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Projektträger Biologie, Energie, Ökologie (BEO)
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Implementing Agreement for a Programme of Research and Development on Wind Energy Conversion Systems

19th Meeting of Experts – Wind Turbine Control Systems, Strategy and Problems

London, May 3-4, 1990

Organized by:
Project Management for Biology, Energy, Ecology (BEO)
Research Centre Jülich (KFA)

On behalf of the
Federal Minister of Research and Technology,
The Fluid Mechanics Department
of the Technical University of Denmark

Scientific Coordination:
M. Pedersen (Techn. Univ. of Denmark)
R. Windheim (BEO-KFA Jülich)

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Introductory note on wind turbine control problems

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For cost effective wind turbines a number of ReQuirements (RQi) must be met. Topics for wind turbine control problems are:

- **RQ1** Optimizing of balance between energy production and power factor.
- **RQ2** Minimizing fatigue damage, due to fluctuations in wind velocity and grid induced fluctuations.
- **RQ3** Safety under extreme external conditions in normal operational modes.

It is the aim of a control strategy to meet as good as possible the first two requirements and to be compliant with the third one. Two types of control are distinguished: passive and active control. Under passive control, stability is acquired from the very nature of selected wind turbine components. Under active control measurements on the wind turbine are processed to turbine inputs.

For example passive power control may be realized by pitch angle adjustment due to centrifugal or aerodynamic forces acting on a rotor blade. Rotational speed can be passively controlled by a direct grid connected AC-generator. Under active control these control aims can be realized by servo-driven pitch angle adjustment and variation of the generator field. Certain control strategies require both passive and active control. E.g. passive power control based on pitching by variable centrifugal forces needs a variable rotational speed at constant electrical power.

Any active control system causes extra costs and risks (it may fail by itself). Hence application is only justified if its benefits supersede the disadvantages related to extra costs and risks. An active control system changes more or less the CHaracteristics (CHi) under fully passive control of a wind turbine, such as:

- **CH1** Concentration of aerodynamic loads around (multiples of) blade passing frequency, due to (nearly) constant rotational speed.
- **CH2** Chance on aerodynamic instabilities in full load at power control by stall.
- **CH3** Impossibility to use fast actuation of electrical conversion system inputs in order to suppress excited eigenfrequencies and effects of transients (start-ups, severe grid disturbances and severe wind gusts).

An active control system will be useful in Pursuing the following Effects (PEi):

PE1 Decreasing of fatigue damage in stationary operation by suppressing excited eigenfrequencies.
PE2 Decreasing fatigue damage during frequent transients (startup, shut-down).
PE3 Guaranteeing of survival under extreme condititons in order to make constructions cheaper (grid fall out, severe wind gust)
PE4 Producing more energy per unit of area.

For active control a complete control concept must be realized. This procedure consists of several STeps (STi)

ST1 Definition of control aims for N+2 modes of operation:
* partial load
* full load
* transiënt stages (N kinds: grid fall out, start-up, etc.).

ST2 Choice of controller design methods for realizing control purposes in 2+N modes of operation.
ST3 Fitting controller design methods on realistic conditions, with respect to:
* accuracy of underlying model knowlegde
* accuracy of sensors and AD-, DA-convertors in case of digital control
* accuracy and speed of controller computational hardware.

ST4 Controller design (for 2+N modes), using an integral model of the wind turbine.
ST5 Design of a protocol for being compliant with a safety strategy, which takes care of action in case of failure of a wind turbine subsystem and emergency situations, not leading to a defined transient stage.
ST6 Implementation of the controllers for different modes of operation and protocol for the safety system on a test facility (control system in experimental environment).
ST7 Realization of the control system for commercial wind turbines (control system in production environment).

Characteristics for wind turbine Controllers and related Problems are (CPi):

CP1 Several control aims per mode of operation and interaction of in- and outputs:
-> multivariable controller design approach.
CP2 Non-linear turbine system behavior and uncertainty in models:
-> robustness problem in controller design.
CP3 Extremely large range of time scales (0.001s up to 10s):
-> numerical problems at implementation.
CP4 Several modes of operation:
-> criteria for switching from one mode-wise controller to another.

Suggestions for discussion on Control problems (SCi) are related tot the seven steps of realization of a complete control concept:

SC1 -> How to translate generally stated requirements in explicit control purposes?

SC2 -> Which approaches for controller design are used and why?
SC3 -> How to adapt a theoretical controller design method for robust implementation; which conflicts arise?
SC4 -> Do converging problems for the used algorithms exist for calculating a specific controller? Is it possible to indicate constraints for reliable application?
SC5 -> Which design techniques for protocols are used for being compliant with a safety strategy?
SC6 -> What kind of problems (software, hardware, mixture of analogue and digital control) arise when combining mode-wise designed controllers and the protocol related to the safety system?
SC7 -> Can commercial requirements be met (e.g. low costs for control system, reliable under all circumstances and in long term)?

APPROPRIATE CONTROL FOR PITCH REGULATED HAWT'S

by

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Abstract : A specification for the control system of a pitch regulated constant speed wind turbine is suggested. A PI controller and a classical controller are designed for a fictitious wind turbine and the performance of the latter is shown to be an improvement over the former. Control related issues of variable speed wind turbines are briefly reviewed.

Acknowledgements: The author gratefully acknowledges the support provided by the Deparment of Energy for the work presented here.

1. Objectives for Active Pitch Control of a Constant Speed Wind Turbine

If active control is to be employed for the regulation of a wind turbine, the role of the control system and its objectives must first be identified. Of course, these are dependent on the configuration of the wind turbine and the means of achieving control. For constant speed grid-connected HAWT's, active pitch control has been widely used to regulate the power generated by the wind turbine when operating above rated windspeed. However, the relevant dynamics include the aerodynamics, structural dynamics, drive-train dynamics, power generation dynamics and control system dynamics, all of which are influenced by the controller. Hence, a specification, much wider than simple power regulation, is required for the controller. A possible specification for the controller might include the following:

(a) It must cause the drive-train to have appropriate dynamics, that is

- (i) induce the required response to changes in mean windspeed and steady increases or decreases in windspeed
- (ii) induce adequate damping and compliance in the drive-train

(b) It must reduce the drive-train torque transients, that is

- (i) increase rejection of the disturbances due to wind turbulence
- (ii) not increase disturbances due to the spectral peaks induced by rotational sampling.

(c) It must reduce the extent of pitch action and prevent saturation of the actuator, that is

- (i) reduce the bandwidth of the closed-loop system to prevent too high a control demand
- (ii) protect the actuator from measurement noise and high frequency spectral components due to rotational sampling.

(d) It must cause the system to be robust in the presence of dynamic uncertainty, that is

(i) induce sufficiently large gain and phase margins

(ii) ensure rapid roll-off to protect the system from high frequency dynamic uncertainty.

(e) It must not unnecessarily aggravate structural loads, that is

(i) reduce off-design structural loads.

(ii) not increase cyclic loads

(f) It must maintain performance over the operational envelope, that is

(i) cater for the wide range of values as a function of windspeed, of the rate of change of torque with respect to pitch angle.

(ii) cater for the transition between below and above rated windspeed.

The different parts of the specification are not complementary but are competing and an appropriate balance must be attained. The relative importance of them depends on the specific dynamics of a wind turbine and the manufacturer's priorities.

PI control is the industrial standard, but it has limited capabilities and cannot address such a complex specification as the preceding one. A classical controller with dynamic shaping over the relevant frequency range is much more appropriate.

Both a PI controller and a classical controller were designed for a fictitious wind turbine [1] constituting an amalgamation of information obtained from the manufacturers [2], [3] and from the literature [4]. It has been deliberately designed to be dynamically very active so as not to minimise the difficulty of the design task. However, none of the individual dynamic components is unrealistic and may be found on existing commercial machines. The control and simulation models used to represent the wind turbine are internally consistent and as such it is physically realisable.

The wind turbine is three-bladed with full-span pitch control (all three blades acting in unison). The power-train is the simplest possible configuration consisting of a gearbox and an induction generator with 1.5% slip and only the generated power is measured. The rated windspeed is 12 m/s above which the turbine generates 300 kW of power from a rotor torque of 81 kNm.

2. PI Controller

A PI controller which performs as well as possible was designed for the wind turbine. That it was appropriately designed, was checked by varying the P and I constants about their nominal values. No improvement could be demonstrated. The theoretical performance of the controller as determined from the control design analysis is summarised by Table 1 and Table 2.

Table 1 ***Standard deviation of aerodynamic torque for PI controller***

Windspeed (m/s)	Standard deviation of aerodynamic torque (kNm)			
	Wind	$3\omega_r$	Measurement noise	wind + $3\omega_r$ + measurement noise
12.0	12.88	4.26	0.01	13.57
15.7	21.17	4.26	0.01	21.59
22.9	46.61	4.26	0.01	46.80

Table 2 ***Standard deviation of pitch acceleration for PI controller***

Windspeed (m/s)	Standard deviation of pitch acceleration (deg/s^2)			
	Wind	$3\omega_r$	Measurement noise	wind + $3\omega_r$ + measurement noise
12.0	1.796	1.641	20.78	20.922
15.7	0.676	0.3006	3.805	3.876
22.9	1.024	0.1526	1.931	2.191

The input aerodynamic torque to the rotor is used as the prime indicator of drive-train loads since it is the most appropriate for comparison to other methods of regulation such as stall regulation. (It is the sum of all loads without amendment by power train dynamics). A breakdown for wind turbulence, rotational sampling and loads due to pitch angle errors induced by measurement noise is given. The acceleration of the pitch angle is used as the second indicator of performance with a similar breakdown since there are strict limits on it in order to prevent the control action being too strong. The performance is assessed for mean windspeeds of 12 m/s, 16 m/s, 22 m/s. The extent of torque transients rises as the windspeed rises but the control action becomes less. The large amount of control action at low windspeed is caused by the high frequency components of the disturbance spectrum, i.e. the measurement noise and rotational sampling peaks.

The PI controller was designed and its theoretical performance determined using simplified linear models of the wind turbine ignoring the transition of the turbine for below rated operation to above rated operation at 12 m/s. To obtain a more realistic evaluation of the controller a full nonlinear stochastic simulation of the wind turbine was employed. The results are shown in Figs. 1, 2 and 4 for a 12 m/s mean windspeed and in Figs. 8,9 and 11 for a 20 m/s mean windspeed. Windspeed, input aerodynamic torque and blade pitch angle are given in both cases. (The performance is consistent with the control design estimates).

3. Classical Control

A classical controller to meet the objectives specified in Section 1 was designed for the wind turbine. Its theoretical performance as determined from the control design analysis is summarised by Table 3 and Table 4. The same indicators of performance are shown as for the PI controller, for mean windspeed of 12 m/s, 16 m/s, 22 m/s.

Table 3 ***Standard deviation of aerodynamic torque for classical controller***

Windspeed (m/s)	Standard deviation of aerodynamic torque (kNm)			
	Wind	$3\omega_r$	Measurement noise	wind + $3\omega_r$ + measurement noise
12.0	2.25	4.58	0.10	5.10
15.7	4.57	4.58	0.10	6.47
22.9	13.29	4.58	0.10	14.05

Table 4 ***Standard deviation of pitch acceleration for classical controller***

Windspeed (m/s)	Standard deviation of pitch acceleration (deg/s^2)			
	Wind	$3\omega_r$	Measurement noise	wind + $3\omega_r$ + measurement noise
12.0	8.989	43.96	43.14	62.252
15.7	3.442	8.049	7.901	11.796
22.9	5.330	4.085	4.010	7.821

The extent of torque transients is still rising with windspeed but at low windspeed the controller operates sufficiently well that these transients are dominated by the spectral peaks in the disturbance spectrum due to rotational sampling. Again the extent of control action decreases with rising windspeed. The high frequency components of the disturbance spectrum, i.e. measurement noise and rotationally induced spectral peaks, are large at all windspeeds but particularly dominate at low windspeed.

Advantage was taken of the transparency of the classical controller to realise an implementation which catered for both the nonlinear nature of the aerodynamics and the transition between below and aboved rated operation of the controller. The performance was verified by simulation. The results are shown in Figs 1, 3, 4, 6 and 7 for a 12 m/s mean windspeed and in Figs. 8, 10, 11, 12, 13 and 14 for a 20 m/s windspeed. The windspeed inputs used were the same as in the PI case to facilitate direct comparison, i.e. the windspeed trace for mean windpseed of 12 m/s is Fig. 1 and the windspeed trace for mean windspeed of 20 m/s is Fig. 4. In both cases plots of input aerodynamics torque, blade pitch angle, blade pitch velocity and blade pitch acceleration are given.

4. Comparison of Control Performance

Comparing Figs. 2 and 3 and Figs. 9 and 10, it is evident that the classical controller achieves a considerable reduction in the extent of the transient components of the aerodynamic torque. Because the hypothetical wind turbine was deliberately chosen to be dynamically active to maximise the difficulty of the control task, the quantititative extent of the torque excursions in Figs 2, 3, 9, and 10 are a little excessive but the relative extent of the excursions is a reasonable indicator of what is possible.

The reduction in torque excursions must not be achieved at the expense of greatly increased control action. Figs. 4 and 5 for mean windspeed of 12 m/s and Figs. 11 and 12 for 20 m/s indicate that activity of the blades is increased by the classical controller over the PI controller but this increase is not excessive. The blade pitch acceleration and velocity remain well within the saturation levels, of 90 deg/s/s and 15 deg/s respectively, shown in Figs. 6 and 7 for mean windspeed 12 m/s and Figs. 13 and 14 for mean windspeed 20 m/s.

The classical controller is judged to give improved performance over the PI controller and is more suited to the control of wind turbines. To obtain more insight into the relative performance a section for a mean windspeed of 12 m/s is shown in great detail, Figs. 15-18. From the aerodynamic torque Fig. 17, it can be seen by comparing the sections a, b and c that the extent of torque excursions increases as the windspeed increases. The blade pitch acceleration decreases ; near 12 m/s windspeed, section a, it sometimes saturates but by 20 m/s, section c, it is very small compared to the saturation level. These observations are in agreement with the trends indicated in Table 3 and Table 4.

The classical controller investigated has in no sense been optimised. Its design is best suited to windspeeds near 16 m/s. At 12 m/s it is perhaps too active and its performance should be reduced to lessen the amount of blade pitching. At 20 m/s the activity of the controller is low and might be increased to reduce the extent of the excursions of the drive-train loads. If a controller were designed to achieve these modifications to performance the increase in loads at high windspeed which is apparent from Figs. 5 and 10 would be reduced. In fact it might be possible to eliminate the increase. Whether it is desirable for the controller to be so designed requires a contextual judgement. It was not done during the investigation reported here since it would have increased the complexity of the control algorithm.

The structure of the classical controller appropriate to cater for the aerodynamic coefficient nonlinearities and the transitions from below to above rated operation is shown in Fig. 19. The transfer functions for the two subsystems of the controller are plotted in Figs. 20 and 21.

5. Variable Speed Wind Turbines - Some Observations

Variable speed operation of horizontal axis wind turbines is perceived to have several potential advantages of which two frequently mentioned ones are:

(i) Additonal energy capture below rated windspeed.

(ii) Additional power-train compliance and associated load alleviation above rated windspeed.

In judging the utility of a variable speed design these advantages must be assessed against the disadvantages which include increased complexity and additional hardware costs.

To assess the performance advantages of a variable speed wind turbine, it is necessary to make comparisons to some constant speed wind turbine but, to avoid introducing bias to the conclusions, the constant speed and variable speed wind turbines must in some sense be equivalent. Unfortunately, what exactly constitutes an equivalent wind turbine is not clear. For example, the criterion for equivalence might be that the drive-train torque rating for both is the same. This enables the variable speed wind turbine to generate more power in high windspeeds by operating with a higher rotor speed. However, a more highly rated generator is required which is less efficient at the more frequently occurring lower windspeeds. In addition, the wind turbine structure experiences greater stress since the centrigual forces on the rotor are greater. Also, the blade bending moments and the thrust forces on the wind turbine are increased by variable speed operation. Whether the increased power generated at high windspeed outweighs these disadvantages requires careful assessment and can not be determined in isolation from the complete design problem for the wind turbines.

The appropriate method for comparison is to design a constant speed wind turbine and a variable speed wind turbine for a site with the objective of generating a specified number of kilowatt-hours per annum at minimum costs. Whether the performance advantages of the variable speed wind turbine outweigh the disadvantages can be judged by comparing the relative costs of the machines.

Because of the preceding considerations when comparing a variable speed wind turbine to a constant speed wind turbine, they are assumed to have the same maximum power rating and rotor diameter. The power production of the variable speed wind turbine rises rapidly with increasing windpseed until rated power is reached. At higher windspeed it is regulated to produce the rated power. To enable the constant speed wind turbine to have the same performance envelope, it is assumed to be pitch regulated. With these assumptions, the energy production for a specific site of the variable speed and constant speed wind turbines are comparable and, particularly since energy capture by the rotor and not energy production is estimated below rated windspeed, the performance advantages may be assessed with a minimum of bias.

With these assumptions it has been shown [5] that the additional energy capture of a variable speed wind turbine on a constant speed wind turbine is only 2 to 3%. Because the advantage is marginal, the loss of energy capture due to poor tracking of the $C_{p\ max}$ curve, which might be 1 to 2% of energy capture, should not be ignored. It may be argued that poor tracking is not important since the peak of the C_p curves is broad and flat but in this case the extra energy capture is small and not economic. If the peak of the C_p curves is sharp to enable the additional energy capture to be worthwhile then poor tracking loses a significant fraction of the additional energy capture. In these circumstances the ability to track the $C_{p\ max}$ curve closely is dependent on the control system.

In addition it has been shown [5] that for a variable speed wind turbine, without pitch regulation above rated windpseed, the extent of the additional power-train compliance is limited. With pitch regulation above rated windspeed employed to regulate the rotor speed, the wind turbine experiences additional structural loads which increase as the range of allowed rotor speed variation increases. To reduce these loads requires the rotor speed variation to be decreased, i.e. the control action has to be improved.

6. Conclusions

For an actively pitch regulated constant speed wind turbine the following have been concluded.

(i) Classical control is much more appropriate than PI control.

(ii) Drive-train load transients can be successfully regulated by classical control.

(iii) Fast control action is not required. (The classical controller is active for frequencies less than 0.5 Hz only)

(iv) The transition from below to above rated operation presents no difficulty. (Figs. 7 and 17 show several transitions).

(v) Active control is capable of adding damping to the power-train. (The open loop system is lightly damped).

Although these benefits have only been demonstrated for a simulation of the wind turbine and some degradation must be expected in actual implementation on a real machine the performance of the classical controller should still be markedly improved over the PI controller.

For a variable speed wind turbine the following have been concluded.

(i) The additional energy capture is only a few percent.

(ii) With a flat C_p curve the additional energy capture is marginal.

(iii) Using a peaked C_p curve to improve energy capture, must be done in conjunction with a controller which achieves good tracking.

(iv) Above rated windspeed, there is little if any increase in compliance when the turbine is not variable pitch.

(v) Above rated windspeed, the controller should act to reduce speed variation to an absolute minimum when the turbine is variable pitch.

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Figure Legends

Figure 1 Windspeed with mean of 12 m/s.

Figure 2 Aerodynamic torque for PI controller with mean windspeed of 12 m/s.

Figure 3 Aerodynamic torque for classical controller with mean windspeed of 12 m/s.

Figure 4 Blade pitch angle for PI controller with mean windspeed of 12 m/s.

Figure 5 Blade pitch angle for classical controller with mean windspeed of 12 m/s.

Figure 6 Actuator pitch velocity demand for classical controller with mean windspeed of 12 m/s.

Figure 7 Actuator pitch acceleration demand for classical controller with mean windspeed of 12 m/s.

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Figure 10 Aerodynamic torque for classical controller with mean windspeed of 20 m/s.

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Figure 12 Blade pitch angle for classical controller with mean windspeed of 20 m/s.

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Figure 15 Section of windspeed with mean of 12 m/s.

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Figure 17 Section of aerodynamic torque for classical controller with mean windspeed of 12 m/s.

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Figure 19 Structure of classical controller

Figure 20 Bode plot for classical controller subsystem S_1

Figure 21 Bode plot for classical controller subsystem S_2

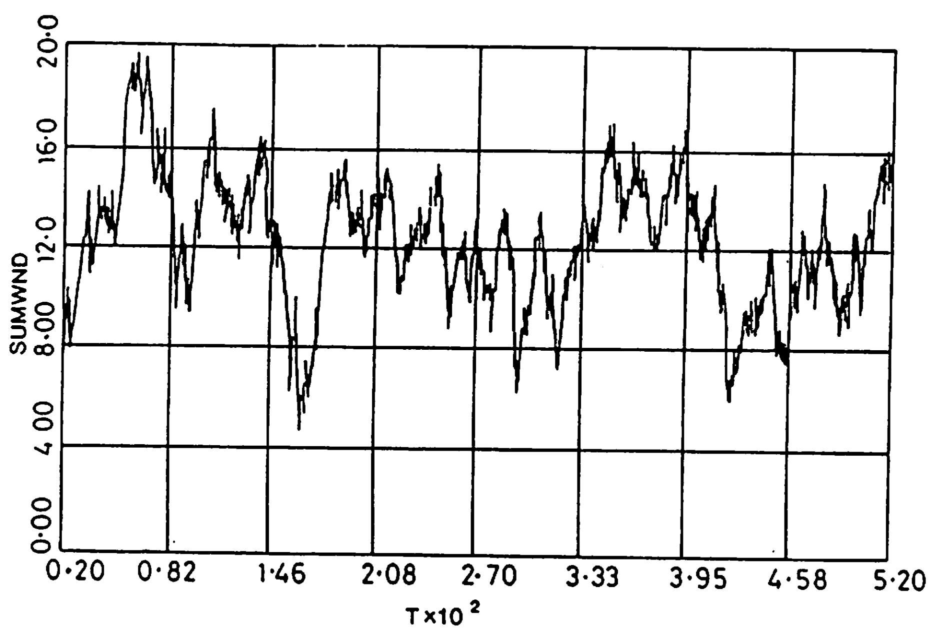


Figure 1 Windspeed with mean of 12 m/s.

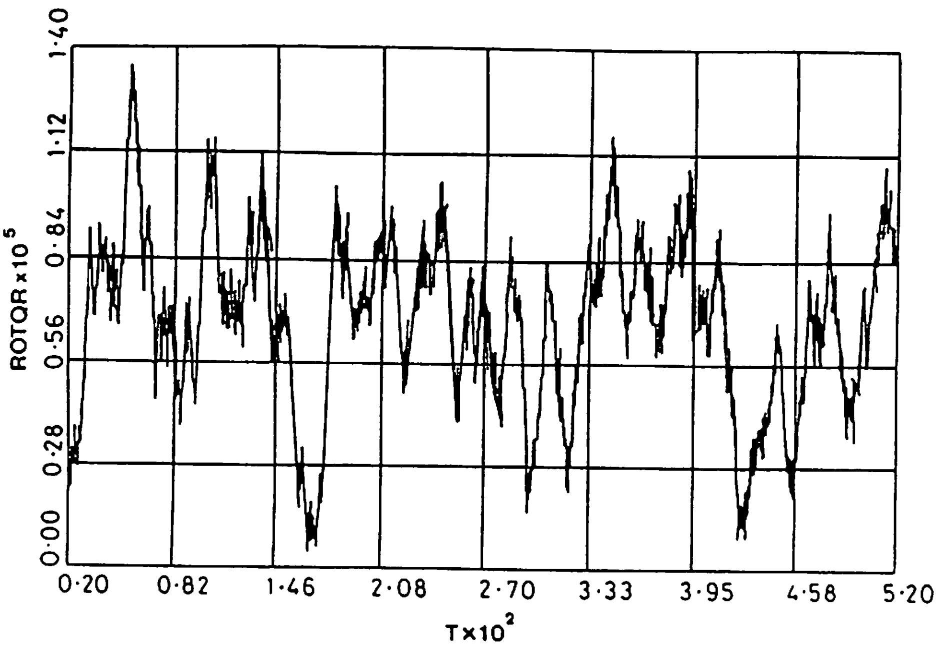


Figure 2 Aerodynamic torque for PI controller with mean windspeed of 12 m/s.

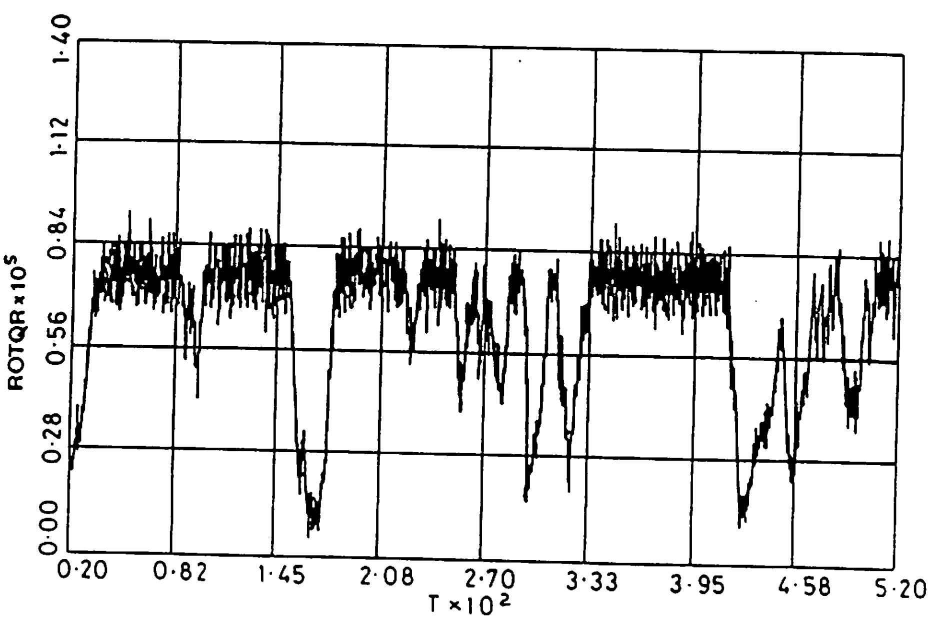


Figure 3 Aerodynamic torque for classical controller with mean windspeed of 12 m/s.

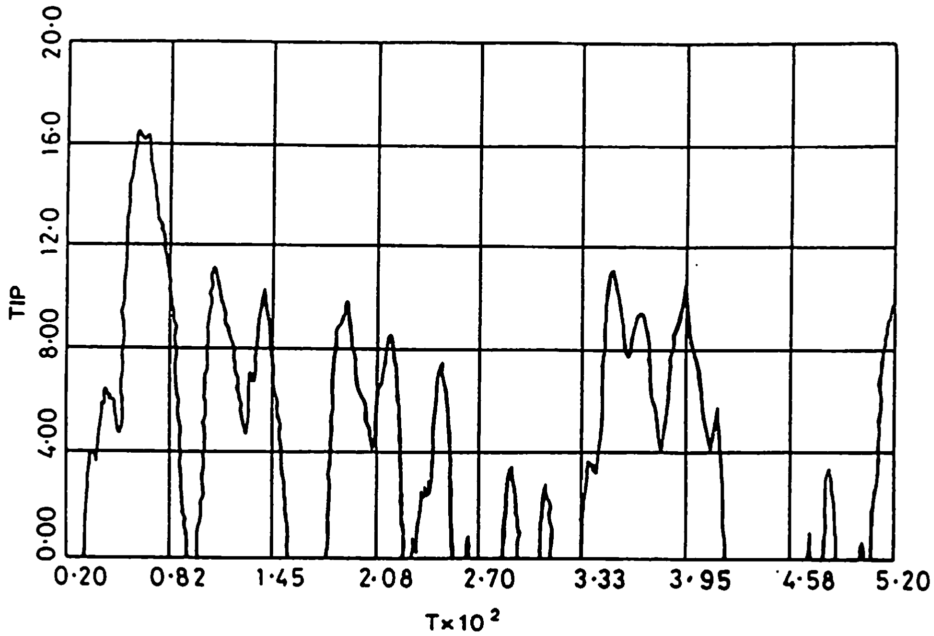


Figure 4 Blade pitch angle for PI controller with mean windspeed of 12 m/s.

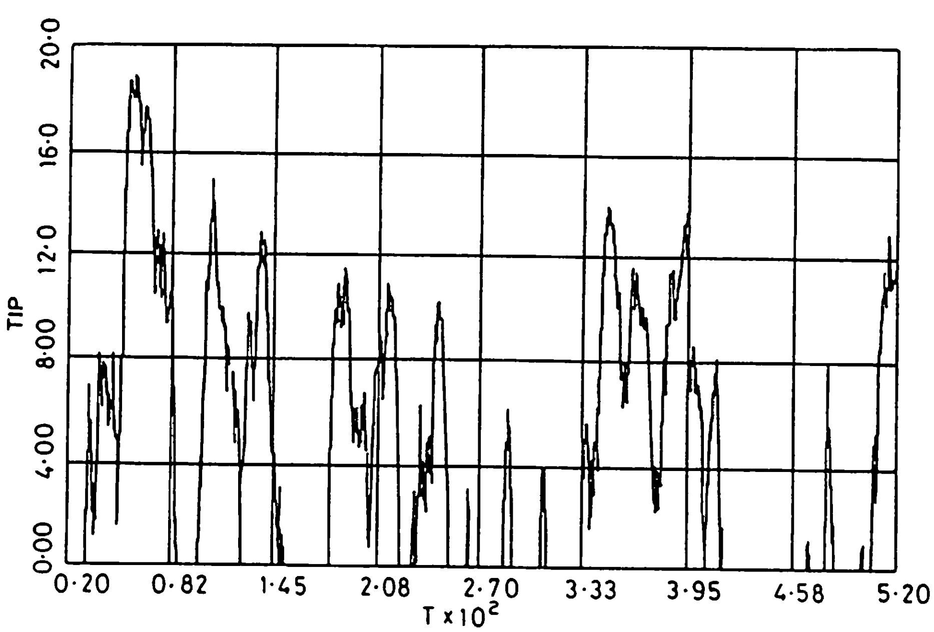


Figure 5 Blade pitch angle for classical controller with mean windspeed of 12 m/s.

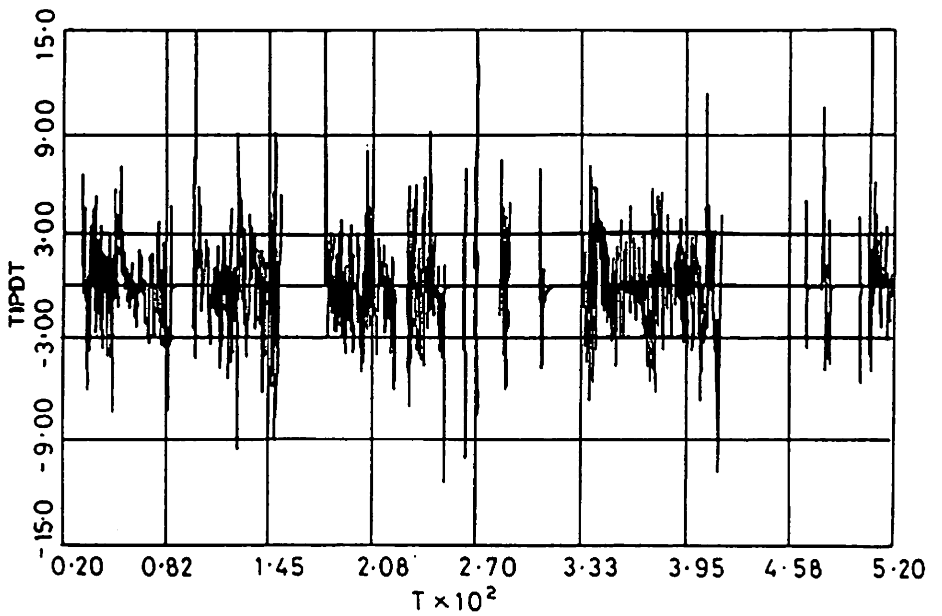


Figure 6 Actuator pitch velocity demand for classical controller with mean windspeed of 12 m/s.

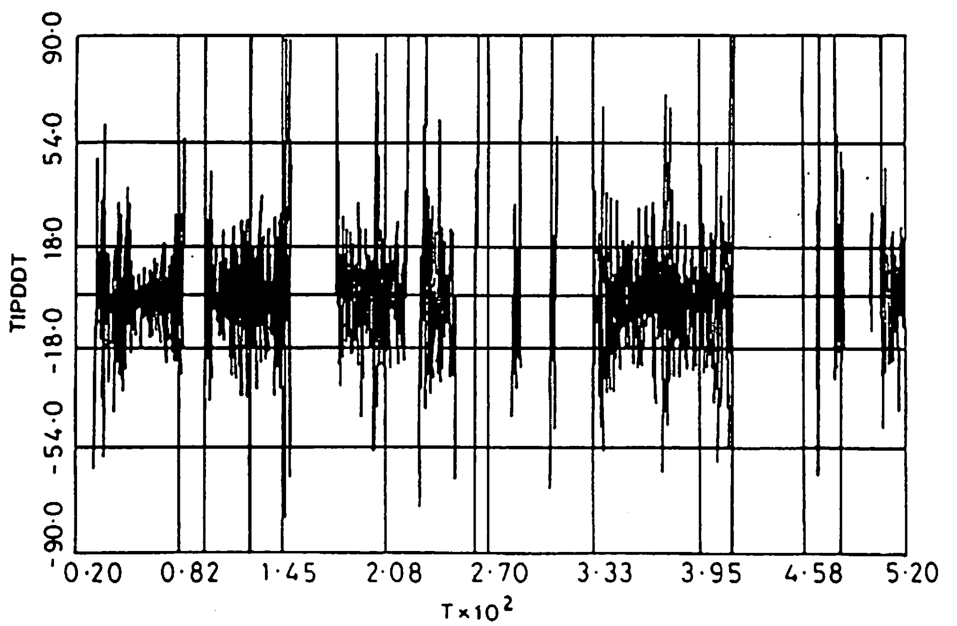


Figure 7 Actuator pitch acceleration demand for classical controller with mean windspeed of 12 m/s.

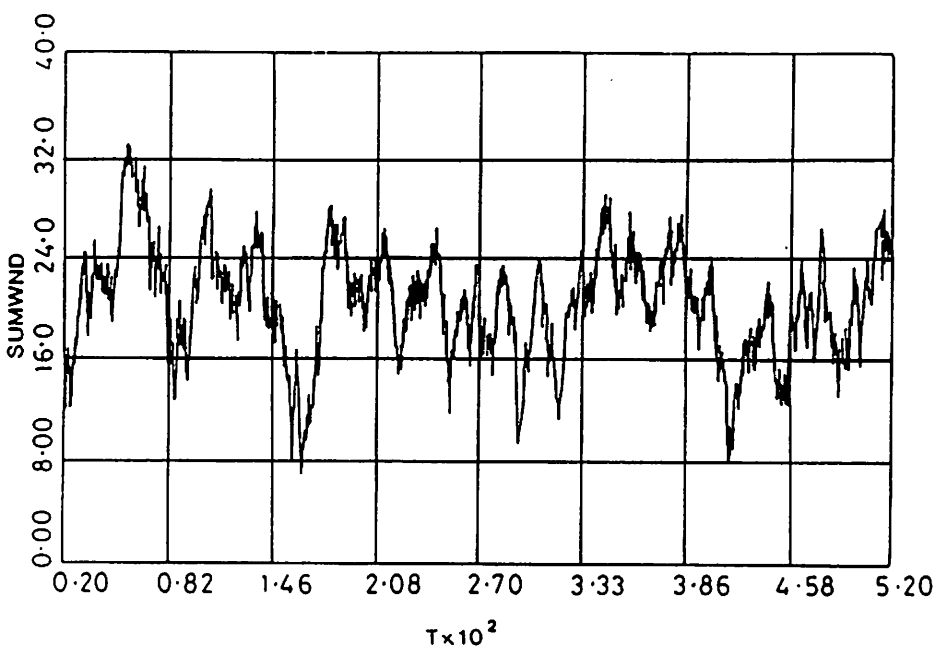


Figure 8. Windspeed with mean of 20 m/s.

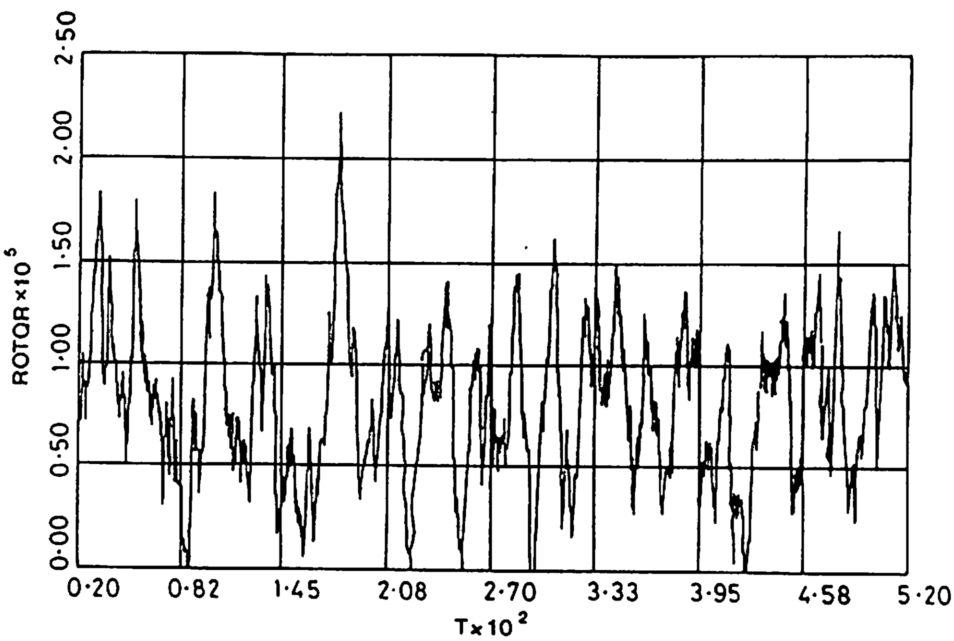


Figure 9 Aerodynamic torque for PI controller with mean windspeed of 20 m/s.

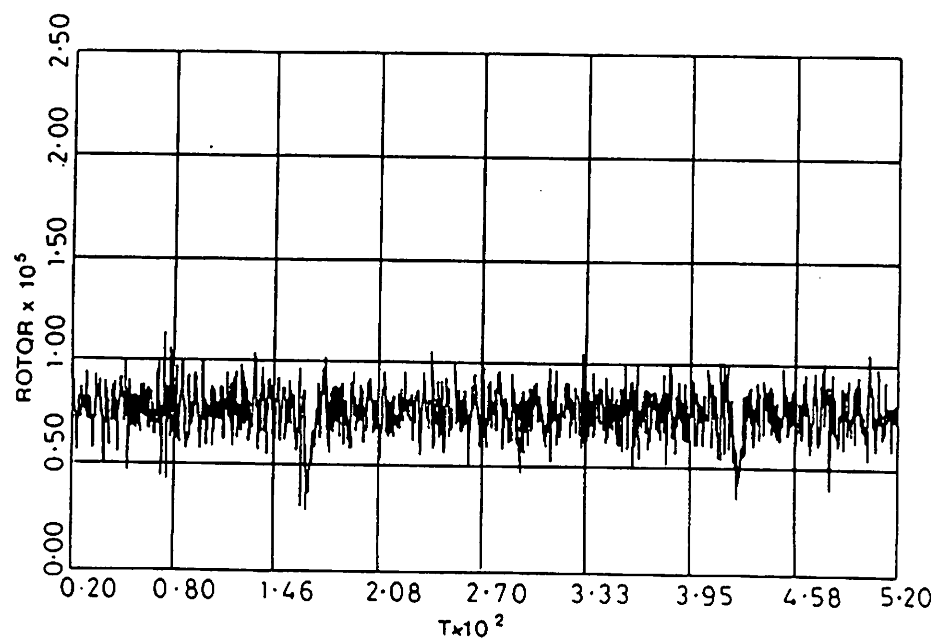


Figure 10 Aerodynamic torque for classical controller with mean windspeed of 20 m/s.

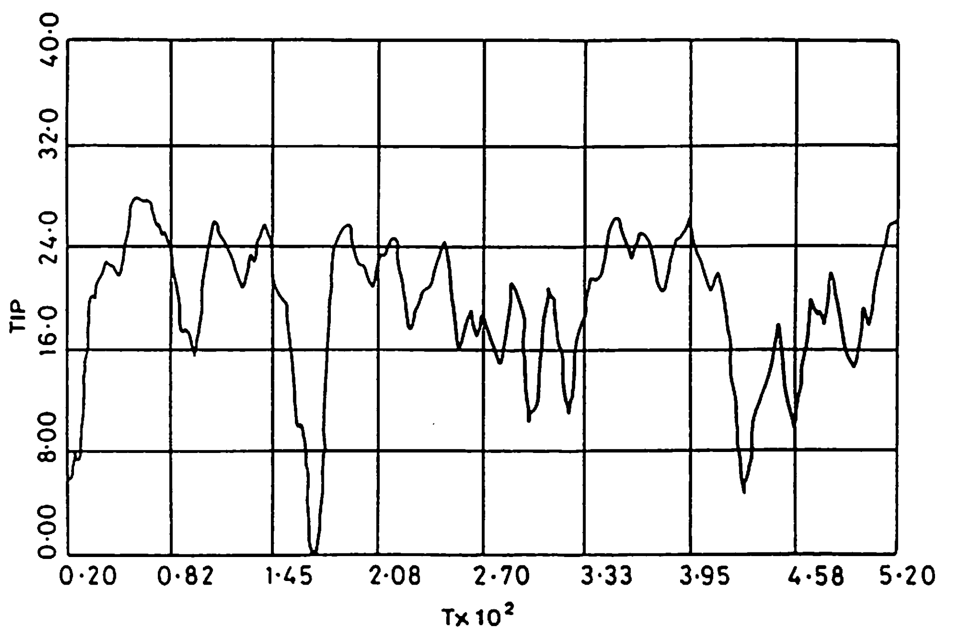


Figure 11 Blade pitch angle for PI controller with mean windspeed of 20 m/s.

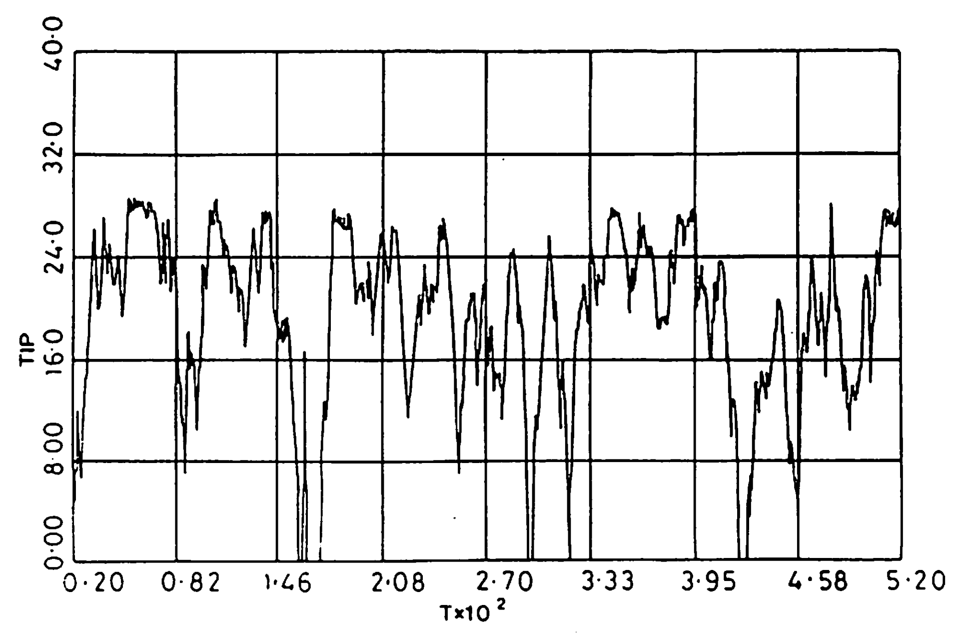


Figure 12 Blade pitch angle for classical controller with mean windspeed of 20 m/s.

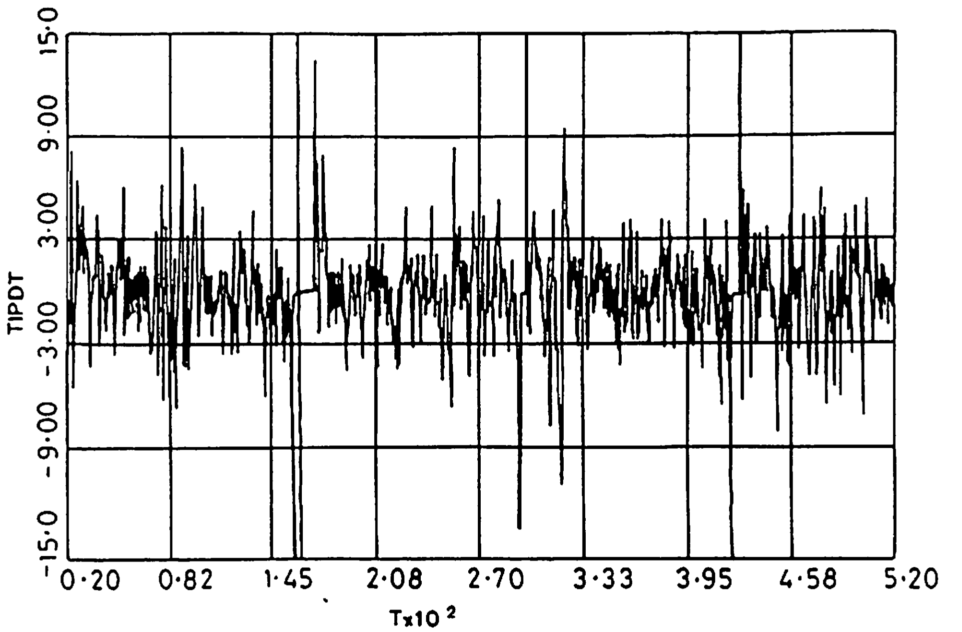


Figure 13 Actuator pitch velocity demand for classical controller with mean windspeed of 20 m/s.

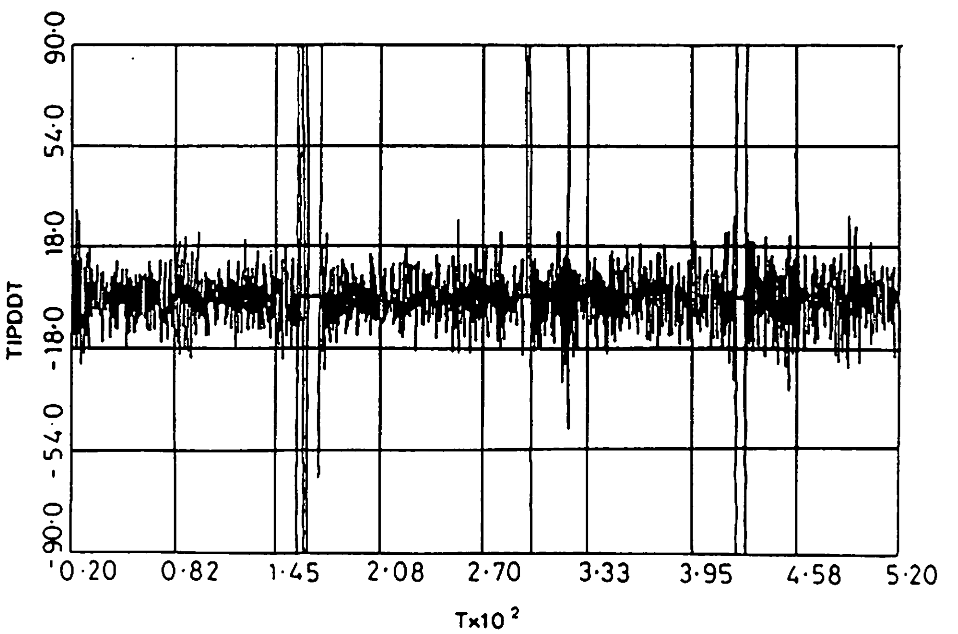


Figure 14 Actuator pitch acceleration demand for classical controller with mean windspeed of 20 m/s.

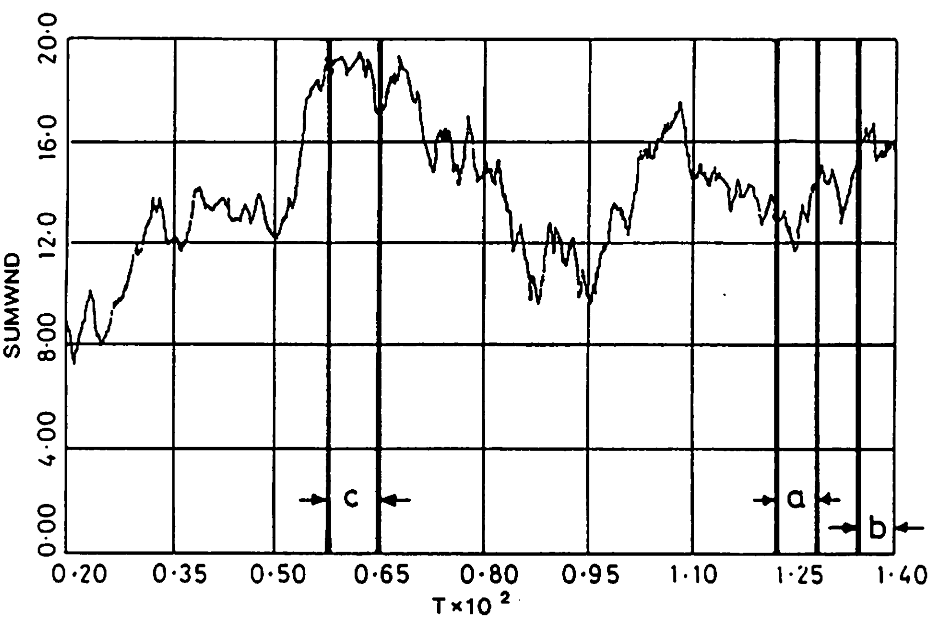


Figure 15 Section of windspeed with mean of 12 m/s.

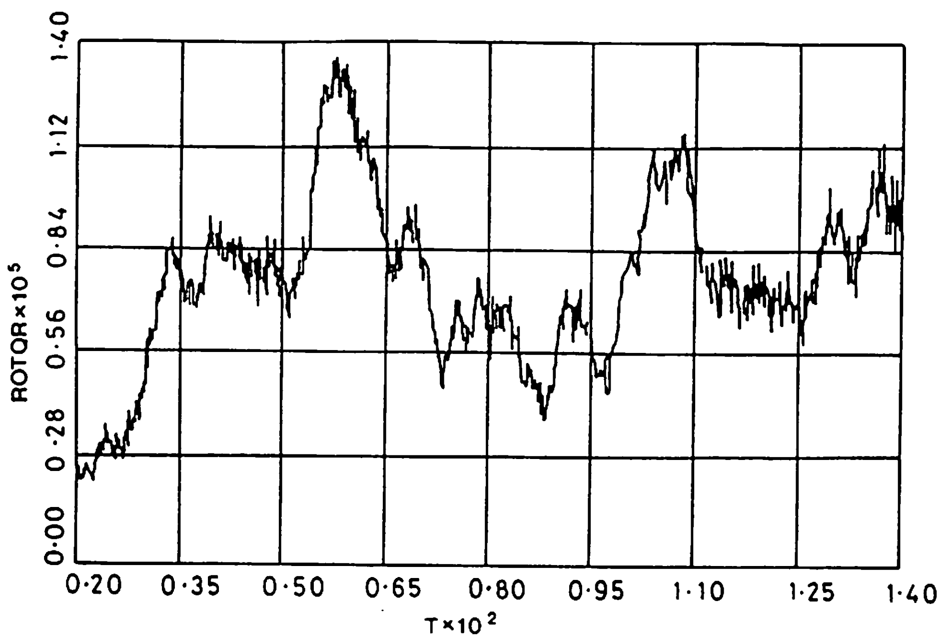


Figure 16 Section of aerodynamic torque for PI controller with mean windspeed of 12 m/s

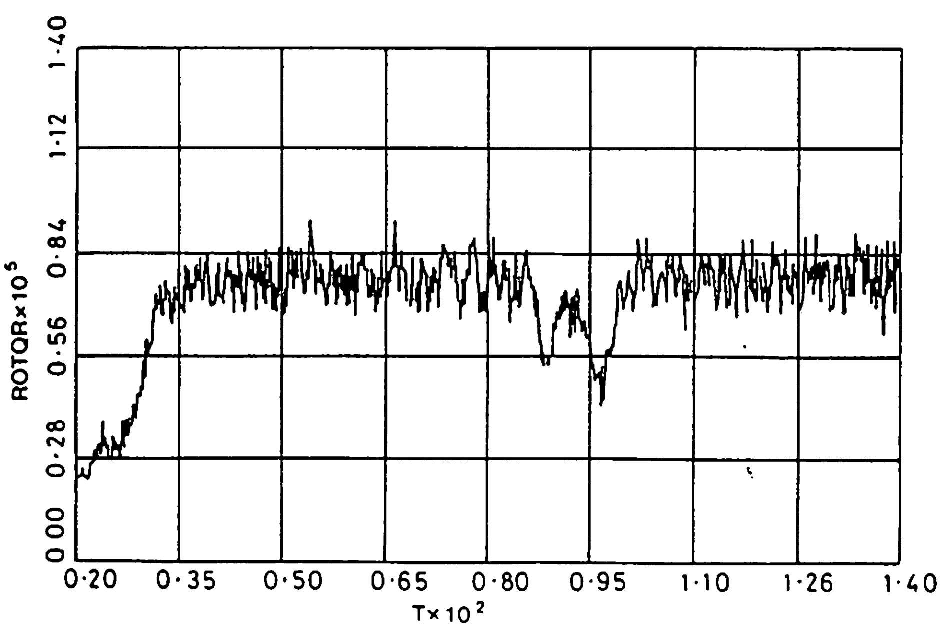


Figure 17 Section of aerodynamic torque for classical controller with mean windspeed of 12 m/s.

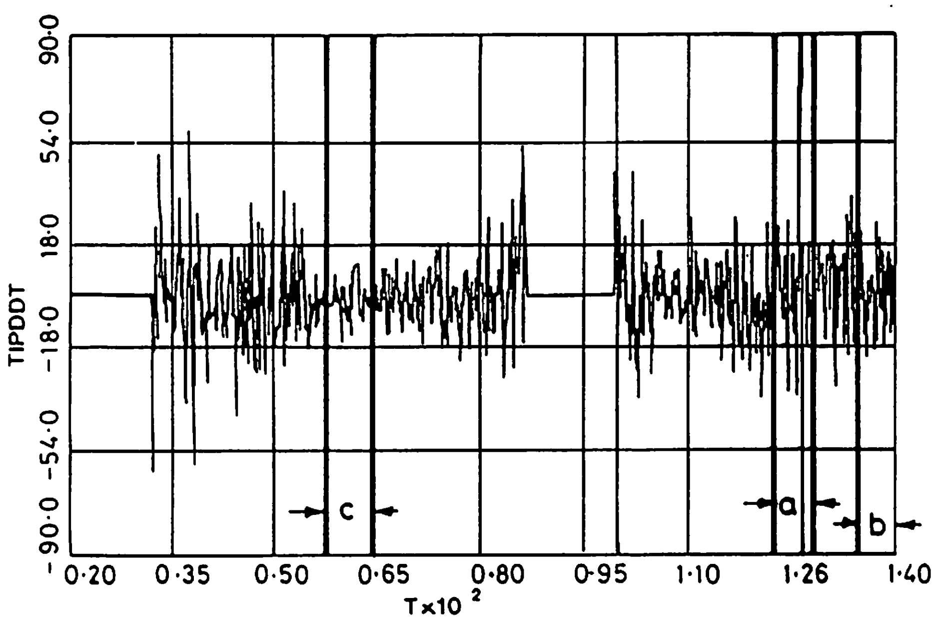


Figure 18 Section of actuator pitch acceleration demand for classical controller with mean windspeed of 12 m/s.

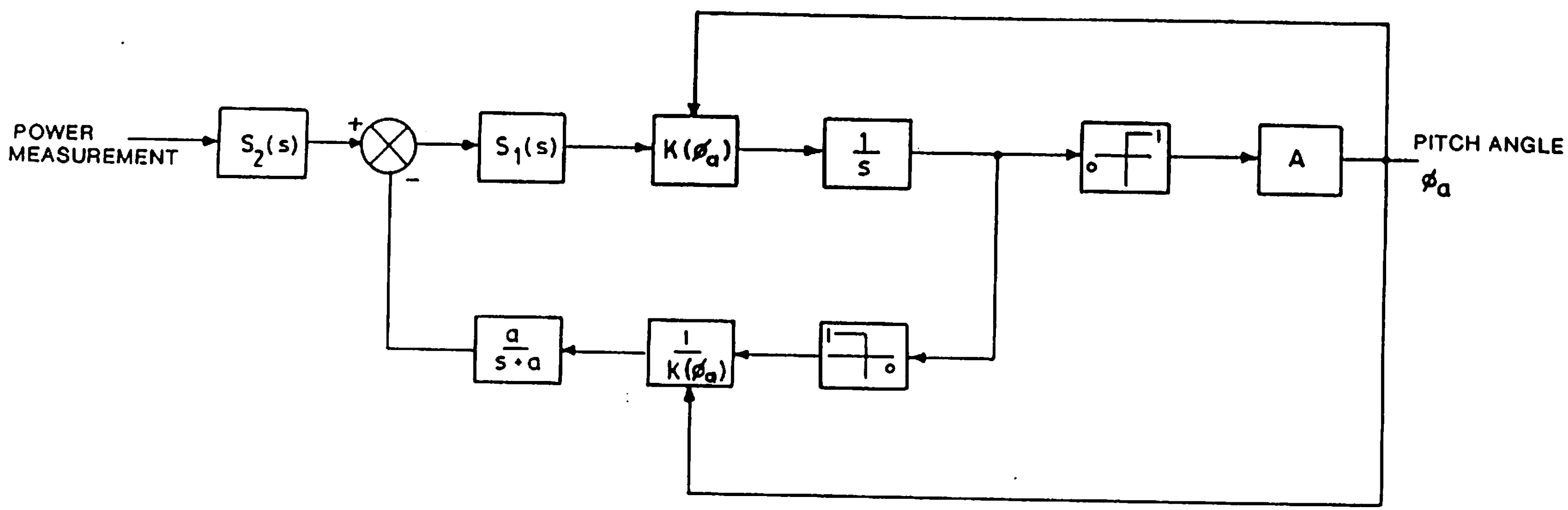


Figure 19 Structure of classical controller

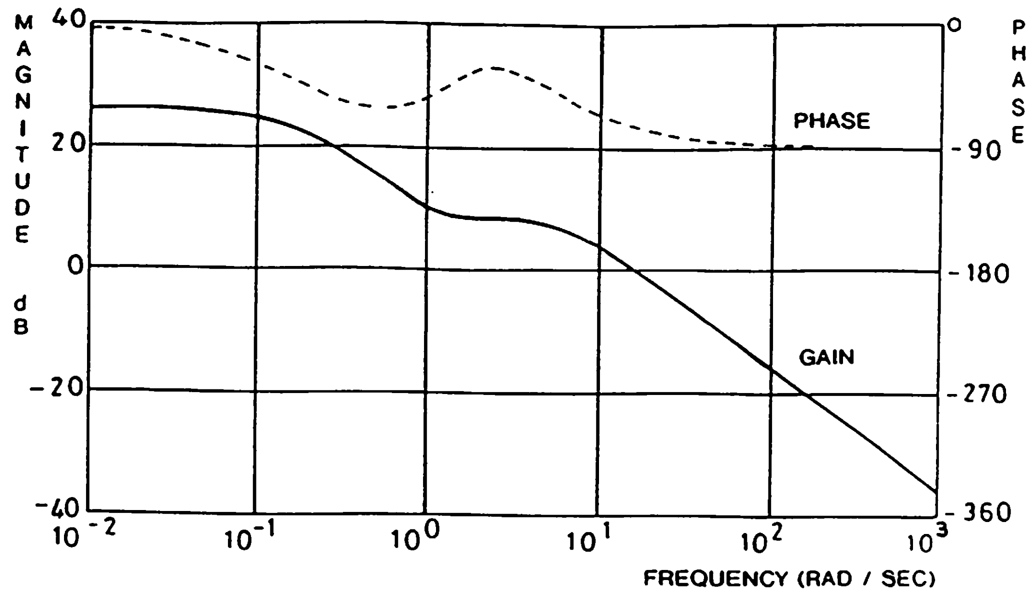


Figure 20 Bode plot for classical controller subsystem S_1

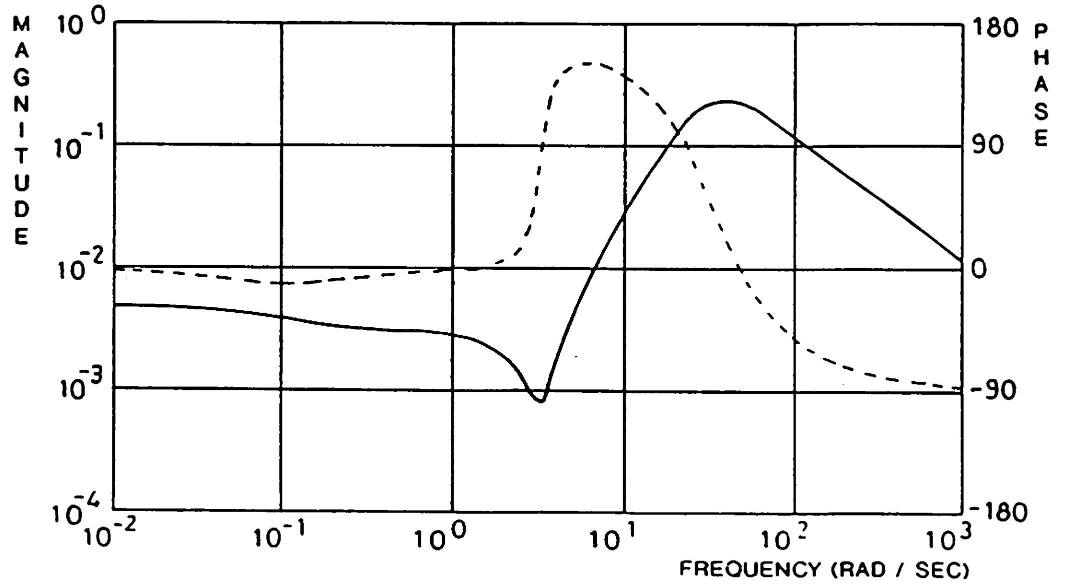


Figure 21 Bode plot for classical controller subsystem S_2

A COMPARISON OF AERODYNAMIC DEVICES FOR CONTROL AND OVERSPEED PROTECTION OF HAWTs

P.Jamieson and P.Agius
James Howden Group Technology

OBJECTIVES

* To review existing aerodynamic options for the regulation of HAWTs - both power limiting and overspeed prevention.

* To present two new ideas;

  SLEDGE - sliding leading edge,

  FLEDGE - flying leading edge.

Emphasis is placed on developing a rational approach to establish the effectiveness of an aerodynamic control device. Overspeed performance is discussed in detail. Consideration is given to control characteristics (active and passive), and particularly how to minimise the size of a device and the associated actuation loads.

OVERSPEED CONTROL

1) An aerodynamic brake need not stop the rotor but it must keep it at a safe speed in all design wind conditions.

2) Typically, this may lead to a criterion such as keeping the rotor below synchronous speed in all windspeeds below 30 m/s (as suggested by Riso).

3) Howden design on the basis that the rotor should be in no more than 30% overspeed in the design maximum gust. Taking into consideration gust factors etc. , this criterion is rather similar to Riso.

The material of the remainder of the presentation refers usually to various types of aerodynamic control devices selected for a 45m diameter rotor for a wind turbine rated at 750 kW generator output. The wind turbine has a tip speed of approximately 70 m/s and runs at 30 rpm.

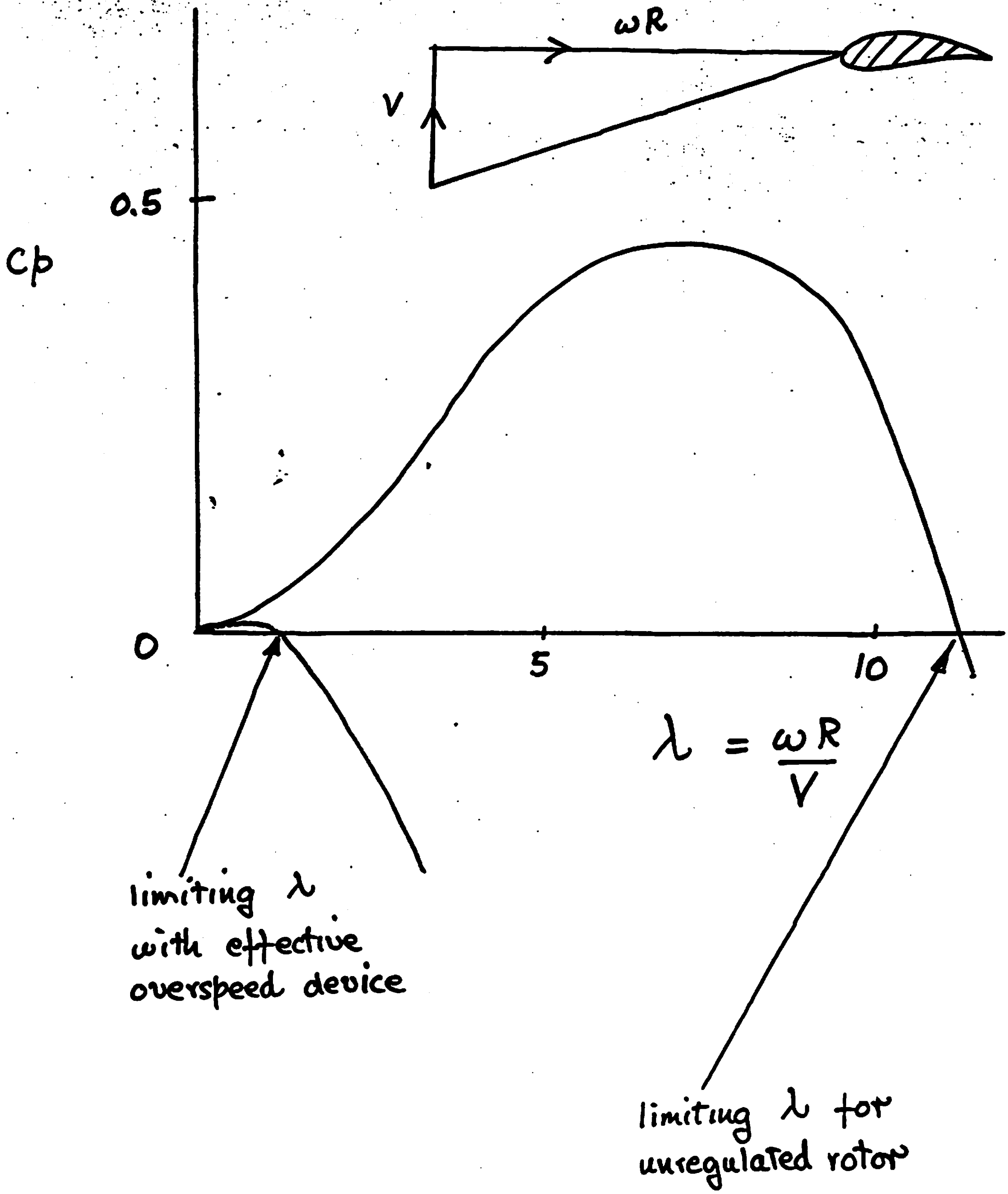
Thus, the maximum allowable rpm is 1.3 x 30 = 39 rpm, and this should only result in a 3 second gust wind speed of 60 m/s. When the airbrakes are operational, the rotor must therefore be limited to a tip speed ratio of;

$$\lambda = \frac{39 \times 2 \times \pi \times 22.5}{60 \times 60}$$

$$= 1.53$$

Thus, in what follows, various aerodynamic devices are sized, in terms of the amount of blade span required, to give a limitng tip speed to wind speed ratio of approximately 1.5.

OVERSPEED CONTROL



CONVENTIONAL AERODYNAMIC DEVICES

1) LEADING EDGE SPOILER

Between 6 and 10m of span is needed to meet the criterion of $\lambda_{max.} = 1.5$. Such spoilers are only effective if very near the leading edge. However ,in such a position, they are likely to trip the boundary layer and degrade performance in normal operation when they are retracted.

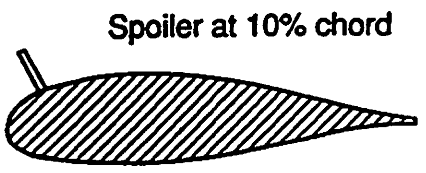
2) VENTED AILERON

About 9m span is needed - structural engineering will not be simple.

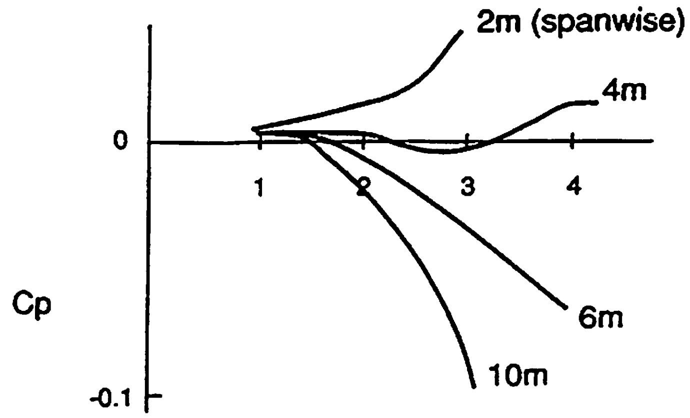
3) ROTATING TIP

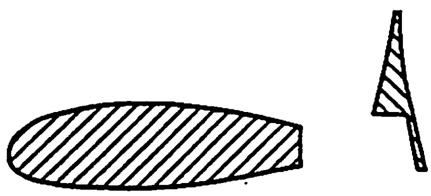
This is highly effective aerodynamically - BUT - quite difficult to engineer within the envelope of the outboard aerofoil sections, because of structural requirements and actuating loads.

CONVENTIONAL AERODYNAMIC DEVICES

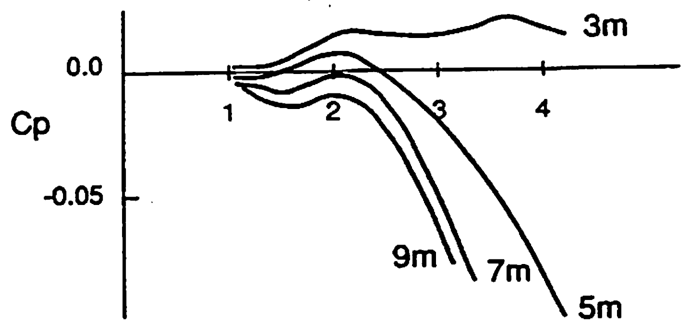


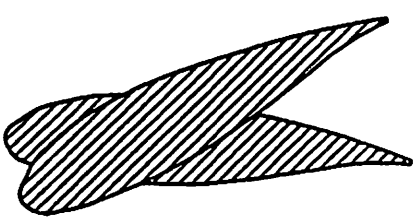
Imperial College type



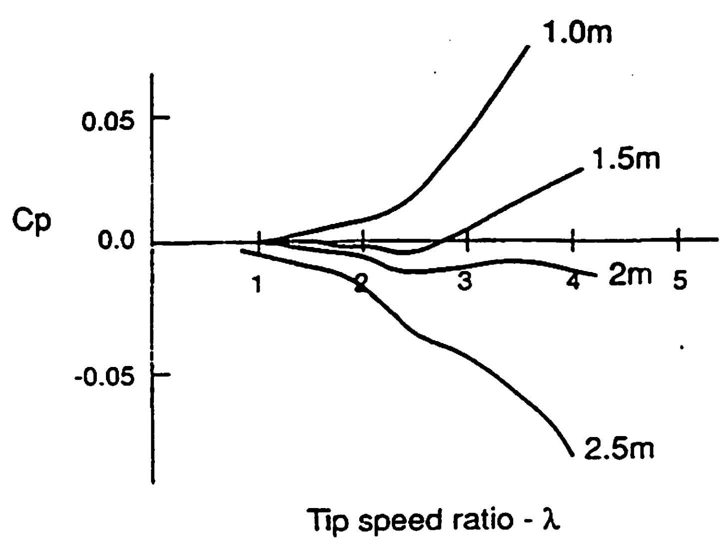


Vented aileron





Rotating tip



THE SLEDGE - SLIDING LEADING EDGE

This is an invention of the author patented by Howden.

The main advantages claimed are;

* natural failsafe operation linked directly to centrifugal force,

* high aerodynamic effectiveness due to a threefold effect, viz.

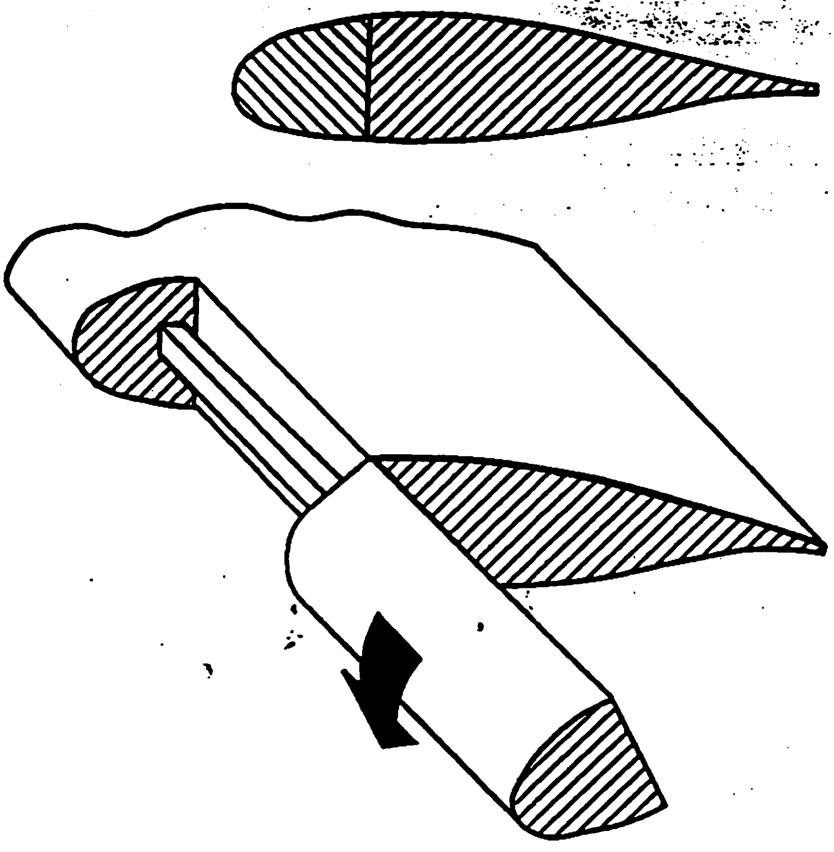
    1) destruction of the positive performance of the blade tip region,
    2) creation of additional drag there,
    3) displacement of the leading edge section radially outward into a region of increased relative wind velocity where it will be particularly effective, using both negative lift and drag to retard the rotor.

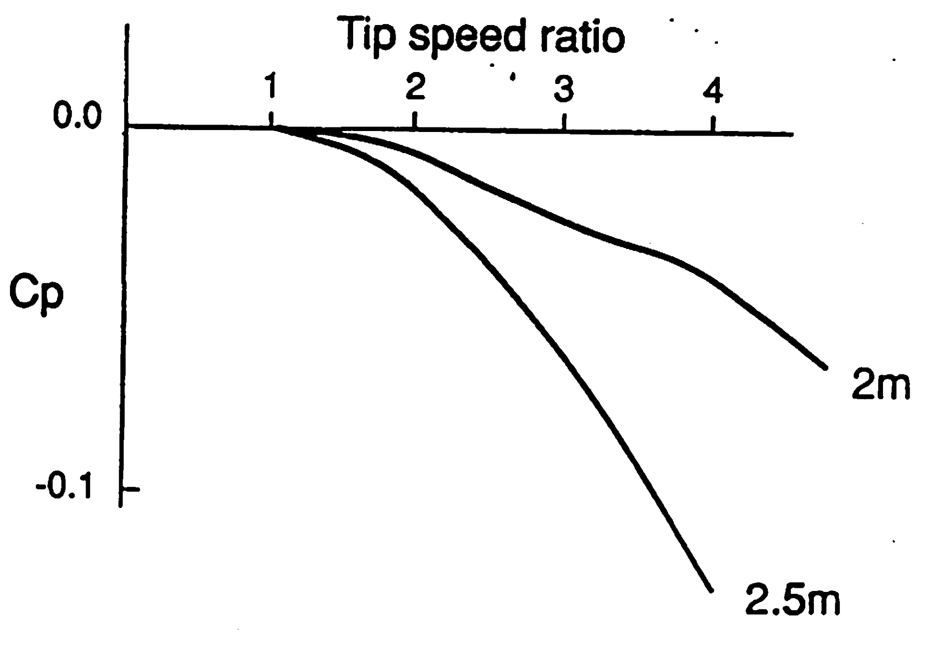
* small control surface area requirement in consequence of the effectiveness, and hence potential for low cost,

* protection from bending loads and dynamic loads due to the strong effect of centrifugal stiffening; also the avoidance of substantial flapwise loadings that are not useful for controlling rotor torque.

Note that the SLEDGE translates and also rotates. Because the 'D' section of the sledge is a highly effective lifting body, a sledge of minimum span is achieved by using this property and rotating the sledge about 50 degrees into negative incidence. The rotation is easy to engineer because of the relatively long travel of the sledge.

THE SLEDGE - SLIDING LEADING EDGE



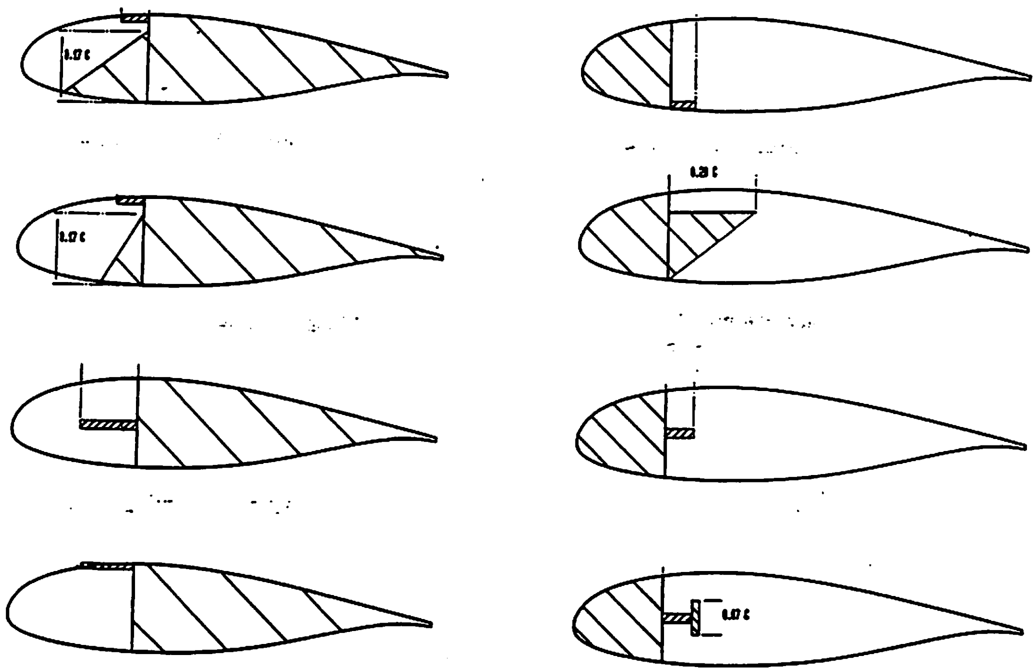


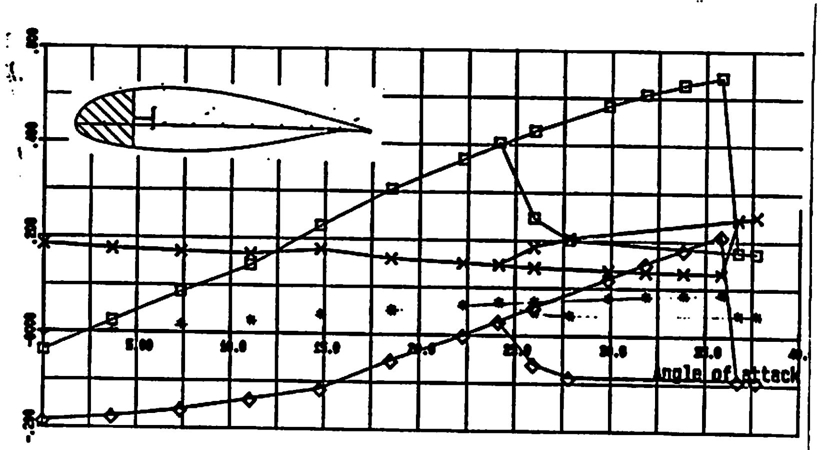
WIND TUNNEL TEST ON THE SLEDGE

An extensive series of wind tunnel tests have been done at Imperial College, London, as part of a project funded by the UK Department of Energy. A report is available on request from ETSU at Harwell Laboritories (E/5A/CON/5119/1989.

The effect of numerous variations of shape of the sections was investigated in two dimensional test in a 1.4 x 1.2m wind tunnel. This was aimed at optimising the shapes aerodynamically and also to be aware of the effects of variations in shape that might result from the practical engineering of the slide mechanism etc. Over 60 configurations were tested. In addition tests were done in a larger wind tunnel on the effects of 3d geometry, on the effects of Reynolds No., and on the effects of imperfect sealing of the joins in the aerofoil section. The following is an example of a few of the configuations tested. In each case lift drag and pitching moment were measured.

SLEDGE – A NOVEL BRAKING DEVICE FOR HORIZONTAL AXIS WIND TURBINES





CONTROL OF WIND TURBINES USING MASS DYNAMICS

Hitherto, control of wind turbines has relied on aerodynamics or on controlling the electrical load. As the sledges deploy they introduce transient Coriolis forces which can be used to control cyclic loads in the wind turbine system.

The following calculation is based on a sledge design for a 45m wind turbine similar to the present Howden HWP 750/45 wind turbine. It is easy to show that the Coriolis term is not negligible.

- I – Rotor inertia (excluding sledges)
- ω – Rotor angular velocity, 3.1 rad/s
- q – Assumed rate of deployment, 2 m/s
- m – Sliding mass, 100 kg
- r – Radius to mass centre of sliding system, 20 m
- y – Displacement of the sledge at maximum velocity, 1 m

Consider the case where the speed is held constant e.g. braking at full power before the contactor comes out;

Torque = rate of change of angular momentum

$$= \frac{d\,[I + 3m(r + y)^2]\,\omega}{dt} \quad \text{assuming a 3 bladed rotor}$$

$$= 6m\,(r + y)\,q\,\omega$$

$$= 6 \times 100 \times 21 \times 2 \times 3.14 \text{ Nm}$$

$$= 79 \text{ kNm}$$

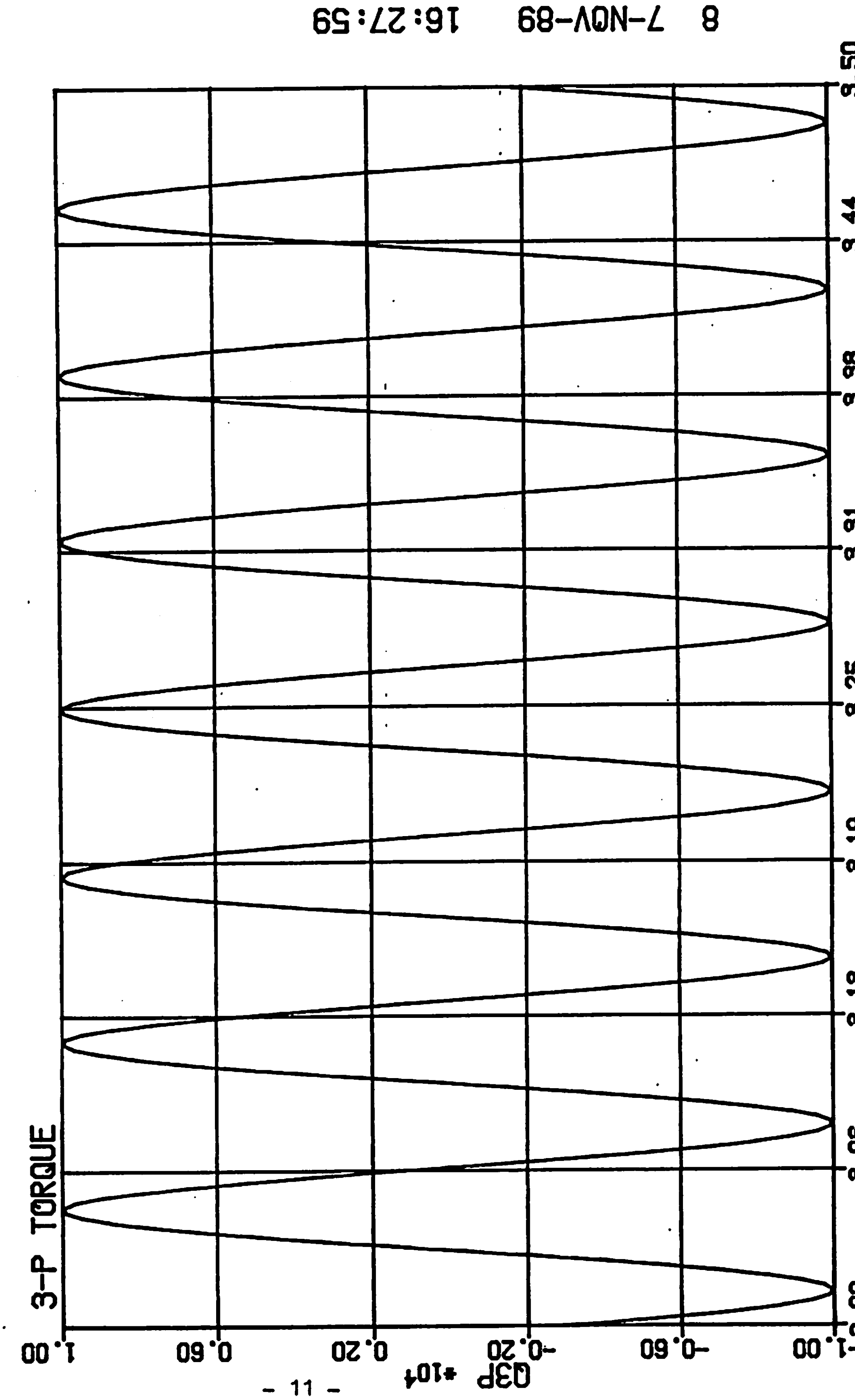
This is far from being a negligible torque change; it corresponds to a power reduction of 250 kW in the nominally 750 kW rated system.

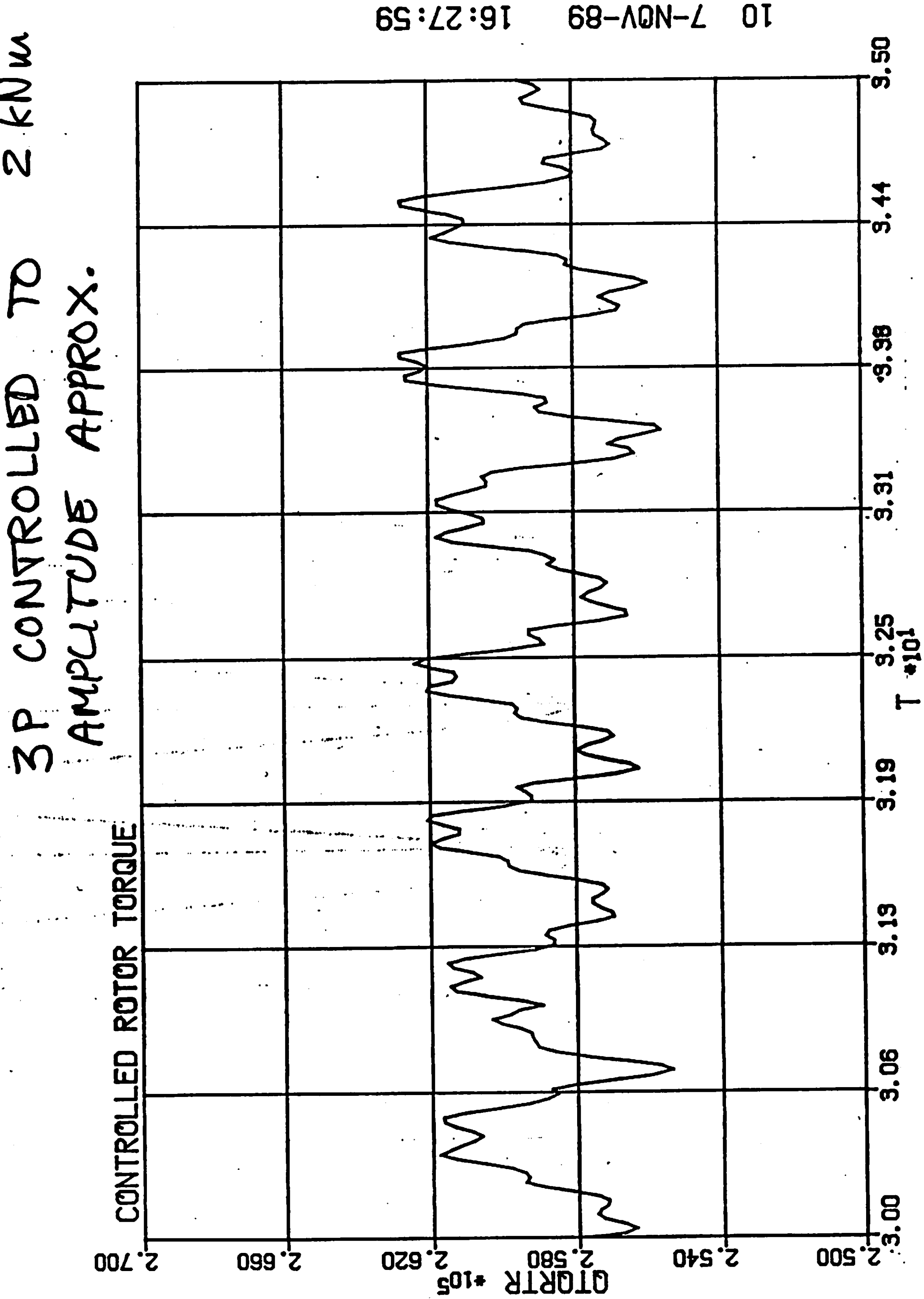
This resulted in a further study for the UK Department of Energy being done on the feasibility of using sledges for control of power fluctuations. The idea is that low frequency coontrol would be done in the usual way using aerodynamics in a slow control loop. Harmonics of 1 2 or 3P (3 x rotor frequency) in the shaft torque could be cancelled using small oscillations of the sledges exploiting the Coriolis forces. This would be done in a fast control loop.

The study has confirmed the feasibility of this and some preliminary results from a simulation model are presented. The model includes turbulent wind input, non-linear aerodynamics, rotor, all drive train components and the control system all modelled in the time domain. These results show how a sinusoidal input at 3P is controlled by the fast loop and the amplitude reduced.

The controller is designed to shape the frequency response of the system to reduce 3P power fluctuations.

INPUT
10kNm
amplitude




3P CONTROLLED TO 2 kNm
AMPLITUDE APPROX.
CONTROLLED ROTOR TORQUE
10 7-NOV-89 16:27:59
QTQRTR *10^5
2.700
2.660
2.620
2.580
2.540
2.500
T *10^1
3.00
3.06
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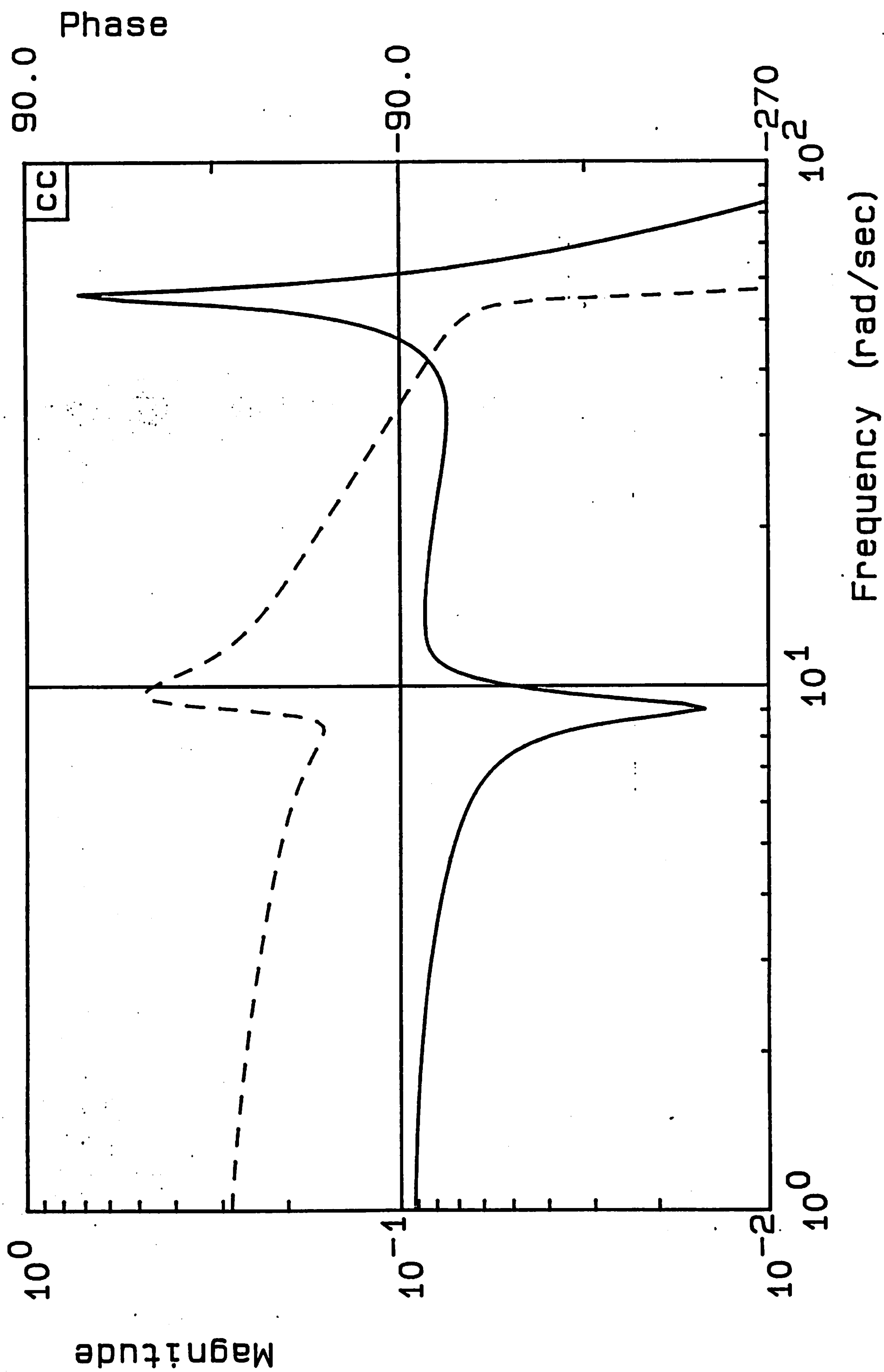
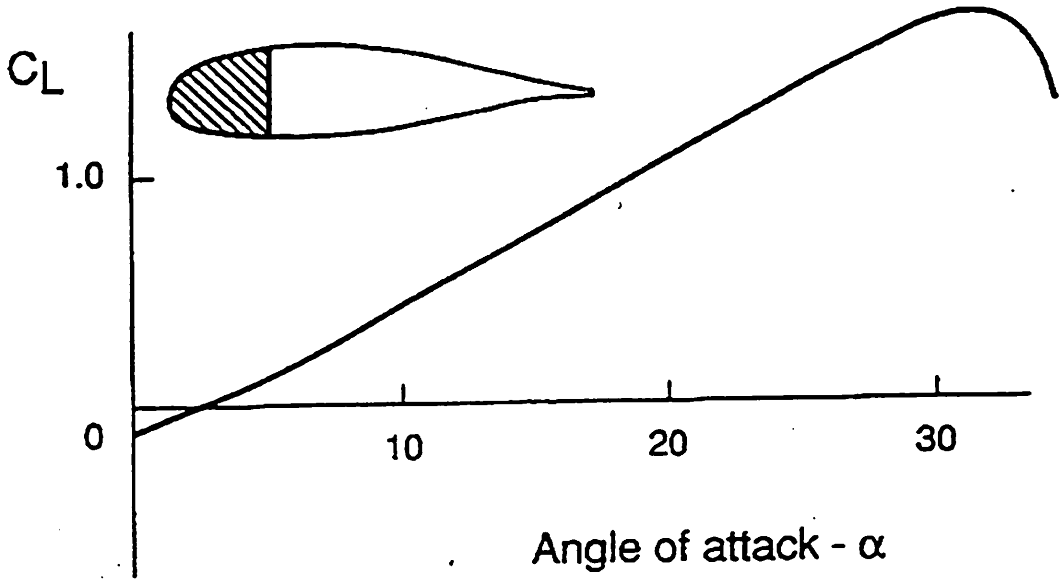
3.31
3.38
3.44
3.50


CC
Phase
90.0
-90.0
-270
Magnitude
10^0
10^{-1}
10^{-2}
Frequency (rad/sec)
10^0
10^1
10^2

LIFT ON THE "D" NOSE SLEDGE SECTION



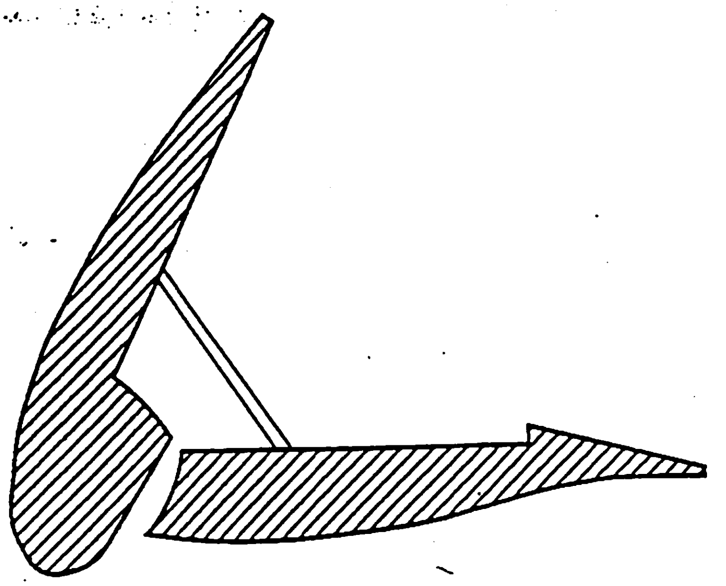
It is thought that the strong suction base at the back of the "D" section is the reason why the flow remains attached at very high incidence.

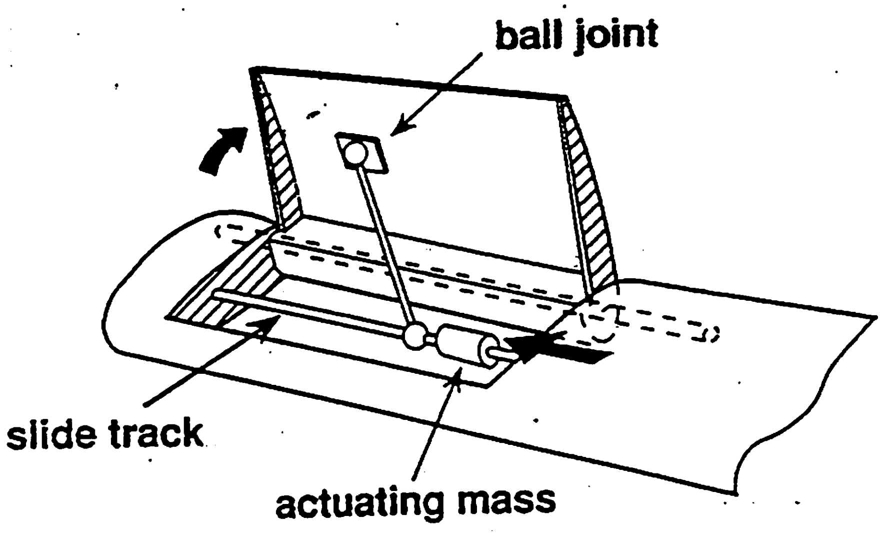
This was intially problematic in that it detracted from the performance of the a purely sliding SLEGDE. However, it led to two further ideas;

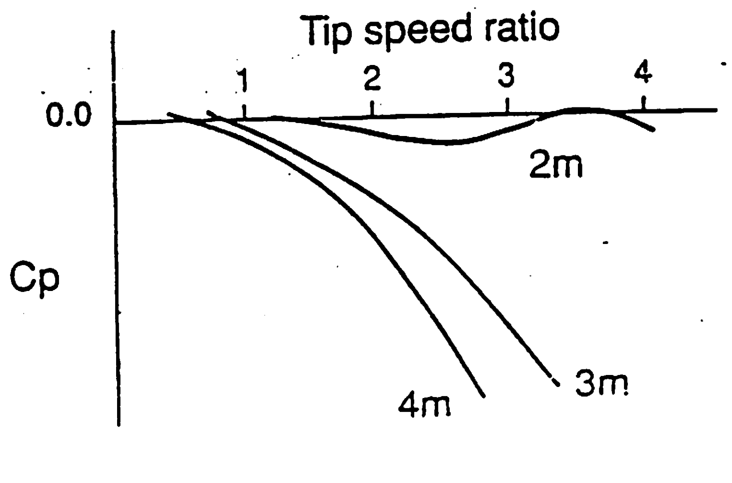
1) Rotating the sledge element as it translates to exploit the lift for regulation thereby leading to a much more compact (shorter span) sledge.

2) A purely rotating leading edge system leading to the FLEDGE idea.

THE FLEDGE - FLYING LEADING EDGE







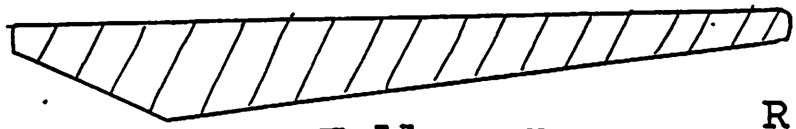
COMPARISON OF AERODYNAMIC REGULATION SYSTEMS FOR A 45 M WIND TURBINE

The following illustrates the requirement for each of 4 conventional systems and 2 new systems - sledge and fledge to meet the braking criterion of $\lambda_b = 1.5$. Of course, in the case of full span, this does not apply. The whole blade is moved as a matter of choice.

The point is that it is possible to provide overspeed protection and control with much smaller control surfaces than has previously been supposed. Thus there is a significant opportunity for engineering much cheaper control systems for wind turbines.

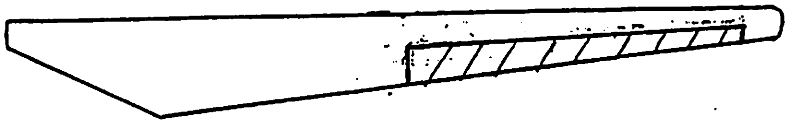
Traditionally the choice has been between full span pitch control or stall regulation. It can be shown that a full span pitch system is sensitive to turbulence in an undesirable way - and frequently full span pitch machines have been de-rated in high winds because of power control problems. Tip control devices exploit the stall regulation of the main blade section which also provides a secure limit on rotor loads. They can do better than stall regulation in avoiding a drive train with very high torque rating. They have low inertia compared to full span pitching, and can easily move as fast as is desirable to control power and load fluctuations.

COMPARISON OF AERODYNAMIC REGULATION SYSTEMS FOR A 45 m WIND TURBINE

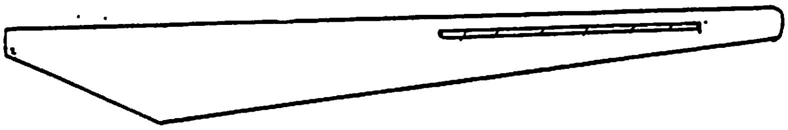


R = 22.5 m

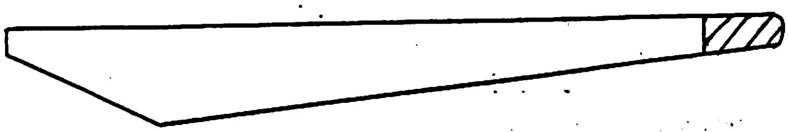
Full span



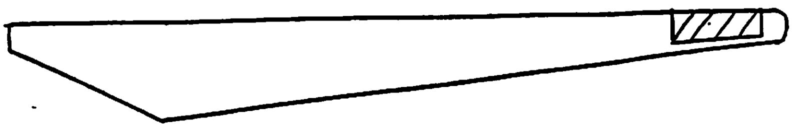
Vented aileron



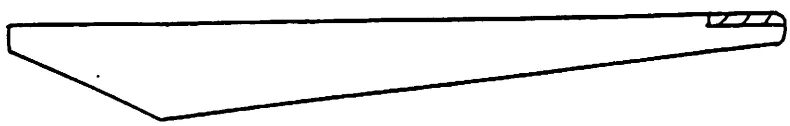
Leading edge spoiler



Rotating tip

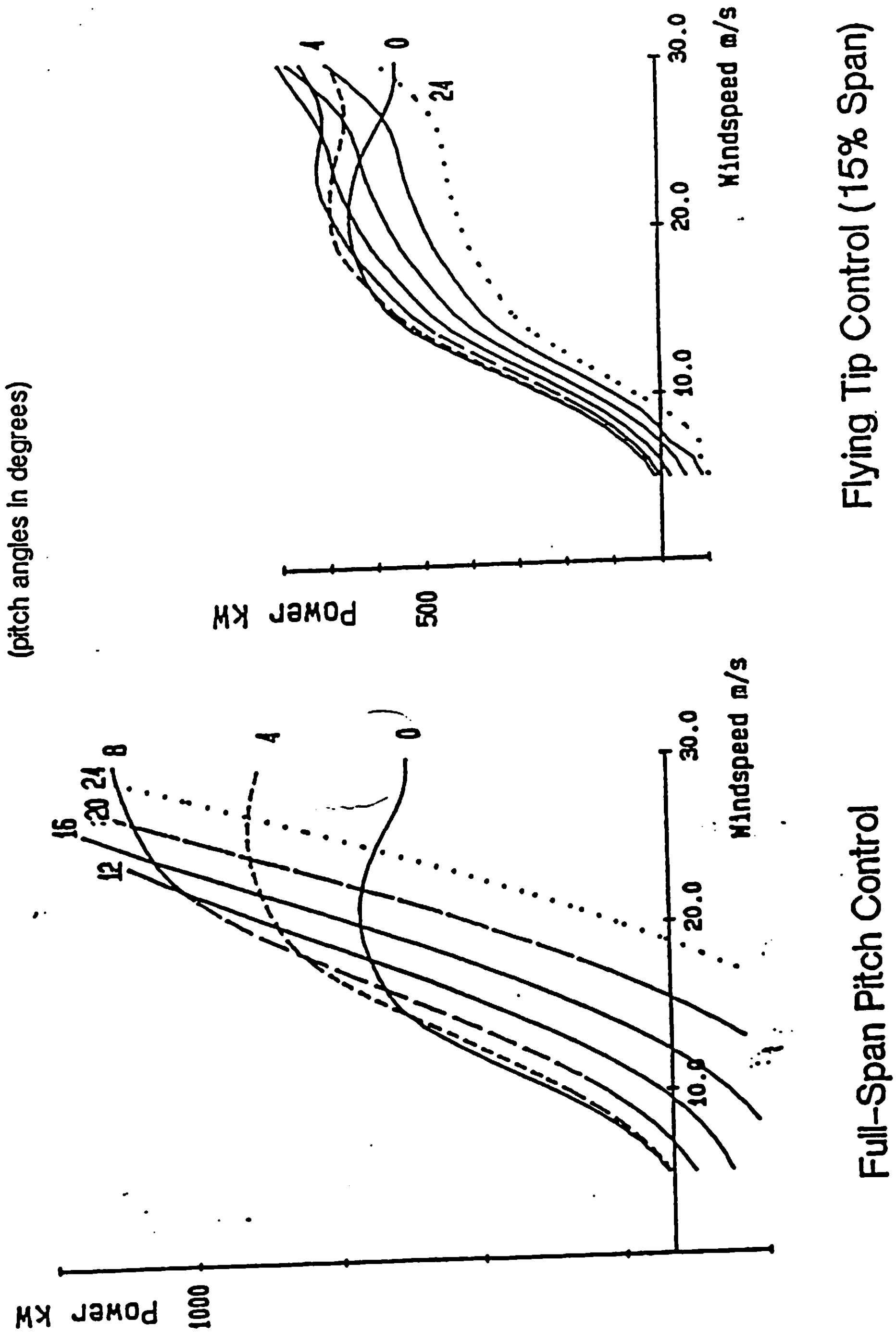


FLEDGE



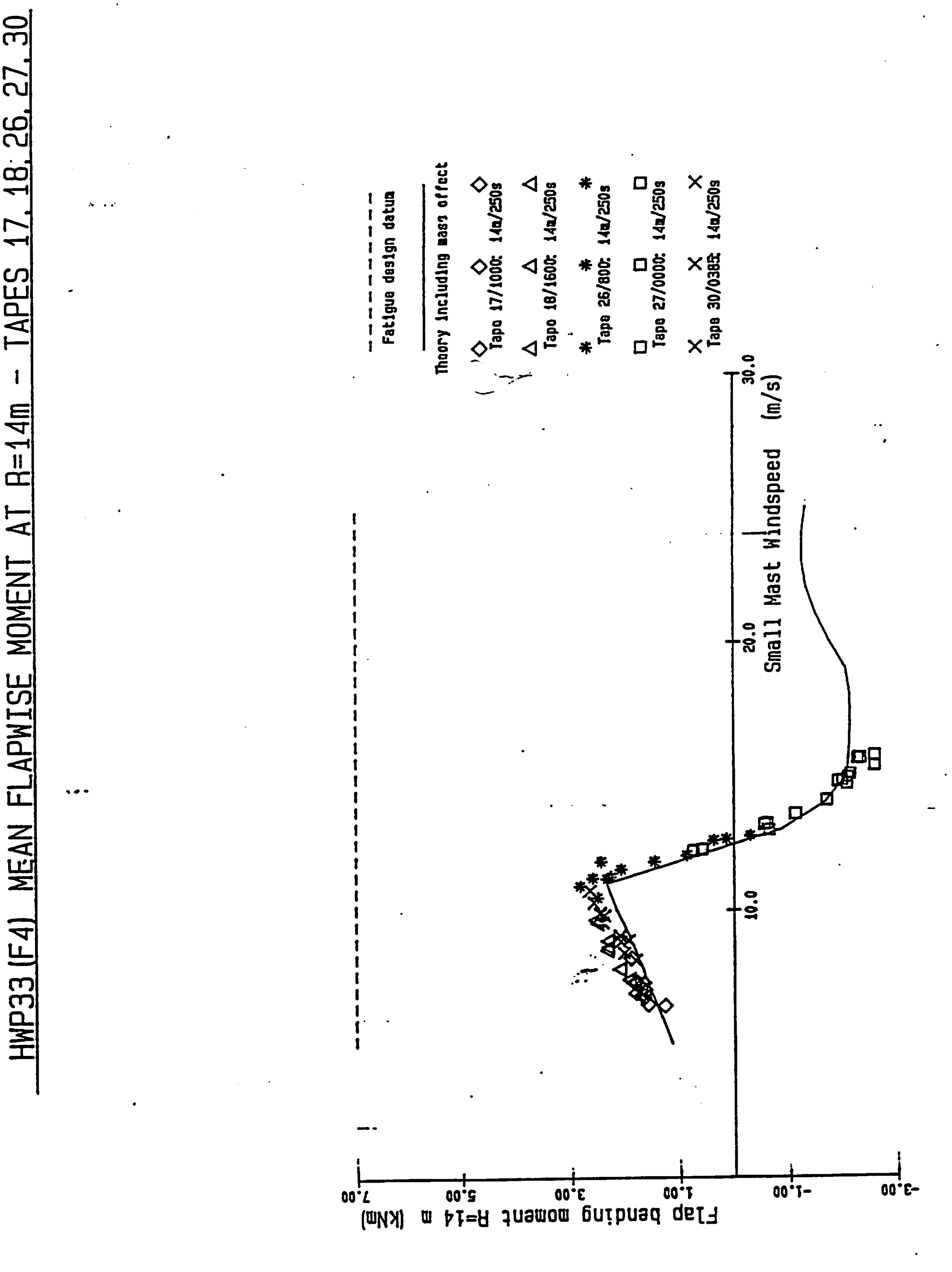
SLEDGE

Partial span control is equivalent to partial stall control. By comparison the off-design load capability of a full span pitch machine is horrific - note the steep curves and extreme power levels in high winds.



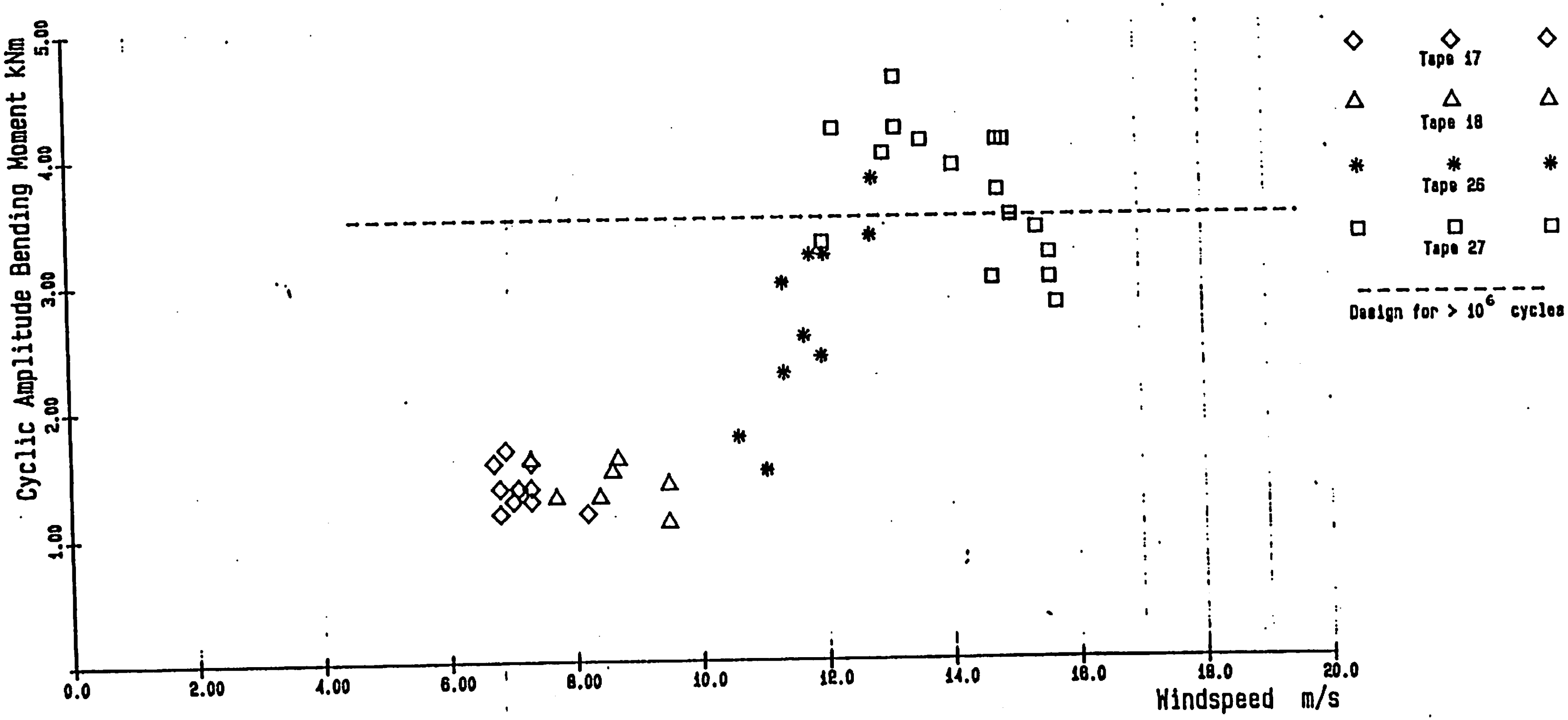
Power Curves for the HWP330/33

Near the tip joint, flying tip loads reverse as the tip is used to unload the blade. Note the steep gradient from rated windspeed 11.4 m/s onwards to 15 m/s. Peak cyclic loads occur (logically) at 13 m/s when operating in the "middle" of this gradient. See next figure.



JAMES HOWDEN & Co. Ltd.

HWP33 (F4) FLAPWISE CYCLIC BENDING : 14m STATION



JAMES HOWDEN & Co. Ltd.

FLEDGE AND SLEDGE - LOADING IMPLICATIONS

COMPARISON OF ACTUATING LOADS FOR A 45M DIAMETER WIND TURBINE

A problem with the standard rotating tip configuration is that the peak pitching loads occur at full deployment. Commonly a powerful spring is used to make tip deployment fail safe, and actuators have to overcome large forces in routine operation for power control.

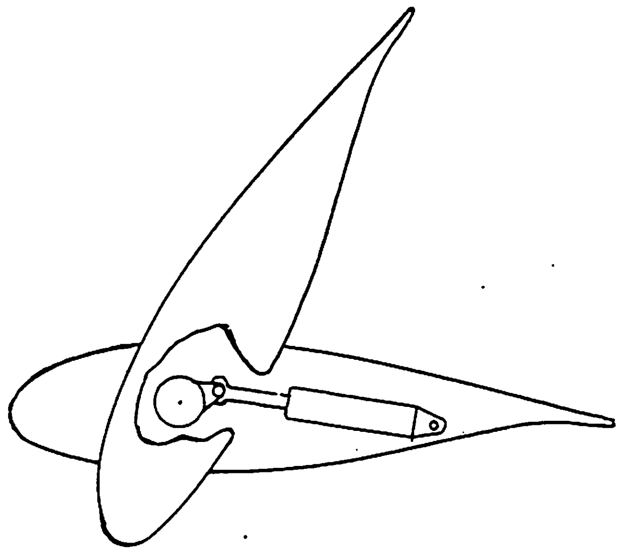
Loads on the SLEDGE depend primarily on the chosen weight. For a 45m wind turbine, with a sledge suited to Coriolis control the centrifugal load will be about 30 kN. If however, the sledge is use only as an airbrake or for conventional aerodynamic control, the loadings depend basically on the lowest weight that can be cost effectively engineered, and cable operation could even be considered. The sledge avoids the association of power (torque) control actions with flapwise loadings. This is always a problem with conventional pitching aerodynamic systems and is a souce of fatigue at the blade joints.

Loads on the FLEDGE are harmoniously integrated with the operating conditions. When the fledge starts to deploy, the actuating mass has a poor mechanical advantage, BUT there is necessarily strong aerodynamic suction assisting deployment. When the fledge opens beyond about 30 degrees, aerodynamic drag forces damp its further deployment reducing shock loading BUT the actuating mass now has a large mechanical advantage and enforces full deployment to the braking position. If the sledge is used for power control, the actuating mass is retained as the failsafe operating mechanism, and routine control is done by pulling a cable attached to the trailing edge of the fledge plate. Actuating loads are remarkably low, and actuators should be very cheap. Also the overhung structure which causes problems for the structural engineering of a standard rotating tip are avoided.

COMPARISON OF ACTUATING LOADS

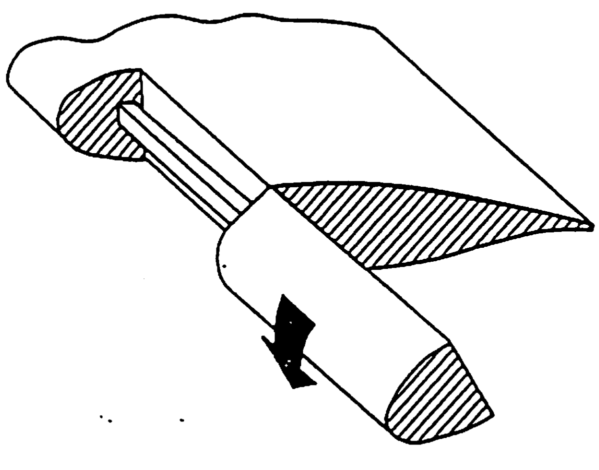
ROTATING TIP

70 kN



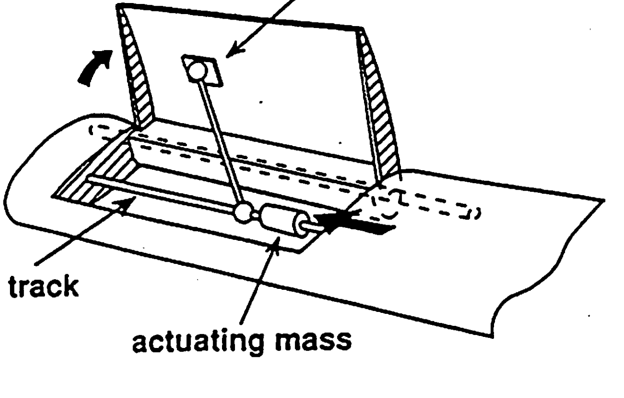
SLEDGE

10 - 30 kN

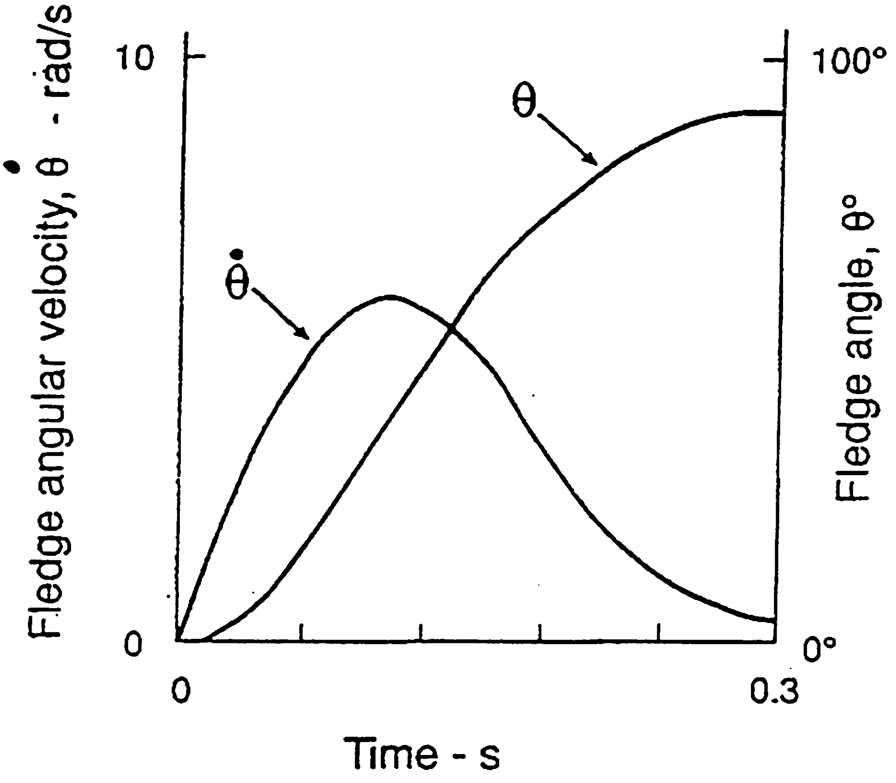


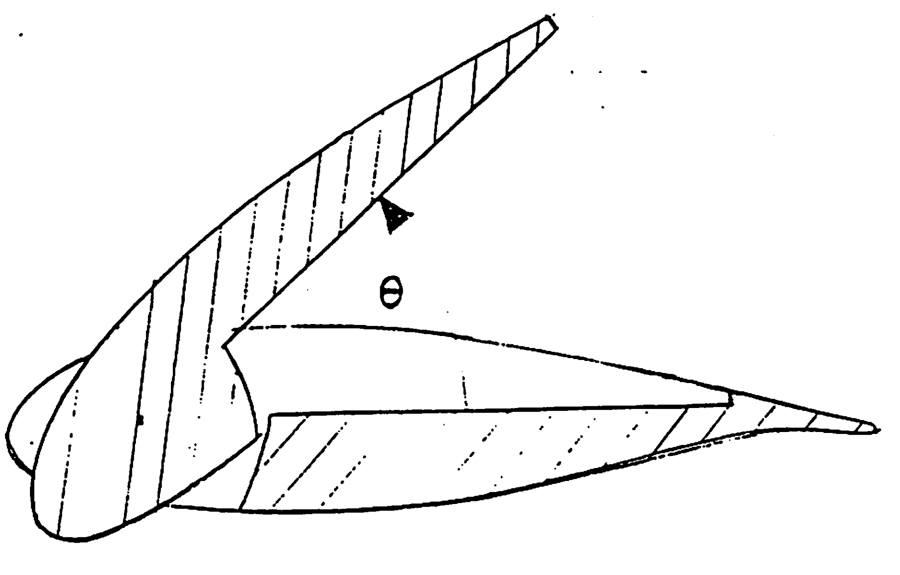
FLEDGE

0.5 - 7 kN



FLEDGE - CONTROL OF SHOCK LOADING





CONCLUSIONS

* Spoilers and ailerons are not very effective.

* Rotating tips are effective but relatively hard to engineer.

* Full span blade pitching requires substantial engineering, and may even be disadvantageous from a control point of view.

* Aerodynamic devices that deal with rotor loads at source can benefit the design of the rest of the machine.

* The SLEDGE brings a new dimension to rotor control - the possible use of mass dynamics via Coriolis forces.

* The FLEDGE offers airbraking with minimum shock loading, and active control with greatly reduced actuating loads compared to conventional full span and partial span systems.

THERE ARE INTERESTING AND SOPHISTICATED CONTROL POSSIBILITIES USING THE HIGHLY RESPONSIVE SLEDGE/FLEDGE SYSTEMS

--- BUT ---

DISCOUNTING THAT, THEY OFFER IMMEDIATE COST REDUCTION ROUTES FOR CONVENTIONAL STALL AND PARTIAL SPAN SYSTEMS.

Wind Turbine Control at WEG

Ervin Bossanyi
Wind Energy Group Ltd, UK

WEG has a broad experience of different control techniques for wind turbines ranging from 200 kW up to 3 MW rating, using both 2-bladed teetered and 3-bladed rotors, synchronous and induction generators, fixed and variable speed, and stall, tip-pitch and full-span pitch-regulated rotors. Experience has also been gained with various forms of compliant drive-trains and with different numerical control algorithms.

It is fair to say that from all these possibilities an optimum wind turbine control concept has not yet conclusively emerged. R & D at WEG is still proceeding along several different lines.

A "potted history" could be written as follows:

Machine	Diameter (m)	Rating (kW)	First Rotation	Comments
MS-1	20	250	1983	Experimental: 2 bladed, hub teetered or fixed, speed variable or fixed, compliant transmission (spring and damper), synchronous generator, tip pitch control
LS-1	60	3000	1987	Experimental: 2 bladed, teetered, compliant transmission using differential gearbox and electrical reaction machine, synchronous generator, tip pitch control
MS-2	25	250	1984	Sold commercially. 3 bladed, full-span pitch control, induction generator. Prototype used to test variants described below.
MS-3	33	300	1988	Sold commercially. 2 bladed teetered rotor, full-span pitch control, two-speed induction generator

All machines are still running at full rating and producing useful data.

The MS-2 prototype at Ifracombe, England has been used to test a number of alternative concepts:

- 3 bladed stall-controlled rotor, emergency braking with parachutes [1]
- advanced pitch control algorithms [2]
- synchronous generator with fluid coupling [3]
- torque-limiting gearbox using hydraulic reaction control [4]

The MS-3 is the current "state-of-the-art" WEG turbine, with the two-bladed teetered rotor emerging as the preferred rotor configuration. This greatly reduces the loads transmitted through the hub to the bearings, pallet, yaw bearing and yaw drive. The MS-3 is significantly more cost-effective than the MS-2, both designs having proved themselves very reliable in operation. The MS-3 concept is currently being used as the basis for a larger 50 metre diameter, 1 MW rated turbine which is nearing completion of the detailed design phase.

However a number of further tests of new variations are currently in progress or planned, for testing on either MS-2, MS-3 or LS-1. These include:

- rigorous design of classical pitch control algorithms
- other pitch control algorithm developments
- torque - limiting generator (an electronic equivalent of the torque - limiting gearbox).
- control algorithms to optimise compliance in a narrow-range variable speed system
- two-bladed, teetered, stall-regulated rotor.

In addition, paper studies on broad range variable speed and yaw control are in progress.

Broad range variable speed needs to be examined very carefully in order to balance a number of relatively small advantages and disadvantage [5]. At the end of the day, the cost of variable speed equipment at present makes the economic case marginal at best, but the position is being kept under review.

Wind turbines require some form of transmission compliance in order to operate stably and generate power into a strong constant-frequency grid. With a stiff transmission and synchronous generator, resonance is likely to occur rather close to the excitation frequency caused by blade passing effects. This problem is alleviated by introducing some form of damping, usually by means of a fluid coupling or by using an induction generator. Both these solutions introduce slip, which implies reduced efficiency. Compliance obtained by a spring and damper system does not suffer the problem of slip energy loss. This could be done with springs and dampers on the gearbox as in MS-1, or in principle by using a synchronous generator with damper windings [6]. However it is difficult to introduce enough damping into a synchronous generator, and mechanical means imply typically of the order of 10^{0} of movement of the low-speed shaft or the gearbox, which is expensive to engineer.

The hydraulically torque-limited gearbox and the electronically torque-limited generator both work on the principle of providing minimal compliance (low slip) below rated wind speed to maximise efficiency, and very high compliance above rated wind speed where efficiency is irrelevant because power has to be discarded anyway (by pitch control). The effect is to allow almost complete suppression of transmission torque peaks above the rated level. The corollary is increased rotor speed, which is controlled by pitching the blades. Because of the high inertia of the rotor acting as a flywheel, it is much easier to control rotor speed than to control rotor torque at fixed speed. In effect, the torque limitation device is relieving the pitch control system of the high-frequency part of its duty, which it cannot normally cope with anyway - hence the need to over-design gearboxes in traditional designs by a considerable margin.

These devices can therefore allow a considerably reduced specification, and hence cost, for the gearbox and the pitch control system. This has to be traded against the capital cost of the torque - limitation equipment, and increased complexity and maintenance. However these devices are likely to be cheaper than full variable speed drives. Although they do not have the same advantage of increased energy capture in low winds as the full variable speed systems, most of this benefit is obtainable anyway by two-speed operation. Nevertheless the economic case for such devices cannot be proven without further experience - hence the current [4] and planned future experimentation with them.

Provided the cost of adequate braking is not too high, a teetered stall-regulated rotor might represent a very simple and elegant solution. The stall-regulated tests on the 3-bladed MS-2 [1] were very successful and indicated a marked reduction in transmission torque loads above rated. However there are still considerable uncertainties surrounding the dynamic behaviour of a stalling teetered rotor, which need to be investigated by experimentation.

As suggested by Leithead [7], a pitch control system with a properly designed response characteristic may be able to control the rotor aerodynamic torque considerably better than is being achieved by simple control algorithms, in which case the problem of having to over-rate gearboxes may be solved very cheaply. A project is currently underway with the University of Strathclyde to examine this possibility using a rigorously designed classical control algorithm.

The earlier adaptive control project [2] investigated the use of some mathematically fairly sophisticated algorithms for pitch control. These were based on a kalman filter used as a state estimator. This means that by making "optimum" use of information available from transducers coupled with knowledge about the dynamics of the system, and the statistical properties of the disturbances, namely wind and transducer signal noise, a "best estimate" of the state of the whole system is obtained. Even if only a power transducer is available, this technique allows simultaneous estimates to be made of unmeasured and even "unmeasurable" quantities such as rotor slip and aerodynamic rotor torque. Any or all of these estimated quantities can be used in a feedback algorithm which then controls the blade pitch.

The feedback element used in the project was initially an "optimal" LQG controller, whose purpose is to calculate at each timestep that control action which achieves the best compromise between a number of objectives during the next timestep. This is done by minimising a cost function which is specified in terms of the predicted system state, and its deviation from some desired state. In principle the system state could include variables representing torques at various points in the drive-train, blade pitch mechanism and tower loadings etc. The LQG optimal controller could then achieve the desired compromise between all these different loadings, the compromise being defined by selecting appropriate weightings in the LQG weighting matrix.

In practice, such weightings are very difficult to define, since they involve the costs of components (which are not independent of each other) and some description of the loadings concerned: minimising peak loadings is easier to define than minimising fatigue loadings. Simple cost functions, for example only trading off gearbox torque against pitch servo action, may actually reduce to feedback laws of simple proportional form. These can be extended to include integral and differential action. The resulting controller may not perform significantly differently from a straightforward PI or PID control algorithm acting on the estimated torque. This is, however, significantly better than a PI(D) controller acting on the raw power signal, which is the normal strategy. Both power variations and servo action can be significantly reduced.

The state estimator and feedback parts of the total control algorithm can in fact be treated independently, and there is a mathematical theorem to the effect that each can be optimised separately, and the whole algorithm will still be optimal. It is clear, then, that while state estimation can bring useful benefits, the feedback law must also be carefully designed. Hence the current project with Strathclyde University to design a control algorithm along classical lines.

Once the full potential of classically designed algorithms has been explored, the value of combining them with state estimators can be assessed. The latter may have a role to play because of the importance of the stochastic nature of the main driving force, the wind, as stochastic effects are dealt with in an explicit fashion. However it would seem that feedback laws based on LQG techniques will only come into their own once it is possible to define cost functions more accurately. This implies the use of slimmed-down structures which rely on the controller to manage the loadings, as an integral part of an advanced design concept.

Multivariable control applications may also present an opportunity for these modern control techniques. Their mathematical formulation is general, in the sense of being directly applicable to multivariable situations, which are not so straightforward with classical techniques. Such applications would principally include variable speed wind turbines in which both the load torque and the aerodynamic torque are being controlled. This applies equally to broad-range variable speed and to narrow-range applications, including such concepts as the torque-limiting gearbox or generator.

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Acknowledgment

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IEA R&D WECS ANNEX XI

WIND TURBINE CONTROL SYSTEM, STRATEGY & PROBLEMS

MEETING 3-4 MAY 1990

CONTROL OF VERTICAL AXIS WIND TURBINES

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EATON COURT, MAYLANDS AVENUE,

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1. CURRENT PRACTICE

The early straight-bladed vertical-axis turbines (e.g. VAWT450, Carmarthen Bay, UK) used a reefing mechanism to move the blades to reduce power in high winds. However current designs employ stall regulation, which results in a fixed H-configuration rotor (fig. 1). With no pitch mechanism and no yaw mechanism, only sequential control of brakes and contactors is required, together with plant supervisory functions.

2. STRATEGY ISSUES

2.1 Stop/start strategies to achieve the optimum trade-off between fatigue damage and energy yield, especially in high winds, are under investigation.

2.2 Similarly at low windspeeds, there is scope for increased energy capture in determining the best strategy for transitions between generating and freewheeling (rotating at below synchronous speed while disconnected from the grid).

2.3 In windfarms, a central controller may be necessary to co-ordinate starting and stopping of turbines to reduce voltage disturbances (flicker).

2.4 For offshore windfarms, for which vertical-axis turbines are well suited, reliability is of prime importance. Duplicate control systems (including duplicate transducers) are under consideration either under a hot or cold standby system, or using majority voting.

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24 May 1990

BLADE PITCH CONTROL OF A SMALL WIND GENERATOR

S. Corsi

SUMMARY

The paper considers wind generator blade pitch control problems and compares the behaviour of two different blade-pitch control system". More precisely the dynamic performances of rotor speed and power-output limitation controls of 50 kW ENEL-FIAT wind generator are shown with reference to the original solution adopting hydromechanical actuators, and the more effective new one which employs electromechanic actuators.

The study and design of the solutions adopted are presented.

Some experimental results on the behaviour of the old control system and the new electromechanic one are also given.

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1. INTRODUCTION

1.1 General Notes on Blade Pitch Control

With regard to grid-connected wind generators provided with continuous variable blade pitch, the control of the blade collective pitch can, in general, be used to pursue several objectives, each of which assumes particular relevance depending upon the type of machine utilized and its application: synchronous or asynchronous generators, electrical and mechanical rating of the unit in relation to the wind characteristics at the installation site, whether the yaw angle be controllable or not, etc..

A first objective might be that of regulating aerogenerator rotor speed during off-load operating conditions (paralleling and load-tripping). This objective takes on particular importance during synchronous generator paralleling.

A second objective, which only concerns synchronous generators, might be a more efficient damping of the unit's electromechanical oscillation relative to the dominant electrical power network. In fact, speed and electrical power oscillation occur because of the continuos torque changes induced by the intrisic wind speed spectrum. Such oscillations can assume amplitudes so high as to cause generators to fall out-of-step or, in any case, to be unacceptable for normal operation.

A third objective might be to aim at the maximum conversion of wind to mechanical power, and thus to the resulting power output. This objective could be of interest in cases in which the aerodynamic profile of the rotor blades involves a non_monotonic torque-versus-blade_pitch characteristic (at a given wind speed).

Nevertheless, it is to be noted that, in the more frequent applications, the mechanical power, for a given wind speed, increases with blade pitch, so that the maximum power conversion corresponds to maximum blade pitch.

The last objective concerns the limitation of power output in high wind speed conditions. This objective, which represents the matter of this paper, assumes great importance, given that design specifications usually optimize aerogenerators for prevailing wind conditions, which are normally in the 10-40 km/h range, though wind speed values can occasionally reach gusts above 100 km/h.

On the other hand, to design an aerogenerator for high wind speeds, appears to be disadvantageous, not only because of low conversion efficiency at normal wind speeds, but also because of the increase both in machine size and in unit costs.

Aerogenerators designed for prevailing wind conditions must be equipped with a suitable power output limiting control system, to avoid possible mechanical strains on the structure and thermal stresses on the electrical generator.

Such a control system could, in principle, be very simply devised by a logic that feathers the blades and trips the electrical generator as the wind speed and/or the power output reaches a pre-set limit. However, optimized exploitation of wind energy is achieved by keeping the aerogenerator running, while limiting the electric power with an automatic regulator that continuously controls the blade pitch.

1.2 The Case Examined

The study and application described here, concerning the blade collective pitch control of wind generators, refer to FIAT-ENEL small aerogenerators installed at the Alta Nurra test field in northern Sardinia, Italy [1].

The field test is located close to the coast line and linked with a 15 kV line of the ENEL distribution network.
The aerogenerators under consideration have the following characteristics:

- 13.5 m diameter, twin-bladed rotor
- 18.5 m from ground, horizontally mounted rotor shaft
- Asynchronous generator, rated 50 kW.

The design specification of these aerogenerators require a power_output-versus-wind_speed characteristic having a first section in which the power output increases up to the rated value with wind speed, and a second section, related to high wind speeds, in which the power output is kept costant.

The first section concerns operation at maximum blade pitch (82^{o} at 70% if the rotor blade radius), while the second section correspons to a progressively decreasing angle of attrack of the rotor blade, and hence requires a suitable blade pitch control system.

As regards the requirements of power output limitation, the original design simply provided for a hydraulic rotor speed regulator consisting of a distributing valve, an hydraulic jack, and a centrifugal tachometer: the tachometer controls the distributing valve, depending on rotor shaft speed, and the valve determines, through the jack, the blade pitch value.

The aforementioned control solution is efficient when regulating generator speed during off-load conditions, while it proves inadequate in limiting power output in a high wind.

In fact, even in the case of an asynchronous generator, which normally works at low slip, rotor speed, when the unit is grid-connected, is virtually determined by the power network frequency and depend only slightly on wind speed. This means that a speed regulator, as a single means of control, is not sufficient to limit aerogenerator power output.

In the absence of a suitable power output limiting regulator the aerogenerators had to be tripped upon reaching a given wind speed threshold, thus not fully exploiting the wind energy available.

A more adequate simple solution to the problem was initially achieved by keeping the original hydromechanical blade pitch control system and by adopting, in addition to the speed regulator, a power output limiting regulator. This operates directly on the basis of the difference between the active power actually delivered and the limiting value required.

Briefly, this solution consists in controlling the distributing valve not only with the cemtrifugal tachometer, but also with a small electrical linear servo-actuator, which, in turn, is controlled by a specially-designed electronic regulator which limits the power delivered. More precisely, the solution provides for the introduction of a power-limiting control loop of integral type, which overlaps the normal speed control loop, which remains active at all times. A definitive solution was given to the problem changing the original hydraulic with a new electromechanic and more effective pitch control system.

In short this solution consists in controlling the blade pitch by a motorized cinematism which completely resplaces the hydraulic original system (see Fig. 1).

The linear servoactuator of the sliding plate position is in turn controlled by a specially-designed electronic device which regulates the rotor speed at no-load working conditions while limits the power-output at high wind speeds.

Use of an integral limiting regulator ensures a nil drop limitation characteristic, and thus operation at constant power output for high wind speed conditions.

2. DESCRIPTION OF THE ORIGINAL BLADE PITCH CONTROL SYSTEM

The original design of the blade collective pitch control system is schematically illustraded in Fig. 1. It consists of an hydromechanical system controlled by a speed governor with a speed feedback provided by a centrifugal tachometer.

The blade pitch is driven by a kinematic linkage that includes a sliding plate on which, in turn, a rotary ring is mounted.

The sliding plate is moved coaxially to the rotor shaft by means of the arm of an hydraulic jack.

The direction of the piston movement depends upon that of the oil flow in the hydraulic circuit. Oil flow direction is, in turn, determined by the position of the valve rod in the 4-way control valve.

The valve rod is connected to one end of a 3-point pivoting lever. The other extremity of the pivoting lever is linked to the sliding plate by means of a rod and cam, which transmit mechanichal feedback from the sliding plate position to the valve rod.

The centrifugal tachometer output rod is connected to the central pivot of the pivoting lever. In normal operation, the oil pressure in the hydraulic circuit is supported by a rotor-driven pump, while a second pump, driven by an electrical motor, operates during start-up. The above control system is a classical hydromechanical speed regulator, in which the equilibrium is reached when the 4-way control-valve rod position is such as to inhibit oil flow.

The valve-rod position is determined by rotor speed through the centrifugal tachometer, and also by rotor blade pitch through the mechanical feedback from the sliding plate.

During aerogenerator star-up, and until the rotor speed reaches a given value n_o, depending on the preloading of the centrifugal tachometer spring, the valve rod is on its upper limit stop and provides oil to push the rotor blades to their max pitch (maximum lift at 82^o).

As rotor speed n (rpm) increases, the centrifugal tachometer progressively draws the valve rod down until it reaches the position at which the valve closes the two oil inlets feeding the hydraulic jack.

Starting from this situation, which correponds to rotor speed n_o, the speed control loop comes out of saturation and becomes active. The rotor speed n_o chosen is close to the grid synchronism value.

The mechanical feedback from the sliding plate (and therefore fron the rotor blade pitch) is necessary to ensure stability of the speed control loop during aerogenerator off-load running.

In fact, in this situation, without the mechanical feedback, the control loop would comprise two cascaded integrators, one corresponding to the dynamic relationship between mechanical torque and machine speed, and the other correponding to the relationship between the position of the 4-way control-valve rod and that of the jack piston. This mechanical feedback, together with the elasticity of centrifugal tachometer spring determines governor drop: that is, the slope of the rotor_speed-versus-blade_pitch characteristic.

The progressive decrease in blade pitch caused by rotor speed increase continues until the rod of the mechanical feedback linkage reaches its lower limit stop (corresponding to a blade pitch of 45^{o}). When the limit stop is reached, the mechanical feedback opens and further velocity increases cause the 4-way control-valve rod to reach its lower limit stop, at which the hydraulic piston pushes the blades into their fully-feathered position.

2.1 Steady-state Characteristics

The aerogenerator steady-state characteristics: active_power-versus-wind_speed, at different grid frequencies, are shown in Fig. 2. From these curves, it becomes apparent that the power output at high wind speeds is far greater than the rated value P_N with consequent heavy mechanical and thermal stresses. This behaviour of the wind generator considered was also confirmed by operation on the ground.

Furthmore, it should be noted that, at high wind speeds, the working point is strongly affected by grid frequency variations, while, at prevaling wind speeds, and for grid frequencies higher than the rated value, the power output levels are very low.

This power reduction occurs because, at high grid frequencies (and thus at high rotor speeds), the speed governor reduces the blade pitch notwithstanding low power levels.

To sum up, Fig. 2 clearly demonstrates that a speed regulator alone is not able to confine the power output supplied to the rated value. Therefore it proves the need to install a special automatic blade-pitch regulator to limit power output at high wind speeds, irrespective of other relevant quantities, such as grid frequency and voltage variations. Such an additional limiting regulator would provide the desired power_output-versus-wind_speed characteristic shown in Fig. 2 (see the the thick line).

3. AN ADDITIONAL POWER OUTPUT LIMITING REGULATOR

3.1 Functional Description

The need to limit the power output at high wind speeds, in the various operating conditions (changes in grid frequency and voltage, etc.), made it necessary, as mentioned previously, to modify the original blade pitch control system. The modification involves an additional power output limiting regulator that, when a pre-set power limit is reached, controls blade collective pitch as a function of the difference between the actual power delivered and the set limiting value.

More precisely, action by the additional regulator closes a power output control loop overlapping the original rotor speed control loop, which is kept active to control rotor speed during generator-paralleling and load-tripping.

Of the possible solutions for such a regulating system, that chosen is relatively simple, cheap, does not require structural modification to the unit, and provides acceptable performances.

As mentioned in section 1.2, the solution chosen provides for control of the hydraulic 4-way control valve, not only by means of the centrifugal techometer, but also through a small electrical linear servo-actuator that replaces the mechanical feedback rod.

Fig. 3 illustrates the modification introduced into the original control system shown in Fig.1.

The servo-actuator is, in turn, controlled by an electronic regulator that limits power output.

The servo-actuator stroke is sufficiently large to control the 4-way control-valve, irrespective of the position of the centrifugal techometer rod. This also allows the power-limiting control loop to conpensate for the action of the rotor speed control loop, which always remains active.

When power output is below the limiting value, the electronic regulator forces the linear servo-acquator to its bottom limit stop, in which position the servo-actuator is fully withdrawn and its length is practically equal to the rod it replaces. In this operating state the power-limiting control loop is not operative, and blade pitch control is totally taken over by the rotor speed control loop. Nevertheless, the speed control loop is, in turn, not operative until rotor speed rises above the threshold value determined by the spring preloading of the centrifugal tachometer. In practice, for speeds below the threshold value, the centrifugal tachometer forces the rotor blades to their max lift position (blade pitch at 82^{o}).

A suitable setting of the spring pre-loading (that is, with the speed threshold equal to rotor operating speed at max grid frequency and at rated power output) allows the speed governor to be operative in off-load operating conditions only. This setting criterion ensures, when the generator is working connected to the grid, the full exploitation of the wind's energy until the power output limit is reached, thus preventig uncalled-for blade pitch reductions due to the speed governor.

3.2 Implementation of the Limiting-Regulator and Experimental Results

Theoretical research was followed by electronic design and construction of the limiting regulator prototype making provision for heavy environmental working conditions (salinity, rain, etc.).

Initially, the prototype was checked on a rotating machine at the manufacturer's test room. During such tests, the limiting power-output-control loop was closed, using a special real-time simulator emulating the wind energy conversion process. Fig. 4 shows the transient response of the power-output-limiting-control loop, following simulated step variations of wind speed.

The prototype was then installed on N. 2 unit of the Alta Nurra power plant (see Fig. 5). Experimental operation over a period of about one year has shown the effectiveness of the solution, and field test have confirmed the theoretical evaluations and simulation results.

Fig. 6 compares, during field operation at relatively high wind speeds, the behaviour of Alta Nurra N. 2 and N. 6 wind generators, equipped with the additional power output limiting regulator, and with the original speed governor alone, respectively. It was found that, during the periods when wind speed exceeds the value (about 12 m/s) corresponding to aerogenerator rated power output, while the average value of the power delivered by N. 2 unit is kept to the limitation value P_N through blade pitch control, the power output of N. 6 unit is appreciably in excess of the rated value P_N.

With reference to the N. 2 unit behaviour during power output limiting operation, it should be mentioned that the fast power-output variations around the limitation value P_N are mainly due to the dynamic performaces of the blade-pitch-hydro-machanic-control system. Nor can this performance, in practice, be improved by acting on the limiting regulator transfer function. Nevertheless, better performaces of the power-output-limiting control loop could be achieved by modifying the hydraulic circuit dynamic characteristics.

In principle, better overall performances could have been achieved by making radical changes in the original blade pitch control system, by replacing, for example, the original hydromechanic system by a more up-to-date electromechanic system.

4. A NEW ELECTROMECHANIC BLADE PITCH CONTROL SYSTEM

4.1 Functional Description

The original hydraulic blade pitch control system has been replaced by a new very effective electromechanic one.

It consists of a high performance motorized linear actuator of the sliding plate position controlled by suitable electronic regulators.

The actuator is obtained by combining together a brushless-high-dynamic-motor with a ball screw and gears of high mechanical accuracy.

The system safety is warranted being the cinematism provided with a high reversibility so that the returning action of two springs acting on the sliding plate are sufficient, when the motor is not feeded, to push the blades into their fully-fea_ thered position.

Through this very fast and accurated actuator it is possible to design a control system having the desired dynamic performance.

Fig. 7 shows the block diagram of the blade-collective pitch control system which consist of three overlapped control loops.

The more internal of them regulates the blade pitch angle (Φ) at the reference value (Φ_{REF}) which is provided by the higher-level regulator.

The dominant time constant of the blade pitch control loop is less than 0.2 s.

The rotor speed regulator, which has a control law of integral type, defines the pitch-angle reference Φ_{REF} on the base of the difference between the rotor speed reference (Ω_{REF}) and measure (Ω). The dominant time constant of the rotor speed control loop is of about 0.3 s.

The rotor speed reference signal (Ω_{REF}) is obtained measuring the network frequency so allowing, during off-load operating condition, the tracking of the network in a very narrow way, and then a soft paralleling.

The more external power output control loop becomes operative when the active power measure (Pe) exceeds the reference value (Pe_{REF}) which establish the maximum limit admitted.

In fact the power limiting regulator, characterized by a control law of integral type, in designed to modify the rotor speed reference with negative values only (the positive values are saturated to zero). Hence during off-load working condition the power output limiting control loop is open while works the rotor speed regulator only. After paralleling the operative rotor speed reference is slightly increased in comparison with the network frequency, so that it can assume lower values exclusively when the power output exceeds the set limiting value.

The dominant time costant of the power output limiting control loop is of about 0.25 s.

This advanced dynamic performance of the control system is a sufficient guarantee (taking into account the wind-power-spectrum of Alta Nurra site) for considerably reducing the power-output variations around the limit value at high wind speed.

The modified aerogenerator main working states are: Standstill, Loadless; On-Load. Besides these continuons working states, the other possible are transient states through which the system pass after a changing state request or following protection interventions which command the emergency stop.

Referring to the influence of the non-linear relation between pitch angle, wind speed and mechanical torque on the main control loop dynamics, and for optimizing the performance of the power_output limiting control loop, has been applied a control law of adaptive type.

4.2 Implementation of the New Electromechanical Blade Pitch Control System and Experimental Results.

A complete design and construction of the mechanical, electronic and electrical parts of the new control system has been made for unit N. 5 (see Fig. 8).

About the motorized actuators substituting the previous hydromechanical four: blade collective-pitch actuator; blade cicling-pitch actuator, rotor brake, and nacelle brake, particolar attention was paid for obtaining very precise and completely reversable cinematisms.

The precision is mainly required to obtain a continuous linear control of the blade-pitch, so evoiding lags after commands and their negative effects on the dynamics.

The reversibility of the cinematisms is very important to warrant the safety of the aerogenerator in the face of possible foult of the a.c. power supplying electronic devices, or control system failures/malfunctions.

The control system is comprehensive of: cinematism position regulators, rotor speed and power output regulators, control logics for protections, consistency tests on commands and state transitions, and checking of the electronic apparatus failures.

The performance of the modified aerogenerator was completely tested in laboratory by reclosing the various control loops through a special real-time simulator emulating the wind generator energy conversion process.

The control loop cut-off frequency was chosen reaching a compromise between the dynamic performance of the control loop and the filtering of 50 and 100 Hz noise.

The completely modified generator was installed at the Alta Nurra test plant, and the first results showed the effectiveness of the solution and the validity of the theoretichal analysis.

Fig. 9 puts in evidence particularities of the modified aerogenerators.

Fig. 10 refers to field tests behaviour of the generator after the implementation of the electromechanic control system.

The performance of the new control solution is very impressive when compared with the previous one either at no-load or during power-output limitation (see Fig. 6) .

The fluttuation of the power-output around the set limiting value and the consequent fatigue sollecitation are now inappreciable.

5. CONCLUSIONS

Wind generator mechanical and thermal stresses can occur at high wind speeds, unless power output is limited to the maximum acceptable value. This limitation can be obtained by controlling blade pitch.

In the case of a grid-connected wind generator, if the blade pitch control system is only sensitive to rotor shaft speed, then it will generally be inadequate for obtaining a power_output-versus-wind_speed characteristic with power-limitation at high wind speeds.

With reference to the Alta Nurra demonstration wind power plant , the initial design of ten 50 kW ENEL-FIAT wind generator simply provided a hydro-mechanical speed governor.

In order to prevent generators from operating at more than their rated power values the units had to be tripped when the wind speed reached a preset value, wich resulted in a substantial reduction in wind energy conversion.

To solve the above-mentioned problem, a preliminary solution was devised and developed that did not require structural modifications to the units. This solution consists in operating the 4-way control valve of the blade-pitch hydraulic control system, not just through the centrifugal tachometer, but also by means of a small linear electric servo-actuator controlled by an electronic regulator that limits the power delivered. In other words, the control solution consists in overlapping a power-output-limiting control loop on the normal rotor speed control loop, which always remains active. A nil drop limitation characteristic is obtained by applying an integral-type control law.

After theoretical research, a limiting regulator prototype was installed on N: 2 unit of the Alta Nurra power plant.

Subsequent experimental operation over a period of about one year showed the effectiveness of the solution, and field test results confirmed the expected performace of the device.

To improve the dynamic performace of the control system a completely new electromechanic actuator of the blade-pitch, and new electronic apparatus for regulating, controlling and protecting the wind generator were designed and developped.

The high mechanical-accuracy actuator is also reversible so

warranting the system safety in case of emergency stop.
Through this fast and precise actuator and the accurate design of blade-pitch, rotor-speed and power-output regulators, the desidered dynamic performace for every working conditions has been obtained. In particular the power-output variations around the set limiting value at high wind speed have been drastically reduced.

Laboratory tests and experimental operation have showed the high effectiveness of the electromechanic control solution, and field test results confirmed the expected improvements in the performance of the control system.

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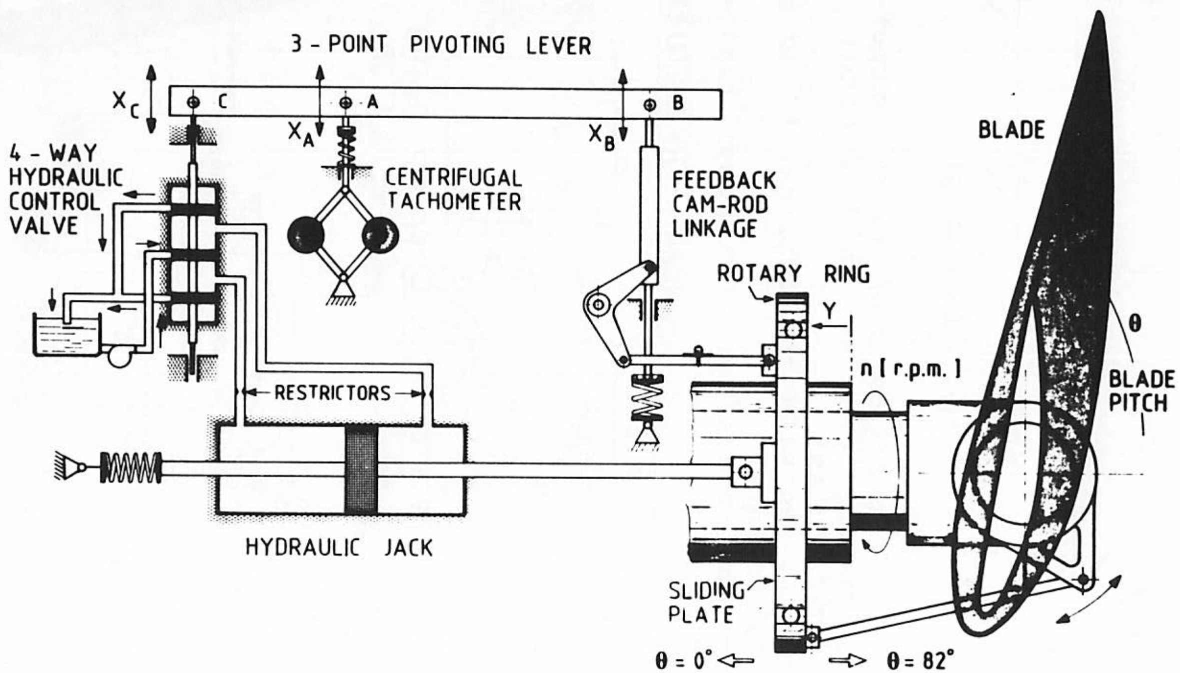


Fig. 1 Schematic diagram of the rotor blade pitch control system according to the original design (rotor speed governor only)

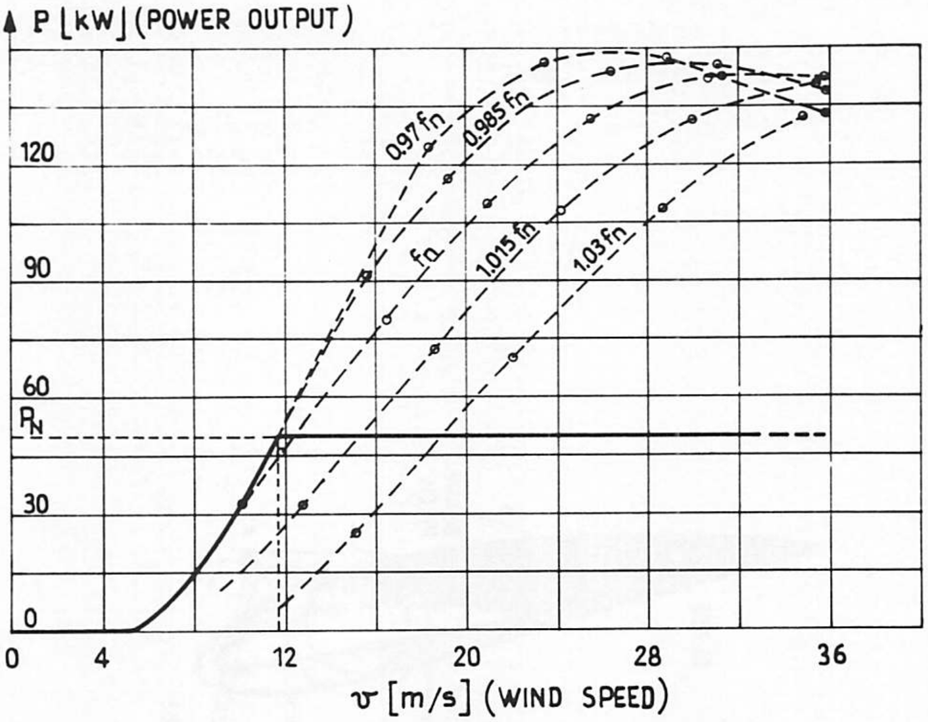


Fig. 2 ENEL-FIAT 50 kW aerogenerator power output-versus-wind-speed characteristics, at grid nominal voltage and different frequencies:
--- characteristics with the original blade pitch control (speed governor only)
___ desired characteristic with power limitation at the rated value P_N.

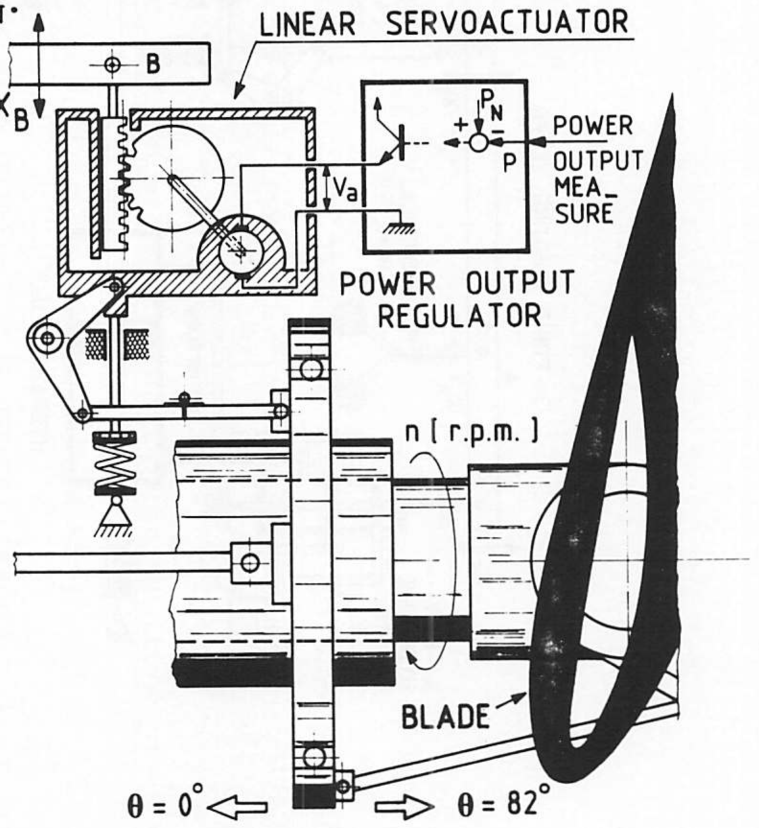


Fig. 3 Schematic diagram of the modified rotor plade pitch control system, replacing the feedback rod by a linear servoactuator controlled through an electronic power-output limiting regulator.

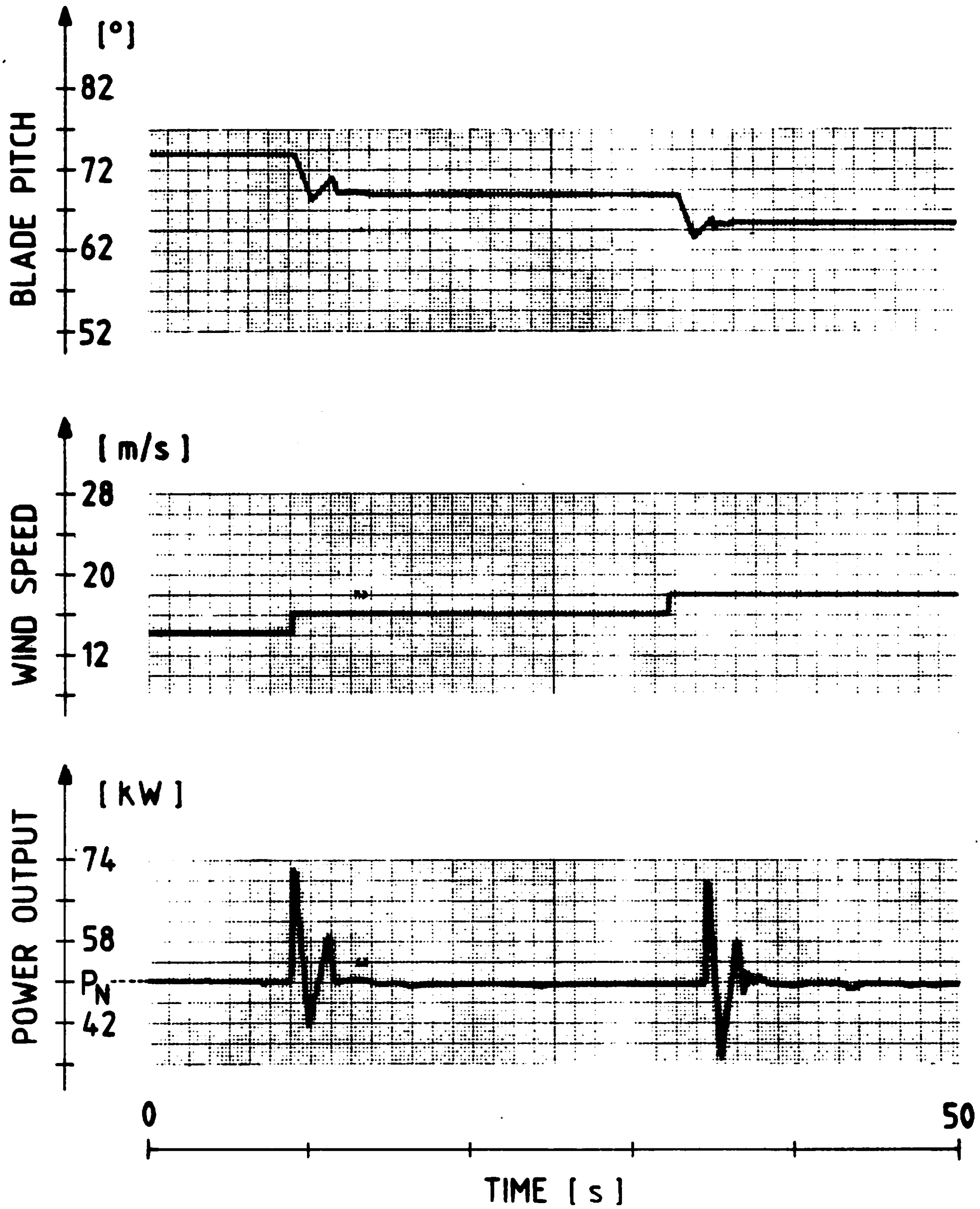
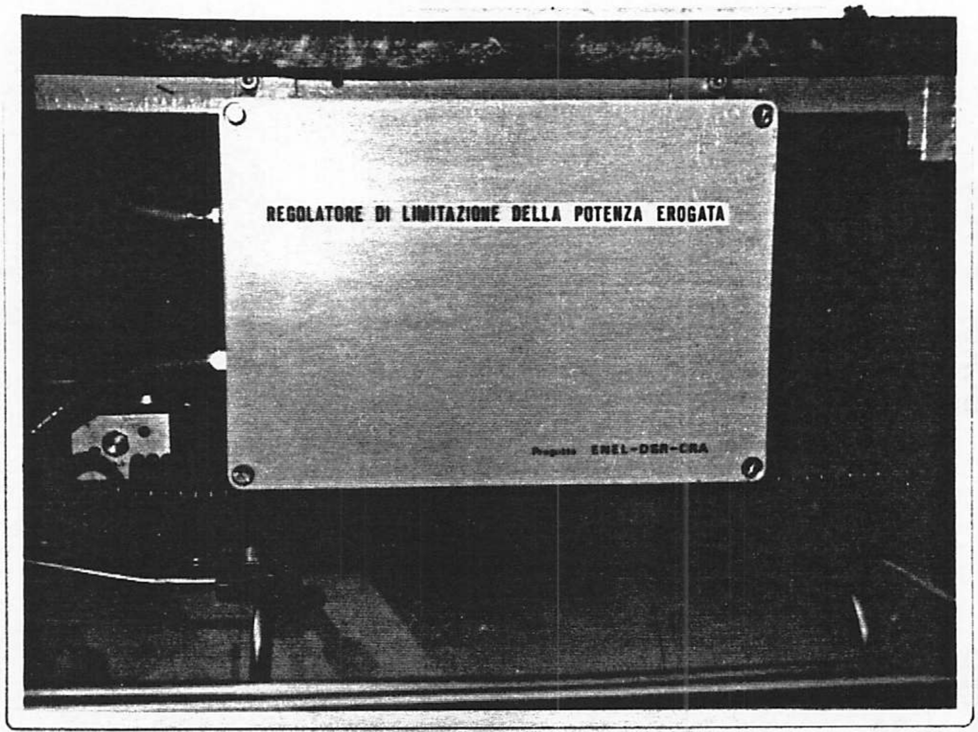


Fig. 4 Reponse transients of the power-output-limiting control loop following stepped wind speed perturbations. Simulation results.



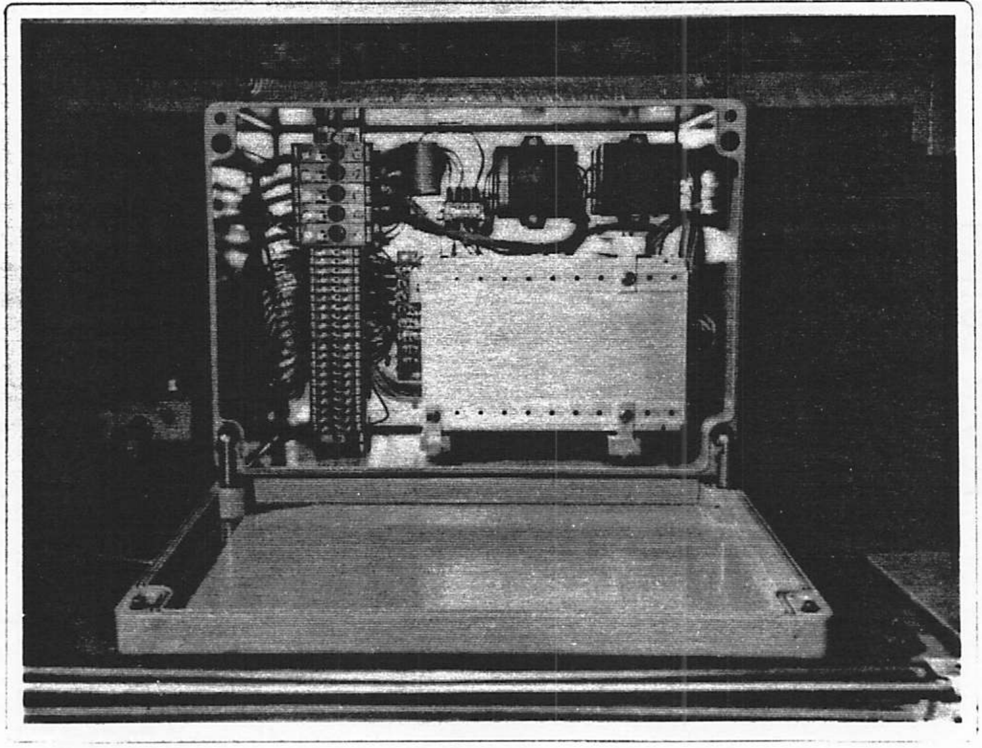


Fig. 5 Installation of the prototype limiting regulator inside the nacelle of the N. 2 Alta Nurra unit.

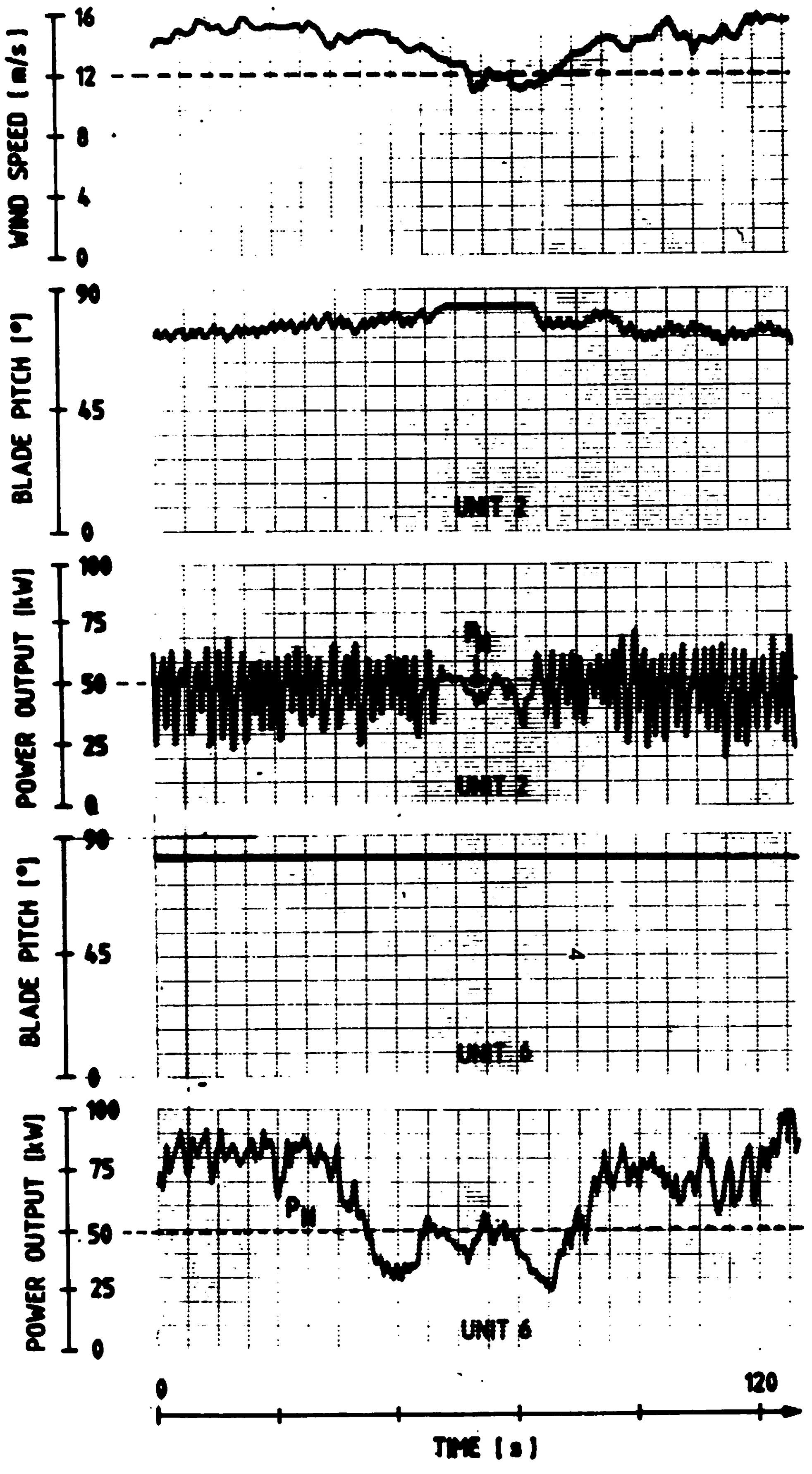


Fig. 6 Alta Nurra power plant on 6th August, 1985. Comparison between unit N. 2, equipped with power output limiting regulator, and unit N. 6, equipped with rotor speed

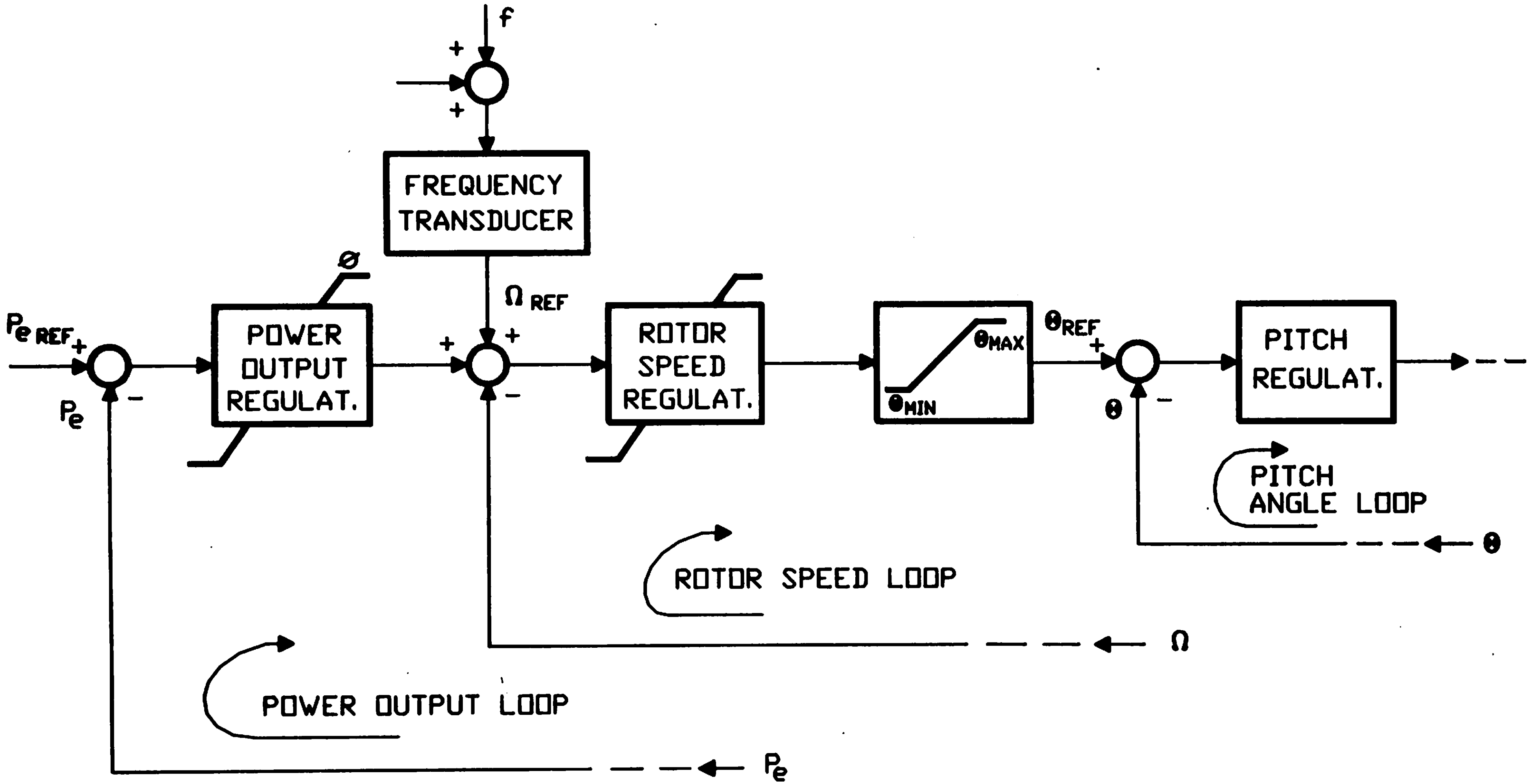
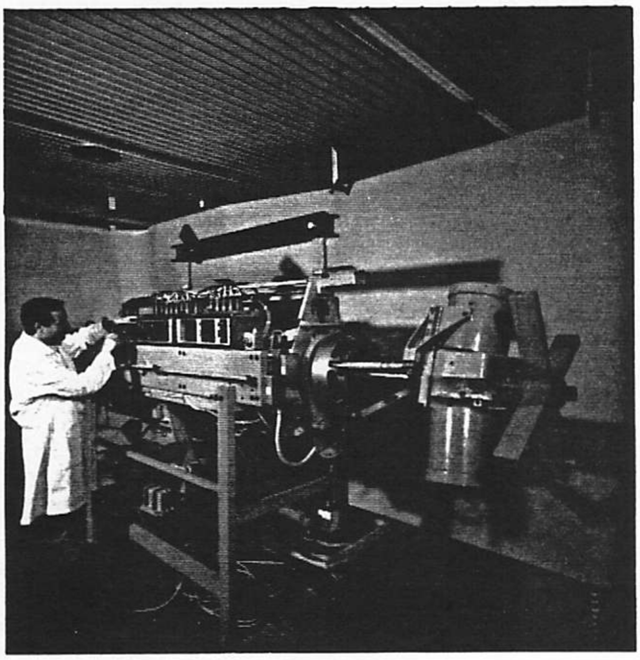


Fig. 7   Block diagram of the blade-collective-pitch control system.



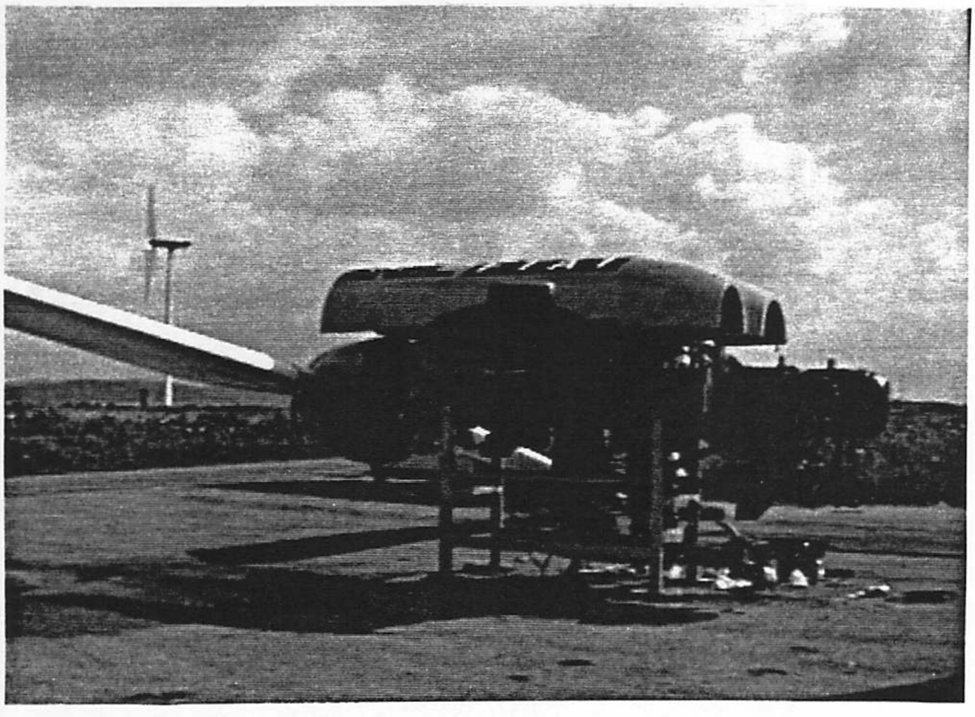
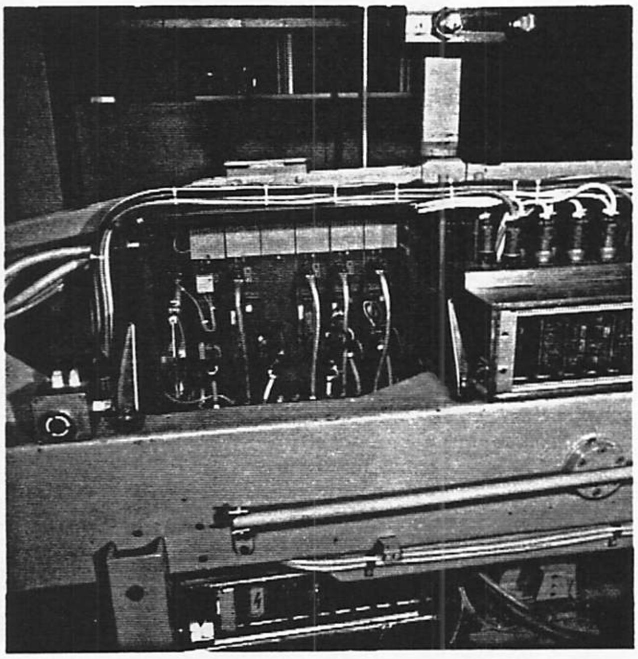


Fig. 8 Overall view of the new electromechanic blade-pitch control system



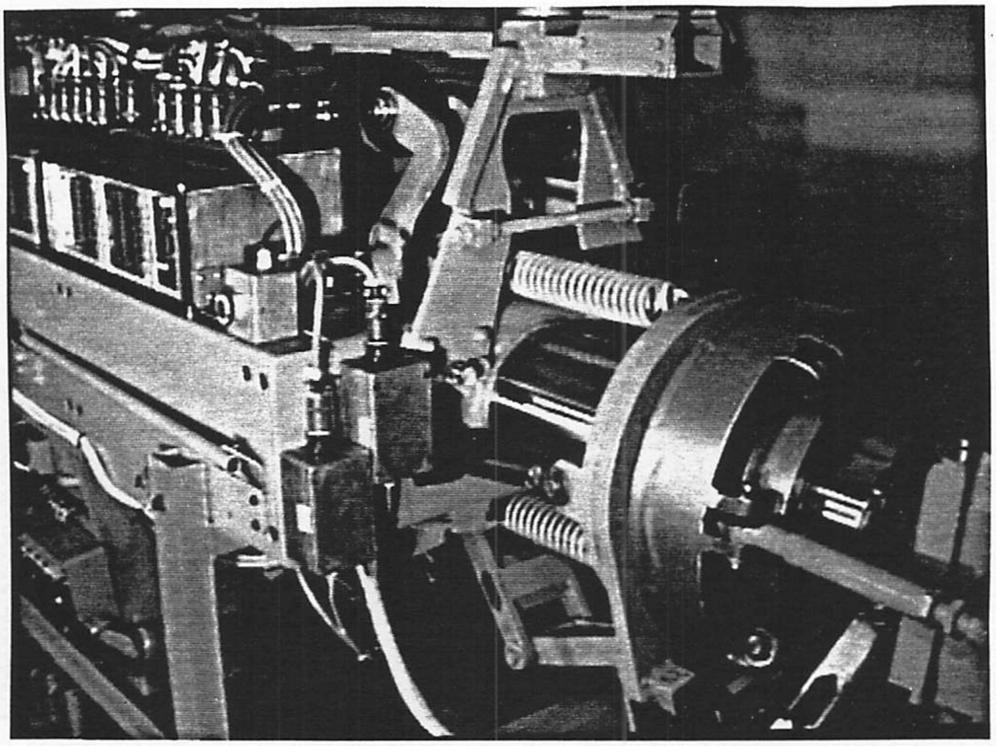


Fig. 9 Particularities of the new electromechanic blade-pitch control system.

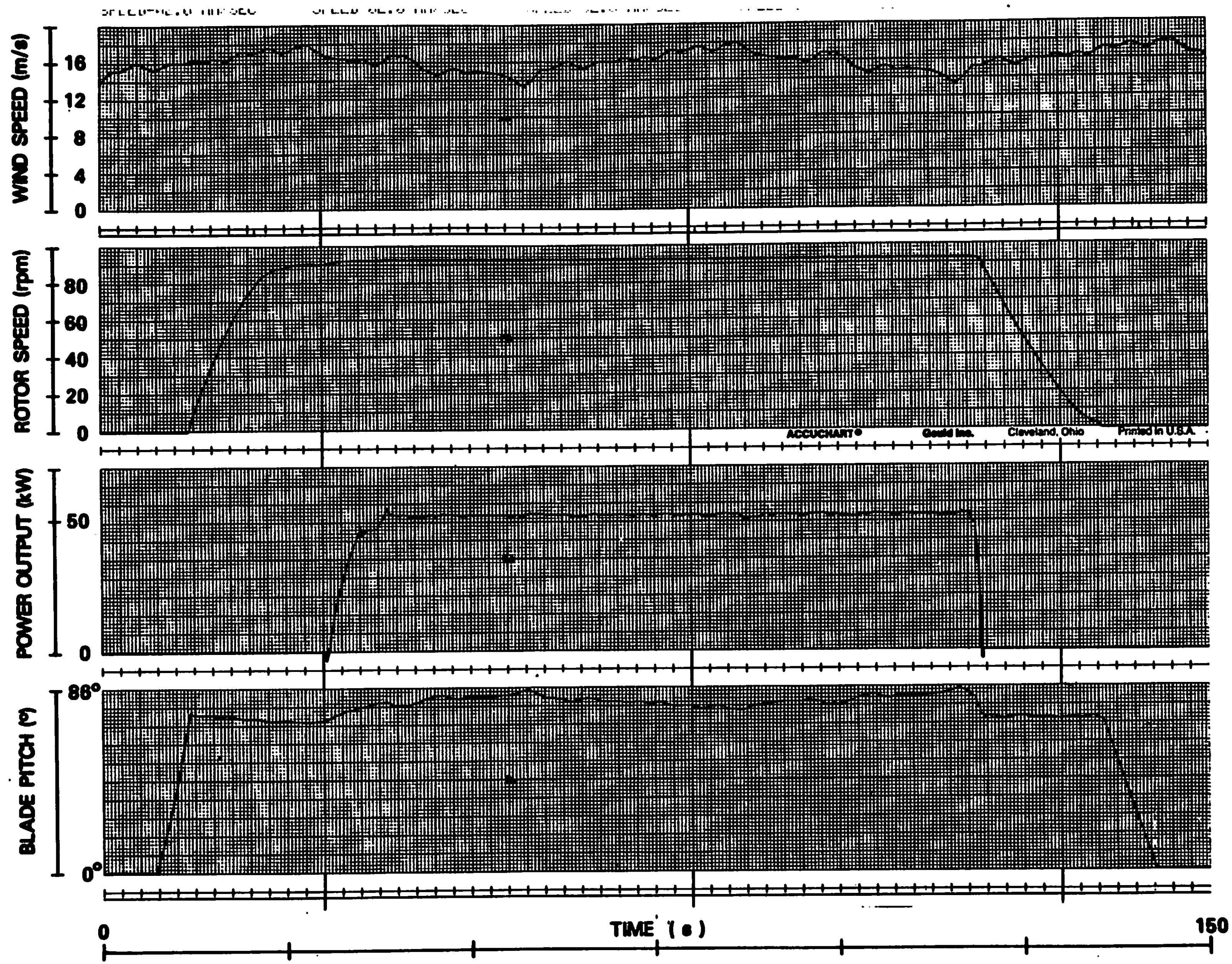


Fig. 10 Starting - Power Output Limiting and Stopping of unit N. 5 equipped with the new electromechanic blade-pitch control system.

Control strategies for single-bladed W.T.G.

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Introduction

Riva Calzoni experience in wind energy conversion has been developing in the single - bladed WTGs since about ten years. This philosophy of construction, which can seem more problematic for the unbalanced loads and for a little loss in aerodynamic efficiency, is able to face several WTG problems, in particular costs effectiveness.

After a general description of the two WECSs M7 and M30, developed by RC with the partecipation of E.N.E.A., this report describes RC control philosophies for different size WTGs. General design considerations (evaluated by simulations results), to prevent structural fatigue loads and to evaluate the influence of the blade-pitch control, are also presented.

M7 General Description

The M7 is a small single-bladed converter with a rotor diameter of 6.5m and a rated power of 5.2kW (fig.1); it is a variable speed system designed for isolated sites. It can be used for water pumping or to support stand-alone electrical systems with a battery set (fig.2).

The passive yawing is operated by two bearings between the nacelle support and the tower. Its intrinsic stability is guaranteed by a convenient displacement between the yaw axis and the leeward rotor plane.

The typical influence of dynamic loads on the single-bladed WTGs has required some particular design solutions and deeper previous studies. Therefore M7 is based on a high flexibility and low weight structure including a blade with a free flap angle and a nacelle softly connected to the tower by means of a hinge junction, which is able to decouple the nodding and partly the rolling. The structural design took into account the eigenfrequencies of the single parts. A simulation code was developed, in order to evaluate the behaviour of the WTG structure, regarded as an 8 d.o.f. system, with a pitch position considered undisturbed by the dynamic loads [1]. The most important parameters of the full system in working conditions, can be optimized by simulating different load cases.

M7 Control System

The M7 pitch-control (fig. 3) is performed by means of a system completely hydro-mechanical fed by a pump directly mounted on the high speed shaft of the gearbox. The differential piston devoted to actuate the pitch position is assembled inside the rotating structure (on the blade axis) and is fed through a rotary connection and tubes inside the low speed shaft. A nut screw couples the translation of the piston with a rotation and drives the blade around the pitch axis. For safety reasons, in low pressure condition, two preloaded springs keep the piston in a position corresponding to a pitch angle of 45 degrees (fig.4)

In working condition the pump creates a flow rate proportional to the rotational speed of the rotor; in the hydraulic control panel, when the speed is over the nominal value, the pitch positioning piston is fed by the overflow rate. In this way the speed actuation is proportional to the rotation overspeed. After some theoretical evaluations in the frequency domain (fig.5), the hydraulic panel was modified to obtain a different relation between actuation and rotation overspeed. This improvement also simplified the hydraulic diagram (only one tube inside the low speed shaft) with consequent costs reduction.

After the simulation of the transient and steady-state operation of the WTG, the effects of structural stiffnesses, dampings and control system parameters were analysed.

M7 Simulation

The control system parameters adjustment by mean of computing the time response to the more critical wind speed variations, is very important, since the control panel is not easily adjustable on site. The parameters, in fact, are strictly related to the design of the actuation mechanical system. Regarding the system response to typical wind variations, two different simulation around very general values were been investigated, in order to ensure a control for every sites conditions. These simulations are:

- The system response to a signal representing the annual wind gust (starting from the nominal windspeed, gust factor = 2, acceleration = 1g) to evaluate the extreme operating condition for the WTG.

- The system response (fig.6) to a stochastic signal with mean value, standard deviation and spectrum of the wind speed[2], to evaluate stability problems with regard to the excitation frequencies of the system (eigenfrequencies).

The first simulation permits to evaluate:
- the max pitch speed actuation, as a result of an optimization relevant to the limit of the centrifugal forces on the rotor;
- the power generator output;
- the axial thrust on the rotor (with flap loads consequence).
These values are related with structural peak loads and with thermal shocks on the generator.

The second simulation is relevant to a stability investigation and is based on more realistic wind input generated with a stochastic approach[3]. This investigation involves the problems of power smoothing quality and structural stability.

M30 General Description

The M30 is a medium size single-bladed converter with a rotor diameter of 33 m and a rated power of 200 kW (fig.7). Two prototypes were manufactured: a double fixed speed system designed for a direct grid connection; a variable speed system with double static conversion. This WTG can be installed in clusters to realize grid-connected or island-connected wind farm. M30 has been produced in partnership with Messerschmidt Bölkow Blohm.

The double speed (1000 r.p.m.; 1500 r.p.m.) is made possible by using two high slip induction generators (6 poles, 2.2% slip - 4 poles, 3.13% slip). The active yawing of the WTG is actuated by means of two hydraulic motors. Its control is one of the functions performed by the Operation Control Computer. The connection of the two generators to the grid is automatically performed by an autonomous sinchronous controller. It is able to optimize the operation efficiency by a voltage reduction in partial load operation due to an appropriate thyristors control.

The pitch regulation is performed to control the rotor speed during transient operations, and the power output, during grid connected operations (see fig. 8-9). The actuation system includes a piston, hydraulically driven, that, moving a shaft inside the low speed shaft, operates the pitch leverage. The M30 structure includes a soft tower stiffly connected (regarding nod and roll) to the nacelle. The blade has a free flapping angle of 31 degrees with two damped limit stops.

M30 Control System

The control system of the WTG is performed by three different functional subsystems: an electro-mechanical one, devoted to safety functions, where a logic circuit relais some safe critical devices, performs an emergency shutdown or requires emergency functions from the computer; the A.S. Controller devoted only to the soft grid-connection; the Operation Control Computer (OCC) that has several tasks like supervision, faults monitoring, logic control and digital regulation. The OCC functions are displayed in the diagram of fig. 10.

The regulation functions are organized in a multi-loop structure, where two digital P.I.D. algorithms alternatively drive the picth position for the rotor speed or the power output control; a rough parameter adaptation is possible with a parameter set dependent on the pitch actual value. The actuation of the pitch is then operated by means of an analog servo loop. The supervisory functions consists of yaw monitoring and logic control.

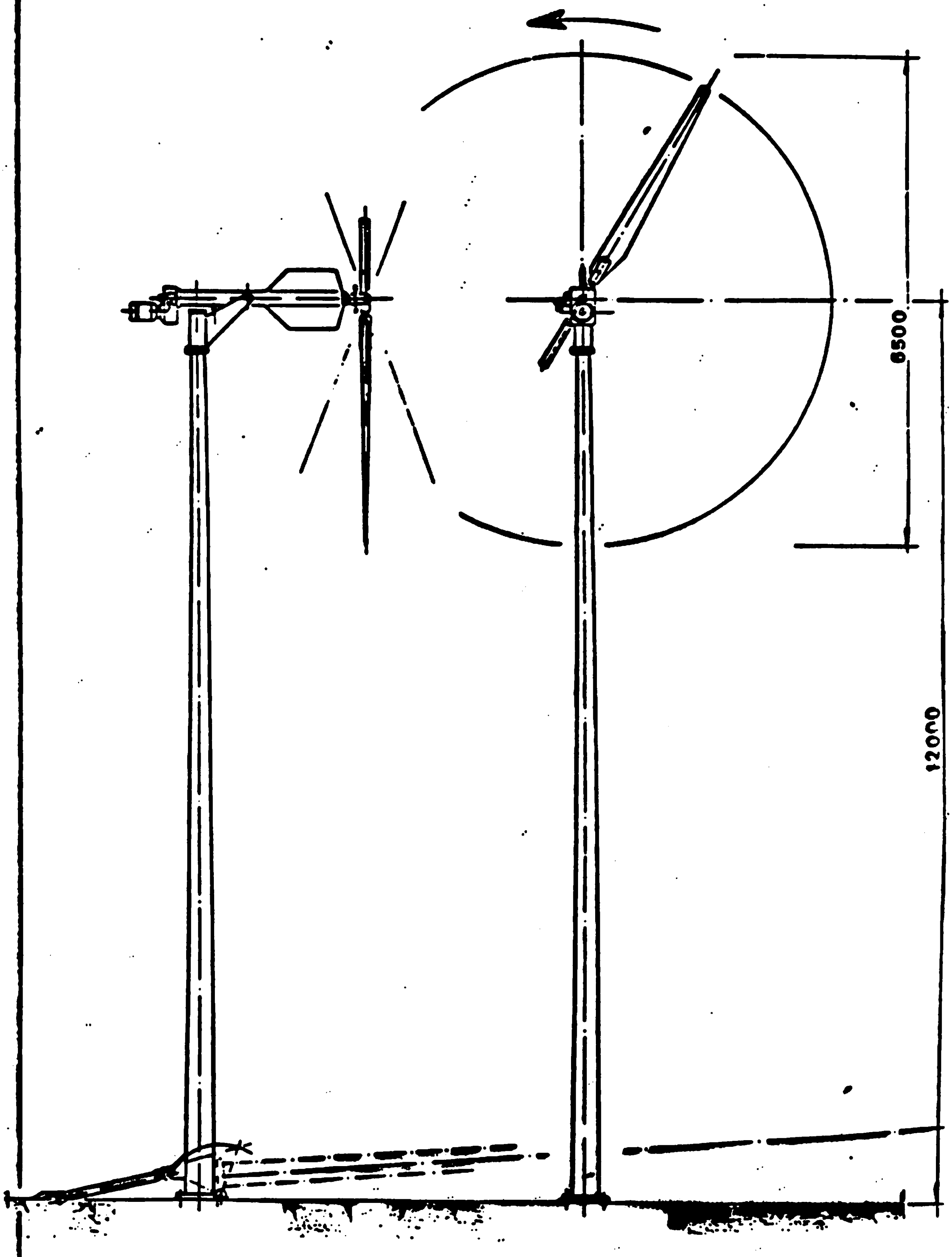
generators switch operations and cut-off, cut-in conditions monitoring. Also the power and the speed set values are controlled by the supervision. The parameters involved in the control functions can be on-line tuned during the commissioning for a site adaptation.

M30 Simulation

After the evaluation of the power output (with approximate estimate of the aerodynamic and mechanical efficiency) versus windspeed and pitch angle, a dynamic simulation (only 3 d.o.f.) has been developed to find out the control system influence on the power output.
The aerodynamical losses were evaluated with the simplifying introduction of additional aerodynamical efficiencies: yaw and tilt misalignment with respect to the wind direction, flap angle position evaluated in rated conditions and counterweight energy losses were considered. The stochastic wind speed was generated in the same way as for M7 simulation.
This simulation is able to check the P.I.D. regulation parameters, with a theoretical optimization of the power output with regard to its stability (fig.11). Because of a strong limitation coming from the maximum allowed pitch speed (due to structural reasons) the optimization of the system behaviour is rather difficult. The power fluctuations are anyway acceptable and don't lead to any oversizing of the generator and drive train components.

RIVA CALZONI s.p.a.
Bologna-Italy

FIG. 1 - M7 WIND TURBINE GENERATOR



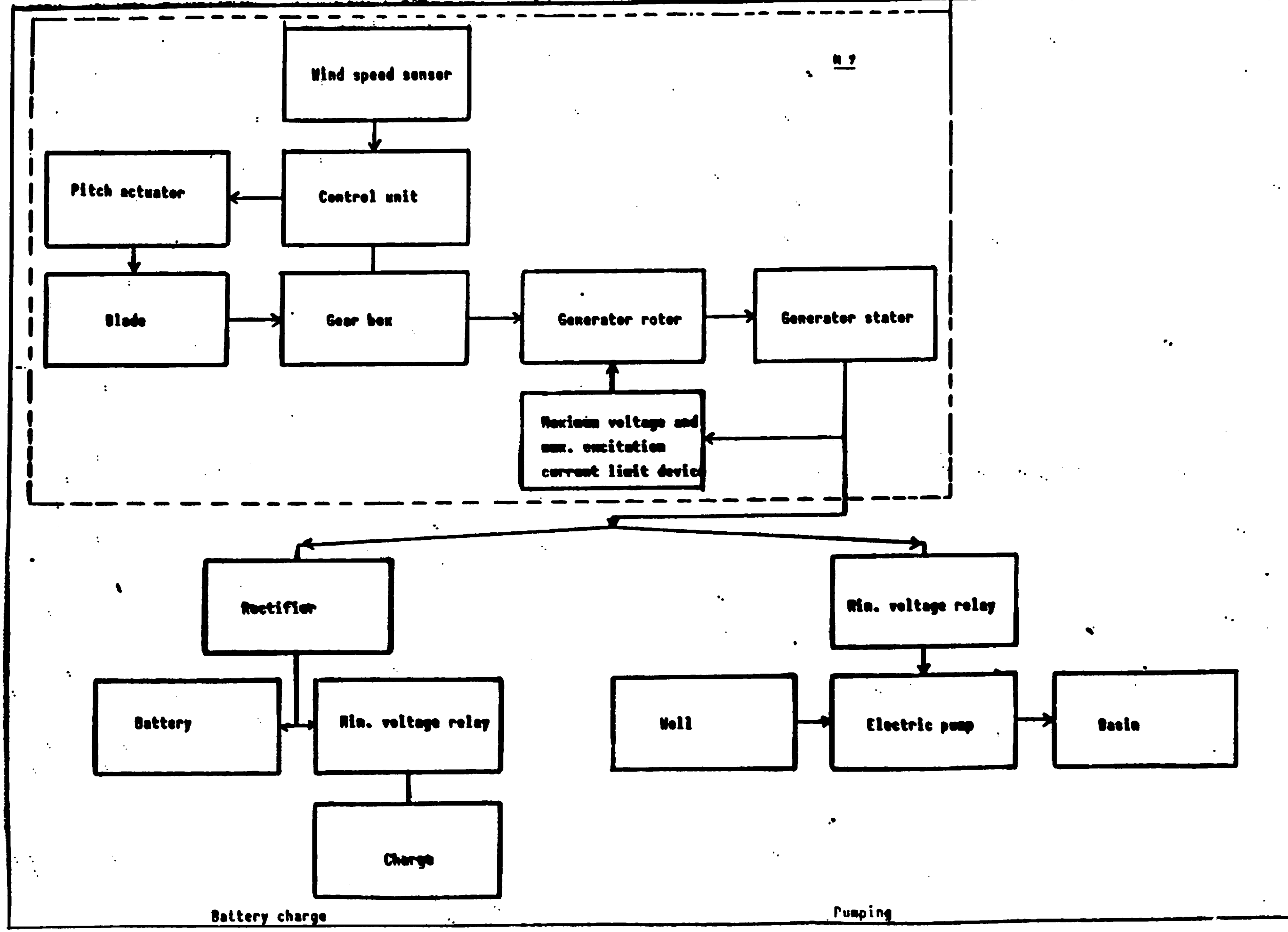


FIG. 2 - DIAGRAM SHOWING M7 OPERATION WITH THE TWO POSSIBLE OUTPUT COUPLINGS

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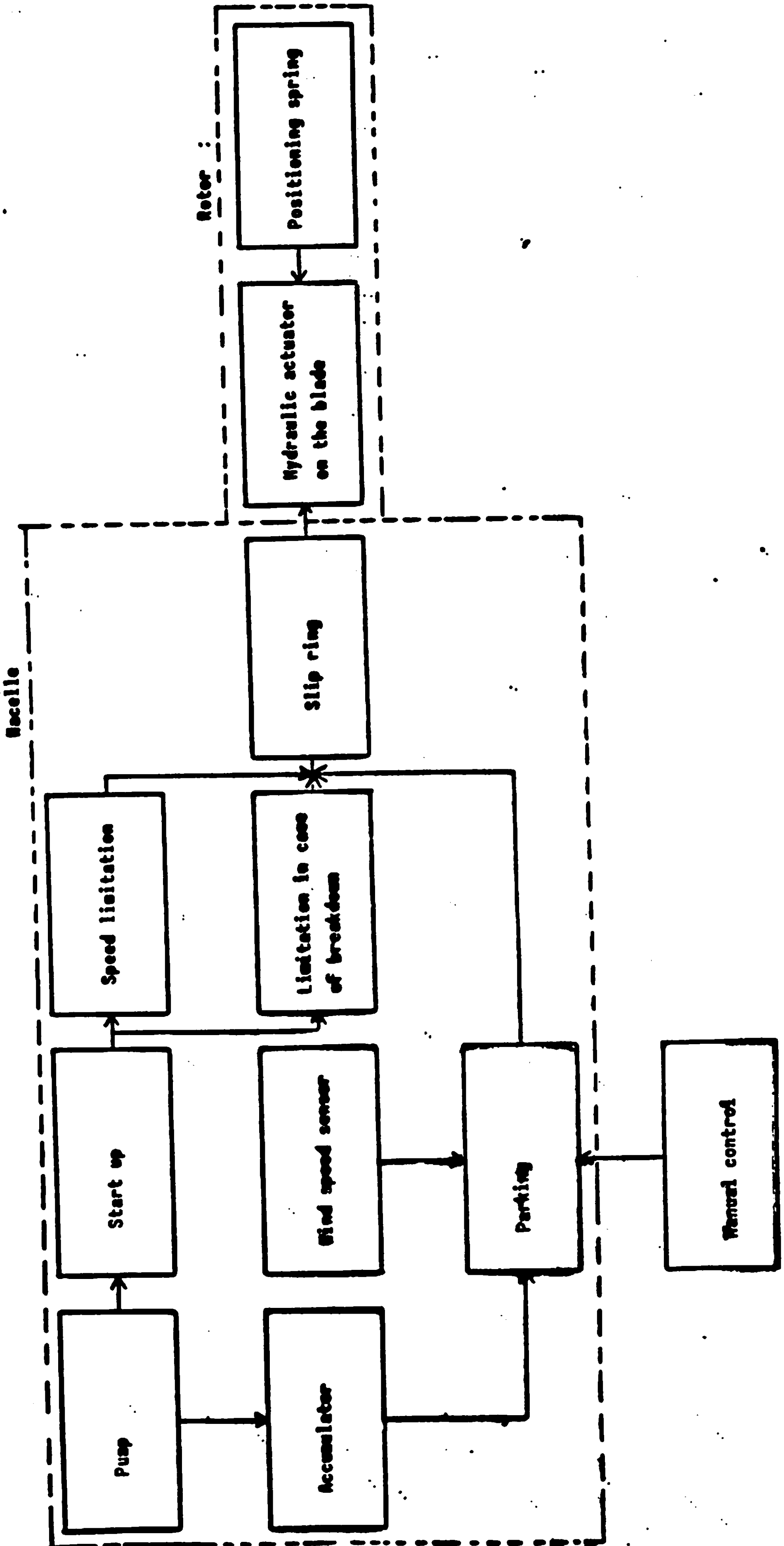
Bologna-Italy

FIG. 3 - M7 CONTROL SYSTEM DIAGRAM



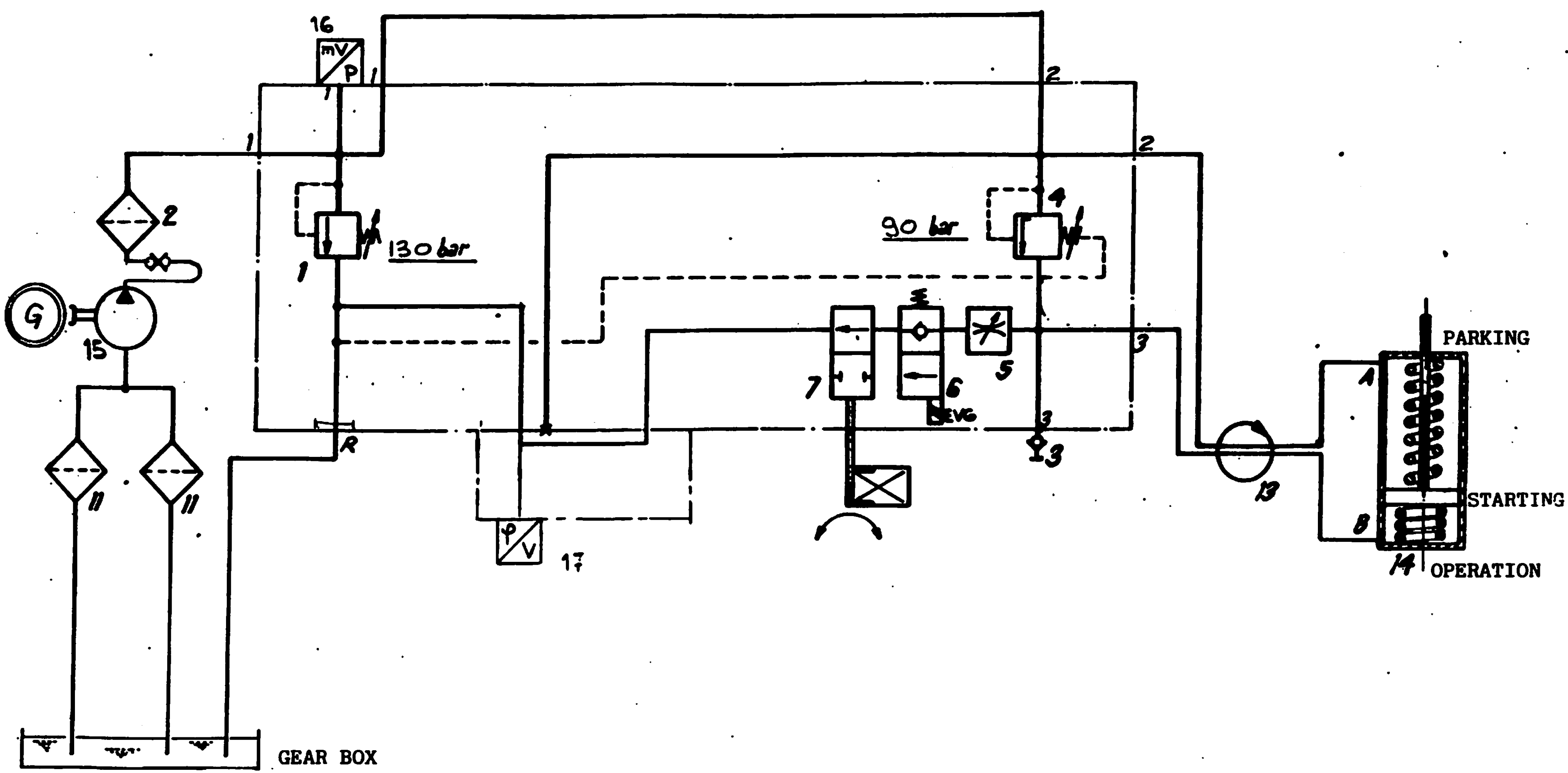


FIG. 4 - HYDRAULIC BLOCK DIAGRAM

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	MOMENT OF INERTIA (Kg. sq.m.)	ACTUATING SPEED (degrees/sec. r.p.m.)
(1)	60	0.05
(2)	50	0.05
(3)	50	0.10

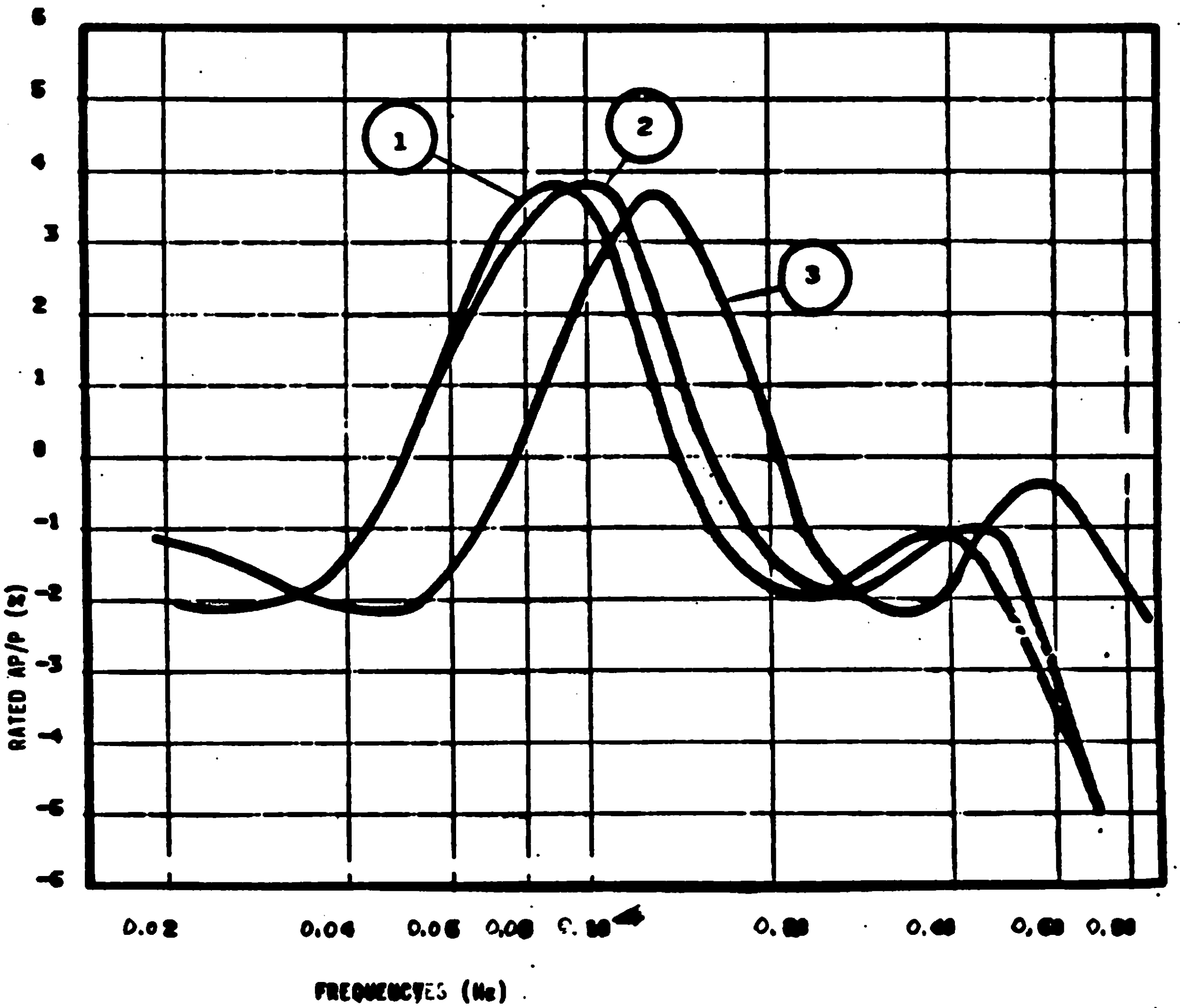
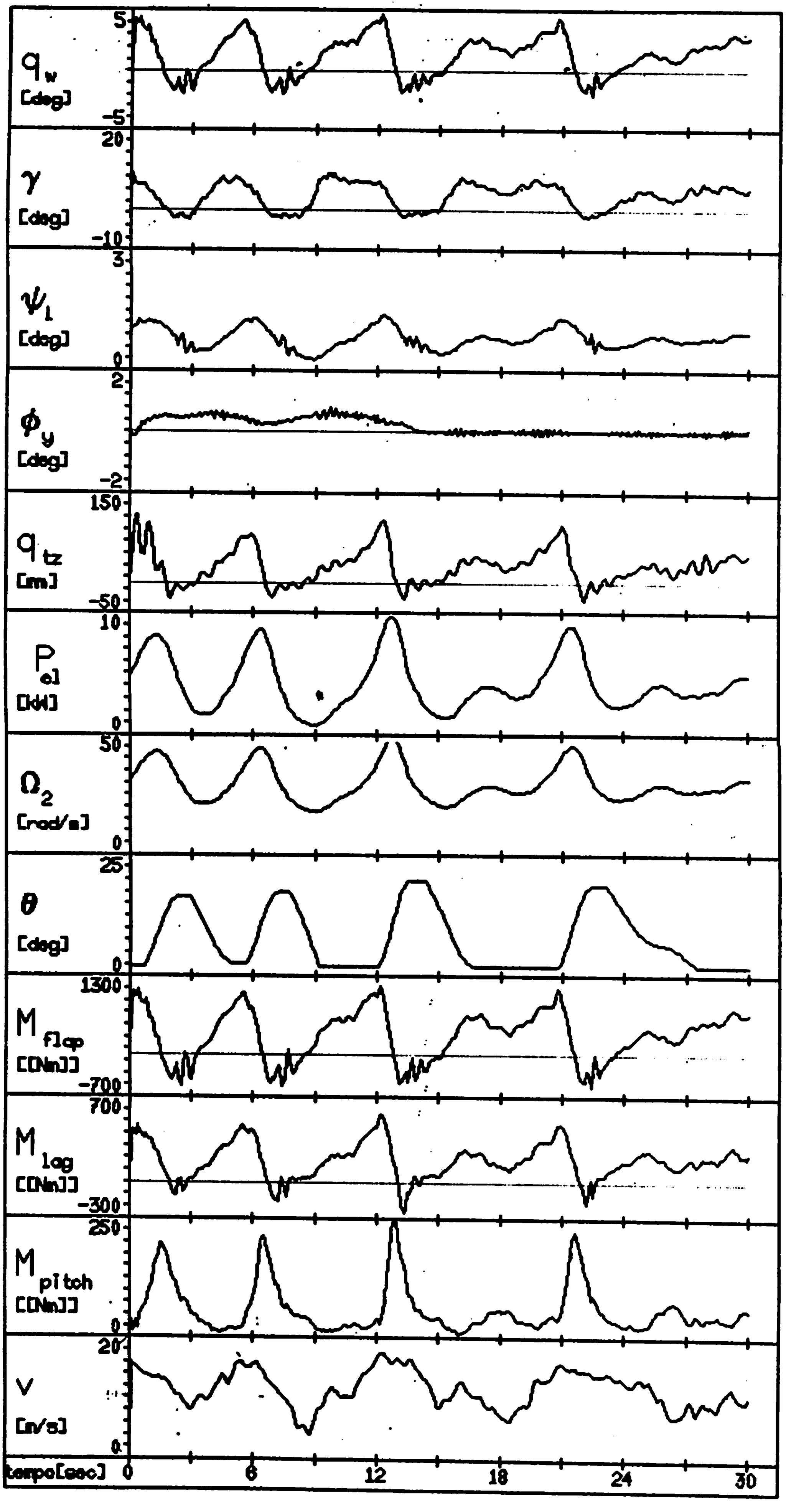


FIG. 5 - PERCENTAGE OF POWER VARIATION SPECTRUM

FIG. 6 - M7 DYNAMIC SIMULATION



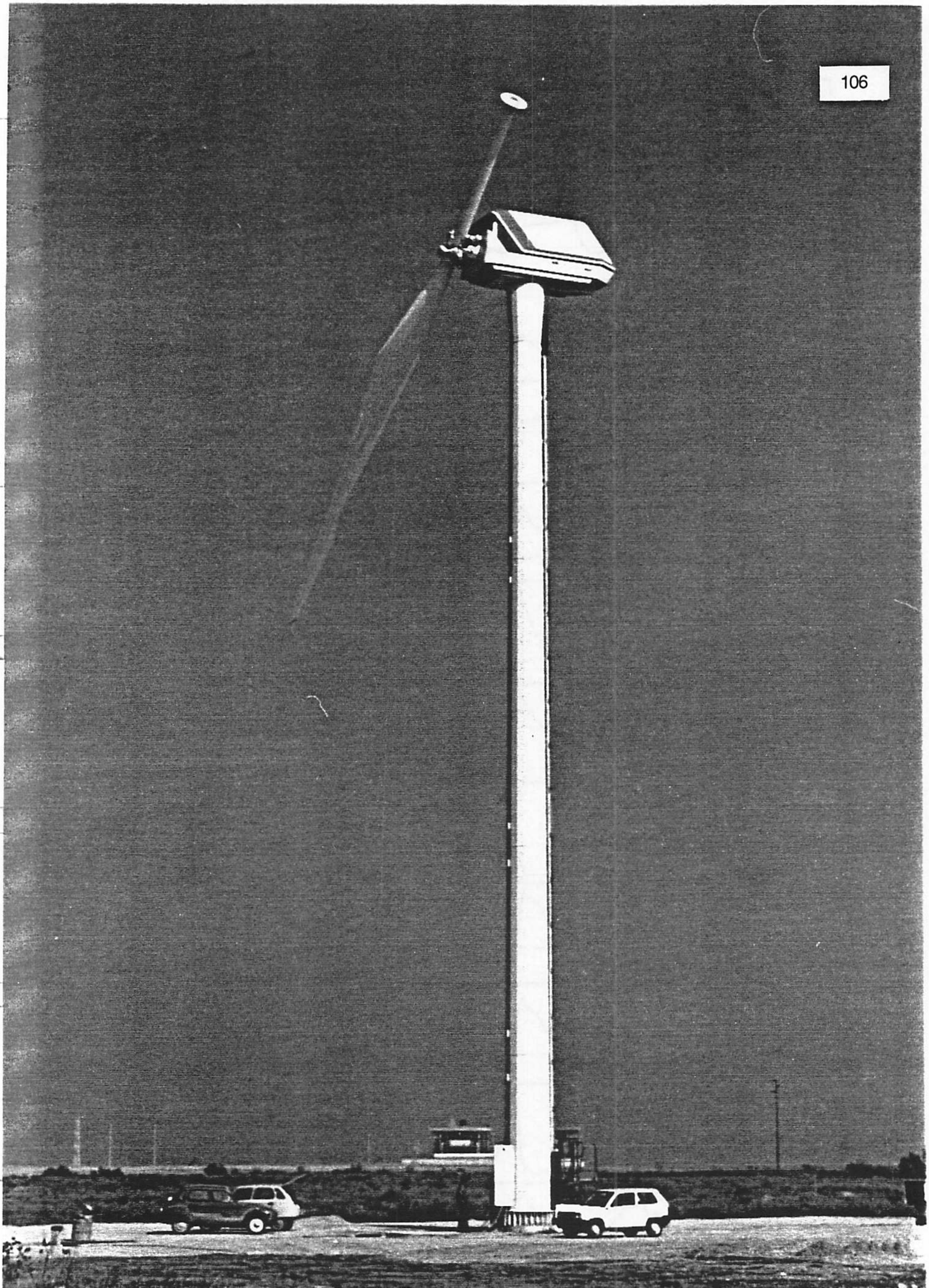
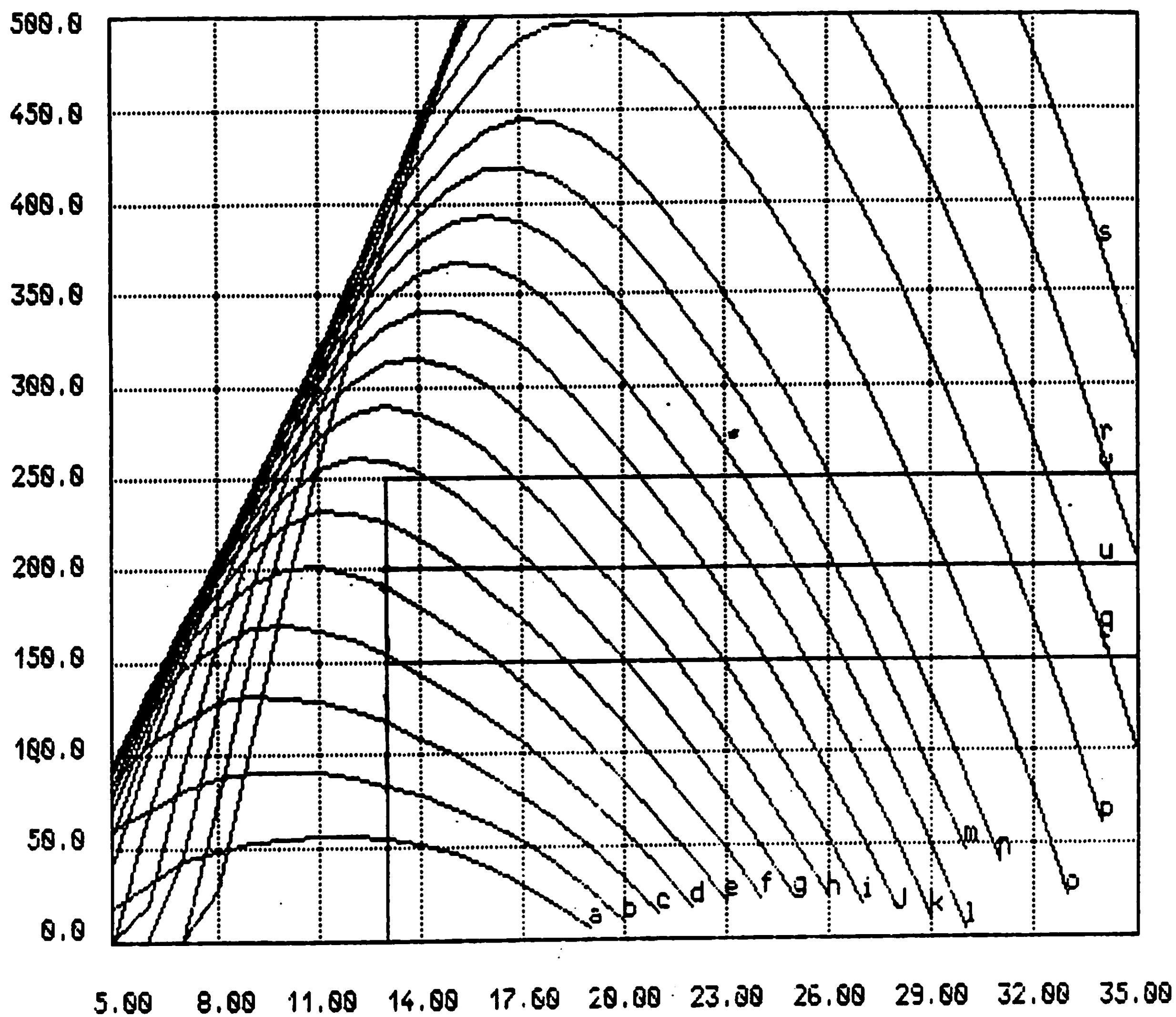


FIG. 7 - M30 INSTALLED IN ALTA NURRA (SARDINIA)

FIG. 8 - ELECTRIC POWER(kW) versus THETA(degrees) - U=Rated



WIND	SPEED(m/sec)
a	7.00
b	8.00
c	9.00
d	10.00
e	11.00
f	12.00
g	13.00
h	14.00
i	15.00
J	16.00
k	17.00
l	18.00
m	19.00
n	20.00
o	22.00
p	24.00
q	26.00
r	28.00
s	30.00
t	Ld=Ra.-25%
u	Ld=Rated
v	Ld=Ra.+25%

BLADE :

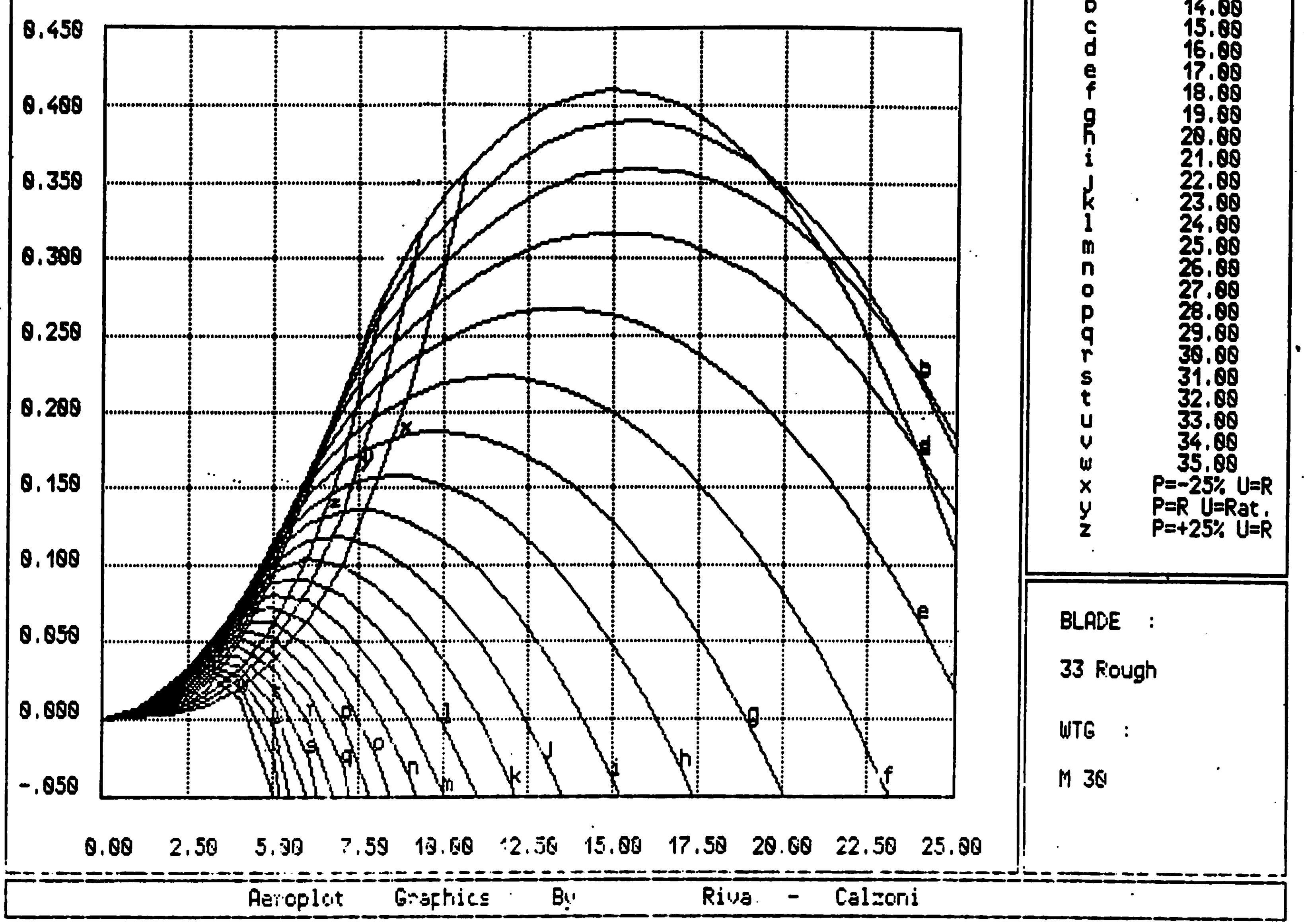
33 Rougt

WTG :

M 30

Aeroplot Graphics By Riva - Calzoni

FIG. 9 - CP Coeff. versus LAMBDA with U = Rated Value



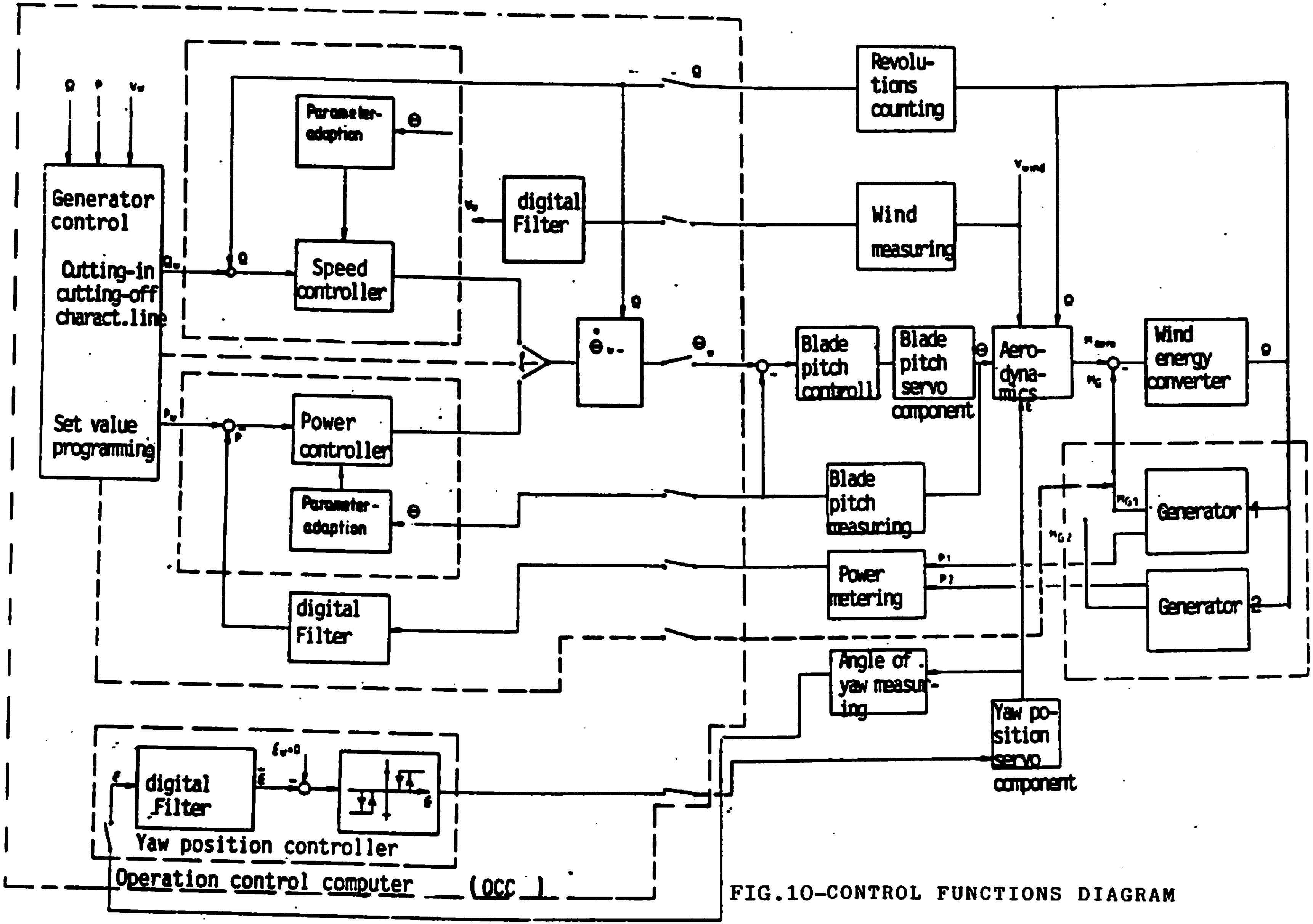


FIG.10-CONTROL FUNCTIONS DIAGRAM

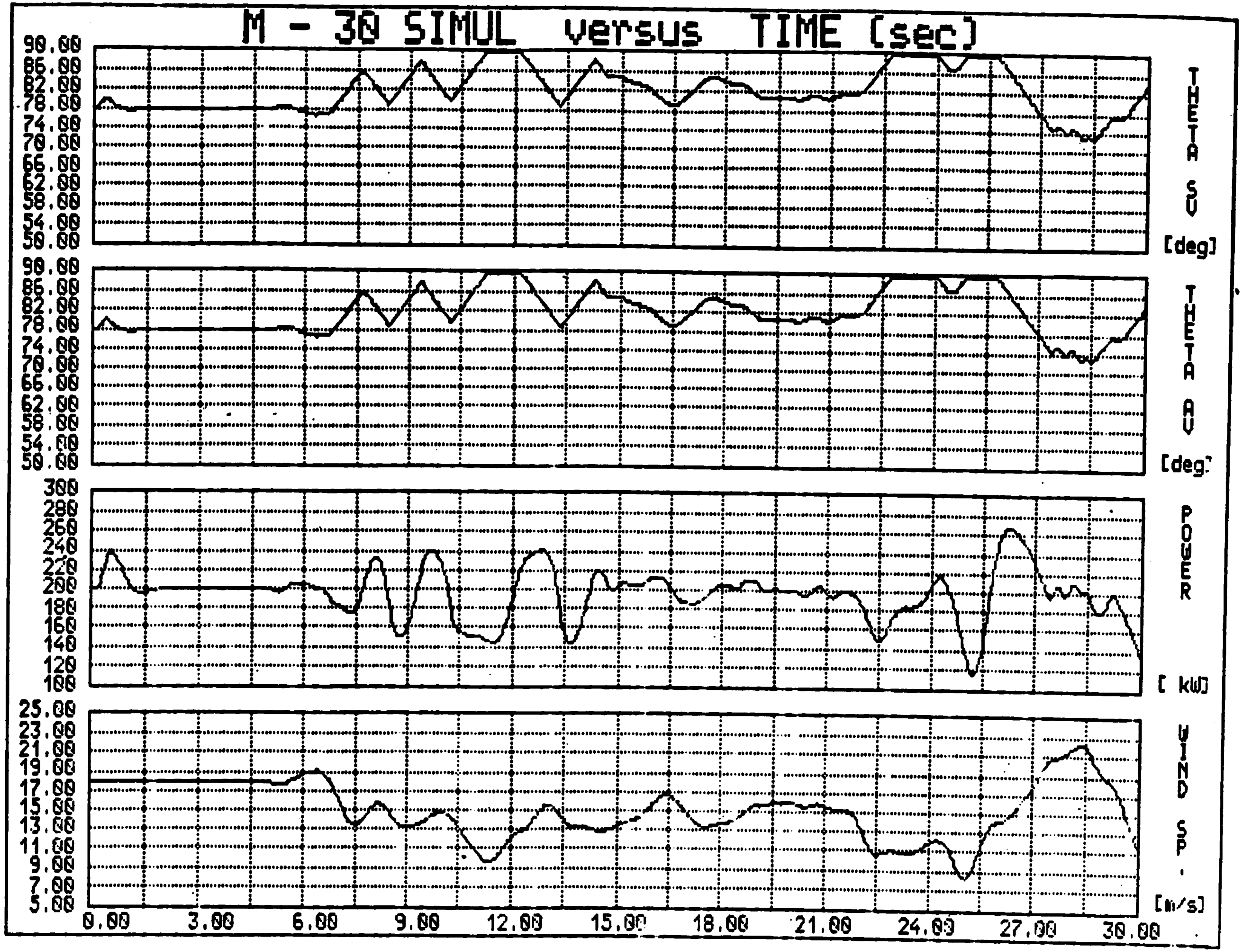


FIG. 11 – P.I.D OPTIMAL REGULATION

FLYGTEKNISKA FÖRSÖKSANSTALTEN

The Aeronautical Research
Institute of Sweden
Aerodynamics Department

1990, June

SOME CONTROL PROBLEMS WITH BROAD RANGE VARIABLE SPEED FOR HORIZONTAL AXIS WIND TURBINES

by
Eskil Ulén, The Aeronautical Research Institute of Sweden
and Göran Dalén, Swedish State Power Board

CONTENTS

SYMBOLS

QA	aerodynamic torque
QSH	shaft torque
QE	electrical torque
QEREF	electrical torque reference
JH	hub inertia
JB	blade inertia
JT	JH + 2 JB
JG	generator inertia
KS	shaft spring constant
GR	"generator" resonance
BR	blade edgewise resonance
RS	rotor speed
GS	generator speed
λ	tip speed ratio
CP	power ratio
R	turbine radius
A	turbin area
IF	field current
IDC	d c current
UDC	d c voltage
P	power
θ	yaw angle
ω	angular frequency
[ ]	references

1 GENERAL

1.1 The Energy situation in Sweden

The energy situation in Sweden is interesting. The cost of electricity is still one of the lowest in Europe. The main reasons for this is that about 50% of the electricity is produced by hydropower and 50% is produced by nuclear power. Thermal power is only used when peak power is required.

This situation is, however, changing. The use of nuclear power will, according to a parliamentary decision, come to an end by the year of 2010. Hydropower still has some additional capacity, but can only be used marginally, due to environmental reasons.

Another aim is to freeze the total emission of carbondioxide in Sweden, which puts high restrictions on development of new coal and oil/gas fired powerplants.

A study of available locations for wind energy in Sweden indicates a future capacity of about 1.5-7 TWh on land and an additional 20 TWh off shore (The total consumption of electricity today in Sweden is 140 TWh).

With this as a background, the possibilities to use wind energy as a complement in the Swedish energy system is presently investigated and several R&D activities are going on.

1.2 Wind Energy - status and development

The wind energy activites at the Swedish state power board are focussed on three main areas:

- technical development,
- demonstration,
- siting and energy potential.

The aim is to make it possible to start a commercial development of wind energy in the middle of the nineties.

The Swedish state power board has been involved in wind energy for more than 10 years.

In 1982, a 2 MW prototype, Näsudden, was erected on the Island of Gotland. This unit will now be rebuilt, and the new unit, Näsudden II, will be erected in the summer of 1991. The new unit can be characterized by the following:

- increased rotor diameter to 80 m
- increased power output to 3 MW
- 2 constant speeds (14 and 21 RPM)
- 2 rotor blades made of CRP/GRP
- new blade pitch mechanism without hydraulics or electronics in the hub.

One importand design driver for this new unit has been to minimize the maintenance, and thereby increase the total availability.

An almost identical unit, Aeolus II, is beeing built for a German power company (Preussen Electra). The main difference between the two units is that Aeolus II will be using a variable speed concept. This will give a very good opportunity to compare variable speed versus constant speed. The units are built in cooperation between Kvaerner Turbin AB in Sweden and MBB in Germany.

Discussions are also taking place regarding the possibilities to build a new prototype in the range of 400 kW. The unit shall be based on a technic that is simple, reliable and, cost effective. Concepts to be investigated are:

- fixed pitch
- variable speed
- stall and/or yaw regulation

A reliable control system is a necessity if this project shall be successful.

Although Swedish Wind Energy is at it's beginning, there is a lot of experience to build on. Both big prototypes (WTS-3 in the south of Sweden and Näsudden I) were equipped with sophisticated Data Aqusition Systems that have gathered important information for Universities, Research Centres, manufactures and the utilities.

Chalmers University of Technology in Gothenburg have a test fascility including a simulation laboratory and a test machine (45 kW) where different control algorithms and power regulation methods can be tested.

An 1.2 MW-machine (WTS1200) has been analyzed in details and partly constructed. An interesting detail was the augmented drive train compliance that was obtained by hydraulics.

Of special interest is power regulation in larger machines, using stall and/or yaw regulation together with variable speed. This may be one step towards what we all want and need; cheap, reliable and cost effective wind turbines.

In this article variable speed in a broad range is discussed. By connecting a frequency converter between the generator and the grid, the speed of the generator will be independent of the grid frequency. By control of the fireing angle of the frequency converter it is possible to control the electrical torque in the generator. This system offers the following advantages [2].

- Operation at near optimal tip speed ratio
- Increased energy production
- Reduced mechanical stresses
- Reduced acoustic noise
- Fast control of electrical torque

2 SIMPLIFIED DRIVE TRAIN DYNAMICS

2.1 Constant speed

The mechanical interface of the coupling to the grid is a damper.

QE = ELECTRICAL TORQUE
GS = GENERATOR SPEED
S = SLIP (0.5-3%)

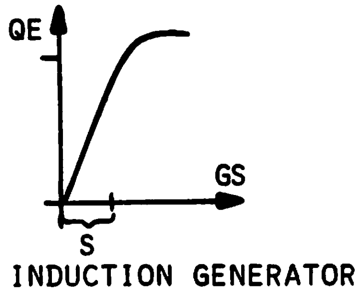


Fig. 2.1

Dynamically a small time constant is added. It is more or less a tight coupling to the grid. Big slip can give damping of the resonance between the turbin inertia and shaft spring but gives a less efficient generator. A limited filtering of torque disturbances to the shaft is obtained. This filtering can be augmented by a flexible gear mounting. This can be done mechanically (WTS3 in Maglarp [1]) or by hydraulics (WTS1200).

2.2 Variable speed

By means of torque feedback in the power electronics it is possible to get an approximately horizontal curve.

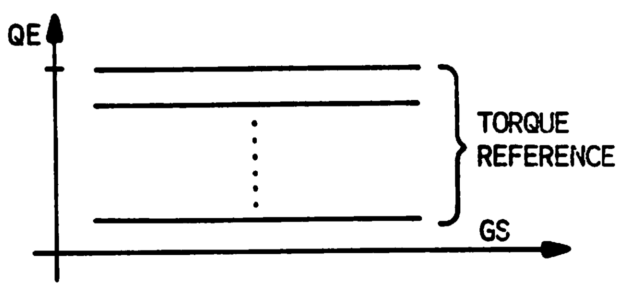
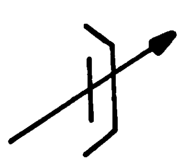


Fig. 2.2

Mechanically the inteface of the coupling to the grid can be seen as a controlled brake.



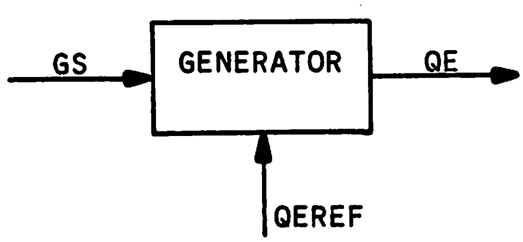
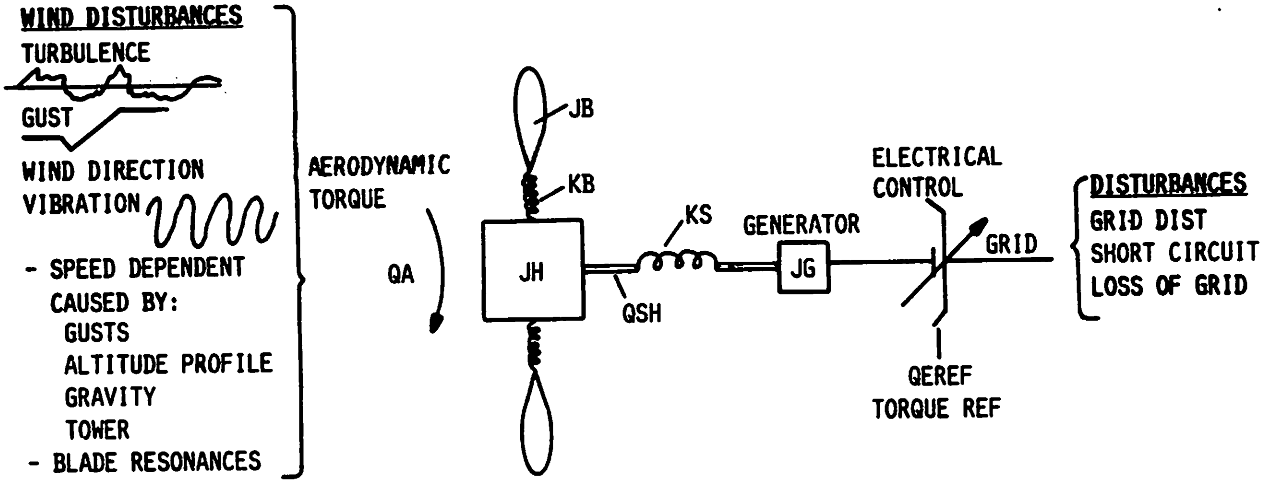


Fig. 2.3

The torque QE is mainly dependent on the QEREF and can be done nearly independent of GS (it can be shown that for a typical design it will be dependent to a small extent on GS). As QE in a typical design varies very slowly below nominal wind speed and is near constant above nominal the coupling to the grid will be very small due to disturbances. They will mainly accelerate the turubin and will not interfear to the shaft [3].

The following figure shows a simplified scheme (reduced to the turbin side of the gear).



QA= aerodynamique torque
JT= JH+2JB=turbin inertia
KS= drive shaft spring constant

JG = generator inertia
QSH= shaft torque
QE = electrical torque

Fig. 2.4

Mainly the following resonances are of interest for control of the torsional dynamics:

GR (generator resonance) $\omega \sim \sqrt{KS/JG}$; supposed JG<<JT
BR (blade resonance) blade structure edgewise
TR tower resonance (axial, lateral)

Also the flapwise movement and the thrust is of interest:

Tycical frequency diagram:

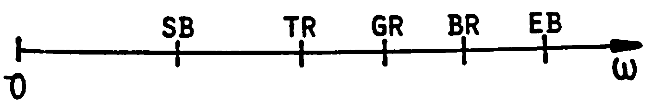


Fig. 2.5

Bandwidth:
SB speed control
EB electrical control

LOW FREQUENCIES (ω<BR,GR)

From the figure:

$$QSH = QE + \frac{JG}{JG + JT} \times QA$$

From control point of view JT,JT/JG and the bandwidth of the torque control loop should be as high as possible.

HIGH FREQUENCIES

Typical Bode-diagrams $\frac{QS\,H}{QA}(\omega)$

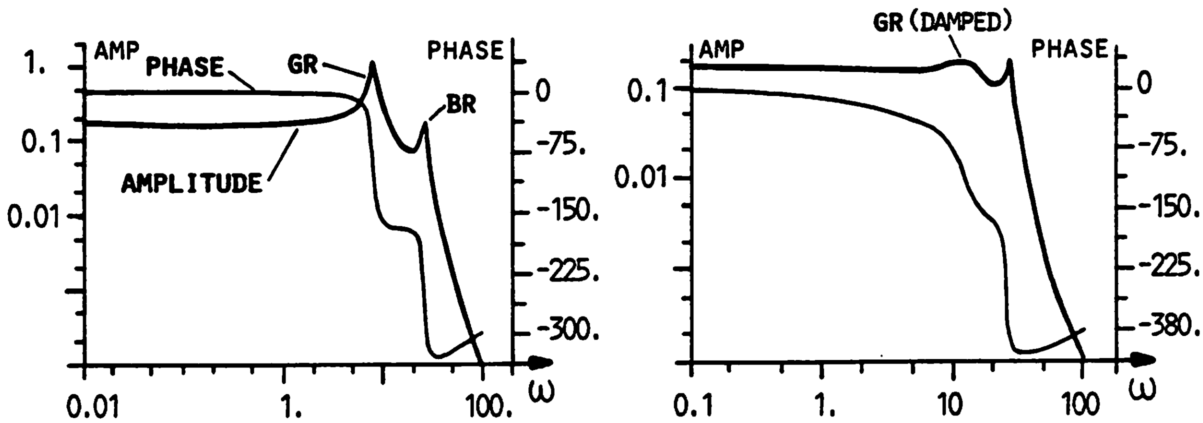


Fig. 2.6

There are several possibilities to get active damping of GR by means of feedback to the electrical control. From a theoretical analysis it is concluded:

Feedback signal

RS - GS	Effective but very difficult to measure with sufficient accuracy
$\dot{G}S$	Gives higher resonance frequency
QSH	Gives good result, the sensor can be placed on the gear mounting and can be high pass filtered for damping purposes

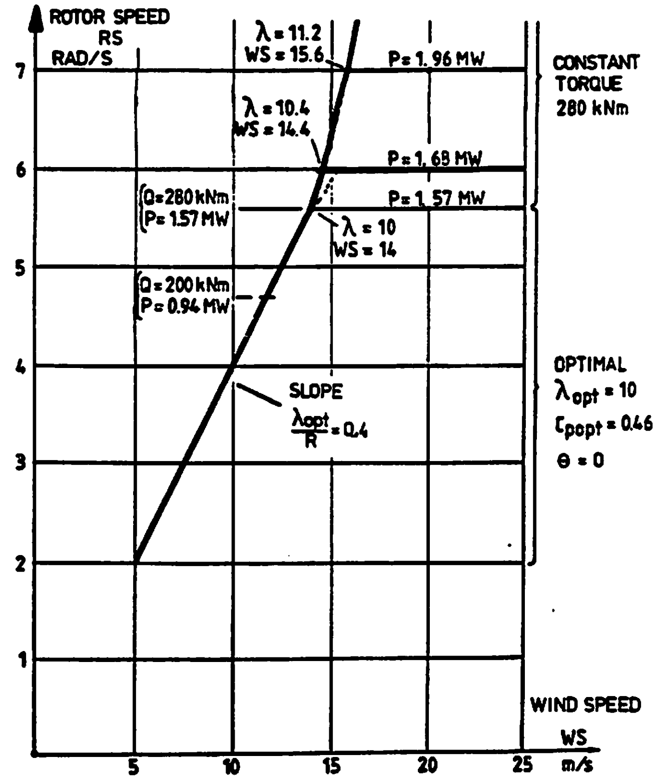
State estimation — Effective but difficult to get roboust
(observer or stationary KALMAN filter).

Concerning BR it is probably enough with the amplitude reduction if JG<<JT.

3 VARIABLE SPEED

3.1 General

The following shows an example [3], [4], [5], [6]



RS=rotor speed
P =power
Q =torque

Fig. 3.1

One can consider three different modes:

Optimal mode (speed and torque varies)
"Free" mode (speed variable, torque constant)
Speed regulation mode (speed control around near max allowable speed, torque constant)

3.2 Optimal model

3.2.1 General

$$P = 0.5 \times \rho \times A \times CP \times WS^3$$

where ρ = air density
A = turbin area
CP= power ratio
WS= wind speed

Q = P/RS gives:

$$Q = 0.5 \times \rho \times A \times R^3 \times CP/\lambda^3 \times RS^2$$

Optimal values gives:

$Q_{opt} = KQ \times RS^2$ where $KQ = 0.5 \times \rho \times A \times R^3 \times CPopt/\lambda^3 opt$

$RSopt = K\omega \times \sqrt{Q}$ where $K\omega = 1/\sqrt{KQ}$

Simplified scheme for the electrical control

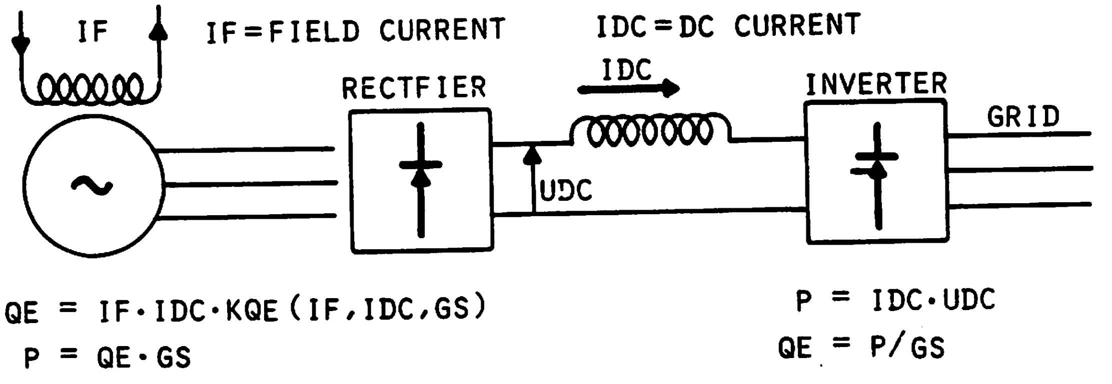


Fig. 3.2

The voltage UDC is not allowed to be higher than a certain value. UDC is roughly proportional to GS and is also dependent on IF and IDC. In the following some measurements at Chalmers University of Technology (CTH) are presented. A 40 kw generator is used [7].

The following figure is an example of field current behaviour.

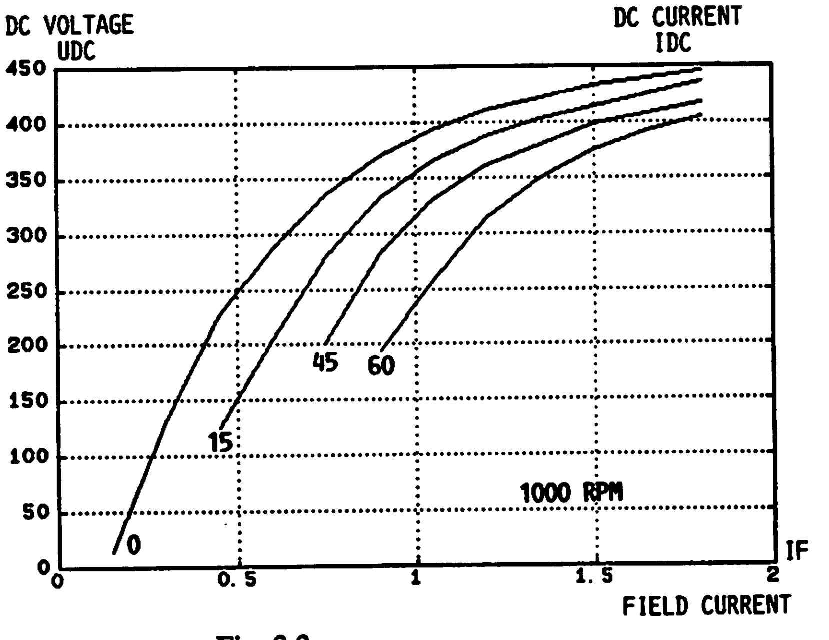


Fig. 3.3

There are several possibilities to include this scheme in a fast control loop. One possibility is to make a polynomial fitting of the derivative of the above function UDC and use this to control the gain in the regularor.

3.2.2 Torque loop

$QEREF = (RS)^2 \times KQ$
(on the turbine side of the gear)

$$KQ = \frac{1}{2} \times \rho \times A \times R^3 \times \frac{CP_{opt}}{\lambda^3_{opt}}$$

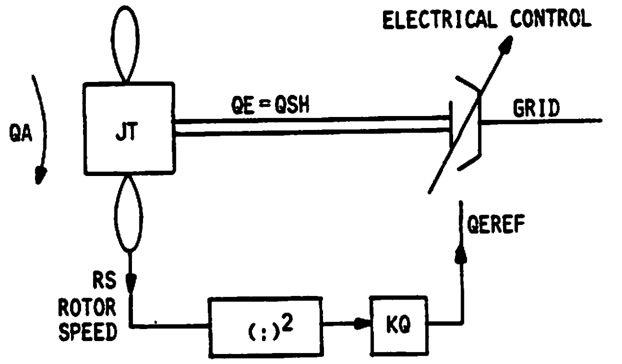


Fig. 3.4

Torque balance between QA and QE. Dynamically it varies with rotor speed and can be modelled by a time constant 5-15 sec. From this point of view a flat optimum of the C_p-λ-curve is desirable.

In this case the torque generated by the electrical control must be determined with a certain accuracy.

QE = IF × IDC × KQE(IF,IDC,GS)

The following diagram is an example of the function KQE

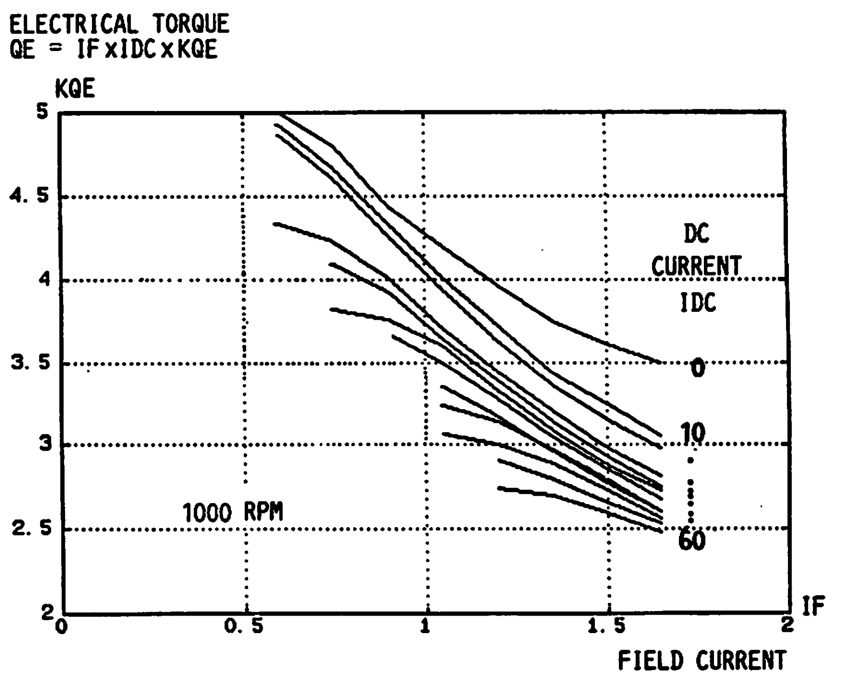


Fig. 3.5

KQE is only to a small extent depending on generator speed GS and this can be calibrated separately. The IF and IDC-dependence can be described by means of polynomial fitting (easy to implement in a computer) and this fitting can be included in the control scheme in order to get acceptable accuracy.

3.2.3 Speed loop

$$RSREF = K\omega \times \sqrt{QE}$$

In this case is QE determined from the power measurement. The measured value of P does not count losses. These losses are relatively great at low power and should be estimated [2].

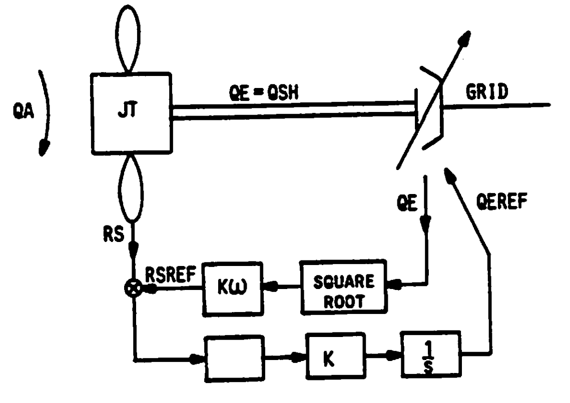


Fig. 3.6

3.3 Speed control loop

3.3.1 Blade control

From practical experiences so far this control will give no serious problems. However, a rather expensive blade servo is needed.

3.3.2 Fixed pitch

3.3.2.1 Yaw control

The following figure is a function fitting coming from a test of a small turbine in a wind tunnel and recalculated to a megawatt maschine.

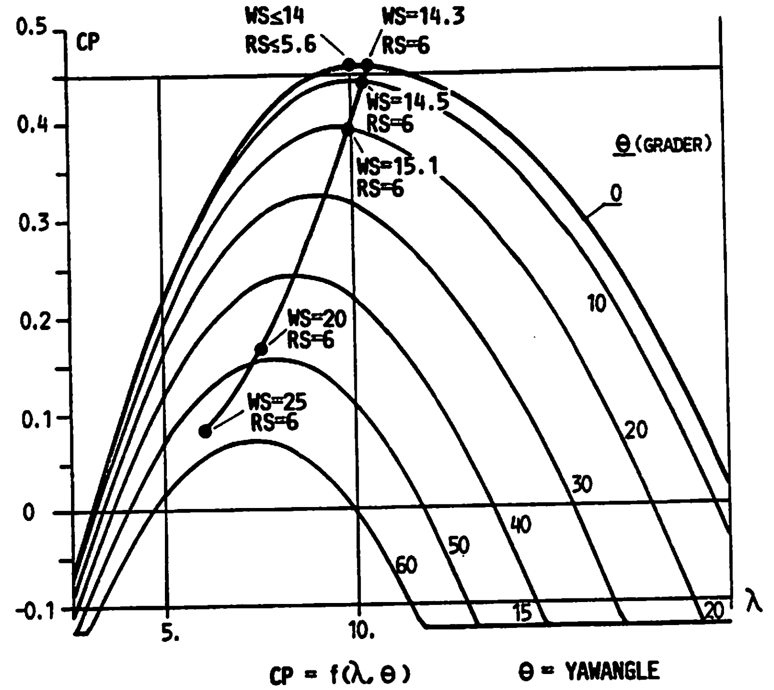


Fig. 3.7 $CP = f(\lambda, \theta)$ θ = YAWANGLE

The torque gain $\Delta QA/\Delta\theta$ will vary rather heavily with wind speed and yaw angle value. Gain scheduling can be used, see Section 4.2 and the θ-range near zero can be avoided by inserting a θ-variation before max speed is obtained. Max yaw-speed is important.

There is a coupling from yaw movement to tower lateral movement when center of gravity differes from yaw axis. This is sometimes called "fish-tail"-movement. It will in fact be a closed loop as indicated in the following scheme.

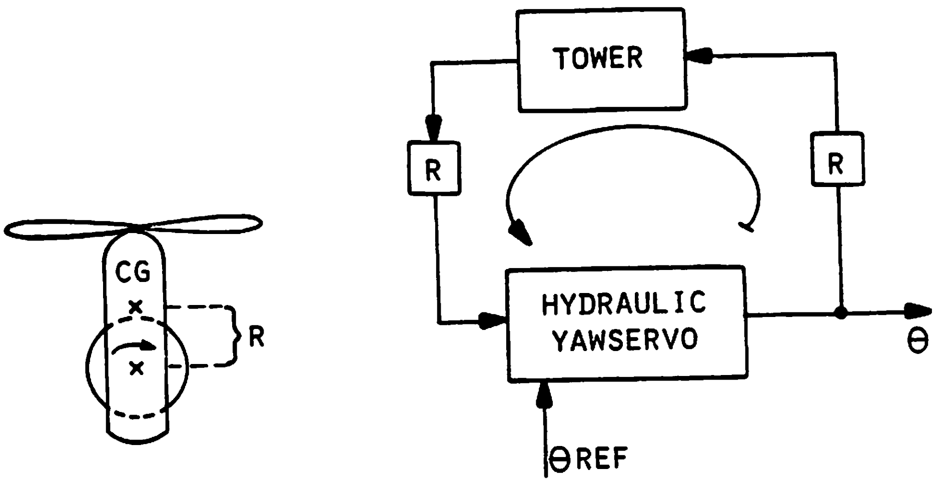


Fig. 3.8

As the tower lateral resonance is very undamped at least for low yaw angles this coupling can be rather serious because it is in practice difficult to get low distance from CG to yaw-axis.

A detailed analysis including a simulation has been carried through. In this analysis torque feedback in the yaw servo is utilized [8]. The following is a simplified scheme.

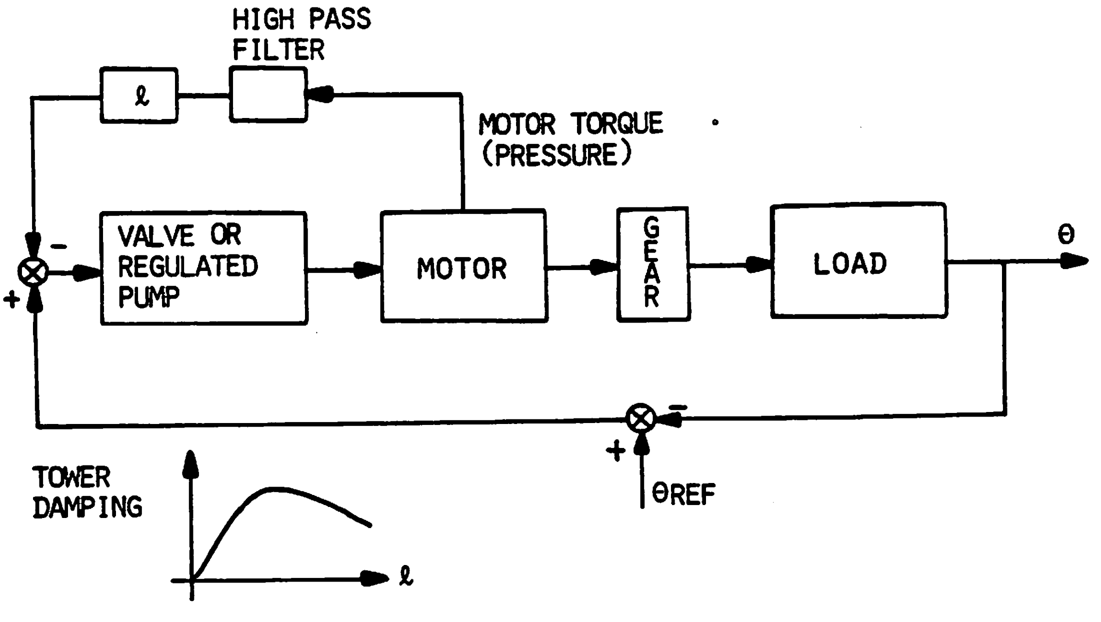


Fig. 3.9

A specific feedback gain gives max damping to the tower resonance. A high pass filter can be introduced to make the feedback mainly effective for the resonance frequency. The following figure shows an example from the simulation.

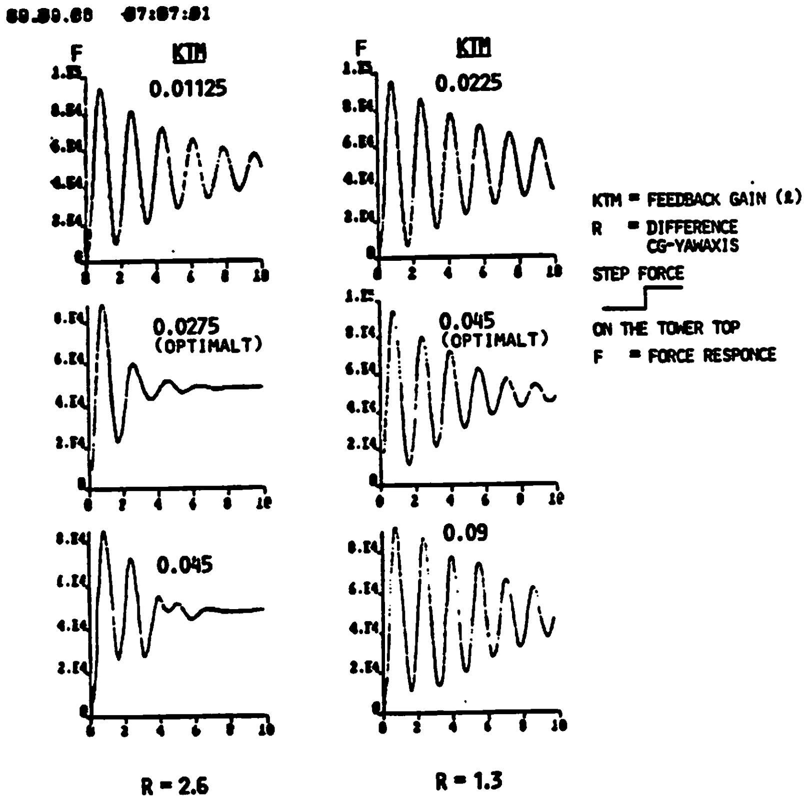


Fig. 3.10

It is also possible to use a combination of orifices and accumulators directly applied over the motor [9].

This will in fact act in the same way if it can be regarded as lineary. However, the orifices must probably be non linear due to temperature drift.

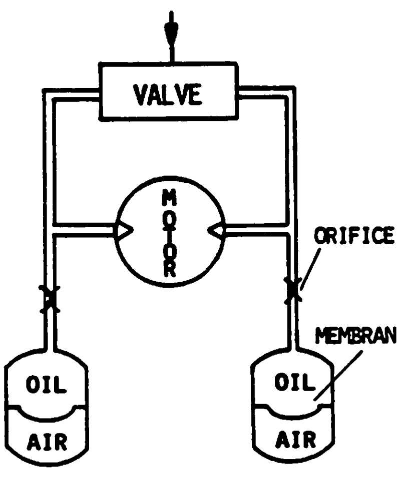


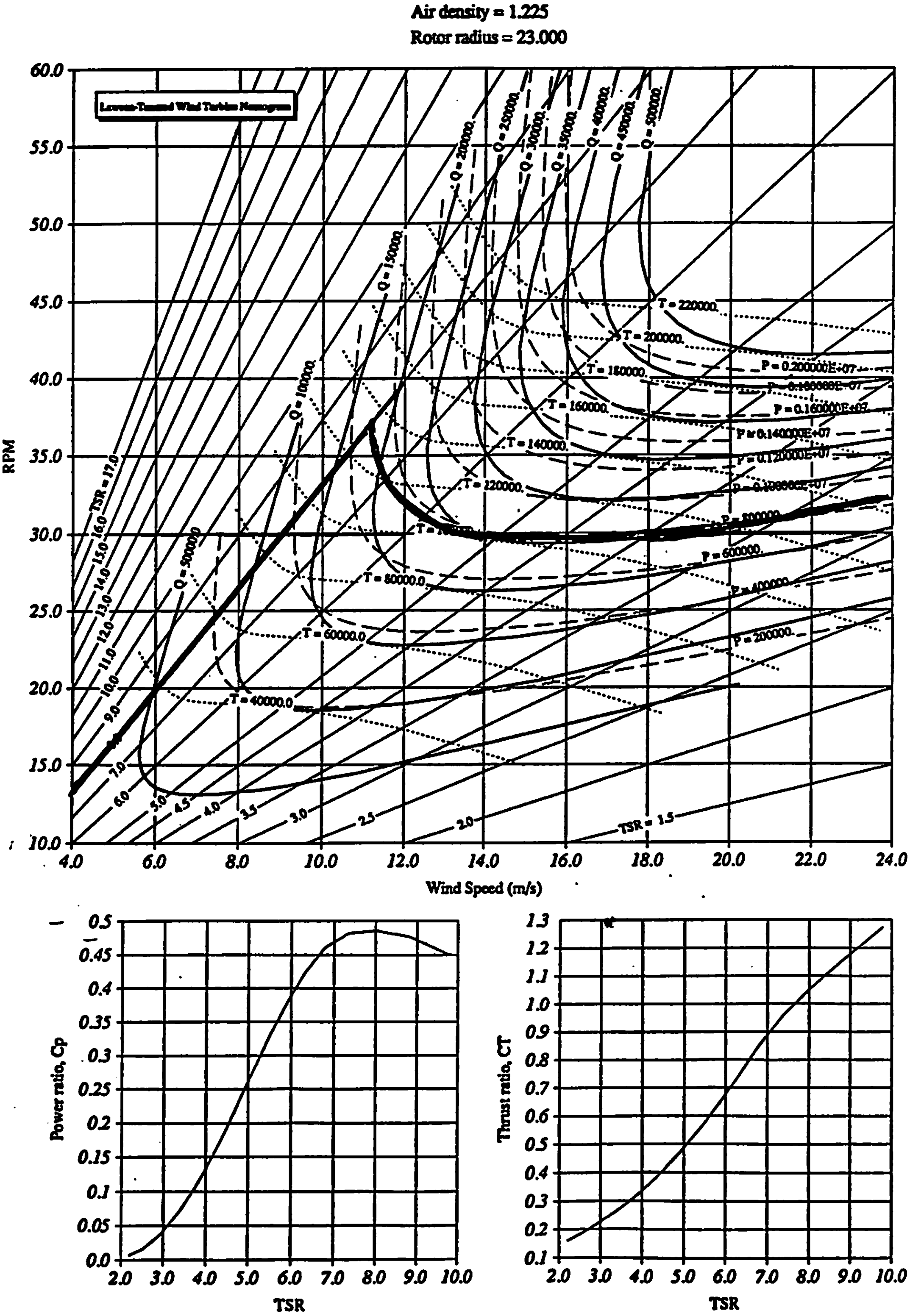
Fig. 3.11

3.3.2.2 Stall control

Stall control has so far mainly been used for small and medium machines. For big machines with variable speed as far as we know very little is done. At FFA a project is started to analyze the possibilities with this concept. The shaping of the blade profile is important.

One idea for variable speed is to go optimal to a lower power than the maximum and then if the wind speed augments somewhat smoothly go to the stall region. For the control strategy the following diagram has shown very helpful (the example shows an 800 Kw maschine, diameter 46 m) [10].

X_800 (Pitch = -4 deg)



TSR = Tip speed ratio

Q = constant torque
P = " power
T = " thrust

Fig. 3.12

4 MORE ELABORATED CONTROL

4.1 Optimal mode

For a change in wind speed the rotor will change its speed with a time constant of 5-10 sec. It could in principle be possible to make this time constant smaller by using feedback to the electrical control. Two possibilities has been investigated by simulation:

- rotor acceleration feedback (derivation and filtering of rotor speed),

If + sign is used a simplified model shows an equivalent smaller rotor inertia.

- state feedback.

Linear quadratic optimization has been used. A cost function using the output vector [RS QSH] and the input QEREF is used.

Neither of this types of feedback has given any growing energy capture for wind turbulence modelled as a stochastic process characterized by a Kaimal spectrum. A more advanced wind prediction used in an optimal feed-forward loop (optimal tracking) could perhaps give better result [11].

4.2 Speed control

QA = f(WS, RS, θ).

Linearization around (WSØ, RSØ, θØ)

$$d\,Q_A = \underbrace{\frac{\Delta QA}{\Delta\theta}}_{\text{Torque gain}}\,d\theta + \underbrace{\frac{\Delta QA}{\Delta RS}}_{\text{Damping}}\,dRS + \underbrace{\frac{\Delta QA}{\Delta WS}}_{\text{Sensibility to wind disturbances}} \times d\,WS$$

$\Delta QA/\Delta\theta$ is included in the control loop and is highly variable with wind speed. It turns out from measurements at WTS 3 and CTH test station that gain scheduling can be used to some extent using the value from a normal wind speed sensor. Concerning WTS 3 the maximum wind speed has been argumented from 20 to 25 m/s and for the test station a gain variation in the computer between 2 and 3 during the wind speed range is possible.

Some preliminary identification tests using measurements from the test station has been performed. As there is very big disturbances a general model (PEM-model) must be used. Off line analysis of 2-4 minutes recordings shows good possibilities for identify $\Delta Q_A / \Delta_\theta$ and a recursive version could perhaps be used on line.

The electrical control can also be included in speed control either by acceleration feedback (-sign gives an equivalent higher inertia) or more advanced with multivariable LQ-design with estimation [12]. However, the load on the drive train will than augment.

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Wind Engineering, Vol. 13, No. 5, 1989.

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Vierwgraphs of the Contribution

Netherlands Energy Research Foundation

ECN

Unit Renewable Energy

- Certification of WT
- Research & Development
  - Aerodynamics
    Rotormodelling
    Fatigue
  - Control &
    Electrical conversion
    - Wind Turbines
    - Photovoltaïc Systems
  - Field experiments
    flexible rotor
- Laboratory

Control of Wind Turbines

Past activities:

- REGHAT :

  Computer program for transient stability analysis of horizontal axis wind turbines

- System identification and control of a 25 m WT

- AWIDIMOD:

  Computer program for dynamic analysis of Autonomous Wind-Diesel Systems

Present activities:

- IRFLET :

  Integral control of a flexible WT

- AWDS (Joule):

  Logistic and dynamic computer program for Autonomous Wind-Diesel Systems

Electrical conversion systems for WT

Past:

- Laboratory tests of a number of E C systems in Dutch WT
- Development of two types of AWDS : AC- AC
  AC - DC - AC

Present:

- Current Source Invertor
- Voltage Source Invertor (Joule project)

Flexhat project :

Design of a flexible turbine

prototype : 300 kW

horizontal axis 2 blades,

- flexible connection of blades to hub

- passive pitch control based on centrifugal forces

- variable speed electrical system : AC - DC - AC

IRFLET

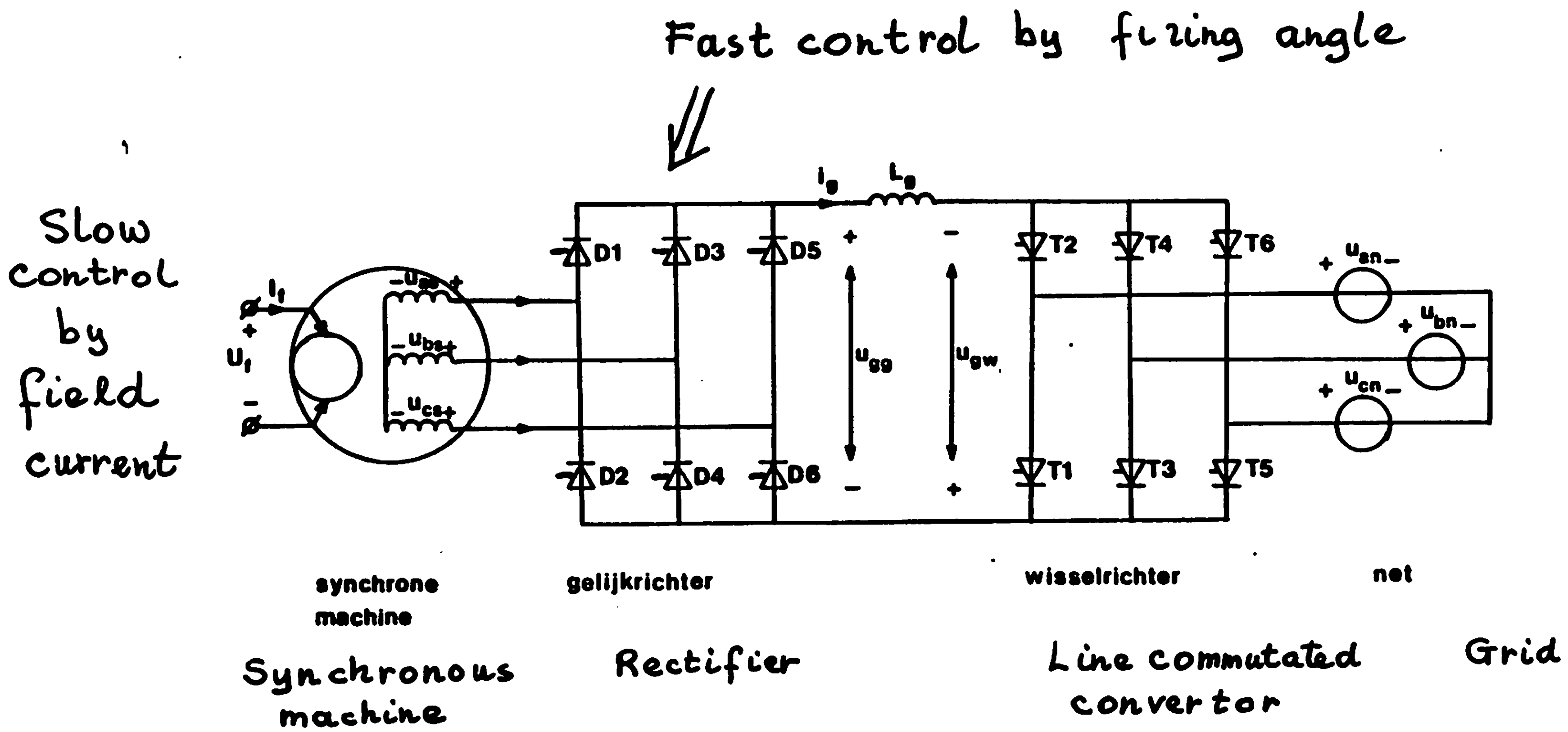
Objective:

Development of a control system for a flexible WT which meets the following requirements:

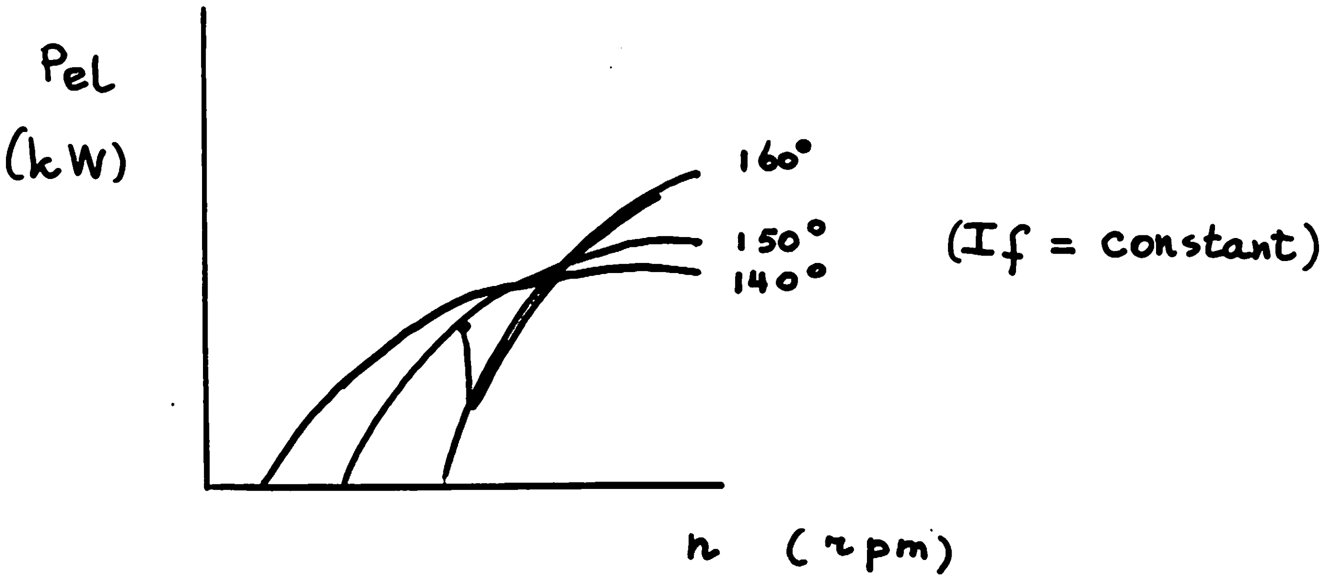
- optimal λ
- active damping of oscillations
- protection against electrical faults (short circuits)

Activities:

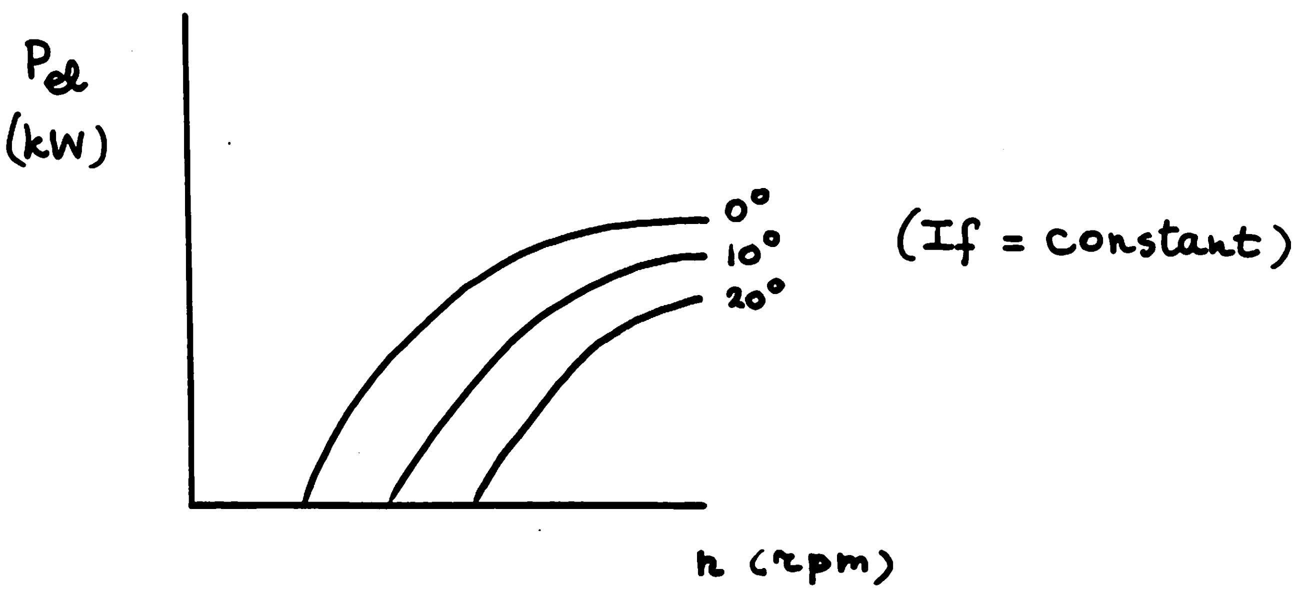
- Modelling of the mechanical and electrical system
- Controller design
- Implementation of controller and safety system in proces computer
- Design of laboratory system
- Experiments



SCHEMA VAN HET ELECTRISCHE SYSTEEM VAN EEN VARIABEL TOEREN WINDTURBINE



Control by firing angle of line commutated convertor at grid



Control by firing angle of rectifier at synchronous machine

IRFLET activities :

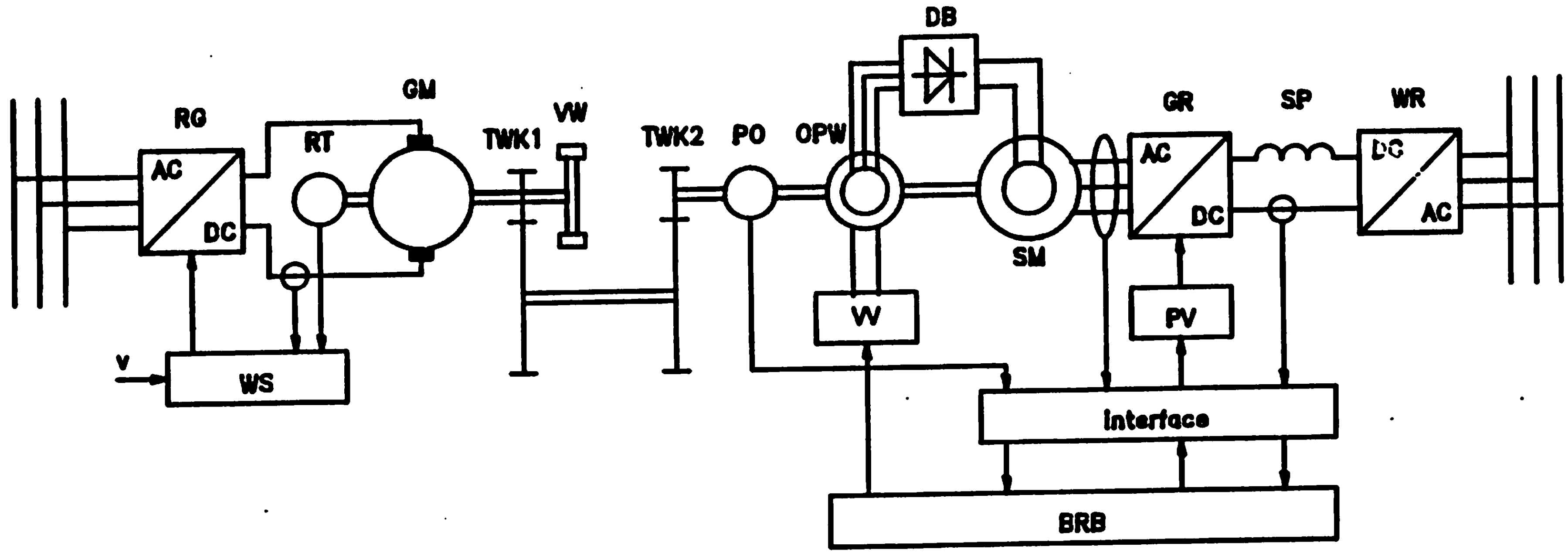
1. Modelling of the mechanical and electrical system

   - inertia, flexibility and damping of mech. components
   - transient model of SM and DC-Link, including
     * saturation
     * model of exciter
   - parameter estimation of SM, transfer function method

$$I(\omega) = \mathcal{F}\{U(\omega)\}$$

2. Design of laboratory system

   - 5 Hz eigenvalue of mechanical system
   - controlled rectifier



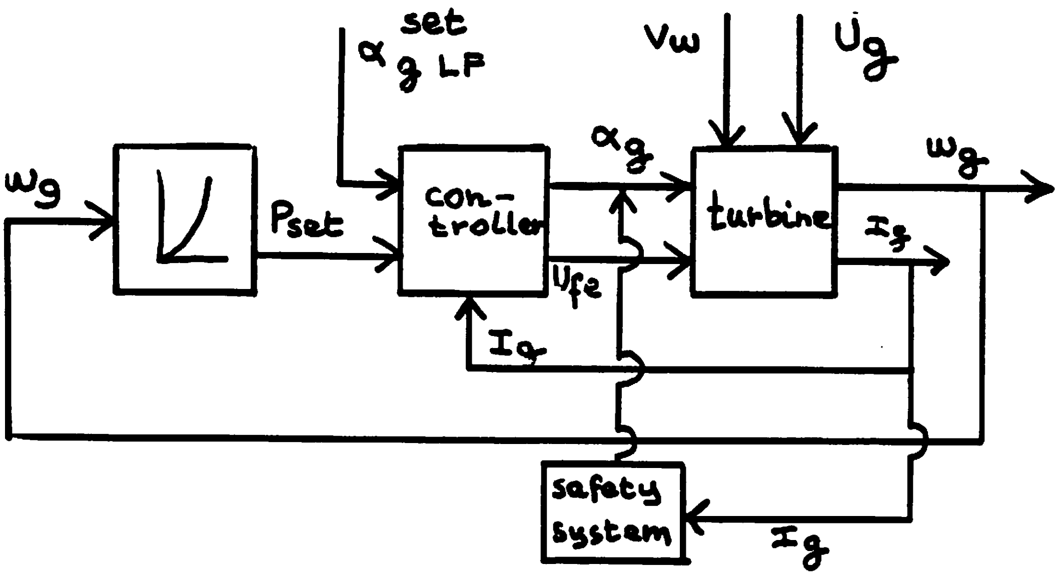
Laboratory system for testing of the designed control & safety system

Controller design & implementation

Objectives

- optimal λ
- active damping of oscillations in drive train
- protection against electrical faults

Control scheme



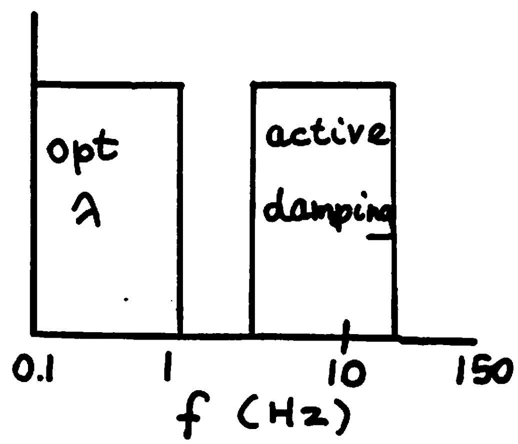
α_g firing angle

ufe exciter field voltage

wg generator speed

Ig DC - current

Pset power setpoint



Controller design (DUT)

1) Linear Quadratic Output Feedback

$$u = F \cdot y$$

$$u = \begin{pmatrix} \alpha_g \\ u_{fe} \end{pmatrix} \qquad y = \begin{pmatrix} \omega_g \\ I_g \\ \int \omega_g \\ \int \alpha_g \end{pmatrix}$$

Criterion function

$$J : \int_0^\infty (x^T Q x + u^T R u) \, dt$$

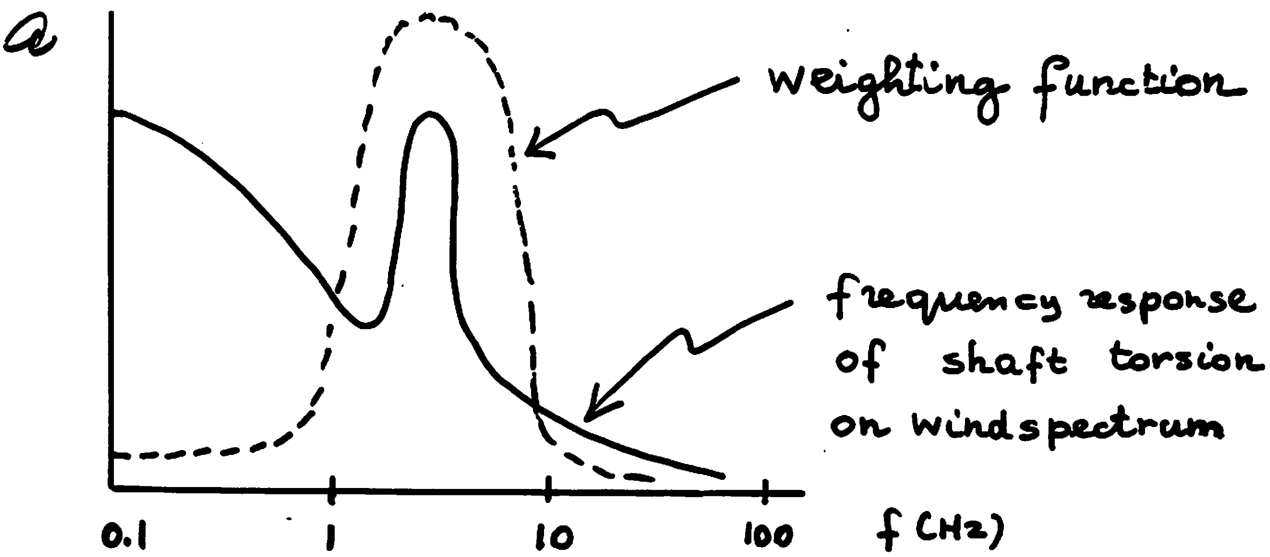
weighted turbine states

weighted turbine inputs

$$x = \begin{pmatrix} \omega_g \\ I_g \\ a \\ \int \omega_g \\ \int \alpha_g \end{pmatrix} \begin{array}{l} \left. \vphantom{\begin{matrix} \omega_g \\ I_g \\ a \end{matrix}} \right\} \text{high frequency contribution} \\ \left. \vphantom{\begin{matrix} \int \omega_g \\ \int \alpha_g \end{matrix}} \right\} \text{low frequency contribution} \end{array}$$

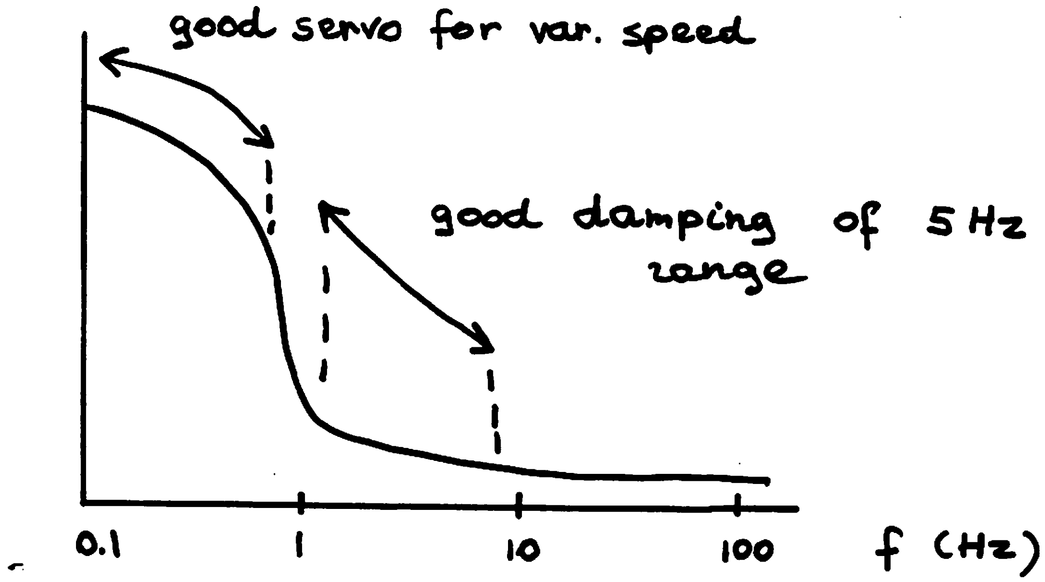
Controller design

2) H_∞ approach



minimization of freq. dependant criterion function

→ dynamic feedback



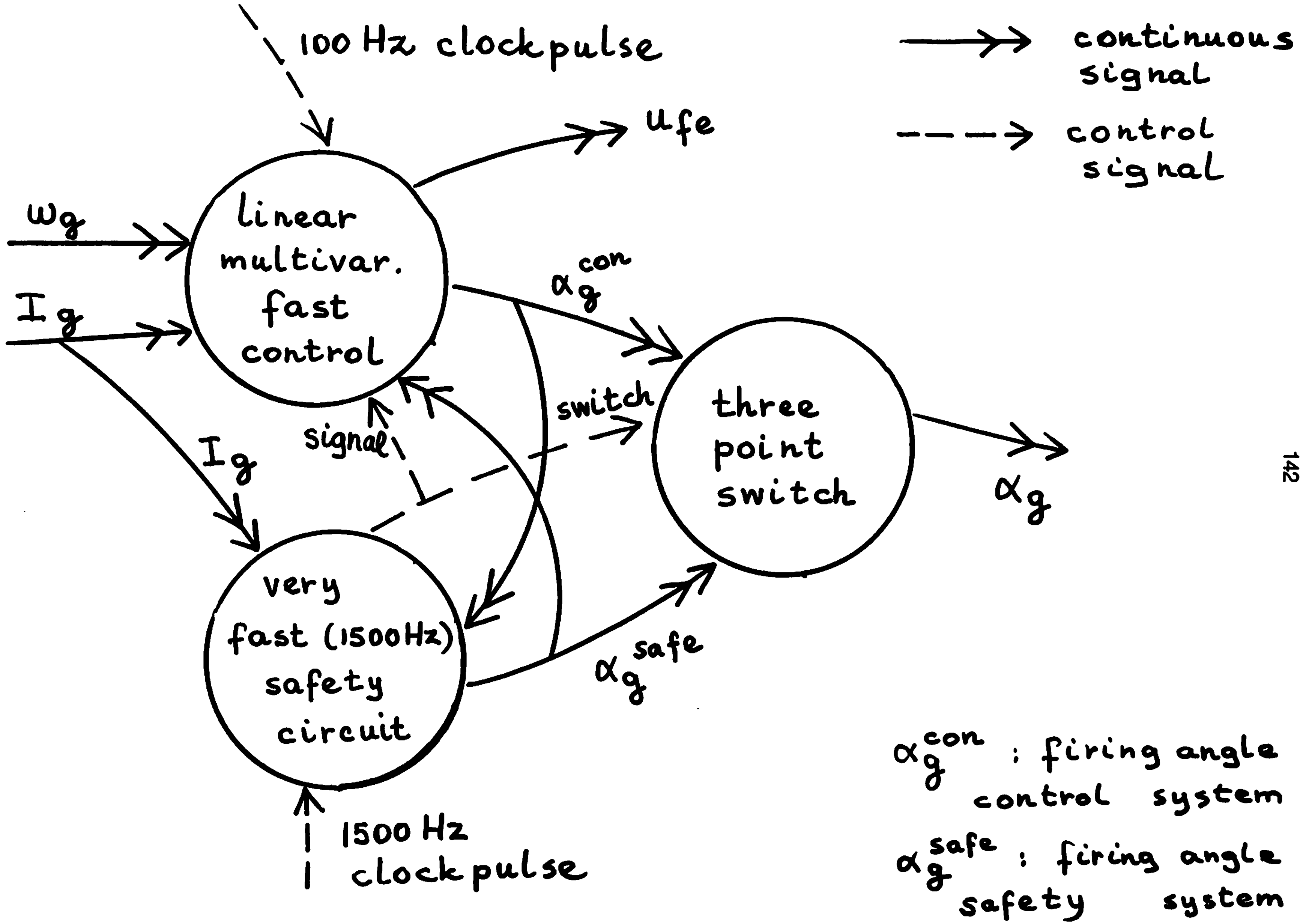


Fig 1: IRFLET Besturings- Regel- en Bewakingssysteem.

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IEA - Implementing Agreement LS WECS
Expert Meetings

1. Seminar on Structural Dynamics, Munich, October 12,1978
2. Control of LS-WECS and Adaptation of Wind Electricity to the Network, Copenhagen, April 4, 1979
3. Data Acquisition and Analysis for LS-WECS, Blowing Rock, North Carolina, Sept. 26 - 27, 1979
4. Rotor Blade Technology with Special Respect to Fatigue Design Problems, Stockholm, April 21 - 22, 1980
5. Environmental and Safety Aspects of the Present LS WECS, Munich, September 25 - 26, 1980
6. Reliability and Maintenance Problems of LS WECS, Aalborg, April 29 - 30, 1981
7. Costings for Wind Turbines, Copenhagen, November 18 - 19, 1981
8. Safety Assurance and Quality Control of LS WECS during Assembly, Erection and Acceptance Testing, Stockholm, May 26 - 27, 1982
9. Structural Design Criteria for LS WECS, Greenford, March 7 - 8, 1983
10. Utility and Operational Experiences and Issues from Mayor Wind Installations, Palo Alto, October 12 - 14, 1983
11. General Environmental Aspects, Munich, May 7 - 9, 1984
12. Aerodynamic Calculational Methods for WECS, Copenhagen, October 29 - 30, 1984
13. Economic Aspects of Wind Turbines, Petten, May 30 - 31, 1985
14. Modelling of Atmospheric Turbulence for Use in WECS Rotor Loading Calculation, Stockholm, December 4 - 5, 1985
15. General Planning and Environmental Issues of LS WECS Installations, Hamburg, December 2, 1987
16. Requirements for Safety Systems for LS WECS, Rome, October 17 - 18, 1988
17. Integrating Wind Turbines into Utility Power Systems, Herndon (Virginia), April 11 - 12, 1989
18. Noise Generating Mechanisms for Wind Turbines, Petten, November 27 - 28, 1989
19. Wind Turbine Control Systems, Strategy and Problems, London, May 3 - 4, 1990
20. Wind characteristics of Relevance for Wind Turbine Design, Stockholm, March 7 - 8, 1991