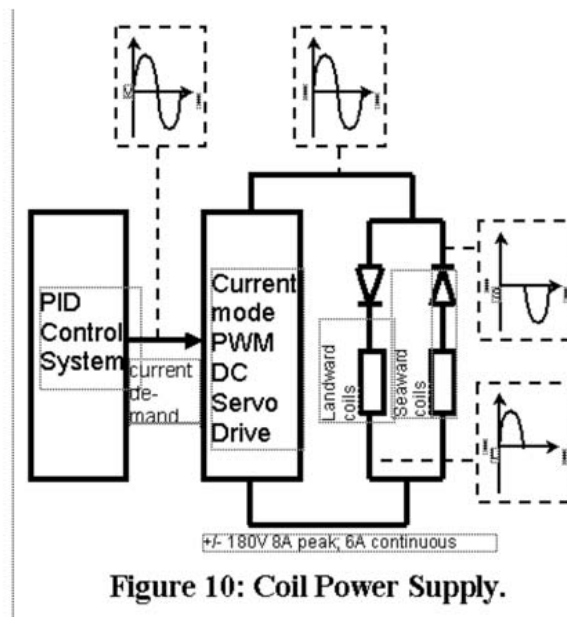


Figure 10: Coil Power Supply.

This is achieved by using a 4 quadrant PWM brushed DC motor servo controller, the output of which is directed to either group of coils with diodes. When the servo is given a positive command, current flows in one coil; when given a negative command, current flows in the other.

Initial trials

A stable control loop has been implemented using the described system, which can be easily tuned by adjusting the PID parameters. Frequency and step response plots are not presented here because without the blade aerodynamic stiffness and bearing friction, the loads on the actuator are not representative. However a number of factors are encouraging:

- Eddy disk torque can be controlled to a bandwidth of at least 50Hz.
- The actuator can supply much more torque than should ever be needed.
- The opto sensors give very high quality position and velocity information.
- Idle losses come only from windage and are of the order of 100 watts.

Conclusions

The requirements for the pitch actuator have prompted the development of a novel system based on eddy current brakes.

This system re-directs large amounts of mechanical power direct from the shaft into pitching the blades while consuming only a small amount of electrical power.

It offers low idle power, long life, high bandwidth and high torque. Tests show that it can easily be integrated into a position control loop.

Acknowledgements

The actuator idea came from Stephen Salter who also did much of the mechanical design.

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References

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