<u>DARRIEUS WIND TURBINE TECHNOLOGY ASSESSMENT STUDY</u>

<u>FINAL REPORT</u>

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Submitted to:

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1.0 LITERATURE REVIEW

1.1 Introduction

The bibliography of technical papers and reports is shown in Appendix A sorted by author.

Categories have been assigned to each bibliography entry in the data base. In addition, each entry includes the location of the document (for example, Schienbein, Malcolm, Indal, etc.) and a notation as to whether it is a key document. The printout in Appendix A does not include the categories and the location information.

Eighteen categories were selected and each bibliography entry has been assigned to as many as four categories, as appropriate. The bibliography covers all aspects of Darrieus wind turbine technology (but mostly restricted to curved blade turbines) as is indicated by the list of categories shown below.

Category Number	Subject
1	Darrieus curved bladed VAWT
2	Straight bladed VAWT
3	Aerodynamic theory/codes/airfoils
4	Power/aerodynamic evaluation
5	Structural theory/codes
6	Structural evaluation
7	Electrical system
8	Control theory/algorithms/evaluation
9	Braking
10	Manufacturing
11	Environmental
12	Mechanical drive train
13	Turbine testing/evaluation
14	Windfarm operations
15	Rotor wake and array effects
16	Other
17	Costs
18	Configurations

The bibliography was assembled from four main sources:

1. Library data base search of periodicals, conference proceedings and laboratory reports carried out through the University of Washington library.

2. R. Lynette and Associates technical library computer data base.

3. Libraries of the two principal contributors to this assessment, Dr. L. Schienbein and Dr. D. Malcolm.

4. Individuals and organizations throughout the world contacted by the project team. The main

resources have been Sandia National Laboratories (USA), the National Research Council (Canada), Indal Technologies (Canada), Energy Mines and Resources Canada (Raj Rangi), NRC Institute for Mechanical Engineering (Marc Chappell), Paul Perroud (France) formerly of CENG, and Dornier Deutsche Aerospace (Germany).

The reports listed in section 1.3 have so far been identified as being particularly relevant (key references).

1.2 Responses to Questionnaires

The project team mailed prepared and mailed approximately 70 questionnaires to individuals and organizations throughout the world. 36 replies were received. Of these, six indicated that they would not complete the questionnaire. A copy of the questionnaire is shown in Appendix B.

In addition, on-site interviews were held with several key individuals and organizations in Canada and the U.S.A., including Sandia National Laboratories, Indal Technologies, Energy Mines and Resources Canada, Adecon Energy Systems, Peter South, C.F. Wood, R. J. (Jack) Templin, and Lavalin Engineers. The responses have been used in the preparation of this report.

The following 30 individuals and organizations responded to the questionnaire (R) or were interviewed in person (I).

I Sandia National Laboratories Wind Energy Division (USA), Henry Dodd, Paul Veers, Dale Berg, Tom Ashwill, Bob Nellums, Emil Kadlec, Don Lobitz

I Adecon Energy Systems (Canada), Peter Jaeggin

I Indal Technologies Inc. (Canada), Vince Lacey

I C.F. Wood, formerly with Indal Technologies

I,R Peter South, formerly with Adecon Energy Systems and the National Research Council of Canada

I John Ereaux, formerly with Adecon Energy Systems, Indal Technologies and Lavalin Engineers

I Lavalin Engineers (Canada), Luc Lainey

I Energy Mines and Resources (Canada), Raj Rangi (also formerly with the National Research Council of Canada)

I,R R. J. Templin (Canada), formerly with the National Research Council of Canada

R G. Cunliffe, formerly with Indal Technologies

R Ion Paraschivoiu (Canada), Ecole Polytechnique

- R Dornier Deutsche Aerospace, Dr. Fritzsche
- R Marc Chappell (Canada), NRC Institute for Mechanical Engineering
- R Moscow Aviation Institute, Wind Energy Department (Russia), Vladimir Dobrovolski
- R Department of Alternative Energy Systems, Moscow Institute of Hydro-Energy (Russia), Dr. Yuly Shpolyansky
- R National Research Council (Canada), Paul Penna
- R Paul G. Migliore (USA)
- R Lewis Feldman (USA), Sail Bladed Turbine
- R Robert Hammett, formerly with Indal Technologies
- R Bechtel Group Inc. (USA), Bob Lessley
- R Bill McEachern (Canada), B.C. Research, formerly with Indal Technologies
- R Electric Power Research Institute (USA), Earl Davis
- R West Texas State University Alternative Energy Institute (USA), Dr. Vaughn Nelson
- R John Payne (USA), Kaiser Aluminum Research Center
- R Paul Perroud (France), formerly with the CENG
- R Dr. J. Kentfield (Canada), University of Calgary
- R David Salt (Canada), Indal Technologies
- R H. Abramovich, Israel Institute of Technology
- R Dr. B. Luft (Canada), formerly with the Alberta Solar and Wind Energy Program
- R Marshall Klingensmith, Aluminum Company of America

The following six individuals and organizations have replied to the inquiry but have indicated that they will definitely not or will likely not provide any information.

Flowind Corporation. Per Bill Archibald (Executive Vice President), FloWind will likely not respond.

Herman Drees (formerly with Flowind) will not respond.

Robert Watson (Sandia National Laboratories and formerly with FloWind) will not respond.

Institut de recherche de Hydro Quebec (IREQ) of Canada. Per Albert Watts they will provide only information already in the public domain

W. Brandt Goldsworthy (USA). A possible conflict of interest prevents discussions on this matter.

R.A.N.N., Inc., Dr. Al Eggers. RANN cannot respond due to corporate policy.

1.3 Results of Discussions - Highlights

C. F. Wood (formerly Vice President of Engineering, Indal Technologies, Canada).

Indal Technologies innovations included strut dampers, blade aerodynamic brakes, integrated brake/gearbox/hydraulic rotor support, a relay logic approach to the control system, bolted blade joints including adhesive, floating upper bearing system, pinned strut to blade connections, rotor lower pivot bearing/coupling system and the three paired guy cable system. Three bladed rotors were abandoned mostly due to installation problems.

Power transmission gear type coupling failures occurred due to torque transients during shifting (open transition) from wye to delta electrical connection configuration. This problem was identified and tested/modelled by IREQ for the Indal 50 kW turbine.

Rotor strut dampers were found to be troublesome and fairly ineffective.

The series 6400 turbine (rated at 500 kW, 24 m diameter) incorporated Indal's learning up to that point, including the experience with the Magdalen Islands 24m turbine, which Indal had built for the NRC. The 6400 was designed to capture part of the windfarm market in the 1982-85 period. During that time, manufacturers like Howden and Westinghouse were having to invest in their own windfarms and had to consider installations of at least 50 turbines.

Peter Jaeggin (Adecon Energy Systems, Canada).

The performance of the Adecon 19m turbine is summarized in a published report that was turned over to the interviewer.

The Adecon SL55 turbine was designed for a belt drive transmission but that project is on hold. This machine is planned to have four blades with the blades connected circumferentially at the strut level. The design survival wind speed is 60 m/s with a safety factor of 1.25.

The Adecon SL38 turbine has the same high speed brake system as the 19m turbine (single caliper disc brake), the same NACA 0018 airfoil, and the same 71.1 cm chord blade. The blades are multi-die aluminum extrusions. It has three rigid support legs with lateral support. The control system is relay based. The total cost of the machine before installation is $750 to $800 CDN per kW.

Peter South (formerly with Adecon Energy Systems and the National Research Council of Canada).

The drive train cost dominates the turbine cost as the machine size increases. Examples were $35,000 for the SL38 gearbox versus $100,000 probably for the SL55. A corresponding belt drive may be only $20,000.

The interconnected blades approach is superior to hinged blades (at the root attachment). A philosophy of blade replacement every five years was promoted. Steel blades may be a viable option for VAWTs. A design parked survival wind speed of 60 m/s is probably far too conservative for many sites.

Steel truss towers are superior to tubular towers.

John Ereaux (formerly Adecon, Indal and Lavalin).

Design should include the damage tolerance approach. The Darrieus has a better visual appearance than HAWTs. Future designs may include lattice steel columns, concrete power modules, aerobrakes, and elastomeric connections.

Sandia National Laboratories (USA)

It is vital to remember that all current curved blade Darrieus turbines are essentially first generation designs. This type of turbine has not evolved to the degree that HAWTs have. HAWTs and VAWTs have had similar "teething" problems but no systematic comparison of the development of the designs has ever been published.

No fatal defects have been found in the Darrieus curved blade design. HAWTs and VAWTs are equally efficient. The unfamiliarity of VAWTs and the need for more analytical capability has contributed to the slow pace of commercial development. Stated differently, it is possible that the price paid for mechanical simplicity is the need for more sophisticated analytical design methods.

Extruded aluminum blades were finally chosen by Sandia for the research machines because they were the least expensive and the most readily available blades. (About 1/10 the cost of the helicopter type blades that were used first on Sandia's 17m research machine.)

The "ECON16" computer model was developed to do turbine optimization studies. In part, results from this model supported the development of turbines of about 34 ms in diameter. Further development of the model (after over 10 years) is now planned as "ECON90". This is, however, a low priority within the current research program

One bladed MW scale designs were briefly considered. This type of machine is still of interest for large scale turbines, as it is for large scale HAWTs. However, Sandia has no plans to pursue this design approach at this time.

The ground level position of the VAWT power transmission permits much flexibility in design approaches. This inherent advantage has yet to be fully exploited (see also the Peter South belt drive and Indal "bull" gear approaches). The development of simple direct drive multi-generator systems could have a large payoff, especially for megawatt scale turbines.

Aerobrake experiments and pumped blade spoiling experiments were carried out. That work was not continued.

The application of probabilistic methods in design will have more impact than the use of more sophisticated optimization models. Point designs are necessary to explore new concepts. Money would be better spent on developing innovative point designs than on developing optimization codes.

A smaller scale research turbine (less than 15m diameter) may be necessary to cost effectively explore innovative rotor and blade concepts, like the "soft" rotor and "soft" support systems.

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2.0 PERFORMANCE

2.1 Power Output Performance

2.1.1 Introduction

Power output performance testing of Darrieus wind turbines has been carried out for the following purposes:

1. To validate and support the development of performance prediction computer codes. (See Section 2.1.3.10).

2. To establish performance curves for marketing and sales of commercial turbines, including measurements for warranty claims.

3. To investigate the fundamental behavior of this type of rotor, primarily the effects of rotor geometry (ie. number of blades, solidity, height to diameter ratio, and airfoil zero lift drag coefficient).

4. To investigate methods of performance tailoring such as the addition of vortex generators, pumped spoiling, blade offset (a form of pitch variation) and blade camber.

5. To investigate the effects of alternative airfoil sections (including the natural laminar flow airfoils, or NLF) and blades of varying chord and varying airfoil section.

6. To validate and improve power loss models (electrical, aerodynamic and mechanical losses) including generator losses, gearbox and bearing losses, parasite aerodynamic drag losses (struts and connections).

7. To investigate the effects of blade contamination (or soiling) during normal operation due to insects, airborne dirt, leading edge erosion and paint flaking.

Table 2.1.1a (field tests) and Table 2.1.1b (wind tunnel tests) show the turbines for which measured power output performance data have been published.

2.1.2 Test Methods

Sandia National Laboratories (SNL) developed and maintained the standard approach for the field testing of many of the prototype commercial vertical axis wind turbines in North America.

Sandia National Laboratories (Akins 1978) developed the "method of bins" performance data acquisition and reduction method. SNL also systematically investigated the effect of the position of the measuring anemometer with respect to the wind turbine and the method of correcting measured power output to standard sea level conditions. The results of this work were largely incorporated into national and international performance testing standards.

SNL developed its own data acquisition and reduction computer software to run on an HP computer hardware system (Stiefeld 1978). This system was later used by FloWind Corporation, Alcoa, Indal Technologies (through Southern California Edison Company and the National Research Council of Canada) and VAWTPOWER in the field performance and structural testing of their prototype commercial turbines. Sandia provided varying levels of consulting support during these tests.

The data acquisition and reduction approach (software and hardware) were upgraded to support the testing of the 34 m research turbine (Ralph 1990, and Berg, Rumsey, Gallo, and Burwinkle 1988).

Table 2.1.1a Power Output Performance Data Available From Field Tests

Turbine	Country	Shaft Power	Electrical Power	Predicted Power	Aero Model
Sandia 2m Research	USA	x		x	SIMOSS, DARTER, PAREP, Multiple Stream Tube
Sandia 5m Research	USA	x		x	VDART, DARTER, PAREP
Sandia 17m Research	USA	x		x	SIMOSS, DARTER, PAREP
Sandia 34m Research	USA	x		x	SLICEIT
DOE 100kW (17m)	USA	x	x	x	
VAWTPOWER 185	USA		x		
FloWind 120 (17m)	USA		x		
FloWind 17m	USA		x		
FloWind 19m	USA		x x x	x x	DMST, 52 rpm DMST, 35 rpm Clean, Dirty
Indal 50kW (11.2m)	Canada		x		
Indal 6400 (24m)	Canada		x	x	CARDAA
NRC/Hydro Quebec 24m	Canada	x	x	x	VAWT AERO, CARDAAV
Lavalin Eole 64m	Canada		x	x	CARDAA
NRC/DAF 6.1m, 2 blades	Canada	x			
CENG D5 (5m)	France		x	x	
CENG D10 (10m)	France	x	x	x	
TEV 100 (23m)	Romania	x	x		
Pionier I (15m), Cantilever	Holland	x	x	x	RIGID ROTOR (NLR)
Alpha Real 19m	Switzer-land		x		
Adecon A19m-150kW	Canada		x	x	

Table 2.1.1b Power Output Performance Data Available From Wind Tunnel Tests

Turbine	Country	Shaft Power	Electrical Power	Predicted Power	Aero Model
Sandia 2m Research	USA	x		x	SIMOSS, DARTER, PAREP, Multiple Streamtube
NRC 0.76m 1,2,3,4,6 blades	Canada	x			
NRC 3.7m, 1 & 3 blades	Canada	x			
NRC/DAF 4.6m, 3 blades	Canada	x		x	

2.1.3 Performance Data

2.1.3.1 Performance Curves

Introductory Comments

Considerable measured performance data and theoretical predictions exist for Darrieus wind turbines. In the following sections selected data are presented and discussed in an attempt to address the key issues.

The vast majority of performance data available are for two-bladed turbines. The only single blade test data appear to be those for the NRC 0.76 and 3.66 m wind tunnel models, although researchers and designers continue to express interest in exploring the apparent advantages of this approach.

Some three-bladed turbine test data are available as a result of testing by the National Research Council (Canada), Sandia Laboratories (USA), CENG (France), Dornier Deutsche Aerospace (Germany) and Alcoa (USA). The three-bladed approach seems to have been abandoned for large commercial designs mainly due to installation complexity. However this approach offers much reduced power ripple (for conventional constant speed rotors) and the ability to achieve optimum rotor solidity using blades of fairly small chord. As a result, the designer may have more options in blade materials and blade manufacturing methods.

Field verification of the performance of stall regulated wind turbines in the post stall regime has always been difficult and costly because of the generally low frequency of high wind speeds at most test sites. Sandia's 17 m and 34 m (continuously variable speed) research turbines and the NRC/Hydro Quebec 24 m research turbine were operated at a number of rotor speeds. This permitted very high quality data to be gathered over the complete range of tip speed ratios, including the post stall region.

Only Sandia Laboratories and FloWind have reported on the performance effect of soiled blades. It is not clear whether any measures were taken to clean rotor blades during the performance testing reported by others.

National Research Council Wind Tunnel Tests

The National Research Council of Canada's wind tunnel tests included the first known basic performance measurements for this type of wind turbine. The later tests included fundamental investigations of the effect of one, two and three blades, rotor solidity and aerobrakes.
The Canadian NRC performed the first organized performance testing work beginning with small scale wind tunnel models. In 1966 South and Rangi carried out wind tunnel tests of two small VAWT rotors having diameters of about 0.76 m diameter. The blades were aluminum extrusions (NACA 0012) and had a chord of 2.54 cm. Testing was carried out in a 1.83 m x 0.91 m wind tunnel at a constant speed of 5.2 m/s. The test Reynolds number of the blade was only about 20,000 but the results were corrected to a full-scale blade Reynolds number of $0.5x10^6$.

The first rotor had three straight blades. Rotor number two had blades curved in the troposkien shape and was tested with one, two, three, four and six blades (South and Rangi 1971). Figure 2.1.3.1a shows the results uncorrected and corrected to the higher Reynolds number. Note that the power coefficient (C_p) was defined as the shaft power output divided by the available energy in the wind stream. In this case the available energy was taken as the Betz limit, therefore the power coefficients should be multiplied by 16/27 to make them consistent with standard usage.

These data represent the first investigation of the effects of solidity on Darrieus turbine performance. A very respectable maximum C_p of 0.41 was achieved with both the two and three-bladed configurations. It is interesting to note how rapidly the maximum C_p increases with solidity.

A 4.27 m diameter rotor was tested in NRC's 9.1 m by 9.1 m wind tunnel in the early 1970s. The blade chords were 15.2 cm and had the NACA 0012 profile. A maximum power coefficient of 0.37 was reached at a tip speed ratio of around 6 (South and Rangi 1972) for the two bladed configuration. See Figure 2.1.3.1b.

Due to manufacturing imperfections, the three blades were not similar. Therefore tests were carried out with each pair of blades. It was found that one pair of blades produced about 10% more power than the other pairs.

It is interesting to note that South and Rangi also tested the blades with preset pitch (or offset). This was done to determine whether it would be possible to restore the initial angle of incidence to zero to compensate for the angle of incidence of 4° (pointing outward) that resulted from the blade forming operation. These data are discussed in Section 2.1.3.6.

Self starting tests were also carried out and it was found that the rotor would only accelerate up to a tip speed ratio of about 0.5. (This is consistent with later field observations.) When the rotor was assisted to a tip speed ratio of 2.6, it then accelerated rapidly to its free running (runaway) tip speed ratio of 8.6. This is also generally consistent with field observations where it has been noted that wind gusts will occasionally result in a free wheeling rotor suddenly accelerating to the runaway tip speed ratio. The reason for this is probably the fact that due to its own inertia the rotor will not maintain a constant tip speed ratio (of about 0.5) in variable winds and will therefore be operating at a continuously varying tip speed ratio. Furthermore the changes in wind direction can interact with the blade azimuth positions to increase rotor torque and speed even when the average tip speed ratio (for one rotor revolution) is below the critical ratio of about 2.6.

Later tests were performed using a 3.66 m diameter rotor. The rotor was configured for three blades and a single counterbalanced blade. The blade chord was 15.2 cm. The rotor was supported with an external frame and had aerobrakes mounted on the blades. Testing was carried out in the 9.1 m x 9.1 m wind tunnel. The blades were manufactured to loose tolerances using six straight segments to approximate the ideal shape.

The tests included basic power output performance, rotor drag and performance with speed limiting flaps (spoilers). Figures 2.1.3.1c and 2.1.3.1d show the performance data for the single blade and three blade configurations. Figure 2.1.3.1e shows the rotor drag coefficient data. Data on the spoiler tests are included in Section 2.1.3.3.

Sandia Research Turbines

The performance testing of the Sandia National Laboratories' 2, 5, 17 and 34 m research turbines has resulted in the most rigorous and exhaustive set of performance data and comparisons to theoretical predictions. SNL routinely presented test and predicted data in nondimensional form to facilitate comparison with other data, including that for HAWTs.

The 34 m test data are discussed in the section on NLF airfoil sections and tapered blades (Section 2.1.3.2).

2 Meter Wind Tunnel and Field Test Turbine

The 2 m turbine was tested both in the wind tunnel and in the field (Blackwell, Sheldahl and Feltz 1976 and Sheldahl 1981). The wind tunnel tests were carried out in the Vought Systems Division Low Speed Wind Tunnel (4.6 x 6.1m).

Both two and three blade configurations were tested at chord Reynolds numbers up to 290,000. The test matrix is reproduced in Figure 2.1.3.1f. Note that the effect of the number of blades at the same solidity was investigated. The blades were machined from aluminum. Blade chords were 59, 73 and 88 mm.

The maximum power coefficient achieved by any of the configurations in the wind tunnel was about 0.35, which was consistent with the NRC tests. At the same solidity and Reynolds number, increasing the number of blades improved the performance at lower tip speed ratios.

The effect of increasing solidity on turbine performance is shown in Figure 2.1.3.1g and the effect of increasing Reynolds number is shown in Figure 2.1.3.1h.

The wind tunnel and field data are compared in Figure 2.1.3.1i for Reynolds numbers of $1.3x10^5$ and $1.5x10^5$. The effect of Reynolds number on maximum power coefficient is again quite evident.

Sheldahl (1981) suggested that the improved field performance at tip speed ratios greater than five might be the result of the wind tunnel blockage correction factor used or it might be a real (but unexplained) difference. The wind tunnel data were compared to the NRC wind tunnel data for the NRC 3.66 m rotor. (See Figure 2.1.3.1j.) The main problem in comparing the data is the unknown combination of wind tunnel speed and rotor speed for the NRC data.

No further field tests of the 2m turbine were carried out, as it was concluded that the credibility of the wind tunnel data had been established.

Sandia 5 Meter Research Turbine

The Sandia 5 m turbine was fabricated in 1974 and operated with two different types of blades. The first tests were carried out with blades consisting of three sections with a circular arc near the equator and straight sections near the mast. (Sheldahl and Blackwell 1977). The sections (NACA 0012 profile) were joined by attachment knuckles. The blade chord was 19 cm for the curved section and 10 cm for the root sections.

It was later determined that this method of blade construction was detrimental to performance. (Sandia's 34 m turbine also uses a type of knuckle to join the blade sections and these knuckles have been the subject of considerable research aimed at reducing their detrimental effect.)

The second set of tests were performed with one-piece blades made of aluminum extrusions having a chord of 15.2 cm (Sheldahl, Klimas and Feltz 1980). The blades were bent to the curved blade shape by incremental bending and then stress-relieved. Tests were carried out with both two and three-blade configurations.

The highest performance (C_p of 0.392) was obtained with three blades at a rotational speed of 150 rpm. This compares to a maximum of 0.273 obtained with the three piece blade configuration during the earlier round of tests. (See Figure 2.1.3.1k.) It was pointed out that this improvement agreed with the wind tunnel results from the 2 m turbine which had continuous blades.

The test Reynolds numbers were limited to about 300,000 by torque limitations on the timing belt and the induction generator. The authors stated that a power coefficient of 0.4 would have been exceeded if the Reynolds number could have been increased.

NRC/Hydro Quebec Magdalen Islands 24 Meter Research Turbine

Performance test data for this turbine operating at 29.4 rpm are shown in Figure 2.1.3.1l (Penna and Kuzina 1984). These data are believed to be the first field data gathered on large scale Darrieus turbines that clearly show the performance in the post stall regime (low tip speed ratios). A complete data set for operation at 36.6 rpm could not be obtained because high wind operation was limited to about 15 m/s. The performance data gathered from this turbine were an important element in the design of the Indal 6400-500kW turbine since the effects of dynamic stall were not included in performance prediction models and peak power output was seriously underestimated by the models. Figure 2.1.3.1m shows the CARDAAV double multiple stream tube model predictions with and without dynamic stall included.

NRC/DAF 6.1 Meter Research Turbine

One of the 6.1 m turbines was used as part of a wind diesel test system (Schienbein 1979) As part of that test program, the effect of the struts and the strut to blade connections were measured.

One rotor ("Rotor 2") was tested without struts and without blade section joints (continuous blades). "Rotor 3" was equipped with aerodynamic struts and clean blade to strut connections. Figure 2.1.3.1n shows that the performance of the strut free rotor is significantly improved over that for the rotor equipped with struts.

Lavalin Eole (64m) Research Turbine

Figures 2.1.3.1o and 2.1.3.1p show the measured electrical power output performance for the Lavalin Eole turbine at fixed rotor speeds of 10 and 11.35 rpm respectively. The maximum power output of in excess of 1.3 MW at about 14.7 m/s (11.35 rpm) is by far the greatest measured power output for any Darrieus wind turbine constructed to date. Measured performance is seen to fall well short of the predicted performance, however. No explanation for this was offered.

The turbine was designed to operate in a variable speed mode up to a rotor speed of 16.3 rpm with the maximum power reaching about 3.6 MW at 17 m/s and then being held constant by decreasing rotor speed at higher wind speeds (Richards 1987). However, fatigue life predictions showed that the turbine should be limited to 13.25 rpm with a nominal cut-out of 15 m/s (about 2 MW maximum power output) in order to operate successfully for the five year duration of the energy purchase agreement. Figure 2.1.3.1q shows the fatigue life curves overlaid on the operating characteristics.

Pionier I (15 Meter) Cantilevered Rotor Research Turbine

This experimental turbine was installed in 1982 and is the only large scale cantilever supported Darrieus wind turbine known to have been constructed. The turbine can operate at both constant and variable speed driving a 93.8 kW DC generator coupled to an inverter.

Measured shaft power coefficient data are shown in Figure 2.1.3.1r along with calculated values using the VAWT AERO and RIGID ROTOR computer codes (Machielse and de Groot 1986). The measurements are in better agreement with the predictions for VAWT AERO because VAWT AERO includes dynamic stall and Reynolds Number effects and is based on double multiple stream tubes.

The measured energy conversion efficiency from shaft power into AC electrical power was 81%. However, with improvements to the generator, inverter and gear box, this was expected to improve to 90%.

Sandia 17 Meter Research Turbine

The Sandia 17 m turbine was tested in four configurations as described in Table 2.1.3.1 (Johnston 1982). The strutted configurations used an "X-brace" strut.

Table 2.1.3.1 Sandia 17 M Turbine Rotor Configurations

Configuration	**Number of Blades**	**Blade Construction**	**Blade Profile**
1. Struts	2	Composite, 0.533 m chord	NACA 0012

Configuration	Number of Blades	Blade Construction	Blade Profile
2. Struts	3	Composite, 0.533 m chord	NACA 0012
3. No Struts	2	Aluminum Extrusion, 0.61 m chord	NACA 0015
4. No Struts	2	Aluminum Extrusion, 0.61 m chord	Sandia NLF, NACA 0015

Measured performance results for configuration 4 are discussed in the next section. Figure 2.1.3.1s shows the effect of the struts on the measured performance for the two blade configurations (Johnston 1982). Note that in addition to the lack of struts, configuration 3 uses blades having the NACA 00015 profile with a chord of 0.61 m. The power loss due to the struts only could not be measured. The effect of the increase in chord length alone could be expected to increase the peak power output by about 14%.

Figure 2.1.3.1t compares the measured performance for configurations 1 and 2 at two rotor speeds (Worstell 1980). These configurations differ only in the number of blades (or solidity). For the same rotor speed it is clear that:

1. The peak turbine power output is higher for three blades than two.

2. The three-bladed rotor exhibits higher power output at high winds and lower output at low winds when compared to the two-bladed rotor.

Both results are predicted by the aerodynamic models.

The maximum performance coefficient (K_{pmax}) was found to increase with increasing Reynolds number (increasing rotor speed in this case). The maximum power coefficient (or aerodynamic efficiency) was roughly the same for both the two and three bladed configurations and occurred at a rotor speed of about 37 rpm (Worstell 1978). See Figure 2.1.3.1u.

Worstell (1981) reported a maximum rotor power coefficient of 0.467 for the 17 m research turbine operating in configuration 3 at 38.7 rpm. Maydew and Klimas (1981) compared the 17 m measured performance to that of the NASA Mod-1 (60.7 m) and the Mod-0A (37.8 m) horizontal axis wind turbines (Figure 2.1.3.1v). They concluded that the maximum average measured power coefficients of the VAWT are about 0 to 15% higher than those of the HAWTs. They concluded that generalizations which refer to the Darrieus VAWT as aerodynamically less efficient than the HAWT must be used carefully.

DOE 100 kW (17m) Commercial Prototype Turbine

Measured shaft power output for the DOE 100 kW turbine is shown in Figure 2.1.3.1w (Nellums 1985). The maximum performance coefficient based on shaft power output was 0.41 at the rotor operating speed of 48.1 rpm.

FloWind 17m and 19m Commercial Turbines

The measured electrical power output for the FloWind 19 m turbine with clean and dirty blades is shown in Section 2.1.3.2.

The electrical power output performance for several FloWind 17 m turbines were measured during an array wake study (Liu et al. 1987). Figure 2.1.3.1x shows the results for three upwind turbines in the test array. The differences in the power output at which the measured performance begins to deviate from the published curve may be the result of differences in the amount of dirt and insect accumulation on the blades.

Indal Technologies 50 kW (11.2 m) and 6400/500kW (24 m) Commercial Prototype Turbines

Schienbein (1980) compared two methods of performance data analysis for the Indal Technologies 50 kW (11.2 m) turbine. He found good agreement between the results of the frequency matching approach and the results of binning. See Figure 2.1.3.1y. Insufficient data were obtained for wind speeds greater than about 15 m/s (tip speed ratios less than about 3) so the maximum power output could not be established.

Wehrey et al. (1985) measured the performance of the Indal Technologies 6400/500 kW turbine. Figure 2.1.3.1z shows reasonably good agreement between the measured and predicted performance at standard sea level conditions. (The measured performance is based on binned three minute averages recorded during 511.8 hours of operation.) The highest recorded three minute average power output was 530.8 kW at 22.6 m/w at the site air density or about 565 kW at standard sea level conditions, providing further support for the predicted performance.

2.1.3.2 NLF (Tailored) Airfoils and Tapered Blades
Background

Commercial Darrieus VAWTs have used blade airfoil sections that were originally designed for aircraft (commonly the NACA 0015 or NACA 0018 symmetrical profiles). However, the operating regime of Darrieus VAWT blades is quite different from that experienced by airfoil sections in aircraft applications (Berg 1990).

VAWT blade sections operate in inherently unsteady conditions due to their rotation in the horizontal plane. VAWT (and HAWT) blade elements often operate in stall whereas stall is assiduously avoided for aircraft. VAWT blades operate at Reynolds numbers of between a few hundred thousand and a few million while subsonic aircraft wing sections operate in a range of Reynolds numbers about 10 times greater.

Systems studies carried out by Kadlec (1978) demonstrated that there was significant potential for decreasing the cost of energy of the Darrieus VAWT by using specially designed airfoil sections in place of the older airfoil sections borrowed from aircraft. Further studies revealed that the cost of energy is reduced when the blade profile in the equatorial region exhibits the following features:

1. Modest values of maximum lift coefficient (C_L).

2. Low drag at low angles of attack and high drag at high angles of attack (α).

3. Sharp stall, and

4. Low thickness to chord ratio.

Blade elements closer to the tower (or central column) should have:

1. High maximum lift coefficients.

2. Gentle stall, and

3. High thickness to chord ratio.

In 1980 Sandia National Laboratories began a program to develop a family of airfoil sections specifically tailored for use in the Darrieus VAWT environment. The airfoil design code PROFILE was used to design airfoil sections which would exhibit the desired characteristics. That program has resulted in the development of the Sandia family of NLF (natural laminar flow) airfoils and their use in conjunction with the thicker NACA 00XX airfoil sections in tapered blades (Gregorek and Klimas, 1985).

The performance of NLF airfoils depends on their ability to maintain a significant extent of laminar flow over a range of angles of attack. For example Gregorek and Klimas (1985) reported that laminar flow extended to as much as 60 to 65% of the chord for the Sandia 0021/50 and 0018/50 airfoils. Sandia believed that in commercial manufacturing practice the blade profile could be held close enough to design specifications to generate a sustained extent of laminar flow and that the blade surface finish would be sufficiently smooth to delay premature transition to turbulent flow.

Test data for the Sandia 34 m rotor have demonstrated that these NLF airfoil sections perform very close to predictions. Earlier test results for the 5 m turbine were disappointing. However the poor performance was attributed to the very low Reynolds numbers associated with the small scale.

Recent test results for the 34 m turbine (Dodd et al. 1991) show a loss of power limiting behavior when the blades are soiled by insect debris. This is a surprising result since it runs counter to predictions and wind tunnel test results for artificially roughened blades. Further wind tunnel tests and analyses are in progress to find an explanation. The 34 m turbine test results are discussed later.

Wind Tunnel Tests

The Ohio State University Aeronautical and Astronautical Research Laboratory performed wind tunnel tests to measure the aerodynamic coefficients of the Sandia 0018/50 and Sandia 0021/50 NLF airfoils and the NACA 0015 and NACA 0021 airfoils in both the static and oscillating modes (Gregorek, Hoffman and Berchak, 1989). The results of the tests are documented in a set of four reports. Table 2.1.3.2a summarizes the main test conditions.

Table 2.1.3.2a Ohio State Wind Tunnel Tests

Airfoil Type	Reynolds Number	Mach No.	Static Tests Angles of Attack	Oscillatory Tests Angles of Attack
Sandia 0021/50	10^6	0.15	-2° to +45°	±10°, ±12°, ±15°, ±20°, ±30°
Sandia 0018/50	1.3 to 2.5×10^6	0.2 to 0.3	-2° to +30°	Amplitude Ranges Up To 45°

Airfoil Type	Reynolds Number	Mach No.	Static Tests Angles of Attack	Oscillatory Tests Angles of Attack
NACA 0021	$1.5x10^6$	0.2	-2° to $+45^\circ$	$\pm5^\circ$, $\pm10^\circ$, $\pm14^\circ$, $\pm20^\circ$, $\pm30^\circ$, $\pm45^\circ$
NACA 0015	$1.5x10^6$	0.2	-2° to $+45^\circ$	Same as NACA 0021

Tests utilized the 0.15 m by 0.30 m and the 0.15 m by 0.56 m transonic wind tunnels. The models (0.15 m chord) were equipped with 56 pressure taps to record airfoil chordwise pressure distributions.

A study of trailing edge treatment was carried out for the SAND 0018/50 using extruded aluminum models. The results of the study are summarized in Figure 2.1.3.2a. It can be seen that the sharp trailing edge significantly reduced the minimum drag coefficient (for symmetrical airfoil sections the drag coefficient at zero lift, or C_{do}). Since the performance of Darrieus rotors is quite sensitive to C_{do}, these data are important to blade designers. Although sharp trailing edges are often difficult and costly to achieve in commercial blade manufacturing, detailed cost and performance analyses are probably warranted.

A study of blade roughness was also carried out using the extruded aluminum models. Carborundum grit of 0.15 mm was distributed around the airfoil leading edge extending to 10% of the chord on the upper and lower surfaces (see Figure 2.1.3.2b). Significant sensitivity to roughness is evident. However the SAND 0018/50 is less sensitive to roughness than the NACA 0018 as shown in Figure 2.1.3.2c, taken from Berg (1990). Figure 2.1.3.2d shows data for the Sandia 0015/47 and NACA 0015 airfoils. The NLF Sandia 0015/47 is less sensitive to roughness than the NACA 0015.

Test Results - Sandia 5 m Research Turbine
Sandia Laboratories carried out the first tests of NLF blades on the 5 m research turbine (Klimas 1984a and 1984b). Both the SAND 0015/47 and the SAND 0018/50 airfoil sections were tested. The results indicated that when compared to the NACA 0015 bladed version of the 5m rotor:

1. Efficiencies (C_p) remained nearly constant over a wide range of tip speed ratios.

2. Maximum power level was lower.

3. Cut-in tip speed ratio was the same.

4. Peak efficiency was lower.

The first two results were considered favorable. The third result was thought to be primarily the result of the low Reynolds numbers. The last result was believed to be correctable by increasing the drag bucket by adding thickness to the airfoil. This gave rise to the SAND 0018/50. Blades of this section were tested on the 5 m turbine. Results were again disappointing. However it was clear that the low Reynolds number was definitely to blame (excessive flow separation). Nevertheless the test results showed that these blade sections could reduce the peak power output while maintaining the performance at lower wind speeds. See Figure 2.1.3.2e and 2.1.3.2f.

An unexpected favorable result of the testing was that the measured lead/lag stresses in the blade did not increase with wind speed as rapidly as with the NACA 0015 section. See Figure 2.1.3.2g.

Test Results - Sandia 17 Meter Research Turbine

Further testing was carried out by Sandia using the 17 m research turbine with two blades having chords of 0.61 m (Gregorek and Klimas 1985, and Klimas and Worstell 1986). The blade sections near the root used the NACA 0015 airfoil and the Sandia 0018/50 airfoil was used in the center portion. Figure 2.1.3.2h (Klimas 1985) shows the test results for this configuration and for the same turbine equipped with blades having the NACA 0015 profile only. The stall regulation effect at 50.6 rpm is clearly shown.

Test Results - Sandia 34 Meter Turbine

The Sandia 34 m turbine was the first curved blade Darrieus turbine rotor originally designed to incorporate step tapered blades using varying blade section profiles and a blade airfoil section specifically designed for VAWTs. The equator and transition sections of that rotor use the SAND 0018/50 airfoil section while the root sections are NACA 0021.

The blade sections were fabricated of multiple aluminum extrusions (two or three) joined along the span. (See Figures 2.1.3.2i and 2.1.3.2j.) A summary of the blade design details is presented in Table 2.1.3.2b.

The five blade sections per blade were joined together using external joints. (See Figure 2.1.3.2k.) The chord changes abruptly at the joints (hence the term step tapered blade) along with a slope discontinuity. Aerodynamic smoothing compound was used to cover recessed bolt heads, to fair portions of the external blade-to-blade joints into the blades and to protect surface mounted transducers and their associated wiring and completion units. The blades were painted.

Table 2.1.3.2b 34 Meter Turbine Blade Data

Blade Section	Length of Section	Airfoil Section	Airfoil Chord Length	No. of Extrusions
Equatorial, Curved	19.1 m, one per blade	SAND 0018/50	0.91m	2
Transition, Curved	7.5 m, two per blade	SAND 0018/50	1.07 m	2
Root, Straight	9.2 m, two per blade	NACA 0021	1.22 m	3

The first rotor power test results compared poorly with the predicted performance (Berg, Davis and Clark, 1989, Berg 1989, and Berg, Klimas and Stephenson, 1990) using the SLICEIT code (double multiple streamtube conservation of momentum approach based on the CARDAA code). That code used the Gormont dynamic stall model as modified by Masse, except for the NLF sections of the blade.

Inspection of the blades revealed that the paint had flaked at the leading edge of the NLF blade sections. This flaking had created forward facing steps near the leading edge with a height of approximately 0.25 mm. These were believed to be very effective boundary layer trips which could be expected to destroy the laminar flow over the blade and result in higher drag and lower lift than predicted. To correct the problem, the paint was removed from the leading edges for a distance of at least one cm or until an area was reached where the paint adhered well. The bare metal was then faired smoothly into the remained painted surface using emery paper. Following this action, power output performance improved greatly at high winds and modestly at low winds, as shown in Figure 2.1.3.2l (Berg, Klimas and Stephenson 1990). The improvement at low winds was due to a decrease in C_{do} while the improvement at high winds was due to a decrease in C_{do} and an increase in C_{lmax}.

In order to improve the performance further, small aerodynamic fairings were installed on the blade-to-blade joints using a lightweight epoxy material with embedded fiberglass tape (Dodd, Berg, Ashwill, Sutherland and Schluter, 1991, and Berg and Migliore, 1989). The fairings seem to have improved the performance in low wind speeds but degraded the performance slightly at high wind speeds (Figure 2.1.3.2m and 2.1.3.2n). Further improvement was expected with larger teardrop shaped fairings, but these were not pursued due to cost. Three fairing designs, the bare blade and the blade joint were wind tunnel tested (Berg and Migliore 1989).

Simulation of the drag of the blade joints resulted in a significant reduction in the predicted performance of the turbine (the original predictions had not included the effect of the joints). Simulation of the effect of the fairings restored the power output performance to that predicted for the turbine with aerodynamically clean joints. See Figures 2.1.3.2o and 2.1.3.2p.

The maximum power output of the turbine increased following the accumulation of insect debris on the blades (Dodd, Berg, Ashwill, Sutherland and Schluter, 1991). This trend is shown clearly in Figure 2.1.3.2q for a rotor speed of 28 rpm. Maximum power output increased by 15% at 18 rpm. Figure 2.1.3.2r shows that rain washing while the turbine was rotating was far more effective at cleaning the blades than was rain washing while the rotor was parked. Figure 2.1.3.2s shows that rain washing (presumably with rotation) did restore the rotor blades to the effectively clean condition.

Schienbein (1987) presented data (acquired by Pacific Gas and Electric Company) for a FloWind 19m turbine with clean and dirty blades (NACA 0015) operating in the Altamont Pass in California (see Figure 2.1.3.2t). Here the rotor power output remained virtually unaffected up to about 200 kW (80% of the maximum output with clean blades). The maximum power output reached only 220 kW with soiled blades compared to 275 kW with clean blades. Predicted annual energy losses due to dirty blades were found to be only between 2 and 7% for various Altamont Pass wind distributions. This is explained by the fact that the available energy is concentrated in the range between about 12 and 18 m/s for these distributions. It can be seen that the power output is not seriously affected by dirty blades in this wind speed range.

The increase in maximum power output for the 34 m turbine due to blade roughness was not predicted. Indeed, a decrease was expected based on the wind tunnel tests with roughened blades. One theory contends that the presence of the roughness due to insect debris delays blade stall and separation of the boundary layer, leading to higher maximum lift and increased maximum power. However this theory does not seem to explain the fact that the power output performance appears to be otherwise unaffected by the roughness at lower wind speeds.

Research is in progress to investigate the effects of naturally occurring roughness on the performance of the Sandia 0018/50 airfoil section (Reuss, Deliance and Gregorek 1992). Models taken from molds of actual blade debris buildup are to be used in wind tunnel tests. Berg and Stephenson (1992) have reported on the successful use of a laser profilimeter to map actual blade surfaces with a resolution of about 0.01 mm. Some initial work on comparing the characteristics of natural roughness and roughness commonly used in wind tunnel tests (grit) suggests that results obtained using wind tunnel roughness techniques may not be valid for naturally occurring roughness (Eggleston and Moroz 1992).

Economics

Cost and performance studies (for example Malcolm 1986) have shown that the cost of energy can be reduced significantly when the NLF sections are used near the equator in place of the conventional sections such as the NACA 0018. This is especially true for operation in wind regimes having annual average wind speeds of about 6 to 7 m/s, which seems to characterize much of the exploitable wind resource. Malcolm reported a 25% decrease in cost of energy (6.2 m/s average wind speed site) for a step tapered rotor with NLF blades versus the baseline rotor, a conventional constant chord blade rotor with NACA 0018 blades (in this case the Indal 6400/500 kW turbine rotor).

However, Malcolm (1986) reported in the same study that the cost of energy for a rotor equipped with a combination of constant chord SAND 0018/50 and NACA 0018 blade sections was only about 2% greater than for the step tapered configuration. In another study (Malcolm 1987) this difference was about 10%.

Doubts concerning the magnitude of the cost of energy benefit for rotors equipped with NLF blades seem to be raised also by results reported by Berg (1990). He reported that for a rotor equipped with constant chord NACA 0018 blades the energy output was 9.4% less and the cost of energy was only 10% more than that for the baseline case, the Sandia 34 m turbine. (That study was limited to extruded aluminum blades.)

The design constraints and methods of estimating cost are different in each of the studies and this must be carefully considered when interpreting the results. However it may be fair to conclude that rotors equipped with conventional airfoils and constant chord blades could still be made quite competitive with rotors equipped with tapered blades and using NLF airfoil sections. Designers have already developed low cost second generation constant chord blade designs for aluminum extrusions and GFRP pultrusions. Even where the new NLF airfoil sections are used, it is not at all certain that the constant chord blade configuration will be abandoned in new designs.

The studies show that the cost of energy benefit of NLF airfoil rotors is most apparent in the lower wind regimes and those with Weibull distribution shape factors greater than 2 (such as trade wind sites).

2.1.3.3 Aerobrakes (Spoilers)

Although investigations of aerobrakes (or spoilers) are reported in the literature, almost no data appear to have been gathered on the actual performance of turbines with spoilers deployed to validate the effectiveness of the spoilers. Several turbines, including the NRC/Hydro Quebec 24 m and the first DAF Indal 50 kW models were equipped with flat plate spoilers at the equator, hinged at the blade trailing edge. These are known to have deployed many times during rotor overspeed events.

The effectiveness of blade mounted plate type aerobrakes depends mainly on the effective installation radius, the chordwise location of the spoilers, and the total drag area. One measure of effectiveness is the tip speed of the rotor while freewheeling with the aerobrakes deployed in the maximum design wind speed.

The aerobrakes on the NRC 3.66 m wind tunnel model turbine were flat plates (at mid-rotor) and hinged at the blade trailing edge. When opened they were at 90° to the blade chord plane. Their drag was sufficient to negate the rotor torque even with a total aerobrake area of less than one percent of the rotor swept area (see Figure 2.1.3.3a). This will bring a free-wheeling rotor down to a relatively low speed under almost all wind conditions.

Schienbein (1980) reported that, in low wind motoring tests of the DAF Indal 50 kW VAWT with spoilers deployed, the actual mechanical power required to overcome the spoiler drag was 56 kW. This is about equal to the maximum shaft power developed by this rotor at its normal operating speed. Undocumented observations of this rotor confirmed that the rotor did not appear to exceed its design rotational speed during rotor freewheeling with the spoilers deployed. The total spoiler plate area was about 1.5 m^2 or 1.2% of the 121 m^2 swept area. The drag of the spoilers when latched was measured at about only 0.2 kW in no wind motoring tests.

The most extensive investigation of alternative aerobrake configurations was carried out by the NRC for the Project Eole 4 MW rotor design. The tests were carried out in the NRC 2 m x 3 m wind tunnel and the results were reported by Penna (1986) and also by Templin and Rangi (1984). The best aerodynamic configurations were found to be plates hinged to open on both sides of the airfoil about the trailing edge (see Figure 2.1.3.3b.) When open, a large gap was exposed ahead of the spoilers that permitted a strong flow of air from the pressure side of the airfoil to the suction side at high angles of attack. This appeared to solve the problem in stalled flow, where one half of the aerobrake loses most of its effectiveness during a large part of rotor rotation. Aerobrake performance tests on Eole were planned but apparently never carried out (Richards 1988).

Aerobrake instability (tendency to open and close cyclically) was frequently observed on the DAF Indal 50 kW turbines that were equipped with spoilers that did not have the gap ahead of the spoiler, as developed by the NRC for the Eole turbine.

Even without the gap, the spoilers on the NRC/Hydro Quebec (Magdalen Islands) turbine managed to limit the freewheeling such that the tip speed was roughly equal to the wind speed. For example, Templin noted that between 1979 and 1984, the Magdalen Islands turbine had been saved three times by the aerobrakes due to cold-weather mechanical brake problems.

Nevertheless, Indal Technologies abandoned the use of aerobrakes when their hydraulic support/braking system proved to be sufficiently reliable for both normal and emergency braking. Indal found that hinged flat plate equator mounted aerobrakes were difficult to maintain, subject to nuisance deployment, difficult to latch in icing conditions and of limited capability. However, a reliable aerobrake design does offer the potential advantage of lower blade, blade connection and column loads when compared to conventional braking on the low speed shaft.

Testing of the advanced Eole aerobrakes would add significantly to the existing base of wind tunnel and limited full scale test results for the more primitive aerobrake designs. Aerobrakes are discussed further in Section 7.

2.1.3.4 Vortex Generators

The root sections of Darrieus blades operate over a much wider range of angle-of-attack and at lower Reynolds numbers than the equator blade region and are responsible for producing power in low winds.

Although VAWT blade section profile and chord variation can be selected to improve the root section performance at lower tip speed ratios, it is also possible to improve performance through the use of vortex generators.

Vortex generators energize the low energy boundary layer and separated flow regions near the surface of the airfoil and can thereby increase the per-rotation average lift to drag ratio of the root sections over certain ranges of wind speed (or tip speed ratio) for constant speed turbines. Ideally, this selected performance improvement should be accompanied by no increase in peak power output.

The investigation of the use of VGs on VAWTs has been stimulated by three factors:

1. Successful application of simple vane type VGs to the Boeing MOD-2 turbine following wind tunnel tests (Miller 1984). Miller presented a good discussion of the basics of the vortex generator physics for that application.

2. Presumed simplicity and rapid payback of the installation cost, based on the counterrotational (or paired) vane design adopted for the MOD-2 tests.

3. Desire to tune the performance of existing Darrieus turbines to maximize performance at the installation sites following some operating experience, rather than more expensive alternatives such as a change in rotor speed or swept area, for example.

The pitch of Darrieus turbine rotor blades on existing turbines cannot be easily adjusted to tune the rotor for the site wind regime or air density. Blade offsetting (chordwise repositioning of the blade at the root connection) which effectively introduces both pitch and twist is possible. (Blade offset is sometimes called "toe-in" and "toe-out".) Gyatt 1986 stated that for HAWTs, a simple increase in blade pitch angle has the same effect as the addition of vortex generators and that vortex generators can alleviate the sensitivity of wind turbine blades to leading edge roughness caused by bugs and dirt. This suggests that vortex generators could be applied to VAWTs to effectively change the blade pitch.

Wind tunnel tests on the NACA 0015 airfoil section support the benefits of VGs with respect to the dirt and bug problem, when leading edge contamination was simulated by a trip wire at the 5% chord position (Rueger, Spring, and Gregorek 1987b) The findings also support the installation of the VGs at the 30% chord position, rather than at 10%, as had been tried by other researchers on actual turbines.

Sandia and SCE Tests

Vortex generators were extensively tested on the DOE 100 kW turbine and the DAF Indal 50 kW turbine (Klimas 1985, and Scheffler and Quinlan 1990).

Klimas reported a 5% increase in annual energy production for the DOE 100 kW turbine with VGs at the 10% chord position in the Bushland, Texas wind regime with approximately 12% of the blade span fitted with VGs. The VGs were of the counterrotational vane type and installed on the inner and outer blade surfaces starting at each blade root connection. The numbers and location of the VG pairs were determined by simulation using postulated lift/drag characteristics for the NACA 0015 in a set of double multiple streamtube performance calculations. No increase in maximum power output of the turbine was reported.

Scheffler and Quinlan found that for the DAF 50 kW turbine:

1. When VGs were installed only at the rotor equator region on both sides of the blade, peak power output fell by 10%.

2. When VGs were extended toward the roots (4.6 m on either side of the equator), peak power increased by 50% over the unmodified rotor. (See Figure 2.1.3.4a.)

3. When the equator VGs were removed, the peak was reduced but performance in low wind speeds was degraded.

Annual energy production improvement for the SCE installation site (near Palm Springs, California) were -7% for case 1 and +10% for case 3. Scheffler and Quinlan estimated that on a multiple unit basis, total costs for fabrication and installation of the VG strips (case 2) would be less than $1000 US per VAWT (DAF 50). This includes the cost of 12 strips each 3.05 m long at US $17 per strip (1986). All tested VG configurations resulted in a decrease in performance over the unmodified rotor below about 22 kW.

FloWind Analyses

It appears that further work is required to investigate the installation of VGs at 30% chord position, alternative patterns, and including inboard-only placement. Schienbein (1987a) reported that the DMST program coupled with modified NACA 0015 lift and drag characteristics predicted 8% annual energy output increase at a 7.15 m/s average Rayleigh site when VGs were installed over 60% of the blade length (starting at the root) and on the inside surface only of a FloWind 19 m turbine. The model predicted a smaller increase (7%) with the VGs installed on both sides of the blade. (Performance predictions using DMST models are sensitive to the airfoil characteristics assumed and to the dynamic stall model used.) In addition, peak power output was predicted to increase by 15 to 20% with no change in performance below about 12 m/s compared to the no VG baseline case.

Of particular interest is the prediction that performance is essentially unchanged when VGs are installed on the outside of the blade only (60% coverage). This suggests that the application of VGs to Darrieus turbines is complex and that significant experimentation is required.

FloWind is known to be interested in the use of VGs to enhance the performance of its fleet but to date no test results have been reported.

Pitching Airfoil Tests with Vortex Generators

Rueger and Gregorek (1991) carried out testing of the NACA 0021 airfoil with VGs undergoing large amplitude pitch oscillations similar to those seen by a Darrieus rotor blade section. The tests were designed to determine the effects of vortex generators on the lift, drag and pitching moment of a wind turbine airfoil oscillating over a range of frequencies and amplitudes. Data are for Reynolds number of about 1.2×10^6.

The significant conclusions and observations were that the vortex generators should be installed at 30% chord rather than 10% chord (the only positions tested). The lift curve slope is greater and the vortex generators remain more effective to higher angles of attack under oscillating conditions when installed at

30% chord.

2.1.3.5 Pumped Spoiling

Background

Pumped spoiling to control aerodynamic stall and hence peak power output was investigated by Sandia National Laboratories (Klimas and Sladky 1985a). This work was stimulated by SNL's system economic simulation results which showed that 10 to 15% reductions in the cost-of-energy are possible by lowering stall regulation power levels. Although this can be achieved by using specially designed airfoil sections and rotor blade geometries (discussed previously) the interest in pumped spoiling could be rekindled for three reasons:

1. Hollow blades (such as extrusions and GFRP pultrusions) remain attractive for Darrieus curved blade designs and these may permit cost effective pumped spoiling systems to be installed.

2. Stall regulation of the SAND 0018/50 airfoil section (designed specifically for this purpose) has been shown to be seriously harmed by bug build-up near the leading edge.

3. SNL and its contractors have carried out enough wind tunnel and turbine testing to permit full scale trials to be readily carried out.

Performance Tests - 5 Meter Research Turbine

Proof of concept tests were carried out on Sandia's 5m research turbine (Klimas and Sladky 1985a). The two blades had a chord of 15 cm and used the NACA 0015 airfoil section. Approximately 2000 total perforations of 1.25 mm diameter and 3.18 mm on centers were drilled at the 0.4 chord location on each side of each blade in equal portions above and below the equator of the turbine. The turbine was run at 175 rpm with various combinations of blade ends left open (to receive air) or taped closed. Figure 2.1.3.5a shows the test results for all blade ends open relative to those with all blade ends closed. It can be seen that the peak power output was reduced. No difference in power production were noted with blade ends closed relative to power measured on the same turbine prior to perforating the blades.

Design Considerations

The practical use of pumped spoiling is fairly complicated because:

1. The location of the jets is dependent on the airfoil characteristics, specifically the airfoil pressure distribution on the suction side. For example it was found that blowing was only effective at the 7.5% chord location on the SAND 0018/50 NLF due to the flat pressure distribution at 30% and 50% chord. On the SNL 5m turbine tests, spoiling had been effective at 40% chord on the NACA 0015.

2. If the perforations are installed on both sides of the airfoil and fed by the same internal duct, spoiling will occur with no centrifugally induced flow ("passive spoiling"). This "cross-talk" extends to perforation rows at different chord locations (Sladky, 1987).

The exact shape of the perforations to optimize the effect has not yet been determined. Furthermore the perforations must remain clean and open after prolonged operation and not increase the blade drag when pumped spoiling is not active.

3. For centrifugally pumped spoiling (without a compressor), the driving pressure is about equal to the dynamic pressure seen by the blade at the openings. This places an upper bound on the potential spoiling power.

Klimas and Sladky (ASME paper) suggested some intriguing variations on the scheme. For example, as an overspeed control, a compressed air bottle could be made to discharge to stop the rotor. They also suggest that if water can be introduced, the rotor moment of inertial change will supplement the aerodynamic spoiling due to the water jet.

Work aimed at developing a sound theoretical design basis should be carried out because pumped spoiling for rotor power regulation and overspeed control has promise for HAWTs as well as VAWTs.

2.1.3.6 Blade Pitch

The effect of blade twist has not been extensively investigated.

Sandia and NRC Tests

Sandia National Laboratories tested blade offset (effective pitch) on the 5 m research turbine and reported large shifts in runaway tip speed ratio, peak power coefficient (efficiency) and maximum power output for effective pitch changes of only one or two degrees (Klimas and Worstell 1981).

Effective pitch angle (at the equator section) ranged from -7° to +3° (7° toe-out to 3° toe-in). Reynolds number was $3.5x10^5$ based on equatorial radius. (Note that the fixed offset method of changing pitch results in varying pitch over the length of the blade due to blade curvature. Pitch decreases from the reference values at the equator as the blade curves to the root angle with respect to the mast.)

Figure 2.1.3.6a shows the performance data for various effective pitch angles. The best performance seems to be for angles of -0.5° and -2° with the performance deteriorating rapidly as pitch either increases or decreases. South and Rangi (1972) reported that the maximum power coefficient increased from 0.344 to 0.367 when the pitch changed from +4° (in this case 4° toe-out) to 0 on the NRC 4.27 m wind tunnel research rotor. This improvement in power coefficient with decreasing pitch magnitude is consistent with the Sandia results.

Klimas and Worstell pointed out that the aerodynamic performance is changed by two sources:

1. The change in incidence angle as a function of circumferential position relative to the free stream wind is changed resulting in altered blade torque variation.
2. The blade's normal force now is able to contribute to the turbine torque since the blade mounting position is significantly offset from the center of pressure for normal forces.

Although not extensively investigated, these results suggest that blade offset could be a powerful tool for the rotor designer.

2.1.3.7 Blade Camber

Blade camber effects on the performance of curved blade Darrieus rotors have not been extensively investigated. Table 2.1.3.7 shows the measured performance characteristics for the Sandia 5 m research turbine (Johnston 1982) using the 0.15 m NACA 0015 blades and the NACA 1515 blades (1% camber at mid chord).

Table 2.1.3.7 Performance Comparison Between Cambered and Symmetrical Blade Section Operation of the Sandia 5 Meter Research Turbine

	Blade	
Characteristic	**NACA 0015**	**NACA 1515**
C_{pmax}	0.328	0.378
K_{pmax}	0.0060	0.0055
X at C_{pmax}	5.5	5.7
X at K_{pmax}	3.2	2.9
X at $C_p = 0$	9.7	10.2

X is the tip speed ratio (ratio of rotor tip speed at the equator to wind speed.
C_{pmax} is the maximum power coefficient (maximum efficiency).
K_{pmax} is the maximum performance coefficient (proportional to maximum power output).
C_p is the power coefficient (efficiency).

All of the performance characteristics of the NACA 1515 rotor, except for the decrease in X at K_{pmax}, are improved over the NACA 0015 rotor. In addition, there is a more gradual transition to stall regulation with the NACA 1515 blades (see Figure 2.1.3.7a).

It can be concluded that blade camber as a means of improving performance should be investigated further.

2.1.3.8 Blade Roughness (Soiling) Effects, Blade Icing Effects and Parasite Drag

Blade Roughness

Measured performance data for Darrieus rotors with blade roughness due to dirt and bug accumulation are sparse. These data have been discussed in Section 2.1.3.2 (Sandia 34m test turbine and the FloWind 19m turbine).

The field test data show that for the NACA 0015 airfoil section the maximum power output decreases with soiled blades (FloWind 19m turbine). This is consistent with the experience with HAWTs. However the 34m test data show an increase in peak power output for a rotor equipped with soiled NLF blades. This surprising result is the subject of current research.

Icing

Very little quantitative information has been published on the effects of ice accumulation on curved blade Darrieus wind turbine behavior.

FloWind's turbines operating near Tehachapi, California at an elevation of about 1400 m ASL are known to experience blade icing during foggy conditions when the ambient temperature is below freezing. Im (1988) has reported on one event with a FloWind 17m turbine. Turbine power output and rotor stress data were acquired from an instrumented turbine during both ice free and iced conditions. The operating conditions and the measured power output are shown in Table 2.1.3.8. The wind speed measured with the rotor iced may be inaccurate since cup anemometers are greatly affected by ice accumulation.

Table 2.1.3.8 FloWind 17 Meter Turbine Icing Data

Parameter	**Ice Free Rotor**	**Iced Rotor**
Turbine Number	151	151
Location	Tehachapi	Tehachapi
Date of Test	86/2/5	86/2/6
Temperature	-0.1°C	-0.4°C
Atmospheric Pressure	853 mbar	848 mbar
Average Wind Speed	17.3 m/s	16.7 m/s
Standard Deviation of Wind Speed	2.64 m/s	2.54 m/s
Average Power Output	115.6 kW	-34.7 kW

Im apparently calculated the predicted performance for the 17m turbine with iced blades based on an assumed blade profile shape and assumed lift/drag data. Unfortunately no details of this analysis were provided. The predicted power output of -29 kW at 17 m/s is quite close to the measured value of -34.7 kW. This close agreement should probably be considered as serendipitous.

Rangi (1985) and Chappell and Templin (1985) have reported on blade icing events on the Indal 6400/500 kW turbine (which was in operation at the Atlantic Wind Test Site in Prince Edward Island, Canada) and the NRC/Hydro Quebec 24 m test turbine (Magdalen Islands). Both turbines are located in the Gulf of St. Lawrence. This area experiences much freezing rain during the winter months.

In the case of the Magdalen Islands turbine, the ice was formed by freezing rain. The power output dropped from about 230 kW to about 90 kW within 15 minutes. The ice thickness on the blades was not measured but the thickness on a nearby fence had reached about 6 mm.

In the Indal 6400 turbine case, a thin coating of 2 to 3 mm had built up on the blades during freezing rain. Power output dropped from about 250 kW to about 100 kW within one to 1.5 hours.

Since the anemometers would have been affected by the ice in both cases, it appears to have been assumed that the average wind speed had not changed significantly during the icing events. Figure 2.3.1.8a suggests that this was a valid assumption in the AWTS case since the power output rose slowly as the ice was shed (although the temperature did not rise above freezing) but the apparent wind speed remained low.

Rangi also reported on icing of the Indal 50 kW VAWT near Swift Current, Saskatchewan in the western prairies. Due to extremely cold temperatures following freezing rain, the turbine could not operate for one week. The turbine automatically started several times but shut down in each case due to negative power production.

Rangi also reported on three other Indal 50 kW turbine installations in Canada (Pacific coast, coast of Hudson Bay and on the Atlantic coast in Newfoundland). During a period of seven years, the longest time that any machine (of the total of four 50 kW and one 500 kW units) was out of production due to ice was about one week, as discussed. For the other four turbines the ice disappeared in one or two days.

It can be concluded that:

1. Very little ice accumulation on Darrieus turbine blades is sufficient to produce large decreases in power output. This is probably also true for HAWTs.
2. Ice accumulation probably has a negligible effect on annual energy production from Darrieus turbines even in areas prone to severe ice build-up, such as the coast and islands of Atlantic Canada.
3. Control algorithms based on changes in average power output and measured wind speed can probably be devised to safely stop turbines as ice accumulates. Preventing on/off cycling of iced rotors is a more challenging control problem.
4. Anemometer icing can be hazardous in cases where the rotor sheds its ice before the control anemometer (for systems that do not use power output as a control input). It is possible for the wind speed to exceed the turbine high wind shutdown speed with the anemometer slowed due to icing. A control system that uses the inputs from two anemometers and constantly compares them (as implemented by FloWind) may solve this problem.

Parasite Drag

Parasite drag associated with struts, blade-to-blade connections and retracted spoilers has been discussed in Section 2.1.3.1 (NRC/DAF 6.1m turbine and the Sandia 17m turbine struts), Section 2.1.3.2 (Sandia 34m turbine blade to blade joints) and Section 2.1.3.3 (Indal 50 kW turbine spoilers).

2.1.3.9 Blade Surface Pressure Data

Current techniques for predicting aerodynamic performance and structural loads for Darrieus wind turbines use two dimensional airfoil section properties and an empirical model to account for the effects of dynamic stall. (These methods are discussed in Section 2.1.3.10.) Furthermore, these models have been validated only indirectly using measurements of rotor torque and power and blade stresses at a few locations. Blade surface pressure measurements have therefore been viewed as being essential for more effective model validation and to guide improvements in the models.

Akins (1989) and Akins, Klimas and Croll (1983) have reported on the acquisition and interpretation of blade surface pressure data on the Sandia 17m research turbine. Akins indicated that the data were to be used to verify existing aerodynamic models, to improve these models and to provide guidance for future experiments and research.

An experimental program was designed and implemented to measure the surface pressures at a single location (at the equator) on one blade (NACA 0015 profile) of the 17m research turbine. Transducers were distributed along the chord of the blade on both on the inner and outer surfaces. Additional instrumentation was used to measure the incident flow field and the rotor position relative to the incident wind. These measurements were limited to a single rotor configuration and a single operating speed (but a range of tip speed ratios). Reliable measurements were not obtained for the local flow angle using the flow-angularity probe. The blade angle of attack had to be calculated using the fixed wake model

(Wilson and Walker 1981) based on the measured relative rotor position and the measured incident wind speed obtained using the flow-angularity probe. This does not appear to have affected the validity of the main conclusions.

Based on an analysis of the pressure data, Akins (1989) concluded that:

1. There is a definite increase in the normal force coefficients beyond those predicted by static airfoil data. This is an indication of either dynamic stall or a phenomenon similar to dynamic stall. This increase was evident for tip-speed ratios from 2.2 to 3.09. See Figure 2.1.3.9a.

2. The measured pressure distributions differ from two-dimensional data. See Figure 2.1.3.9b.

3. At low tip-speed ratios downwind interference effects were observed. These effects momentarily reduce the normal force coefficient and may be caused by wake crossings. These effects were not observed for tip-speed ratios greater than 3.09. See Figure 2.1.3.9a.

4. Very clear upwind/downwind differences were found.

5. There was no evidence of any increase in normal force coefficient beyond that predicted by two-dimensional static section data for the downwind portion of the rotation of the turbine.

Tangential force coefficients were also calculated from the measured pressure distributions and compared to predictions based on a fixed wake analysis (Wilson and Walker 1981) and static airfoil data and a dynamic stall model based on the Gormont/Templin/ Masse model (Masse 1981). The agreement with the predicted values is poor. See Figure 2.1.3.9c.

Given these results, Akins finally and properly concluded that future pressure distribution measurements on an operating vertical axis wind turbine should be aimed at verifying the anomalous pressure distributions observed in this study. The importance of this cannot be overstated, since most of the recent Darrieus rotor design work has made use of the advanced double multiple stream tube (DMST) performance prediction models that use static airfoil data and empirical dynamic stall models. These models have been able to accurately predict power output performance in many cases. These models have also been used to generate the input loads to the structural response analysis models, such as FFEVD, the results of which have not always agreed with field data (see Section 2.2).

It is possible that the success to date may be the result of "tuning" of the models for rotor designs that use the NACA 0015 and 0018 airfoils based on measured data for several turbines, including the Sandia, NRC and Indal turbines. However, this success is still puzzling, since Akin's measurements were for the NACA 0015 airfoil section.

If Akin's results are accurate, it suggests that the current understanding of Darrieus wind turbine blade aerodynamics is quite poor. This is a sobering realization for turbine designers and those tasked with developing and improving performance and structural loads models. Therefore further pressure measurements on an operating wind turbine are necessary.

2.1.3.10 Performance Prediction Models

Background

The Darrieus wind turbine rotor flow field is far more complex than that of a horizontal axis wind turbine rotor. The analysis of this flow has captured and held the interest of many aerodynamicists throughout the world, as evidenced by the extensive literature on this subject. Darrieus rotor aerodynamic analysis continues as an active area of research in conjunction with structural dynamic analysis.
A complete review of this subject is well beyond the scope of this report.

The aerodynamic analysis of the Darrieus wind turbine has two design objectives:

1. To develop models to predict accurately the aerodynamic performance or power output of the turbine as a function of wind speed.

2. To use the calculated aerodynamic loads to accurately calculate the structural dynamic response of the turbine.

Computer codes based on models that satisfy both objectives and that will run quickly on personal computers (such as the double multiple stream tube model) have been widely adopted by designers and hence have continued to be developed.

Models

The mathematical models that have been developed to predict the performance of VAWTs can be classified as:

1. Momentum models. These replace the rotor with one or two actuator disks, divide the flow field into one or multiple streamtubes and apply the momentum equation in each streamtube (Strickland 1975, Wilson, Lissaman and Walker 1976, Shankar 1976, Templin 1974, Lapin 1975, Paraschivoiu 1981, Loth and McCoy 1983, and Fortunato and De Martino 1989).

   This approach has resulted in the development of a number of computer models that incorporate two actuator disks in tandem, multiple streamtubes, wind shear, dynamic stall and Reynolds number effects (for example Paraschivoiu 1988). These are called the double multiple streamtube models (DMST). Recently, this approach has been extended to include the three dimensional spatially varying incident wind (Brahimi and Paraschivoiu 1991).

   The momentum models continue to be developed and improved.

2. Vortex models where the blades are represented by distributed bound vortices (Holme 1976, Wilson 1978, Fanucci and Walters 1976, Strickland 1980, Strickland, Smith and Sun 1981 and Strickland, Webster and Nguyen 1979). Vortex models can be further classified as free wake and fixed wake (Wilson and Walker 1981).

3. Fluid dynamic models which solve the viscous or inviscid governing equations of the fluid motion with finite volume or finite difference methods (Rajagopalan and Fanucci 1985, Fortunato, Dadone and Trifoni 1991 and Rajagopalan 1986).

4. The local circulation model (LCM) which utilizes a momentum balance between the force on the blade and the change in wind momentum as it passes through the rotor (Azuma and Kimura 1983, and Masse 1986). This is similar to the momentum methods, but the blade is represented as a superposition of imaginary blades of different spans with elliptical circulation distributions. The local circulation model can be forumulated to analyze unsteady flow.

Comparisons

Comparisons presented here are restricted to rotor aerodynamic power output performance. Blade normal and tangential load distributions are not considered here, although their accuracy is vital to structural dynamic analysis.

Figure 2.1.3.10a shows a comparison of the theoretical predictions and experimental performance data for the Sandia 17m research turbine (Paraschivoiu, Fraunie and Beguier 1985). In this figure CARDAAV and CARDAAX are Paraschivoiu's double-multiple streamtube codes (Paraschivoiu and Delclaux 1983, and Paraschivoiu, Fraunie and Beguier 1985). CARDAAX contains lateral streamtube expansion effects. VDART3 is Strickland's three-dimensional free wake code (Strickland, Webster and Nguyen 1979) and MCL is Masse's local circulation method code (Masse 1986). The results for the MCL code show the best overall agreement with measurements.

Figure 2.1.3.10b shows a comparison of the predicted performance based on the fixed wake model (Wilson and Walker 1981) and the measured performance of the Sandia 17m research wind turbine.

Touryan, Strickland and Berg (1987) contend that the local circulation model yields better answers than the momentum models, avoids the convergence problems of the vortex models and, with an appropriate wake model requires far less computer time than the vortex models. Notwithstanding this strong endorsement, the double multiple streamtube momentum models have been used the most extensively for Darrieus wind turbine design.

2.2 Structural Response

2.2.1 Introduction

The ability to predict the full spectrum of loads and stresses at key points in the rotor is essential for the accurate and efficient design of the wind turbine (either VAWT or HAWT). There has been an evolution of mathematical models for the structural response of Darrieus rotors, with increasing sophistication and accuracy. However, the problem is sufficiently complex that complete success is still some distance away.

There are several reasons for the complexity: the first is the need to consider the rotation of the structure; the second is the aerodynamic origin of most of the loads and the reliance on an adequate aerodynamic model; a third is the presence of stochastic behaviour (due to atmospheric turbulence) in addition to the deterministic response. A description of the evolution of the design and analysis methods is presented in Section 3. In this section some description of the validation of these methods is given.

Although there have been approximately 30 different types of Darrieus rotors built and operated, there are very few that have been instrumented and for which the structural response has been identified for certain operating or other conditions. There are even fewer for which the measured response has been systematically compared with predictions of mathematical models and the results made public. The following machines can be (partially) included in the latter category.

NRC/Hydro Quebec (Magdalen Islands) 24m
DOE 100kW (17m)
FloWind 17m
Pionier I
Indal 6400
Sandia/DOE 34m Test Bed

Structural response data for several other machines may exist (FloWind 19m, Alpha Real, DZ12, Adecon 19m, Adecon SL380) but the data are either not in the public domain or have not been made available for other reasons. Some information is available on the structural response of the VAWTPOWER 185 in the appendices of Vosberg 1986.

The type and extent of the information collected and disseminated also varies considerably. The data may fall into the following categories.

operating	measured ("field")
	estimated ("model")
stationary	measured
	estimated
braking	measured
	estimated

Table 2.2.1 summarizes the data available for these machines.

Table 2.2.1. Structural Response Data Available

machine	**operating stresses**	**operating modes**	**stationary stresses**	**stationary modes**	**braking stresses**
NRC/Hydro Quebec 24m	field	field model		field model	field
DOE 100kW (17m)	field model	model		model	
FloWind 17m	field	-------------not publicly available------------			
Pionier I	field model	field model			
Indal 6400	field model	field model		field model	field
SNL/DOE 34m	field model	field model		field model	

2.2.2 NRC/Hydro Quebec 24m (Magdalen Islands)

Much of the structural response data for the NRC/Hydro Quebec 24m machine is found in Penna & Kuzina 1984. The rotor was well instrumented, with strain gauges on the blades at the root, at the strut-blade connection and at mid-rotor. In addition there were strain gauges on the central column, accelerometers at the upper bearing and at mid-rotor, and displacement transducers at the strut-column connection.

A great deal of data was collected and stored on analogue tape which is catalogued in Penna & Kuzina 1984. The data covers the various conditions of possible operation: at 29.4 and 36.6 rpm, and either with the strut-column connection fixed, free, damped or double damped. However, the interpretation and evaluation of the data is not so extensive. A sample of two of the spectra from blade leading edge strain gauges is shown in Figure 2.2.1. In Figure 2.2.2 the variation with wind speed of "peak-to-peak" stresses is illustrated.

While this collection of data did not include any comparison with mathematical models, it did provide field data for later comparison. It also showed how sensitive the response could be to the type of restraint offered to the blade and struts. A comparison of the predicted and measured natural frequencies was presented in DAF Indal 1984c. The agreement with the NASTRAN-based technique developed at Sandia National Laboratories was shown to be very good (see Figure 2.2.3).

The same report (DAF Indal 1984c) concluded that the pseudo-static approach to the modelling of cyclic stresses in the blades predicted values significantly lower than those measured.

2.2.3 DOE 100kW (17m "low cost")

Three of these machines were installed between 1980 and 1981. The machine installed at Bushland, Texas, was the most thoroughly instrumented and tested. It was operated at several rotor speeds but mainly at 48.1 rpm, since severe resonance was encountered at 51.5 rpm. Information on the estimated natural frequencies can be found in Nellums 1985 and Lobitz & Sullivan 1984. There appears to have been no effort to measure the operating natural frequencies on this machine but the mathematical method was instead validated on other machines (Carne et al. 1982).

Lobitz & Sullivan 1984 present measured and estimated rms stresses for two different cable stiffnesses at a number of locations including the central column, blade root edgewise, and the blade root flatwise stresses. Figure 2.2.4 shows some of these results. In general the procedure using the aerodynamic code CARDAA (a double multiple streamtube model with dynamic stall) and the NASTRAN-based frequency response tends to underpredict cyclic stresses at low wind speeds and to considerably overpredict at high wind speeds.

The evaluation of the Sandia 17m rotor cyclic stresses did not examine the contribution of the individual harmonics but combined them all into a single rms. It did recognize that a considerable amount of the cyclic response was contributed at non harmonic frequencies and must be due to non-steady effects.

Some measurements of the natural frequencies of the stationary rotor and also some values of the mean blade stresses under operating conditions are included in Lobitz & Sullivan 1984.

2.2.4 Pionier I

The Pionier I is the only non-North American machine on which much information is available. It was funded and operated through the Energy Research Foundation (ECN) of the Netherlands. It is also the only cantilever Darrieus rotor that has been widely described in the public literature.

Machielse, Rieffe & Peeters 1986a,b gives results of the measured and estimated natural frequencies of the stationary and operating rotor. Figure 2.2.5 shows that the NASTRAN-based procedure used to calculate the natural frequencies underpredicted the fundamental blade flatwise and blade out-of-plane modes by about 10%. The fundamental "tower" mode is considerably overpredicted (possibly because it assumed a rigidly mounted cantilever instead of one mounted on a floating pontoon).

The same report gives one example of a spectrum of blade flatwise bending but does not compare it with any estimated values.

The paper by Machielse & deGroot 1986 gives some more information on the structural response. It gives the measured "effective" flatwise and edgewise bending moments at the blade root at increasing wind speed and also gives some estimated results (from a single streamtube model) as shown in Figure 2.2.6. Unfortunately the measurements extend to wind speeds of only 12m/s whereas the estimated values are for wind speeds of 15 and 18m/s. The latter were calculated using a single streamtube aerodynamic model (which Sandia, in their own studies, found to seriously underestimate the response).

2.2.5 FloWind 17m

Although the FloWind 17m was the first Darrieus rotor wind turbine to be successfully commercialized, there is little publicly available information on its structural response. A great deal of information was collected through instrumentation and with the assistance of Sandia National Laboratories, but this has remained largely proprietary.

The FloWind 17m was based on the DOE 100kW (17m "low cost") design but did have a number of structurally significant alterations. One difference was the addition of horizontal struts made of bars and linkages. These were designed to be in tension during operation so that cyclic flatwise bending of the blade was restrained. The presence of these struts and the extent to which they did restrain in-plane motion (and still allowed out-of-plane motion) would possibly add some uncertainty to the modelling.

The only report containing structural response data which is in the public domain is that by Hiester, Mercer & Tremoulet 1983. That report gives a plot of estimated natural frequency vs. rotor speed and suggests good agreement with measured data (see Figure 2.2.7). It also gives frequency spectra of measured blade stresses at a number of locations on the blade (Figure 2.2.8). Unfortunately no comparison is given with estimated values.

2.2.6 Indal 6400

This structural response of this machine is possibly the most extensively documented. Although only two machines were erected they were both instrumented and were subject to studies funded by public bodies so that the results are largely in the public domain. Because of this it has been used as a baseline for researchers who might require comparison with field data.

Malcolm 1985b presents some measured and estimated structural response from the machine at Southern California Edison's Devers test site, a sample of which is included as Figure 2.2.9. This is in the form of the harmonics as well as the total rms of cyclic stressees versus windspeed. These data are derived from an unpublished report (DAF Indal 1984a) which is a complete set of all harmonics from all rotor gauges and a comparison with predicted values.

The 1P and 3P harmonics tend to dominate the response but whereas the 1P response may sometimes be underpredicted, the 3P response is often overpredicted. The result can be that the rms is predicted correctly - a serendipitous outcome. This report (Malcolm 1985b) also points to the possible contribution of turbulence on a vertical axis wind turbine.

Following the blade failure on the Prince Edward Island machine in November 1985, the rotor was modified to incorporate flexible horizontal struts and a modified strut-blade connection. A comparison of the measured and predicted operating natural frequencies of the modified Indal 6400 on Prince Edward Island (PEI) is given in Indal Technologies 1988b. The agreement is shown to be very good for those modes that could be identified (by noting non-harmonic peaks in the frequency response spectra and the deformations implied). The same report compares the harmonic and rms stresses at low and high wind speeds and shows the same trends as mentioned above.

The (modified) Indal 6400 on PEI was used as the basis of a study into the effect of turbulence on fatigue stresses (Indal Technologies 1989c, Malcolm 1989b, Malcolm 1990a, Penna 1989a,b). The study examined the influence of turbulence intensity on the measured and the predicted fatigue stress histograms and especially on the tail end of the distribution. Figure 2.2.10 shows some of the predicted results and Figure 2.2.11 shows some of the corresponding experimental results. The agreement was encouraging and confirmed the greater effect of turbulence on the in-plane blade bending. It also confirmed the ability to translate the frequency response spectrum into fatigue distributions by converting to the time domain with random variations in the magnitude and phase of the components of the spectrum.

2.2.7 Sandia/DOE 34m Test Bed

The purpose of the Sandia/DOE 34m /Test Bed wind turbine was to provide information on new airfoil profiles and other new rotor configurations. It is, therefore, exceptionally well instrumented in addition to having a variable speed drive train.

Since 1988, when the rotor became operational, much data on the structural response has been collected and evaluated. It has been collected at a number of rotor speeds which is advantageous in that it has offered more insights, but disadvantageous since no single set of results is complete. In addition, the collection of structural response has concentrated nearly exclusively on the rms of cyclic stresses during operation. There is little data on the response at separate harmonics and little information on response during braking or when the rotor is stationary.

The availability of data from the 34m Test Bed has allowed it to be used to validate recent codes which incorporate turbulent flow, aeroelastic damping, and alternative aerodynamic models.

Malcolm 1989a presents the predicted results of different turbulent flows on the 34m Test Bed and the effects of prescribed modal damping and of aeroelastic damping. These results are interesting but preceded any experimental data. This report was, however, one of the first to show the effects of turbulence on the modal components of loads used for the solution. This did much to explain some of the effects of turbulence on the response. Some examples are included as Figure 2.2.12.

Ashwill 1989 and Ashwill 1990b present a comparison of predicted and measured natural frequencies which are in close agreement (see Figure 2.2.13). Comparison of blade stresses is given in Ashwill & Veers 1990 and in Ashwill 1991. The comparison is made difficult by the presence of data from different rotor speeds and the need to compare with predictions with and without aeroelastic damping. The task is further complicated by the choice of 34rpm as one operating speed and the close proximity of a resonant condition, making results very sensitive to small changes. Figures 2.2.14 and 2.2.15 illustrate some results from these papers.

The only attempt to generate structural response predictions using an aerodynamic code more sophisticated than the DMST model was carried out at the Hydro Quebec Research Institute and used the 34m Test Bed for validation. The results are presented in Masse & Pastorel 1989, 1990a. The intention was to use the local circulation model in combination with turbulent flow to compare structural response prediction with those presented in Malcolm 1989a, and Ashwill & Veers 1990. Unfortunately there are a number of deficiencies in the work by Masse & Pastorel which make the comparison difficult. These include the treatment of stochastic loads as fully correlated and the exclusion of aeroelastic effects. However, it is possible to conclude that the local circulation model is likely to reduce 3P response in high winds which will agree more closely with field data.

2.2.8 Summary

Documentation of the structural response has concentrated largely on the fatigue stresses, represented by the rms of the cyclic stresses, to the exclusion of other behaviour. This is only partially justifiable. The understanding of the composition of the full frequency spectrum and the contribution of transient loads is also important.

The use of the DMST aerodynamic code to define structural loads has gained only limited success. At higher windspeeds, and in the presence of dynamic stall, DMST predictions agree less well with field data.

The inclusion of turbulent flow (in longitudinal and lateral directions and with correct correlation) in conjunction with aeroelastic effects has been a major advance. However, their inclusion makes the modelling task that much more time consuming and introduces additional parameters to be defined. It has also emphasized the need to improve the aerodynamic model used to generate loads and to systematically validate it against the full spectrum of measured response.

In short, there is a requirement to make some further aerodynamic model improvements and to combine these into an inexpensive and easy-to-use structural computer code which will include turbulence and aeroelastic effects.

2.3 Reliability and Availability

2.3.1 Prototype, Experimental and Test Bed Turbines

2.3.1.1 Introduction

Reliability and/or availability data are available for a number of Darrieus turbines including:

1. Commercial prototypes such as the DOE 100 kW, VAWTPOWER 185, Indal 50 kW, Indal 6400-500kW, Projet Eole and the Adecon A19m-150 kW. Normally these turbines are expected to operate in the fully automatic mode and to log significant hours of operation so that design and manufacturing defects are uncovered before committing to sales and commercial production. The DOE 100 kW is included here, although it seems to have been mainly intended to facilitate technology transfer to private industry.

2 Experimental turbines such as the NRC/Hydro Quebec 24m, CENG VAWT D10, and the Pionier I. These might be better termed "proof of concept" turbines, constructed and operated to evaluate specific design features prior to the design of a commercial turbine. Continuous operation may or may not be emphasized.

3 Test beds such as the Sandia 34m. Test beds are designed so that configurations can be quickly and easily changed to investigate the basic physics of wind turbines. For example, the Sandia 34m test bed is equipped with a variable speed drive system to permit, among other things, performance tests of new blade profiles and blade shapes over a wide range of Reynolds numbers. Test beds are normally operated on a limited basis and only for specific

tests.

2.3.1.2 DOE 100 kW VAWT

Background

The primary objectives of the 100 kW VAWT program were to provide data for validating structural and aerodynamic technology, and to provide hardware facilitating the transfer of technology from Sandia to the private sector (Nellums 1985). Design began in 78/05 and the last of three turbines went into service in 81/06.

The DOE 100 turbine is probably the most thoroughly tested and exhaustively documented VAWT design and is a model for other wind turbine design programs. FloWind based its own commercial 17m turbine design (and some of the 19m design) on the DOE 100 kW and was the main commercial organization involved in the technology transfer.

Operating Data

Two turbines were operated on a long term basis, but, of the two, only the Bushland, Texas turbine operated at a reasonably high average power output (28.7 kW). The Bushland turbine was reported to have reached 7865 hours of synchronous run time and to have generated 219 MWh by the end of 1984. The goal was 10,000 hours of operation. Operation and evaluation began in March 1981 (Davis 1985) but early operation was primarily for structural testing and did not emphasize continuous operation. Davis reported that, since the turbine was a test machine, operation was always conservative.

During the one year period March 14, 1983 to March 11, 1984, availability was about 70% (Nellums 1985). By far the most significant contributors to downtime, in order of importance, were:

1. Occasional slow actuation or minor rotor overspeed during brake actuation requiring manual resetting. This was believed to be due to inherent design sensitivity.

2. Fatigue crack in the blade root as a result of an inclusion in the original weld (after 6560 hours of operation).

3. Bypass contactor melted in the motor starter circuit. Lightning was suspected.

4. A second fatigue crack in the blade root believed to be caused by a sharp stress concentration.

The cracks in the blade root signalled the need for a design change. Nellums (1985) suggested that although welded joints are low in cost, weld strength is well below extruded blade strength. (This appears to have been the only welded Darrieus blade joint ever put into service.) A mechanically clamped or a pinned joint (eliminate flatwise bending moment at the root) were suggested as alternatives. Moving the struts toward the equator (as in the FloWind, Indal and Alpha Real designs) could also reduce the flatwise stresses at the blade root connection.

Nellums (1985) discussed a number of design improvements for all subsystems, in anticipation of transferring the design and test information.

Blade to Blade Connection

The blade to blade connection design deserves special comment because it is in relation to the behavior of this design at Bushland versus the experience of FloWind that the one weakness in the Bushland evaluation is identified. That weakness is the relatively low rotor loading (expressed as the ratio of annual energy production to rotor area) at Bushland compared to loadings experienced on FloWind's 17m turbines in the California wind farms. Rotor loading is often a reliable measure of relative rate of fatigue damage accumulation within an existing windfarm and may be a good measure when comparing turbines installed at different sites.

The original FloWind 17m turbine blade to blade joint design was a close copy of the DOE 100 kW design but FloWind used bolts in place of rivets. FloWind reported cracks in the blade and the blade connecting bars after an average of about 5000 to 6000 operating hours (indirectly based on data reported by Schienbein 1989). In the two cases reported by Schienbein, the rotor loadings were 1100 kWh/m^2 (Altamont Pass) and 750 kWh/m^2 (Tehachapi). The Bushland turbine did not encounter cracks at that joint. The average rotor loading for the Bushland machine was estimated to be less than 500 kWh/m^2 versus an average of about 700 to 800 kWh/m^2 for FloWind's fleet of 17m and 19m turbines.

The Bushland site is less than ideal from the point of view of early identification of significant structural design weaknesses. It also appears that FloWind's 17m prototype tests were not carried out at aggressive sites and were not operated long enough, since the blade joint problem only became apparent after all 309 turbines of this type were in operation.

VAWTPOWER experienced the same type of blade failure on its commercial units in California (it also used the DOE 100kW joint design). Furthermore, the prototype was operated near Albuquerque, New Mexico, which does not have a strong wind resource.

2.3.1.3 Indal 50 kW Turbines

<u>Southern California Edison Turbine</u>

Availability

Cumulative availability for the single Indal 50 kW turbine at the SCE site in the San Gorgonio Pass was 93.6% for the first 14 months of operation (through May 1983) and averaged 94.7% from January 1, 1983 through June 30, 1984, including operating time lost due to shutdown of the test site for rewiring of the site (Bechtel 1986, Wehrey & Yinger 1984). Availability for July 1982 through May 1983 was 100%.

Unscheduled Maintenance Events

Wehrey and Yinger reported only two relatively major maintenance events. Both occurred within two months of commissioning.

1. Upper rotor bearing failure after one month of operation due to the bearing not being greased at the factory. The bearing was replaced.

2. Upper strut-to-blade connection pin and strut end fitting wore out after two months of operation. The parts were replaced.

Schienbein and Malcolm (1983) listed two additional problems that resulted in the down time during the first year of operation:

1. The hydraulic pressure exhibited instability. The pressure regulating valve was found to be defective and it was replaced.

2. Below normal network voltage resulted in the wind turbine failing to reach the required rotor speed in the preset time during start-up. The problem was corrected by increasing the preset starting time to allow for occasional below normal network voltage.

1990 Inspection

The wind turbine was inspected in 90/08 when the machine was taken out of service by Southern California Edison, following more than eight years of operation (Schienbein 1990b) The turbine was operated and maintained by the SCE personnel at the Devers Substation for the entire period. The final meter readings for this turbine were:

Electrical Energy Generated	444,192	kWh
Turbine Generating Hours	21,645	
Rotor Idling Hours	2,361	

The control logic permitted the rotor to freewheel, or idle, at wind speeds below cut-in with the generator disconnected. This reduced the number of start/stop cycles. This logic is often referred to as the "Canadian Coast".

A visual survey of the blades showed black streaking from the strut to blade joints. This is often evidence of fretting (strut connection casting to blade) and could portend the development of fatigue cracks in the blade at that location. No cracks were found, although the blade connections were not disassembled during the inspection.

The Indal 50 kW at the Devers Substation appears to have operated extremely reliably (after early problems were corrected) and to have been well maintained.

<u>Christopher Point (Canada) Turbine</u>

Operating Data

Rangi and Penna (1989) and Rangi and Kimbell (1986) reported on the performance and operational history of the Indal 50 kW turbine installed at Christopher Point, Vancouver Island. The turbine began automatic unattended operation in 81/03 and produced 351.9 MWh in 21,079 generating hours during the 90 month period to 88/08. This turbine is still operating, making it possibly the oldest Darrieus wind turbine still in continuous operation.

Availability

The availability of the turbine for the first 25 months was 65.7%. However, the availability was adversely affected by the control system malfunctions and the fact that the site was visited only once every two weeks. The control system was completely replaced and the availability increased to 90.9% for the period from 83/04 to 88/08. During this period there was a fire in the motor/generator. Five months elapsed before the motor/generator was rewound and returned to service (This was considered to be unnecessarily long since a new generator could have been installed within two weeks.) After the installation of the rewound generator in 85/03 the availability increased to 99.3% during the period 85/04 to 88/08.

Problems and Solutions

1. The blade-mounted aerobrakes were removed after nine months of operation following the loss of one plate due to a hinge pin coming out. The overspeed protection provided by the centrifugal dump valve and the hydraulic rotor lift system was deemed to be sufficient.

2. A slow oil leak from the labyrinth seal on the bull gear was detected. The labyrinth seal was run dry and the leak was eliminated with no adverse effects to the seal or the bearing system.

3. The gear teeth of the flexible coupling that transmits the torque from the rotor to the bull gear were found to be badly worn after about 11 months of operation. The excessive wear was due to poor alignment between the rotor and the bull gear. The bull gear was found to be not level. This problem was corrected.

4. The control system caused many nuisance shutdowns within the first two years of operation. The system was completely replaced and the problems disappeared.

5. The motor/generator caught fire in 84/10. No definite cause was found for the fire. It was speculated that the windings may have shorted due to the high humidity and salt laden air at the site. The generator was rewound and placed back into service. However it was discovered that the torque output of the rewound generator was lower than the original specifications. A new generator was installed in late March 1985 and had operated without fault through the remainder of the period reported.

<u>San Luis Reservoir (California) Turbine</u>

Schienbein and Malcolm (1983) reported that this sister to the SCE turbine had been on-line for 5,032 hours and generated 64,547 kWh between March 13, 1981 (start-up) and December 31, 1982. The availability in that period exceeded 90%. Downtime was due to:

1. A loose nut temporarily shorted two control system printed circuit boards.

2. Failure of the hydraulic fluid drain line connection on the overspeed dump valve resulted in the hydraulic reservoir being pumped dry.

3. Loss of hydraulic fluid due to excessive travel of the bull gear resulted in a loss of the fluid as an aerosol through the baffles.

4. The generator thermal overload breaker tripped during light winds as a result of startup/shutdown cycling. This problem was corrected by increasing the startup threshold wind speed.

The down time periods were lengthened by the time required to dispatch field service personnel from Ontario, Canada to California to service the machine under warranty.

Christopher Point, San Luis and SCE Turbines - Some Observations

The availabilities reported for these turbines are outstanding, given the difficulties of supporting these widely separated turbines (distances of up to 4500 km).

All three machines operated very reliably following the correction of early problems, many of which were associated with manufacturing and installation errors. Most, if not all, problems were minor and appear to have been solved quickly and decisively. However, the causes of some problems (such as the generator fire) were never satisfactorily determined.

The SCE turbine operated at a somewhat higher average power output (20.5 kW versus 16.7 kW) than the Christopher Point turbine but for about the same number of hours. Given these facts and the close similarity of design, a direct and very useful comparison of the effects of extended operation in the dry desert environment to those of a fairly mild coastal marine environment would be possible. (This is particularly important since marine corrosion played a major role in the failure of the blade on the Indal 6400 turbine at the Atlantic Wind Test Site.) Unfortunately, no structural inspection data have been reported for the Christopher Point turbine to compare to the results of the SCE rotor inspection.

2.3.1.4 Indal 6400/500kW

Operating Data

Two prototype 6400/500kW turbines were installed and operated (Schienbein 1985b). As of June 10, 1985 the first unit, commissioned at the Southern California Edison Site (Devers Substation) in late 1983, had accumulated 2,065 hours of on-line operation and had generated 329,040 kWh. That unit had also reached a recorded maximum power output of 545 kW (three minute average). A second unit installed at the Atlantic Wind Test Site (Canada) and commissioned in early 1984 had logged 1,490 operating hours and generated 144,783 kWh as of the same date. Both turbines were extensively instrumented to measure power output performance and structural loads.

Design Innovations

The design included four main innovations:

1. The rotor rotates on a hydraulic lower and upper bearings. High pressure hydraulic fluid is piped to the upper bearing. The rotor is supported by the high pressure fluid. The upper thrust bearing moves vertically to maintain constant guy cable tension.

2. The rotor drives two generators through a single stage/twin pinion transmission. The bull

gear also serves as the brake disc, engaging 16 brake pads when the hydraulic support pressure is released.

3. Two independent, centrifugally actuated overspeed valves are mounted on the rotor column. The valves override the control system in the event that the primary overspeed protection system fails.

4. Tailored braking (two stage) was implemented in the control logic and is a beneficial outcome of the hydraulic support/brake concept. Brake torque could be reduced to about 50% of maximum during the final stage of braking. This two stage approach significantly reduced the fatigue damage accumulation due to braking. Brake stops were reduced through the "coasting" (idling) logic that permitted the rotor to freewheel at wind speeds below cut-in.

Availability and Main Problems

The availability of the SCE unit exceeded 85% for the first three months of 1985. Availability was affected most by two component problems:

1. Cracks in the weld joining the top cap of the lower bearing stub shaft to the cylindrical body permitted high pressure oil to leak back to the reservoir. This reduced the rotor support pressure sufficiently to cause shutdowns. The problem was finally solved by using a solid steel shaft. The replacement of the stub shaft required a crew of two for about two working days, indicating a maintainability problem with this design.

2. Broken strands were detected in the guy wire terminations in 85/04. Analysis showed that the looped/thimble swaged type terminations were inadequate for the loads imposed and that spelter sockets should be used. (It is interesting to note that the looped/thimble terminations had been quite satisfactory for the 50 kW turbines.) The six cables were replaced with ones having the socket terminations. This work was done by a crew of three within three working days without the use of a crane.

Availability was also affected by the inability of the "hard-wired" relay-based controller to cope with the rapid rotor accelerations during startup, generator switching and braking initiation. (This approach had worked very well on the Indal 50 kW turbine but the 6400 turbine rotor was far more responsive and had a higher ratio of aerodynamic torque to inertia than did the 50 kW rotor.) For example, during restart in high wind speeds, the time delay would result in the generator switching on-line at too high a rotor speed. The controller would have sensed an overspeed (in some cases the centrifugal valves would also open) and the turbine would automatically shutdown (overspeed fault condition) and require a manual reset. Although modifications were carried out to introduce anticipation and time delays, it was concluded that a microprocessor based controller should be developed.

Brothers (1991) reported that the microprocessor based control system had been installed in the AWTS turbine, along with an improved hydraulic system.

Blade Failure

The turbine installed at the Atlantic Wind Test Site ("Aquilo") experienced a blade failure on November 15, 1983 (Dainty and Chow 1986). At the time of the failure the turbine had logged 2750 hours in the power generating mode.

The blade fracture occurred at the upper strut attachment as a result of high cycle fatigue. The crack initiation time was reduced and the crack growth rates were accelerated by the combination of the corrosive coastal environment and the high number of cycles. The crack initiated at a stress concentration at a bolt hole located on the outer blade surface. Intergranular corrosion, fretting and sharp edges due to machining exacerbated the problem.

It was found that the rotor had operated in an unfavorable stress configuration (so-called "intermediate configuration") for 524 hours prior to the failure. In the original configuration some rotor out of plane motion, with damping, had been permitted by the strut-to-mast connection design. However, the strut damper system proved to be unreliable so the struts were locked and about 180 kg of ballast was added to each blade to compensate for the decrease in strut compliance. This configuration was termed the "intermediate configuration". Analyses of the strain gage data showed that in this configuration some rms stresses were increased by 70% at wind speeds of about 20 m/s (Indal Technologies 1988b). As a result, the rate of fatigue damage accumulation increased greatly.

Indal designed new flexible struts and connections, including the use of doubler plates bonded to the blades (at the strut attachment) to reduce the local blade stresses. The turbine was rebuilt with the new assemblies and test data confirmed that the blade stresses in the region of the strut attachments had been significantly reduced (Indal Technologies 1988b).

2.3.1.5 Adecon A19m-150kW

Introduction

The prototype A19m turbine was tested and evaluated at the Atlantic Wind Test Site from 89/08 to 90/02 when a brake failure, due to the deterioration of the brake disc surface in the marine environment, resulted in the loss of the rotor (Adecon Energy Systems 1990d). The Adecon design includes three main innovations in vertical axis technology.

1. The use of lightweight multiple extrusion blades.

2. An external three-legged rigid frame to support the rotor.

3. Hinged blade to mast (root) connections and hinged connections at the inboard strut ends. Their purpose is to reduce torque ripple and blade stress.

Problems and Solutions

The turbine accumulated 370 hours of operation before the rotor was destroyed during overspeed. The event was determined to have occurred due to a rust buildup on the brake disk surface. The rust was subsequently polished when the brake engaged creating a layer of high thermal resistance. The resulting high temperature at the friction surface greatly reduced the braking torque.

The main problems uncovered during the limited operation are summarized below:

1. Blade damper assemblies were found to require intensive maintenance, particularly the nitrogen charged accumulators. Improvements are required to obtain reliable long-term operation.

2. The brake was deficient in both torque consistency and in the amount of maintenance required. Excessive pad wear rate was attributed to aggressive brake disc corrosion in the marine environment. The use of a single brake caliper/disc on the high speed shaft does not appear to be a sound philosophy.

3. The service life of the upper lateral load links supporting the rotor was only 370 hours.

4. Vertical members on the external frame were observed to be excited during operation of the turbine at higher wind speeds. Motion was most evident on the downwind leg. Stiffening cables were installed but their effectiveness was not determined.

5. Loose bolts were found on the unit. This is not uncommon on prototypes. All critical structural bolts should probably have been retorqued after about 100 or 200 hours of operation. The need to do this was highlighted by the failure of a blade to strut connection after only 260 hours of operation when the bolts connecting the strut beams to the blade attachment plate loosened and separated causing the strut to become detached. The blade bent around the column during the next shutdown event. Several of the bolts that had loosened were reported as being tight several hours before the incident.

It was concluded (Adecon Energy Systems 1990d) that the operating problems observed could be solved relatively easily and would not substantially affect the machine's cost. Due to the destruction of the rotor, the improvements were not implemented on the prototype. Serious uncertainty remains about the long-term reliability and maintenance costs of Darrieus turbines that incorporate hinged blade connections (with dampers) and high speed shaft brakes.

2.3.1.6 Polymarin Pionier I

The experimental vertical axis wind turbine PIONIER I was installed in Amsterdam in 1982. The turbine rotor is cantilevered (15 m in diameter) and has two GFRP blades. A grid connected AC-DC system with a DC generator of 93.8 kW enables operation at variable and constant rotor speed (Machielse and de Groot 1986).

The owner of the turbine, the Electricity Board of Amsterdam, collected operational data for four years starting in the second half of 1982. The turbine was coupled to the grid for 14,000 hours during that period with an average wind speed of 4.3 m/s. Very little testing had been carried out prior to the start of continuous operation and several problems were encountered almost immediately (Machielse 1984b):

1. The nitrogen pressure system leaked. Since the pressurized nitrogen was used to lift and support dead weights that are part of the emergency braking system, this was obviously a nuisance.

2. The steel rolls of the rotor alignment bearing were noisy, difficult to adjust and wore quickly. The steel rolls were replaced with eight nylon rolls and adjusted to zero clearance.

3. High normal forces on the first gear box bearing seemed to be responsible for the high gear box temperatures experienced at low power levels. Cleaning of the Waldron coupling solved this problem.

4. The electrical overspeed detection system failed on one occasion and the turbine operated for several seconds near its critical speed for tower excitation (78 rpm) and experienced large vibrations. As a result of this episode the electrical overspeed activation was made fail safe, the overspeed limit was decreased, and some emergency conditions were changed to normal stops. In addition, the control logic was changed so that the generator remained connected to the grid during emergency braking. Furthermore the generator was used as an electrical brake to supplement the emergency brake if the deceleration rate was below the set point.

Following these changes, the turbine was reported to be running well.

2.3.1.7 VAWTPOWER 185

The prototype VAWTPOWER 185 (17 m/185kW) was fabricated in mid 1983. Testing began in late 1983 and apparently continued into 1984, although 14 production turbines were already installed in a California wind farm by the end of 1983 (Leigh and Edgel 1984).

Very little information on the prototype test and evaluation has been published. Leigh and Edgel reported that the control system responded properly to two potential runaway conditions caused by utility interconnect problems. Some performance data had been gathered by late 1983.

2.3.1.8 CENG VAWT D10-2

The CENG VAWT D10 is a three bladed turbine having a diameter of 10 m and a rated output of 30 kW at 13.5 m/s. Unit number 2 (D10-2) was installed on Amsterdam Island in the Indian Ocean in December 1986 (Perroud, Bertrand and Plantevin 1988). The purpose of the installation was to demonstrate diesel fuel saving at a remote site and to expose the turbine to marine climate conditions.

Operation in 1987 was restricted to winds less than 15 m/s because the braking system was believed to be inadequate for the peak power developed by the turbine (36 kW at about 20 m/s). A new brake was installed in December 1987. It included a 980 mm diameter disc with a thickness of 27 mm (more than twice that of the original brake) and three independent calipers providing a combined torque of about four times that of the original brake.

2.3.2 Commercial Turbine Installations (Wind Farms)

2.3.2.1 Introduction

Commercial turbines of the Darrieus curved blade type do not exist beyond the first generation, which can be characterized by the FloWind 17m and 19m series. This is in sharp contrast to HAWT development and must be borne in mind when interpreting operating data.

Windfarms of Darrieus rotors have been developed by only two companies, FloWind Corporation and VAWTPOWER, both in the United States. FloWind manufactured and installed more than 10 times as many turbines as VAWTPOWER (see Table 8.1.2).

VAWTPOWER is no longer in business and one remaining wind farm of VAWTPOWER machines is being operated by an independent wind farm operations and maintenance firm with no apparent links to the original designers and with no qualified VAWT engineering support.

FloWind, however, has operated and maintained its own fleet continuously since the turbines were installed during the period December 1983 through December 1985. Furthermore, FloWind has been able to provide on-going engineering support for its products (albeit with a small staff) even through 2.5 years in Chapter 11 bankruptcy protection. This record of original design, manufacture, continuous operation and uninterrupted support is noteworthy, because only one other U.S. company (U.S. Windpower) has been able to achieve this.

2.3.2.2 Operating and Availability Data

Operating Hours and Production

Schienbein (1991) reported operating data for the original 60 FloWind 17m turbines that entered service in 83/12 at the Altamont Pass (see Table 2.3.2.2). Availability averaged 90.3% for this group of turbines during the first six months of operation (Bechtel 1986).

Table 2.3.2.2 Operating Data for the Original 60 FloWind 17m Turbines

	Date	
	88/29/02	**91/03/01**
Accumulated Operating Hours	965,363	1,584,229
Average Hours/Turbine	16,089	26,404
Total Energy (MWh)	33,813	57,937
Average Power/Turbine (kW)	35	37

Through November 1990, Schienbein (1991) reported that FloWind's 510 17 and 19 m turbines in California had generated electricity for 9,058,090 hours and produced a total of 460,170 MWh. At that time 42% of the fleet had operated for more than 20,000 hours and 14.5% had operated for more than 25,000 hours.

Main Fleet Problems and Solutions

Schienbein (1991) described, in summary form, the main turbine problems (both 17m and 19m designs) encountered to 91/04 and the management of those problems (blade joint cracks, generators, brakes and gearboxes). The first three were reported to be fleetwide while the gearbox problem was reported to be limited to those from a particular manufacturer.

1. Fatigue cracks at the blade joints caused by stress concentration, fretting, manufacturing defects and aggressive wind microsites. This problem was being treated by reducing stress concentrations and minimizing fretting in retrofit designs. Blade sections were being patched or replaced. Elimination of the blade joints by using a one piece extruded blade was considered as a longer term solution.

2. Generator winding failures due to moisture, dirt and inadequate insulation (open drip proof design). Generators were being rewound and vacuum epoxy dipped. In the longer term, possible partial enclosure of the base structure was being considered for some turbines.

3. Excessive brake pad wear and misalignment difficulties were experienced due to misapplication of the caliper brake design. On-going preventative maintenance was required. A retrofit was being considered.

4. Low speed shaft bearing failures were reported in gearboxes from one manufacturer and affected about one quarter of the fleet. The failures were caused by the bearing being undersized. Properly sized bearings were being retrofit on a running basis.

Controller Subsystem

Wallace (1991) reported on Flowind's controller problems and solutions. The problems experienced were classified as being hardware, software or interactive.

Hardware problems included controller backup battery failures resulting from discharge during long grid outages at the Tehachapi wind farm where severe winter storms (near hurricane force) are common. This situation was clearly not anticipated by the designers. Some controller power supplies failed as a result of the battery failures.

Software problems included an emergency shutdown and latch condition (requiring manual reset) without any indication of the cause (no annunciator lamp lights). This was rectified by analyzing and modifying the software program.

Interactive problems included the difficulty in reliably detecting a grid outage when the turbine generator and capacitors interact and maintain the voltage within acceptable limits.

The problems reflect FloWind's unique first generation commercial hardware and software approach. (Intel 8085 microprocessor, software code in EPROM, and hardwired relay based override protection logic.) FloWind continues to use the original controllers, with some modifications, attesting to the basic soundness of this system.

Wallace (1991) reported that the custom controller with a relatively simple microprocessor has worked well, although requiring some debugging. He pointed out that fairly complex and often subtle interactive problems (hardware and software elements of the control and protection system) will not be automatically eliminated by adopting an off-the-shelf controller. This was demonstrated, for example, by the modification of the controller for generator enhanced braking (generator on line during braking until the generator rpm falls below synchronous). The modification was made after it was found that lag in brake application (after the generator had been taken off line) varied from turbine to turbine due to ambient temperature, air in the brake lines and the permitted range of allowable brake pressure. This had resulted in nuisance rotor overspeeds at the most aggressive sites.

2.3.2.3 Structural Fatigue

FloWind's rotor structural fatigue problems were encountered first in 1985 when cracks were found in the welds of the 19m turbine struts. As of 91/04, FloWind reported that all known fatigue problems had been analyzed and retrofits had been designed. All retrofit designs were at that time deployed or scheduled to be in service by 91/08. The complete repair and retrofit program was to be completed within 1.5 to 2 years (or by about the end of 1993). Some blade replacement and rotor rebuilding was planned. A key objective was to repair and retrofit on a "running" basis without lowering the rotors. FloWind has not reported any results from this program.

Solt (1991,1992) highlighted the magnitude of the fatigue problem for HAWTs and VAWTs by showing that, as of approximately 92/01, FloWind's most productive and longest running turbines had each generated on average 2400 MWh of electrical energy and had operated for 34,000 hours. This implies about 10^{10} one-per-revolution blade fatigue cycles.

Schienbein (1991) reported considerable variability in microsite effects within FloWind's windfarms, stating that the "fatigue life of a rotor at a given site could be less than 1/8 that at another site within the same windfarm". He also stated that rotor structural fatigue has been concentrated at the blade joints.

2.3.2.4 Structural Life Management

FloWind was one of the pioneers in adopting and developing turbine rotor structural life management practices in support of the maintenance of its fleet (Schienbein 1987, 1988, 1990). The emphasis was on the least cost, maximum revenue solution that combined life extension through inspection and crack stopping with the development and deployment of blade repairs, or patching.

Solt (1992) reported that almost 50% of FloWind's fleet had at least one blade crack as of 92/01. Each crack was being monitored and tracked. Solt (1991,1992) has provided considerable detail on the techniques used to address the structural problems and to extend the life of the first generation rotor structures to maximize the return from the windfarms. The extruded aluminum blades offered significant advantages in such a program, primarily because crack growth was relatively slow and cold worked crack stopping was fairly easy to implement with commercially available materials and tools. Furthermore, visual crack detection (with and without dye penetrant) of very small cracks is possible with practice, visual evidence of fretting is apparent and surface scratches and gouges can be "dressed" and made more benign.

The program of inspection, crack stopping, blade repair and blade replacement has succeeded in maintaining high availabilities for the fleet of 510 17m and 19m turbines with over 10,500,000 lifetime operating hours as of about 92/01. Solt reported, for example, on five turbines that had operated for between 5000 and 7500 hours since crack discovery through an inspection and crack stopping program not unlike that practiced in the aircraft maintenance industry. This program bought time for the company to design, develop and implement blade repairs and to upgrade new blade designs.

Two blade-to-blade joints per blade were used in FloWind's design to allow shipping in a standard 12 m container and due to limitations on handling of very long continuous blade pieces (Mercer, Tremoulet, MacClendon, 1983). The rotor blades could have been manufactured without joints. Indeed the extruded blades for the Sandia 17m research turbine (slightly longer than the blades for the FloWind 17m turbine) were manufactured in that way and installed in 1976.

2.3.2.5 Blade Repairs (Patches)

FloWind did develop and deploy both bolted and bonded blade repairs (patches). Schienbein (1990) stated FloWind's rationale for blade repair as follows:

1. The cost to replace cracked blade sections is higher than the repair costs.

2. Blade connections are fatigue sensitive but the remainder of blade is not, so continuing to repair the high damage regions (joints) makes sense.

3. Repair technology exists but needs to be further developed and adapted to the repair of curved extruded aluminum blades.

(This rationale also applies to GFRP HAWT blades, evidenced by on-going programs of repair and limited replacement of thousands of HAWT blades in California.)

Schienbein reported (1990) that a blade patch for the root connection was in service as of 90/03 and that field trials of a strut to blade patch were scheduled for the fall of 1990. A blade-to-blade joint patch was at the conceptual design stage as of 90/06. As of 92/01, Solt reported the latter design was complete and that a new blade-to-blade joint design (using fasteners) was in field trials, as of the summer of 1991. Solt (1991) reported that a bolted blade patch was being used to repair the blades at the 19m strut to blade connection. Schienbein discussed FloWind's interest in adhesively bonded blade repairs. FloWind is believed to have deployed a large number of bolted and bonded blade repair doublers, however no data on this program have been reported.

VAWTPOWER is known to have deployed a number of adhesively bonded blade to blade joint reinforcing doublers during 1986 on part of its fleet in operation near Palm Springs, California. This followed failure of a splice joint after 1200 hours of operation on turbine A-5 at the EDCC Karen Windfarm.

2.3.2.6 Interpretation of FloWind's Experience

Schienbein provided a summary interpretation of FloWind's experience as of 88/04 (Schienbein 1988). This interpretation would appear to be equally applicable to HAWTs, in the light of structural problems evidenced on that type of turbine. His comments highlight the fact that FloWind's windfarm VAWTs are first generation commercial designs that did not have the benefit of long term prototype evaluation. Limited operation of test turbines (for periods of 10,000 hours or less) and prototypes will not expose high cycle fatigue problems. This has been amply demonstrated.

Schienbein suggests that the original designers of FloWind's VAWTs probably:

> -Had insufficient understanding of the load transfer mechanisms in blade joints resulting in underprediction of the stress concentration factors.
>
> -Had little knowledge of the increase of some cyclic loads due to turbulence and the violent wind microsites that many turbines would find themselves in.
>
> -Used insufficient "factors of safety" to account for accelerated crack initiation caused by manufacturing defects, inadvertent damage, fretting and possibly corrosion.
>
> -Used inadequate specification of critical parts permitting serious "built-in" flaws (while parts still "meet the spec").
>
> -Operated the turbines at very high cut-off wind speeds to maximize energy capture only, without optimizing energy capture and fatigue damage accumulation.

In addition Schienbein suggested that there was little evidence of an integrated design, production planning and quality assurance program. Quality assurance programs were overtaxed by compressed design, procurement, manufacturing and installation schedules.

Schienbein pointed out that the problems encountered have very little to do with the VAWT concept, since blade fatigue, gearbox, generator and brake problems have been encountered on both HAWTs and VAWTs in the California windfarms (for example, Aerostar HAWT blades) and have been reported on in formal papers and in industry publications throughout the past eight years.

FloWind concluded (Schienbein 1988) that the FloWind 17m and 19m turbines were providing a wealth of structural life information that will help in designing the next generation of commercial vertical axis wind turbines. Schienbein (1989) pointed out that failure data from the turbine fleet provide the actual bounds of the fatigue lives ("baseline"). Design improvements can be tested and evaluated in relation to the as-built designs resulting in a significant advantage in turbine product development.

2.3.2.7 Conclusions

FloWind's experience is unique and extremely valuable. It is hoped that FloWind will continue to report on its experiences. FloWind's rotor retrofit and repair designs may point in part to improvements that could be implemented in second generation designs using aluminum blades.

It is clear that, as in airframe design, WECS structural designs (HAWTs and VAWTs) continue to improve due in large part to the learning from the fairly long term commercial fleet operating experience now available. It has been argued that improvement would have been achieved more rapidly and at lower cost if prototype tests had been backed up by high cycle laboratory fatigue testing of the critical blade joints (that is, design backed up by adequate development). It is fair to state that in the period when the existing first generation turbines were designed and deployed, the industry did not follow such an approach.

In the case of both HAWTs and VAWTs, the structural fatigue problems encountered in the commercial fleets do not invalidate the concepts, although they have most certainly led to the demise of certain turbine designs and also manufacturers. These problems are part of the normal product evolution. It is expected that as the industry matures, most designs will be evaluated for structural adequacy through balanced programs that include more analysis, fatigue testing, prototype testing and the interpretation and application of long term fleet operating experience.

It has been suggested that the unusualness of the VAWT approach (and therefore the lack of technical and financial support that probably resulted), and the need for fairly sophisticated analytical methods for design, has been largely responsible for the lag in the development of newer commercial designs. No inherent flaw in the concept exists that would prohibit competitive designs from being brought to fruition.

3.0 DESIGN PROCEDURE

3.1 Historical Review

In the 26 years since the Darrieus rotor was rediscovered there has been a tremendous improvement in the methods and available tools for their structural analysis and design. This evolution has followed the improved understanding of the (often complex) aerodynamic and structural dynamic behaviour. This section reviews the design approaches and safety factors used in a number of Darrieus rotor turbines. The review is approximately chronological.

Early wind tunnel tests in Canada (South & Rangi 1972) were concerned with aerodynamic performance only and with confirming aeroelastic stability and buckling under survival wind loading (Templin & South 1976).

3.1.1 Sandia 5m

The initiative in understanding the structural behaviour of Darrieus rotors was largely taken at Sandia National Laboratories beginning in 1975. Banas & Sullivan, 1976, includes results of analysis of the 5m test machine. Natural frequencies for a number or rotor speeds were calculated but not confirmed. Flatwise bending stresses due to centrifugal action were calculated and were found to be high for the straight-circular-straight configuration selected.

3.1.2 Sandia 17m R&D

Sandia further developed analysis and design tools for their three-bladed 17m test machine during 1975-'76. Initially only the flatwise behaviour of the blades was considered and eigenvectors of the isolated blades were calculated (Banas & Sullivan 1976). Aerodynamic loads were obtained from a single streamtube model. Pseudo static forces were used in the design of the tower and foundations. The structural analysis program SAP4 was later used (Weingarten & Blackwell 1976) to extract natural frequencies and mode shapes for the complete rotor. Although this analysis included centrifugal stiffening, it did not include the rotational effects of softening and Coriolis coupling.

The main criteria used in the design of the 17m R&D machine were

- operation at 52.5rpm with a wind speed of 26.8 m/s.
- runaway condition of 75rpm in a windspeed of 35.8 m/s.
- stationary rotor in a windspeed of 67.0 m/s.
- fundamental blade flatwise natural frequency to be above the 3P frequency.

The configuration chosen for the rotor is shown in Figure 3.1.1 and consisted of an aspect ratio of 1.0, three blades and a system of diagonal struts for increased stiffness. The central column was a 25 mm thick steel tube with an overall diameter of 508 mm. Stress analysis showed a maximum stress of 114 MPa at the blade root connection under the operating conditions. The lowest natural frequency was predicted at 2.75 Hz.

The tubular column was chosen over an equivalent truss due to flexural and torsional stiffness requirements, drag and shadow effects, and appearance.

3.1.3 DOE 100 kW (17m "Low-Cost")

The Sandia 17m R&D machine was not considered to be a prototype for a commercial design. Design for DOE 100 kW (the so-called "17m low-cost") rotor was a product of the economic optimization studies that began in 1976 (Sullivan & Nellums 1979, Sullivan 1979, Grover & Kadlec 1979). Subsequently a number of changes to the configuration were made but most of the design criteria remained the same.

The details of the design and analysis for that machine are given in the report by Alcoa (1980). The analysis used ANSYS for the pseudo-static analysis of the blades and to extract the natural frequencies of the complete rotor (including the differential stiffening terms). The report pointed to the need for laboratory fatigue tests to confirm the strength of the blade root connection.

A safety factor of 1.80 was used against local buckling and 1.0 against yield in the survival wind condition. The program MARC was used to analyze the blade deformation in this large-displacement problem.

3.1.4 DAF Indal 50 kW and NRC/Hydro Quebec 24m

The design of the DAF Indal 50 kW machines which were fabricated between 1977 and 1982 used criteria similar to those adopted for the DOE 100 kW. The design of the NRC/Hydro-Quebec/DAF Indal machine for the Magdalen Islands was also based on these unproven criteria. The ability to predict either the natural frequencies or the true fatigue spectrum to which components would be subject did not exist.

The Magdalen Islands machine did benefit from a number of wind tunnel and model tests carried out by the NRC in Ottawa on flutter and survival wind stability (Templin & South 1976b). Those tests established non-dimensional parameter values limiting the stability of rotors of aspect ratio 1.5 both and without horizontal struts.

3.1.5 Alvawt 500

This three-bladed machine, designed and manufactured by Alcoa, had rotor dimensions 25 x 37m and was first erected at the Southern California Test site near Palm Springs. Its design followed the procedure common to other Alcoa turbines to that date. It suffered a catastrophic failure in 1980 due to a control error which led to an overspeed condition. A second machine was installed at Newport, Oregon.

3.1.6 Rotating Frame Effects

The development of VAWTDYN (Lobitz & Sullivan 1980) showed the importance of including all rotating frame effects in the structural dynamic analysis. A procedure, labelled "FEVD", was then developed to include these effects in finite element analysis using the DMAP facility of NASTRAN (Lobitz 1981a). This led to improved abilities to predict not only natural frequencies but also the forced response ("FFEVD") of operating rotors (Lobitz & Sullivan 1983, Malcolm 1983).

3.1.7 Indal 6400

The improved analysis tools also led to improved design approaches. The Indal 6400 wind turbine was a derivative of the Magdalen Islands machine but was designed more systematically. The rotor design was tuned to avoid natural frequencies associated with out-of-plane ("butterfly") motion of the blades.

Initial design of the rotor was done on a pseudo-static basis with loads obtained from a single streamtube model. Peak loads were compared with material yield or strength using common safety factors. The estimated one-per-rev fatigue forces were compared with the 10^7 cycles fatigue strength of the connections and details.

Fatigue tests were carried out on samples designed to simulate the blade root and splice connections. The stress vs. cycles ("S-N") curve obtained was used with the LIFE code developed at Sandia, to predict an operating life for these components.

Later predictions of all cyclic stresses were made using a DMST model and Sandia's NASTRAN-based frequency response procedure and compared with measured values (Malcolm 1985, DAF Indal 1984a). In addition to the basic load cases of operating and survival wind loads, the transient forces induced during stopping and starting were considered.

In November 1985 the Indal 6400 on Prince Edward Island suffered a fatigue failure in one blade at an upper strut connection. The failure was induced by fretting and was exacerbated by operation near a resonant condition. The machine was modified and was successfully put back into operation.

3.1.8 FloWind

The FloWind 17m was a derivative of the DOE 100 kW rotor and was designed using much the same tools. In 1986 the FloWind 17m and 19m rotors suffered the start of a series of blade fatigue failures (Schienbein 1988). It is understood that these were induced by fretting at the blade connections (Solt 1991). A program of structural life management has kept most of the turbines in operation.

3.1.9 VAWTPOWER 185

This machine was also a derivative of the DOE 17m "low-cost" turbine (Leigh & Edgel 1984) and featured similar details. These machines experienced fatigue problems at the blade splice, very similar to those experienced by the FloWind 17m turbine.

3.1.10 Projet Eole

The 64m diameter rotor, known as Project Eole, was designed between 1981 and 1984. The detailed structural design was carried out by Shawinigan Engineers which later became part of Lavalin Inc. Codes such as GAROS from Omega GmbH in Germany were used to calculate the natural frequency and modal damping and the forced response was checked using Sandia's FFEVD program and NASTRAN procedure (Benjamet 1985).

It is known that great care was taken in the detailed design of the steel weldments to avoid fatigue cracks (Narduzzi 1985, Mai Phat 1985). However, it is not known what factors were applied to the estimated loads and material strengths before calculating design requirements.

The steel blades of Eole suffered some fatigue damage in 1991. This failure was attributed to inclusions in one of the longitudinal blade welds.

3.1.11 Sandia 34m

This machine was designed to operate in the Texas panhandle in a mean wind of 6.3 m/s at 10 m and having a Rayleigh distribution. The normal wind shear exponent was selected as 0.143 and a survival windspeed of 67 m/s with no wind shear was chosen.

The design adhered to the following procedures and cases.

Operating

-Aerodynamic loads were calculated from CARDAA (a DMST code) in conjunction with FFEVD/NASTRAN. Flutter speed was checked using NASTRAN and aeroelastic terms.

-The rms of the cyclic blade (nominal) stresses at cutout windspeed 20 m/s and at the highest rotor speed (38 rpm) was kept to less than 18.6 MPa. An endurance limit in the 6063-T6 aluminum of 68.8 MPa was assumed. Stress concentration factors of between 2.0 and 4.0 were assumed to exist in the blade connections.

-Fatigue life was calculated using these estimates and the LIFE(2) code.

-Fatigue stresses in the central column were not critical.

Survival Wind

-The expected stresses were kept to within the material yield strengths.

-The cables were assumed to carry tensions 60% above "nominal" to allow for future modifications.

Runaway

-Rotor speeds of 60 and 70 rpm were considered and stresses checked against yield stresses.

Single Blade

-Bearings were sized to carry loads corresponding to single blade operation.

Safety Factors

-Load factors were not used, but instead some conservatism was (believed to be) included in the specified loads.

Braking

-A maximum brake torque equal to 4.0 times the mean aerodynamic torque at cutout at the maximum operating speed was specified.

3.1.12 Adecon 19m

No fatigue failures have been reported on any of the Adecon machines. There was a catastrophic failure due to brake failure and overspeed (Adecon 1990). There has been no information on performance and number of hours of operation. There is little known about the design procedures followed by Adecon.

3.1.13 LavalinTech L24

The most complete set of design criteria available is that prepared by LavalinTech (LavalinTech 1991). Those design specifications include the basic load cases and combinations and also the safety factors to be used in conjunction with the various loads. The criteria follow the guidelines set out in the Canadian Standards Association document CSA F416. A copy of those design specifications are included with this report as Section 3.2.7.

For the design of the L24 for fatigue a (zero turbulence) frequency response analysis of the rotor was carried out at a number of wind speeds and the rms of the cyclic stresses at critical locations was calculated. The fatigue life of the specially designed root and splice connections was calculated using the fatigue strength from laboratory tests.

Turbulence was not included due to restraints on the program but it was considered that increases due to turbulence were balanced by the lack of aeroelastic damping in the model. However, the shape of the distribution of fatigue stresses did reflect the influence of turbulence in the use of specially modified "LIFE" computer code.

The design of the L24 also considered possible peak stresses during operating conditions.

3.2 Design Codes

A great deal has been learnt about the dynamic and aerodynamic behavior of Darrieus rotor wind turbines over the last two decades. However, only part of that understanding has been applied to the design process. This is because there has been little detailed design of a new machine during the last few years. The only exceptions have been the LavalinTech L24 and the CENEMESA 23, but those machines have not been built.

3.2.1 Relevant Codes

Until 1985 there were no codes or standards written especially for wind turbine design. The choice of design procedures was very subject to the background and preferences of the principal engineer. The following codes are now available.

CAN/CSA -F416-87
Wind Energy Conversion Systems - Safety, Design and Operation Criteria

CAN/CSA-F417-M91
Wind Energy Conversion Systems - Performance

AWEA Standards 3.1-1988
Design Criteria - Recommended Practices. Wind Energy Conversion Systems

IEC-TC88-draft Feb 1992
Safety of Wind Turbine Generator Systems

Technical Basis for the Type Approval and Cerification of Windmills in Denmark, May 1, 1991, National Engineering Laboratory, Riso.

Germanischer Lloyd, July 1989
Preliminary Regulations for the Certification of Wind Energy Conversion Systems.

Det Norske Veritas, 1989
Tentative Tules for Classification of Wind Turbine Power Plants.

ECN-91-001, Feb. 1991 (Netherlands)
Regulations for the Type Cerification of Wind Turbines: Technical Criteria.

Other codes relevant to the appropriate material have been used. For example, the following codes for construction in aluminum and steel have been used.

Strength of Aluminum,
Aluminum Company of Canada

Aluminum Construction Manual
The Aluminum Association

CAN3-S16.1-M85
Steel Structures for Buildings

Uniform Building Code

ANSI Building Code Requirements for Minimum Design Loads on Building and Other Structures.

ACI Building Code Requirements for Reinforced Concrete.

Aluminum Association Standards and Data

AISC Manual of Steel Construction.

British Standard BS5400
Steel, Concrete and Composite Bridges. Part 10. Code of Practice for Fatigue

Det Norske Veritas, 1977
Rules for the Design, Construction and Inspection of Offshore Structures, Appendix C

3.2.2 Environmental Conditions

Until recently the specifications for the wind regime and the environmental conditions have been somewhat arbitrary. A peak survival windpeed has been specified but any specification of the operating windspeed distribution was meaningless without a procedure to relate it to the distribution of fatigue stresses. A Rayleigh distribution with a mean of 8 m/s has been common on more recent LavalinTech designs.

The majority of wind turbines have not been designed for marine environments with the exception of the machines that have been installed in Canada, where most have been installed near the coastline. The extra protection has mainly involved additional painting and galvanizing of the steel parts.

Turbulence does not play as important a role for vertical axis wind turbines as it does for HAWTs. This is partly because the aerodynamic nature of VAWTs causes them to experience fluctuating loads even in steady flow. Turbulence levels have not, therefore, normally been specified in Darrieus rotor design. However, as discussed below, turbulence should not be ignored.

3.2.3 Peak Loads

Loads associated with the 50 or 100 year return extreme wind have been based on wind speeds of between 53 and 67m/s. These loads have been calculated by pseudo statics (not including any dynamic effects) and have used drag coefficients on the blades of up to 2.00.

Other peak loads have occurred during emergency braking conditions or during severe icing conditions. Most Canadian designers have assumed up to 50mm of ice on all surfaces while it is not clear what others have used.

The peak loads due to emergency braking or due to a torque spike from electrical transients have received very little attention in the available literature. However, there has been evidence from both FloWind and from Indal Technologies that such conditions can be severe. Indal Technologies assumed an electrical transient equal to 4.0 times the rated power torque for design of the 6400 turbine. LavalinTech used a value of 3.2 times rated torque.

The maximum force values during an overspeed condition have been considered for many machines. The DOE 100 kW was designed to overspeed from its normal 52rpm to a speed of 75rpm, although this value was selected rather arbitrarily. Later machines, such as the Indal 6400, have not considered an overspeed greater than 10% of normal speed.

Peak loads occurring due to statistical variation of operational loads has also received little attention. However, the specifications prepared by LavalinTech for design of the L24 do recognize this possibility and suggest that values of the mean plus 4.0 standard deviations is appropriate.

3.2.4 Fatigue Loads

As mentioned above, a VAWT experiences cyclic loads even when operating in steady winds. This fact led to early efforts to identify the natural frequencies of the rotor in order to avoid possible resonances. This, in turn, led to more reliable models for the frequency response of the operating rotor.

However, the lack of reliable structural aerodynamic models led to considerable underdesign on early machines. The procedure for design of the DOE 100 kW rotor was to calculate the centrifugal plus/minus pseudo static flatwise aerodynamic forces to obtain the range of flatwise fatigue stresses. The out-of-plane blade forces were obtained by pseudo-static application of the tangential aerodynamic forces. This approach led to considerable underestimation of blade forces, in part due to neglect of inertial effects, and in part because the effects of dynamic stall were not included.

Most machines designed after 1983 used the improved frequency response procedure with the inclusion of dynamic stall in their considerations (Indal 6400, Eole, FloWind 19m, Sandia/DOE 34m, Lavalin L24). These machines also had the advantage of the calculation of fatigue lives using the LIFE code (Veers 1981b, 1983b). It is understood that Adecon used their own techniques for their rotors.

3.2.5 Atmospheric Turbulence

The several fatigue failures of blades has encouraged the closer examination of the fatigue spectrum to which rotor blades were subject and also of the fatigue strength of typical connection details. It was realized that atmospheric turbulence could add considerably to the fatigue loading (Malcolm 1987a,b,c) and this was confirmed by careful examination of test data (Malcolm 1989a,b, Ashwill & Veers 1990a,b).

It was also realized that poor detailing of connections and environmental effects could substantially reduce the fatigue strength and it highlighted the need for laboratory testing of all connections.

Unfortunately the procedure to incorporate atmospheric turbulence in the structural response was not a simple one but required considerable time and computer resources. It has not been integrated into the initial design of any Darrieus rotor wind turbine.

The inclusion of turbulence in the frequency response of a rotor has emphasized the need to be able to also model the aeroelastic damping forces in order to avoid overpredicting response at natural frequencies.

3.2.6 Safety Factors

The approach to the ensuring of adequate reliability has varied over time and between designers. Some parts have been designed against yield while others have been designed against ultimate failure. Some designers have used a limit states approach while others have used the allowable stress technique.

The most recent codes (IEC draft Feb. 1992) recommend that a separate system of partial safety factors be used for the loads and for the resistance values. This has been followed only for the most recent designs (LavalinTech 1991g). Elsewhere the level of safety is not always specified. We know that the DOE 100 kW used a safety factor of 1.80 against local buckling during survival wind loading but we do not have information about other peak load conditions.

For wind turbine rotors the safety factors against fatigue loading are usually more critical. This was recognized by Alcoa (Alcoa Laboratories 1980) who recommended that extensive fatigue testing be done on a number of connection details.

Even if this is done it is still necessary to recognize the possible variation in loading spectra and in fatigue strength. It is unlikely that any existing Darrieus rotors would satisfy the requirements of the present IEC draft code in these matters.

Some companies and individuals have developed their own design procedures and allowable stresses. Adecon have used the MIL specification for the strength of aluminum in fatigue and have adjusted it for the expected stress concentrations. For the fatigue strength of steel details, they have used the S-N curves from Det Norske Veritas and claim to have kept stresses below one half of the welded strength at 10^8 cycles.

Indal Technologies did carry out a series of fatigue tests on the blade connection for their 6400/500 machine which were used in confirming the safety and were used as a basis in later studies. Instead of adopting certain load and resistance factors in fatigue design, Indal chose to require a fatigue life of 100 years using specified loads and mean strengths.

3.2.7 LavalinTech Design Specifications

The design specifications use by LavalinTech in their design of the L24 are reproduced below.

1. These guidelines are intended to ensure a uniform approach to the design of all components of the rotor (and drive train). They are mostly consistent with CSA F416-M1987, "SWECS - Safety, Design and Operation Criteria". These guidelines shall be updated whenever necessary.

2. The following <u>nominal values</u> shall be used:

Rotor Speed	50 rpm
Rated Power (electrical)	240 kW
Rated Torque (electrical)	45.8 kNm
Rated Torque (aerodynamic)	50.0 kNm
Average Cable Tension (operation)	64.0 kNm (each of six)
Wind Regime	8.0 m/s mean with Rayleigh distribution
High Wind Cut-off	20 m/s at mid rotor
Maximum Normal Braking	1.6 times rated aerodynamic torque
Maximum Emergency Braking	3.2 times rated aerodynamic torque

The rotor shall be designed for "Canadian conditions" implying a range of temperatures from -30°C to +30°C. Standard sea level density shall be used (1.225 kg/m^3).

3. The following loading cases shall be considered:

   a) Dead Load of all Components.

b) Survival Wind Loading: Mid-rotor windspeed = 62 m/s with no additional gust factor. Associated vertical wind shear exponent = 0.07.

c) Ice Loading on all Exposed Surfaces: The maximum ice thickness = 50 mm with a density = 900 kg/m^3. This is equivalent to 45 kg/m^2.

d) Static (mean) and Cyclic Loads During Normal Operation: These loads will be more fully defined later.

e) Loads Associated with a 10% Overspeed During Operation.

f) Maximum Static and Cyclic Loads in a 3-Second Gust While Operating at Rated Power: This gust is approximately 50% greater than the mean windspeed.

g) Loads from Normal Stops and Starts.

h) Emergency Stops and Electrical Transients: Transient torques emanating from the generator shall have the same peak value as the emergency braking torque.

i) Seismic Loads: These are not likely to be critical for this structure.

4. The following combinations of the above loads shall be considered.
   - i a + d (dead load + normal operation).
   - ii a + d + g (dead load + normal operation + normal starts/stops).
   - iii a + b (dead load + survival wind).
   - iv a + ½b + c (dead load + ½ survival wind load + full ice load).
   - v a + e + f (dead load + overspeed + gust).
   - vi a + e + f + h (dead load + overspeed + gust + emergency braking).

5. Frequency of loads. All loads shall be considered to occur once only or infrequently unless specified otherwise. Loading combination, ii) (dead load + normal operation + stops/starts) shall be considered to occur over a 20 year operating life at an average frequency of 5/day.

6. The normal design procedure shall follow the "limit states" approach commonly described in CSA codes. For static or infrequent loads the load factor shall be 1.25 for dead load and for survival wind load, and 1.50 for all other loads. A load factor of 0.9 shall be applied to the dead load if this results in a more critical condition.

The corresponding resistance factor for steel or aluminum parts shall be 0.9.

The static design of connections shall use a resistance factor of 0.67 (in accordance with CSA steel and aluminum codes).

7. Design for repeated (fatigue) loads shall be based on the specified loads. The S-N curves used in design shall correspond to 90% of specimens exceeding that life. The predicted life of all components must be at least 100 years.

8. Identification of peak loads from aerodynamic forces during normal operation shall be based on the rms of the cyclic response multiplied by 3.0 for blade out-of-plane bending and 4.0 for blade in-plane bending. Elsewhere the rms should be multiplied by 2.5.

9. Structural analysis of the operating rotor should ideally include effects of atmospheric turbulence (of at least 15% intensity) and aeroelastic damping of the structure. However, this process is time consuming and still somewhat uncertain. A more economical procedure is to use a deterministic analysis but to exclude the aeroelastic damping (these two approximations will tend to balance each other). Overall structural damping of G = 0.04 should be included.

10. <u>Overspeed Condition</u>

The overspeed of 10% greater than the nominal (50 rpm) speed is a condition that can occur only due to failure of the control system or of the regular braking system. Its duration should, therefore, be limited to a maximum of 10 seconds on any one occasion. The number of such occurrences should be assumed equal to twice per year and the total number of associated stress cycles calculated accordingly.

During those 10 seconds the number of cyclic stresses is limited and the same statistical ratio of peak to rms values as used in normal operation (see item 8) is probably not justified. It is suggested that peak stresses due to aerodynamic loads be based on mean stresses plus 2.0 times the rms.

11. <u>Braking Loads</u>
The structural response of the rotor to braking loads shall be based on the assumption that the normal braking torque or the emergency braking torque increases from zero to the specified value in a linear manner over 0.5 seconds.

These are "specified" loads (loads to be expected with some certainty) and a load factor of 1.5 should be associated with these loads.

12. <u>Operational Loads</u>

The procedure of using multiples of cyclic rms as a possible peak value is described in item 8 above. These are values already associated with low probability of occurrence but a load factor should still be used to represent uncertainty of geometry and structural response. A factor of 1.25 is suggested.

13. <u>Load Combinations</u>

The combination of statistically uncorrelated loads should follow the rule that the peak value of one load may coincide with the "normal" value of other loads (Turkstra's principle). The "normal" value of a load might be interpreted as a value that is not exceeded more than 50% of the time.

The normal value of a variable that follows a Rayleigh distribution is approximately equal to the mean value of that variable, or 1.25 times the rms. It is suggested that for this design the usual load factors be applied to these "normal" loads.

14. Design for Fatigue

The most accurate method of designing for fatigue (of the methods available to us) involves determining the rms stress of response at a number of wind speeds, identifying the typical distribution of stress excursions at a given wind speed, selecting an appropriate S-N curve, and incorporating this information into the LIFE6 program.

A slightly shorter route is to use the LIFE5 program which assumes a Rayleigh distribution of stress excursions at any wind speed. A study carried out by Indal Technologies showed that for the Indal 6400 this distribution was appropriate for in-plane blade response but was somewhat conservative for out-of-plane blade response. The latter may be accommodated in the LIFE5 program using a factor of 0.9 on the rms values, or in the LIFE7 program by selecting a Weibull distribution of fatigue stresses with a shape factor = 2.5.

An even shorter but more approximate method of fatigue evaluation may be used for initial design. This is based on using analysis results at one windspeed only (say 20 m/s). It is also based on the observation that according to the Rayleigh distribution one in a hundred cycles will exceed a range of 6.0 times the rms; but such high winds will occur only one twentieth of one half of twenty years. The number of such cycles is, therefore (at a 2 Hz frequency).

$$N = 20 \times 365 \times 24 \times 3600 \times 2 \times 1/100 \times 1/20 \times 1/2 = 315000$$

It will be reasonable, therefore, to design for 5×10^5 cycles with a range of 6 x rms. This is usually equivalent to 10^7 cycles with a range of roughly 4 x rms.

For locations where the distribution of peak stresses is known to be narrower, the 10^7 strength may be compared with a multiple of 3.5 x rms of a design and should be followed by an evaluation using LIFE5 or LIFE7.

Many published S-N curves do not extend beyond 10^7 cycles or imply an endurance limit at or near 10^7 cycles. This limit can be questioned when loading amplitudes vary, especially with aluminum. It is recommended that the log-log gradient of the S-N curve be extrapolated to 10^8 and then levelled out, or that the log-log gradient beyond 10^7 cycles be reduced by 2. This latter shape is consistent with the 1972 Det Norske Veritas code for steel structures.

4.0 DESIGN RATIONALE

The choice of configuration and materials for most Darrieus rotors has been driven by the wish to minimize the final cost of energy. The only machines for which this has not been so have been those intended solely for research.

The list of design variables is a long one. It includes

- total swept area
- height/diameter ratio
- number of blades
- blade airfoil profile(s)
- rotor speed
- blade chord(s)
- blade material and construction
- type and diameter of central column
- location and construction of horizontal struts
- number, size and preload of guy cables
- type and location of brakes
- standard or tailored gearbox
- type of motor/generator
- foundations
- type of drive shaft

Table 4.1 attempts to summarize, in chronological order, some of the key features associated with a number of wind turbines. In the subsequent sections comments will be made on the rationale behind the selection of these configurations.

4.1 Rotor Size

The question of what is the optimum size of a wind turbine has preoccupied the minds of Darrieus rotor designers as much as it has those of HAWT designers. The reasoning and controlling influences concerning optimum size are similar for both types of machines.

4.1.1 Past Practice and Experience

The earliest rotors were small and were used for wind tunnel tests. No attempt was made to study their economic optimum. The question of what was the optimum size was studied in a number of projects. Sullivan (1979a,b,c,d) built a model to examine the economic optimum configuration in the 15-20 m diameter range. Many assumptions were made and the results were used to define the DOE 100 kW (17m "low-cost" machine).

At about the same time Shawinigan Engineering (1978a,b) carried out a study on behalf of the National Research Council of Canada (NRCC) to identify the optimum large machine. This study indicated that the optimum lay somewhere beyond 10,000m^2 swept area. It was this study that was used to justify the construction of the 4000m^2 Eole wind turbine.

Table 4.1 Summary of Features of Darrieus Rotor Wind Turbines

machine /date	country	type (1) no. built	height x diam (m)	rpm	blades# x chord (m)	blade type	horiz. struts	central column	guy cables	airfoil	brakes see (2)	aero data see (3)	strutural data (4)	comments
NRC 14' 1972	Canada	R 1	6.0x4.3	100- 160	2/3x 0.152	extrusions	no	truss		NACA0012				wind tunnel
DAF 9 1973	Canada	R 3	9.1x6.1		2x.152	extrusion	yes	tube						1/4 scale model of Magdalen I
DAF 50 1975	Canada	R 12	17x11	71-90	2x.355	extrusion	yes	tube	3/4x22	NACA0015	L	F		
NRC/HQ 1977	Canada	R 1	36.5x24	28-36	2x.71	extrusion	yes	al. tube	4x28	NACA0018	A, H	F	OF	runaway, rebuilt
Sandia2m 1975	USA	R 1	2.0x2.0	180-520	2/3x.07		no	truss				F, M		wind tunnel tests
Sandia5m 1975	USA	R 1	5.0x5.0	95-380	2/3x.152	extrusion	no	tube	4x?	NACA0015		F, M		
Sandia17 R&D	USA	R 1	17x17	50-75	3x.50 2x.61	helicopter	X type	truss	4x25	NACA0012	L	F, M		
Tumac 1980	USA	R 1	12x9.5	46,58,70	3x0.24	fibreglass/polyester	no	tube	cantilever	NACA0015	E,L			
DOE 100kW 17m low-cost	USA	P 3	25x17	52	2x.61	extrusion	no	tube	4x22	NACA0012	L	F, M	OF,OMNM	
Alvawt60 1980	USA	P 1	19x13	64	2/3x.35	extrusions	mini	tube	3	NACA0015				accident
Alvawt300 1980	USA		37x25	35	2x.74	extrusion	mini	tube	3	NACA0015				
Alvawt500 1980	USA	P 1	37x25	41	3x.74	extrusion	mini	tube	3	NACA0015				
DornierDZ12 1981	Germany	P	18x12	90	3x0.30	extrusion	no	tube						
CENG D5 1981	France	R 4	5.0x5.0	0-250	2x0.25	frp (5)	no	tube		NACA0015				
TEV100	Romania	R	34.5x23	31	2x0.74	extrusion	mini	tube	3x	NACA0015				

machine /date	country	type (1) no. built	height x diam (m)	rpm	blades# x chord (m)	blade type	horiz. struts	central column	guy cables	airfoil	brakes see (2)	aero data see (3)	strutural data (4)	comments
FloWind17 1982	USA	C 319	23x17	53.5	2x.61	extrusion 6063-T6	cable	tube	3x2x22	NACA0015	L	F	OF	
PionierI 1982	Netherlands	R 1	15x15	30-60	2x.75	gfrp	no	tube	cant.	NACA0012	L	F	OF	
Indal6400 1983	Canada	P 2	36.5x24	45	2x.737	extrusion 6061-T6	yes	tube	3x2x28	NACA0015	L	F, M	OF, OM NF, NM	
FloWind19 1984	USA	C 202	25x19	52.5	2x.71	extrusion 6063-T6	yes	tube	3x2x28	NACA0015	L		OF	
Vawtpower 185	USA	C 40	24x17.5	50.5	2x.74	extrusion 6063-T6	no	tube	3x?	NACA0015	L			
Vawtpower 250	USA	P 1	27x22	50	2x/74	extrusion 6063-T6	no	tube	3x2x32	NACA0015	L			
FloWind25 1985	USA	P 2	30x25	42	2x.75	extrusion 6063-T6	yes	tube	3x2x?	NACA0015	L			
Sandia 34 1986	USA	R 1	44x34	28-38	2x.91 /1.22	extrusion 6063-T6	no	al tube	3x2x	SNLA1850 NACA0015	E L	F, M	OF, OM NF, NM	R&D test bed
Eole 1987	Canada	P 1	97x65	11-15	2x2.4	steel 480MPa	yes	tube	6x95	NACA0018	E, A, L	F, M		
Adecon19 1986	Canada	C 2-10	25x19	49	2x.60	extrusions	yes	truss	truss	NACA0018	H	F		hinged blades
Adecon SL38 1991	Canada	P 1	38x19	49?	2x.60	extrusions	yes	truss	truss	NACA0018	A, H			
Adecon SL55 1992	Canada	P 1	55x27	?	4x.0.60	extrusions	yes	truss	truss + cables	NACA0018				under construction
Lavalin24 1991	Canada	P 0	31x24	50	2x.74	extrusion	yes	tube	3x2x28	SNLA1850	L	M	OM, NM	not built
Alpha Real 1988	Switzerland	P ?	24x19.2	33/50	2x.72	extrusion	yes	3x?		NACA0012				
CENG D10 1986	France		10x10	93	3x.30	extrusion 6063-T5	no	tube	3x22	NACA0015				hinged blades
Cenemesa23 1990	Spain	P 0	30x23	52	2x variable	frp	no	tube	3x	SNLA1850 NACA0021				incomplete
GL20 1992	Canada USA	P 0	20x37	59	2x.74	frp pultrusion	yes	truss	3x2x12	SNLA1850	L			design only

Notes:

(1) R = research
P = prototype
C = commercial

(2) A = aerodynamic
E = electrical
L = low speed mechanical
H = high speed mechanical

(3) F = field data
M = model data

(4) O = operating
S = stationary
N = natural frequencies
M = model predictions

In 1981 DAF Indal Ltd began a study also aimed at identifying an optimum size. Their conclusions (DAF Indal 1981, 1982a,b,c,d,e,f,g, 1983, 1984a,b) were based on a number of point designs and on the detailed evaluation of alternatives and costs. The final recommendation was for a machine rated at about 1500 kW (1700m^2)

There has been an overall trend towards larger machines but the progression has been very irregular. Figure 4.1.1 summarizes this trend in swept area by year of installation (or year of design for machines that have not been built). The two machines that are of much greater size than all others are Projet Eole and the Sandia/DOE 34m Test Bed. The latter was designed mainly as a research machine and should not be considered a commercial prototype whereas Eole was intended as a commercial prototype.

Important experience has been obtained at both ends of the size spectrum. FloWind found that the manufacturing and installation costs of the 19m were not greatly in excess of those for the smaller 17m wind turbine. This meant that the 19m was more cost effective overall. This can be explained by the continued use of standard, available parts and components and by the use of similar infrastructure, monitoring, and interconnection details for each. In addition, the resources required for the maintenance and repair of each machine were almost identical.

Experience with the 4000m^2 Projet Eole has demonstrated the difficulties associated with a very large machine. The initial engineering requirements were formidable; not only were new problems encountered but the value and exposure of the project demanded greater precision and reliability. Installation, especially at a relatively remote location, required special and expensive equipment. Finally the correction of any problems that did arise involved considerable expense and resources, and the lost operating time implied greater loss of revenue.

4.1.2 Scale Effects

The effects of scale on the blades and central column of a Darrieus rotor wind turbine have been studied by Malcolm (1990c). He concluded that, assuming similar construction and materials, their mass (and likely cost) was proportional to the cube of the rotor size (diameter). This also held for the cost of the major drive train costs (gearbox and couplings) so that it was likely that the total manufactured cost would increase with the third power of diameter.

This conclusion is probably optimistic since large components are normally less available and require special manufacturing techniques. It is also less likely that large components will enjoy the cost benefits of mass production. The other major costs are for transportation, site preparation and infrastructure. These are likely to increase at least with the square of the diameter.

The energy capture, however, is proportional to the swept area and to the cube of the wind velocity. In a region with "normal" vertical wind shear this will result in the total energy increasing with the diameter to the power of 2.4. There is, therefore, an overall negative benefit of increased size.

The conclusion concerning the negative benefit of size is given some support if the total mass (of rotor and guy cables) per swept area is compared to the swept area for actual and planned machines. This is done in Table 4.1.1 and Figure 4.1.2 which show an overall trend towards less efficient use of mass with increasing size. Only a limited number of machines are included in this comparison since it was not possible to obtain the necessary data for all models. While total mass is not necessarily directly related to

total cost because of different materials and manufacturing techniques, there is, however, usually some correlation.
There are, of course, other factors that might favour large machines. These might be the cost of land, the cost of maintenance and the cost of the roads and infrastructure. However, there is not yet evidence that megawatt size machines may be favoured in this way.

Table 4.1.1 Rotor Mass and Rotor Size

turbine	**swept area** m^2	**total mass**[1] **kg**	**mass/area** kg/m^2
FloWind 17m	241	7254	30.1
FloWind 19m	315	10,962	34.8
Adecon 19m	316	9100	28.8
[2]Cenemesa 23m	460	9752	21.2
[2]GL20	471	8014	17.0
[2]Lavalin 24m	481	12,890	26.8
[3]Magdalen Is 24m	478	14,961	31.3
Indal 6400	495	17,770	35.9
[3]Sandia 34m	955	72,198	75.6
[2]Lavalin 1MW	2000	11,100	55.5
Eole	4000	300,000	75.0

Note: 1. Mass includes rotor and guy cables (or rigid frame).
2. These turbines have been (partially) designed but not built.
3. Research machines.

4.1.3 Future Trends

Several studies have been made into the most cost-effective size of HAWTs, such as those by Molly[1] and Milborrow[2]. The former reviewed available costs from California at that time and concluded that the optimum appeared to lie at rotors of 20-30m diameter. A conservative projection placed an optimum diameter between 30 and 40m.

The most common size of HAWT being installed in utility-connected windfarms in North America and Europe at present are those with diameters of 25-35m. The reason for this is that these are the largest machines which are available and which have proven reliability. They promise to be more cost-effective than earlier generations of 15-20 m diameter machines. Budtz[3] set out some of the rationale followed by

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[1] Molly, J.P.,"Maximum Economic Size of Wind Energy Converters", Proceedings EWEC'89, Glasgow, July 1989.

[2] Milborrow, D.J.,"Will Multi-megawatt Wind Turbines Ever be Economical?", Proceedings EWEC '89, Glasgow, July 1989

[3]Budtz, L.,"Vestas V39 - 500kW, the Cost Effective Turbine Design", Proceedings of AWEA

Vestas-DWT in establishing the V39 (500 kW) as an optimum for the 1990s.

The trend in the size of Darrieus rotors is likely to follow that of horizontal axis wind turbines since the same laws of scale apply and they require the same infrastructure costs.

4.2 Height/Diameter Ratio

The aspect ratio of Darrieus rotors has gradually increased. Early machines had aspect ratios of close to 1.0, perhaps because this shape can minimize the length of blade and column for a certain swept area. Later designs have acknowledged, among other things, that the cost of the drive train (approximately 50% of the total machine cost, DAF Indal 1984c) is very dependent on the value of the low speed torque; if the aspect ratio is increased then the rotor speed increases (to maintain the same relative wind speed and tip speed ratio) and the torque decreases if the power is constant.

The influence of gravity is related to the aspect ratio. Lower aspect ratios (with or without horizontal struts) lead to greater stresses due to gravity. In addition, the precise blade profile can be tuned to minimize gravity and radial aerodynamic effects (Eggers, Ashley, & Digumarthi, 1991).

Increasing the aspect ratio is another way of increasing the average height of the rotor and capturing more energy (in most sites). However, it must be balanced against the added material required in the rotor and guy cable system (for the same swept area) to maintain the necessary stiffness of the rotor. These objections can be removed if the principles in the concept of a dynamically soft rotor (LavalinTech/R Lynette 1992) are followed.

4.3 Number of Blades

Several early machines were constructed with three blades (NRC 4.3m, Sandia 2m, for example), no doubt in the belief that both the torque and structural response would thereby be more benign. While these motives were justifiable, it was later realized that the material and installation cost of a two-bladed machine were considerably lower.

For a given solidity it is structurally advantageous to have fewer blades of larger chord rather than more blades of smaller chord. This is because the section modulus (and hence the bending stresses) are dependent on the square of the chord size whereas the aerodynamic loads are dependent on only the first power of the chord (for a given rotor size). In addition a one- or two-bladed rotor lends itself to easy assembly while lying on the ground prior to erection.

The structural dynamics of two- and three-bladed machines are quite different from each other, due mainly to the different relationships between column motion and out-of-plane blade motion. Two-bladed rotors can experience severe "butterfly" excitation (Malcolm 1984a, Lobitz 1981a,b) which has been responsible for damaging fatigue stresses in a number of rotors. The structural modes and response of three-bladed rotors have been less well documented. However, a three-bladed rotor is structurally non-directional (like the three-guy cable system) and recent design work indicates that the response of three blades (of a certain chord) is more benign than two blades of the same chord. The largest three-bladed machine was the Alcoa 500kW (Alvawt 1980) which pre-dated the improved understanding of Darrieus structural dynamics.

Conference, Palm Springs, September 1991.

An attempt at calculating the structural response and the natural frequencies of a three-bladed rotor was presented by Curtis (1982a,b). Several of the rotating frame influences were not included and there is no mention of any field measurements to confirm the calculations. This machine (the Tumac 4kW) was also a cantilevered rotor and was funded by the U.S. Department of Energy through Rockwell International Corporation.

The choice of blade chord, and hence the number of blades, can also be influenced by the choice of blade construction. For example, the largest single aluminum extrusion available is approximately 0.76m so that for machines greater than about 25 m diameter there might be a choice of combining multiple extrusions or of increasing the number of blades. The (planned) Adecon SL55 will have four blades partly because of the availability of extrusions from earlier machines.

The advantages of both approaches are summarized in Table 4.3.1.

Table 4.3.1 Advantages of Fewer or More Blades

item	more blades	fewer blades
construction cost	higher	lower
assembly costs	higher	lower
choice of fabrication techniques	better	poorer
strength/weight ratio	poorer	better
torque ripple	better	poorer
structural dynamics	better	poorer

4.4 Blade Airfoil Profile

Until recently all Darrieus rotor blades have used a NACA00xx symmetric airfoil. The reasons for this were the high lift obtained and good stall characteristics combined with low drag and the amount of data already available for these airfoils. Earlier rotors used mainly the thinner NACA0012 and NACA0015 airfoils. However the requirements of increased flatwise strength has led some manufacturers to choose NACA0018 profiles.

In the early 1980s it was realized that to improve the cost effectiveness of a wind turbine it was necessary to maximize the energy capture while minimizing the cost of all components, including the drive train (Kadlec 1980a,b, Kadlec 1983). This implied minimizing the peak low speed torque by avoiding airfoils with high lift coefficients. This led to the development of a family of airfoils at Ohio State University based on laminar flow over the leading section of the blade and earlier stalling (Hoffman & Gregorek 1989). These airfoils were tested on the DOE 100 kW rotor and were included in the Sandia/DOE 34m Test Bed (Berg 1985).

While several studies have confirmed the potential improvements to be obtained by using the laminar flow, or "tailored", airfoils (Malcolm 1986), the test results have been mixed (Ashwill 1991). The maximum power appears to have been successfully attenuated except in the presence of insect accumulation ,when attenuation was diminished (see Section 2.1.3.2).

The Sandia family of tailored airfoils has been specified in some of the latest designs in Canada and the US/Spain (LavalinTech 1990, 1991g, LavalinTech/R Lynette 1992a,b, Schienbein 1989). These designs have not yet been built.

4.5 Rotor Speed

The choice of rotor speed is controlled by, among other factors, the wind regime, the solidity, and the machine power rating. In principle it is possible to extract more energy with the least blade area by increasing the rotor speed. However, this can lead to blades that will not withstand the aerodynamic and inertial loads.

An illustration of this is the NRC/Hydro Quebec (Magdalen Islands) 24m machine which was run at speeds of between 28 and 36 rpm. Later this same configuration was uprated to 45 rpm and rated at 500kW (to become the Indal 6400). This was satisfactory when developers wished to increase the rating of machines but was effective in increasing total energy capture only in sufficiently high wind regimes.

Increasing the rotor speed does decrease the low speed torque and hence reduces the cost of the drive train. This principle has been utilized by the recent Adecon SL38 and SL55 designs.

Other designs have been constrained by other considerations. For example, the CENEMESA23 was designed to use an existing (FloWind 19m) power module and the rotor speed was therefore predetermined.

4.6 Blade Chord

Blade solidity (defined as the developed surface area of all blades divided by the swept area) is one of the key design parameters which, as has already been mentioned, has to be combined and balanced with the other major variables. For minimum cost, solidity should be kept low. However, the lowest values compatible with structural integrity (using existing fabrication techniques such as aluminum extrusions) appear to be about 0.10.

Ideally the blade chord should be varied , from a minimum at mid-rotor to a maximum at the roots, for maximizing energy capture (Kadlec 1983). Such a shape is also a good one for structural purposes and it has been incorporated into the Sandia/DOE 34m Test Bed. However, fabricating a continuous taper or even a series of steps greatly increases the rotor cost. For that reason no taper has been specified in some of the latest commercial designs.

4.7 Blade Material and Construction

Table 4.1 includes a brief description of the type of blade used on each turbine.

The first blades were made from stretched and formed steel sheet or from helicopter-like combinations of aluminum extrusions and fibreglass. The former were difficult to form to a smooth airfoil while the latter were expensive. Laminated wood was also tried on early machines (Butler and Blackwell, 1977). The use of multi-cell aluminum one-piece extrusions offered a combination which promised to solve many problems. They have been adopted for most machines from the DAF 9kW onwards.

The choice of blade material is influenced by the ability to fabricate an inexpensive and fatigue resistant connection at the roots and spices. Extruded aluminum alloys, such as 6063-T6, do not have a high fatigue strength compared with aircraft standard alloys or with high strength bolted steel connections. This has led to a number of the fatigue failures, although most failures could have been avoided with improved connection details. Those details have, therefore, received much attention. The single cover plates and tight fitting bolts, combined with an epoxy adhesive used on the Indal 6400 has proved successful (although expensive).

The solution to the low fatigue strength of mechanical connections to aluminum extrusions may lie in the use of adhesive connections. VAWTPOWER (Vosburgh 1986) retrofitted blade splices with bonded aluminum cover plates and FloWind has carried out blade patching and retrofits with adhesives. However, the technology of bonding to aluminum in a corrosive environment requires some further development.

The use by Adecon of thinner skin extrusions, which are bonded together lengthwise, has resulted in lower overall weight. However, the adequacy of their aluminum inserts and mechanical fasteners for blade connections has not yet been confirmed by prolonged operational experience.

The Sandia/DOE 34m rotor required blades larger than any which could be extruded from a single aluminum die. Therefore, two or three extrusions were connected lengthwise by a series of recessed bolts. The blade splices coincided with a change in chord size which was accomplished through bolting both blades to a common, slightly tapered, aluminum block.

For their L24 design LavalinTech (1991g) adopted a commercial blind fastener, tight-fitting holes, and material cold working to improve the strength of the connections to aluminum.

The proven fatigue strength of high strength bolts in steel construction was one reason for the choice of a steel-core blade for the 96m x 64m Eole machine (Shawinigan 1985). However, that type of blade construction is heavy and was one reason for the high mass-to-swept area ratio of that rotor.

4.8 Type and Diameter of Central Column

The principal choices of column type are closed tube or open truss (usually three- or four-sided). Some of the very early rotors used three-sided trusses, perhaps because they were light and available. From 1975 onwards most central columns were steel tubes, following the choice for the Sandia 17m research machine. This is perhaps unfortunate because, although it was realized that the truss is more efficient structurally, " the truss radii [overall width] were judged to be too large because of blade consumption [sic] near the tower axis, wind drag and tower shadow effects, and unsightliness" (Weingarten & Blackwell 1976).

On larger turbines the required diameter for a closed tube is beyond those readily obtainable and the lower curvature of the tube wall means that local reinforcement may be necessary. This results in even higher costs for the closed tube.

The choice of column type was one of the parameters studied by DAF Indal (1984b). The final recommended configuration was for a 58m x 39m rotor with a four-sided truss column of side 2.4m made up of tubular members.

The Sandia/DOE 34m Test Bed has a central column which is a 3.0m diameter aluminum tube with internal reinforcement. This choice was not made for commercial reasons but was intended to maximize the bending stiffness while limiting the weight. This choice also lessened the variation in guy cable tension with temperature changes.

It was recognized (DAF Indal 1984b) that the ideal shape for the central tubular column was a taper with smaller diameter at the upper and lower ends and a maximum near the mid-rotor. While such a design may reduce the total weight, it may increase the cost. It was incorporated into the Indal 6400 rotor for which the columns were made by a brake press technique rather than spiral welding.

The only manufacturer to use a truss column recently has been Adecon Energy Systems who has thereby reduced the mass-to-swept area ratio. It is not known if the additional drag from such a column has resulted in reduced performance or undesirable response.

One reason for the dismissal of truss columns as too wide and unsightly has been the insistence on maintaining a high value for the first rotor natural frequencies. This has meant that the design of the column has been governed by stiffness rather than by strength. If this condition is removed, as it is in the dynamically soft rotor concept (LavalinTech/R.Lynette 1992), then the side dimension of a truss can be reduced considerably.

4.9 Horizontal Struts

Horizontal struts are members connecting the central column to the blades at a location at least 0.10 of the rotor height away from the roots. There are also "mini" struts on some machines. These are horizontal members which are very close to the root connections. Although these mini struts are separate members, they will not be included in this discussion. Mini struts can facilitate the root connection but do not reduce the (bending) loads that have to be carried in that region.

Horizontal struts play several roles. They stabilize the blades during survival winds, they transfer part of the torque into the central column, they reduce operating mean and fatigue stresses in the blades, and they strongly influence some of the natural frequencies of the rotor. These are all useful features. However, they also add weight and cost and they introduce additional drag and energy loss. Furthermore they represent one more set of connections to the blade and potential failure locations. They can best be classed as a necessary evil.

Perusal of Table 4.1 shows that those machines designed in the USA have mostly been without struts whereas of those developed in Canada nearly all have had struts. This may reflect the relative influence of engineers with either an aeronautical or a structural leaning. One exception to this rule is the FloWind 19m rotor. In Europe, the Alpha Real rotor has struts whereas the Dornier, Pionier I, and the TEV 100 machines have not.

The selection or rejection of horizontal struts cannot be made independently of the other major parameters. If a small blade chord (and low solidity) is required, then struts may be necessary to maintain structural integrity. If a more sophisticated (and expensive) blade with a tapered chord is selected, then the need for struts can be avoided.

Horizontal struts have usually consisted of a steel truss firmly attached to the (tubular) central column, tapered to the blade chord dimension and with some aerodynamic fairing on the outer half. Struts on the DAF 50, the FloWind 19m ("D" type), and Eole were of this type. The Indal 6400 had struts which were steel trusses but which were free to move out of plane through a set of springs. This system was later replaced by a single steel I-beam which passed through a hole in the tubular column.

While the strut-column connection has been one problem, the strut-blade connection has proven even more troublesome. Ideally this connection can be designed not to transmit any in-plane bending (i.e. to have a horizontal out-of-plane pin). The DAF 50, the Indal 6400, and the L24 design all incorporated such a detail, which may have increased the cost but which was structurally advantageous. The FloWind 19m and Eole both incorporated a rigid strut-blade connection. This has possibly led to some problems for the former, and probably added further to the costs of the latter.

Both types of strut-blade connection have drawbacks. The answer may lie in including the desired flexibility in the material of the strut rather than in a mechanical hinge. This has been proposed in a recent report by LavalinTech/R.Lynette (1992b) to the Canadian government.

Adecon Energy Systems has included struts on all of its machines. On the Adecon 19m, the strut-blade connection was effected by encircling the blade, and on the SL38 the strut trails the blade in order to lessen aerodynamic interference on the blade.
It has been found that if struts are placed further away from the blade roots than 0.18 of the rotor height, then the aerodynamic drag losses become significant, especially at the strut-blade interface. A preferred location is at about 0.15 of the rotor height.

Cable struts have also been used (principally on the FloWind 17m and VAWTPOWER 185). These struts do not restrain the blade from out-of-plane motion but are effective in stabilizing the blade under survival wind loading. Such struts might be possible candidates in a fibre composite blade, which is able to carry all operating loads without struts but which may be so flexible that they might otherwise be unstable in survival winds.

4.10 Guy Cables

Pretensioned guy cables have been the most common method of supporting the rotor. The alternative to cables has been a rigid external structure (the Adecon 19) or a cantilever design (the Tumac, and the Pionier I).

In theory, a set of guy cables is the most efficient method of restraining a tall structure; it may occupy a greater area but it will be lighter than an equivalent cantilever design. In practice there are some other disadvantages:

1. More foundations must be included;

2. The tension must be maintained fairly constant;

3. Cables are especially awkward on sloping ground;

4. The cables must have sufficient clearance from the blades;

5. The cables must be angled at 35° to the horizontal for maximum stiffness;

6. Care must be taken not to excite the transverse oscillations of the cables.

Some early machines used a set of four guy cables but this has been replaced by sets of three (or three pairs for redundancy) since such a system is self-equilibrating. It also reduces the total length of cable required and the number of anchorages.

The cable size and preload have a strong influence on the frequency of the natural modes of the rotor which involve in- or out-of-plane motion of the central column. Designs have aimed at keeping these fundamental frequencies (at operating speeds) well above the one-per-revolution frequency which is the major source of excitation. This has resulted in the cable system always being controlled by stiffness rather than by strength.

For very large machines, such as Eole, it has been found difficult to avoid excitation of the fundamental transverse natural mode of vibration. The solution for Eole was to give the cables intermediate support from truss towers. For the Sandia 34m Test Bed very heavy cables were used (partly in order to overdesign the machine for research purposes).

The cable tension can also affect the control strategy. For example the Indal 6400 cable tension is low enough to allow the rotor to coast in low winds and to allow the rotor speed to decrease without exciting the cable natural frequency. The FloWind 19m, however, has a relatively high cable preload which increases the cable natural frequency but precludes the possibility of coasting and a drop in rotor speed. The external structure used for the Adecon 19 did result in more modest anchorage requirements and did lessen the downward thrust on the rotor and bearings. However, it was considered unsightly and may have affected the local airflow.

Cantilever systems have the disadvantages of heavier (although fewer) foundations, and larger bearings. The central column/tower must be designed to carry full bending effects as well as cyclic loading.

4.11 Type and Location of Brakes

Braking philosophy is closely allied to safety philosophy and the ability to ensure that the wind turbine does not pose a threat to people or property. A (curved blade) Darrieus rotor does not enjoy the same number of options as most HAWTs. For example, it is not possible to change the pitch of the blades or to yaw the machine out of the wind. But a number of options still remain and include the following.

aerodynamic brakes
low speed mechanical brakes
high speed mechanical brakes
electrical braking

One of the columns of Table 4.1 indicates the type of braking system(s) used by each machine.

Most manufacturers have chosen some kind of mechanical brake on the low speed shaft (implying the low speed side of the gearbox) and most of those have not included any other braking system. However, there has been some form of redundancy, or fail-safe operation, in those brakes.

The later versions of the DAF Indal 50 kW and the Indal 6400 featured a hydraulically supported bullgear which could be lowered onto a set of brake pads. Any fault would trigger the control system to release the hydraulic pressure and hence activate the braking system. Such a release was also activated by an independent centrifugally operated overspeed switch.

The FloWind 17m and 19m turbines use a single brake disc but three independent sets of brake calipers, each of which can stop the machine. In addition, there are two independent accumulators.

A single high speed brake relies on the integrity of more connections but it can appear attractive financially. This was employed on the Adecon 19m and the NRC/Hydro Quebec 24m and is selected for the Adecon SL38. However, the latter also includes paddle-type aerodynamic brakes for overspeed control.

Aerodynamic brakes are attractive because they need not cause large forces to be aplied to other components. Early machines such as the NRC/HQ 230kW (Magdalen Islands) and the DAF Indal 50kW used blade-mounted spoilers at mid-rotor which could be activated in the event of overspeed. However, it was found difficult to ensure the reliability of these devices and an unreliable device cannot be used as an emergency measure.

Electrical braking is normally associated with variable speed machines of which there have been few - Pionier I, Eole, and the Sandia 34m. These machines have used the specialized electrical drive to also control the rotor speed for operation and in all cases the rotor is finally stopped and parked with a mechanical brake.

4.12 Gearbox

The gearbox of a Darrieus wind turbine is normally ground mounted and the options available are, therefore, less restricted by size, weight, and maintenance constraints. The following gearboxes, or speed-increasers, have been used.

planetary or helical gearboxes
parallel or right angle gearboxes
tailored bullgear and pinion
belt drive
gearbox and lower bearing combination
direct drive

The choice is an important one since the gearbox can be approximately 25% of the total cost of the wind turbine (before installation). The least expensive solution may be to use a standard (helical parallel shaft) available gearbox, but tailored systems can eliminate some other components.

Adecon, VAWTPOWER and Alpha Real have used the low speed gearbox bearing to act as the lower bearing for the rotor. This simplifies the support system but the additional cost of modifying the gearbox can be considerable. It also may result in higher torque ripple in the drive train since there is no (relatively) soft low speed shaft to accommodate the two-per-rev torque impulse. In addition, some scheme has to be included for supporting the rotor while the gearbox is removed.

As mentioned in Section 5.11, Indal combined the brake disc with a tailored bullgear system. While the concept was simple, it resulted in the wind turbine manufacturer also having to design and perfect a gearbox.

Belt drives were included in the early DAF Indal 50 kW and were also present in the NRC/Hudro Quebec 24m. Adecon proposed a single-stage kevlar belt drive for their SL55 model. Such a scheme does make good use of the ground-based nature of the Darrieus drive train and it could drastically reduce costs of the speed increaser. However, belt drives are normally associated with high energy losses, and high reliability may not come immediately.

The 4000 m^2 Eole rotor uses a direct drive (Beausoleil 1985, Richards 1985a) which is combined with a static frequency converter system. While this drive has worked well, with no undue energy loss, its cost precludes it from being an example that others might want to follow.

4.13 Type of Motor/Generator

The type of motor/generator depends largely on how the wind turbine energy is to be used. A small machine for remote use might store energy in batteries and require a dc generator. However most turbines discussed in this report have been connected to some kind of utility grid and have therefore had to supply power at a defined voltage and frequency. The three most common methods of doing this are:

synchronous generator
asynchronous (induction) generator
direct drive and frequency converter

With very few exceptions all the wind turbines reviewed in this study have used an induction motor/generator. The main reasons for this choice have been the greater compliance or softness (compared with a synchronous generator) offered to the drive train, and the general availability of such units. The slip of the induction generator is important for a two-bladed rotor because it may be necessary to attenuate the two-per-rev torque pulse (both for the drive train and for the utility).

The problem of torque ripple can also be solved by adoption of a frequency inverter, or some other variable speed system. Such systems are still expensive and have only been justified for R&D machines (such as the Sandia 34m) or for a large machine such as Eole.

4.14 Summary

The trends and philosophies that have shaped the Darrieus rotor industry may be summarized as:

1. It is difficult for any study, even one focusing on point designs, to fully anticipate the engineering and organizational problems associated with large machines. If larger machines are ever to become viable they must, therefore, be approached incrementally.

2. The prediction of increasing mass per swept area with size appears to be supported by experience. It is, therefore, important that larger machines make use of innovative design concepts and application of materials.

3. There is a trend to increased height/daimeter ratios in order to decrease torque and drive train costs.

4. While three bladed machines may decrease torque ripple and out-of-plane blade fatigue stresses, all other arguments favour two-bladed rotors.

5. The available selection of proven airfoils has been limited.

6. While energy capture can, in theory, be increaseed by adopting tapered-chord blades, practical considerations have favoured constant chord blades and horizontal struts.

7. The adoption of a steel truss in place of a steel tube can offer lighter design and other structural advantages.

8. Guy cables may pose installation problems but they do allow an overall lighter and less expensive design.

9. Innovative designs, such as those incorporating out-of-plane hinged blades have suffered from inadequate financing and insufficient testing.

5.0 OFFSHORE LOCATION

5.1 Introduction

Ljungstrom (1979a, 1979b, 1980a, 1980b and 1986), Bergkvist, Ljungstrom and Stenstrom (1982) and Hardell and Ljungstrom (1978) have reported on the design of a class of large scale cantilever supported Darrieus turbines aimed at off-shore installations. Schematics of two early designs (Hardell and Ljungstom 1978) are shown in Figure 5.1a. A later concept (named LDB) is shown in Figure 5.1b.

Lindley, Simpson, Hassan and Milborrow (1980) reported on a technical and economic assessment of the generation of electricity using very large wind turbine generators located in shallow waters off the coast of the United Kingdom. Kerr (1986) has described the results of a design study for support structures for an offshore array of straight-bladed vertical axis wind turbines having cantilever support.

Graves Smith and Jolly (1986) have provided a good summary of the design process for a very large scale offshore HAWT, including a discussion of how the design criteria were derived from historical wind and sea condition data and the seabed conditions for the proposed installation site.

Hasted and Olsen (1989) and Jespersen (1991) have described the design and construction of the first offshore windfarm in the world in Denmark (near Vindeby), consisting of 11 horizontal axis wind turbines each having a rating of 450 kW.

Although no curved blade Darrieus wind turbines have yet been installed offshore, much useful information on the construction of support structures, installation techniques, marinisation of the turbines and operations problems can be derived from the description of the Danish installation and the various design studies.

5.2 Size and Cost

Most design studies have focused on very large turbines (several MW capacity and rotor diameters often well in excess of 100 m) installed in fairly deep water (up to 50 m). However the only offshore windfarm constructed to date (Vindeby) uses a relatively small 450 kW HAWT having a rotor diameter of only 35 m and installed in water depths of only 2.1 to 5.1 m.

Few wind turbines exist with rotor diameters greater than 50 m and therefore cost estimates for large machines must be treated with caution. For example, the Eole Darrieus curved blade VAWT in Canada has a rotor diameter of only 64 m and the largest wind turbine currently in operation (MOD-5B HAWT) has a rotor diameter of 97.6 m, a hub height of 61 m and is rated at 3.2 MW. The Eole turbine is roughly at the lower end of the size range of turbines that have been considered in most offshore studies.

Not surprisingly, the costs of the offshore wind turbine supporting structures (manufacture and installation) and to a lesser extent the costs of the power collection system dominate the total cost of the offshore installations. For example, the already operating Vindeby offshore windfarm is expected to produce energy at a cost 50% higher than for an equivalent wind farm on land, even though the energy production is expected to be 60% higher than on land.

However, it appears that the cost of the support structure for an offshore turbine does not increase in proportion to the turbine cost and output. Therefore most design studies have finally concentrated on evaluating the design and economics of offshore clusters of very large turbines.

Those involved in designing, installing and commissioning the Vindeby wind farm have already concluded that major development of off shore wind farms is only possible if big machines of simple and reliable design are available on the market (Jespersen 1991). It was also stated that the 1 MW stall-regulated HAWT being developed by ELKRAFT would support more cost effective offshore wind farms.

Ljungstrom (1986) reported on the results of a preliminary performance and cost of energy analysis of several innovative wind turbines (existing and proposed) in 100 unit series installed both on land and offshore. All but one of the turbines had swept areas of between 3500 and 5350 m^2. These results are shown in Figure 5.2. The LDB-70/LS (Figure 5.1b), 3X-70 Triol, and the IST-3X-70 (3S-70 Triol on an inclined shaft) are all curved blade Darrieus turbine concepts.

5.3 Blade Materials and Construction

Graves Smith and Jolly considered high carbon steels, aluminum alloys, titanium and GRP as possible blade spar materials. Aluminum alloy N8 was selected after considering corrosion resistance, weight and fabrication problems. A stiffened aluminum sheath was proposed for providing the airfoil section. The spar joints were considered to be the most fatigue critical and bonding was selected over welding or rivetting to reduce crack initiation. (Bonding has already been used in Darrieus blade joint designs.)

Their conclusions are significant for offshore Darrieus turbines since aluminum blades have already proven to be the most cost effective choice for these turbines in existing designs. Due to its good corrosion resistance, the aluminum blade (even single section extrusions) could also be a strong candidate for offshore Darrieus turbines. Indeed Ljungstom's LDB rotor concept uses two aluminum blade elements arranged in parallel and interconnected by transverse struts. It is claimed that this approach allows fairly small constant chord partly extruded aluminum blades to be used cost effectively on MW scale Darrieus turbines.

However, the fatigue strength of bonded aluminum joints is known to be significantly reduced by exposure to salt water. Furthermore, the lower portion of the rotor blades for a Darrieus turbine installed offshore will likely be more exposed to the salt water than will the blades of a horizontal axis wind turbine. Therefore considerable analyses and tests would have to be carried out to qualify bonded aluminum blade joints for offshore applications.

5.4 Support Towers and Rotating Masts

Offshore Darrieus rotors will likely be cantilever supported or supported using a combined cantilever and guy support system as proposed by Ljungstrom (1986) (see Figure 5.lb). The overall cost of energy advantage of the latter approach is not clear. This approach may be dictated by the desire to use a large diameter rotating base ring (integrated with the tower and guy cables) to support the rotor and to drive multiple generators arranged around the periphery. Therefore it is difficult to separate the advantages of a partly cable supported rotating mast from the cost advantages claimed for the entire system concept.

Graves Smith and Jolly (1986) studied "soft" and "hard" (stiff) reinforced concrete tower design approaches for the offshore HAWT. The results of this study merit careful consideration for offshore

Darrieus VAWT designs where the cantilevered fixed tower (equivalent to the lower portion of a HAWT tower) and perhaps even the rotating mast could be constructed of concrete. Concrete has already been found to be the most cost effective tower material for straight bladed cantilevered VAWTs.

Graves Smith and Jolly (1986) chose reinforced concrete as the HAWT tower material rather than steel because of the proven durability of concrete in marine environments and because prestressed concrete can cause extensive microcracking at the anchorage points. However, this usually leads to heavier sections. This may not be a concern for the nonrotating tower.

Kerr(1986) noted that the durability of well constructed reinforced and prestressed concrete structures in the marine environment is now well established and that steel structures require well-engineered and maintained corrosion protection systems and tend to have higher maintenance costs particularly for areas in the splash zone. This is cause for equal concern in both HAWT and VAWT offshore turbine designs that use steel towers and base structures.

Graves Smith and Jolly (1986) found that the costs of their "hard" and "soft" HAWT tower designs were virtually the same. However the "hard" (or stiff) tower design eliminated the risk of resonance due to aging of the concrete. (Progressive hydration of the cement causes an increase in the elastic modulus and hence the natural frequency of the soft tower could shift significantly over the life of the structure.)

Lattice steel towers may not be favored for durability reasons (especially in the splash zone) and also because the enclosed and sealed power module would have to be a separate structure not already integrated into the tower itself.

5.5 Supporting Structure

Kerr (1986) pointed out that the support structure (substructure and superstructure) has to meet three basic requirements: strength, dynamic response and space for the power plant. The major environmental loads on the structure can be expected to be due to waves, turbine wind loads and sea ice in some locations.

The supporting structure cost may dominate the total installed cost of an offshore VAWT as it does for a HAWT. To minimize this cost, methods and cost of transportation to site and emplacement on the seabed are of major importance.

The turbine supporting structure design (foundation and platform) for an offshore curved blade Darrieus wind turbine will be site specific as it is for HAWTs and other VAWTs. All foundation design approaches (such as gravity and pile type) that have been implemented or proposed for HAWTs are also applicable to cantilever supported Darrieus VAWTs.

Kilar, Tiller and Ancona (1980) reported on the findings of a comprehensive assessment of offshore wind energy conversion systems. However, most of the analysis focused on HAWTs since "more credible engineering information is available on the popular HA WTG than the VA WTG plant". Figure 5.5a (from Kilar, Tiller and Ancona 1980) shows a guy supported curved blade Darrieus rotor mounted on a support structure. Two types of guy anchors were suggested:

1. Guy cable anchored to the sea bottom for the case of a bottom mounted platform.

2. Guy cable attached to outrigger for a floating platform.

Cantilever supported turbines were not discussed. It seems unlikely that a guy cable supported turbine could be more cost effective than a cantilever supported turbine for a floating platform. However, bottom anchored cables and a bottom mounted platform may be a reasonable shallow water option.

Hardell and Ljungstrom (1978) reported that their "L" shaped (the two blades are arranged at 90° instead of at 180° as for most conventional Darrieus turbines), double blade ("biplane") Darrieus rotor design resulted in lower design wind loads making it possible to use a narrower foundation tower base diameter and a reduced base slab diameter. They claimed that this also had the advantage of reducing wave impact and pack ice loads.

The support structure design must address the minimum clearance between the rotor and the maximum design wave crest. If Kerr's criteria for HAWTs is adopted (4 m minimum clearance), the base of the Darrieus rotor (the lower blade to mast connection point) should be at least 4 m above the maximum design wave crest height.

5.6 Power Module - Power Plant

The turbine power module (enclosing the power train, generator, controls, transformers and switchgear) will have to be airtight and include a dehydration system. This is more critical for Darrieus VAWTs because the power module for a Darrieus VAWT will still be more severely exposed to the sea water than will be the case for an equivalent HAWT.

The generator and gearbox should in addition be totally enclosed and use a heat-exchanger system for cooling air.

5.7 Corrosion Protection

All exposed steel surfaces will have to be protected to the highest corrosion class. Special attention must be given to structural joints and the shaft seals.

5.8 Installation

On land, the foundations for VAWTS are most cost effectively constructed in situ. The turbine is usually subassembled at the factory, with final assembly at the installation site. For some VAWT designs, like the Indal 50 kW and 6400 turbines, the base structure and the power plant are fully integrated and assembled at the factory. Other VAWT designs have used lattice base structures (stub towers) assembled on site with the power train and generator subsystem factory assembled and then installed on the foundation. In either case, the rotor is assembled on site and lifted into place, resting on a bearing installed at the top of the base support structure.

The preceding approach would be prohibitively time consuming and expensive for offshore turbines. (A significant part of the cost is the fact that favorable weather and sea conditions for construction at good wind sites are quite limited.) The on land methods of prefabricating the supporting structure and the turbine, transporting to site and emplacement are quite different for offshore wind turbines. The most cost effective solution for a specific project will be greatly influenced by the size of the turbine, the size of the wind farm project (ie. the number of turbines and the area of the wind farm), the "windows" of favorable construction weather, the water depth and the distance from shore.

It seems likely that any systems of preassembly, towing and installation that are feasible for offshore HAWTs and particularly for straight bladed VAWTs will probably be equally feasible for offshore Darrieus VAWTs. For example, Kerr (1986) concluded that it should be possible to float out and install a gravity foundation structure using a detachable and reusable steel buoyancy aid with the complete wind turbine (including rotor) already installed.

5.9 Operations and Maintenance

Offshore turbines require quite different maintenance strategies than are required for on shore turbines because:

1. Personnel, parts and equipment must be transported in service vessels. These costs could be major components of operations and maintenance costs. This is not generally true for land based wind farms.

2. Less time is available for service and maintenance operations compared to land based turbines due to adverse sea and weather conditions.

3. Large cranes cannot be as readily moved into place to support major parts replacements.

Both offshore HAWTs and VAWTs will be equally affected by these conditions. There will be some maintenance cost advantage for those Darrieus designs where the power module is located at the level of the supporting structure platform, just as there is for these turbines when installed on land.

6.0 ENVIRONMENTAL IMPACT

This section considers the impacts on the human environment (mainly noise, land use, TV interference and visual impact) and the impacts on the natural environment. Turbine wake effects are included here because they are an environmental effect of turbine operation, even though turbine wakes may only affect other turbines.

6.1 Acoustic Emissions (Noise)

6.1.1 Introduction

Background

Although measured wind turbine acoustic emissions can be, and often are, extensively quantified for engineering design purposes, it is ultimately the potential for community annoyance, as defined by sound pressure levels in the audible frequencies, that is of the most importance in commercial deployment.

Noise generation by large wind turbines came to the forefront as an environmental concern during the testing of the 2 MW MOD-1 horizontal axis wind turbine near Boone, North Carolina. Based on this experience, the noise problem from wind turbines was determined to consist of the noise generation itself, atmospheric propagation of the noise, the responses of structures to acoustic loadings and the human responses to the noise, both outdoors and indoors.

Noise Sources

Wind turbine noise can be broadly characterized as having impulsive components ("thumping") due to blade/tower interactions (and possibly blade/blade interactions for VAWTs) and nonimpulsive components ("swishing") due to unsteady flow over the blades. The characteristics of the resulting noise components are modified before reaching an observer due to atmospheric propagation and terrain effects.

The background (ambient) noise level is a very important factor in determining annoyance from a wind turbine installation. For example, Dube (1990) reported that the 4 MW Eole VAWT did not appear to increase the background noise levels produced by the wind, in the absence of other extraneous noise sources, beyond a 300 m radius from that turbine.

Wind turbine noise sources are aerodynamic and mechanical.

Mechanically-related noise can be treated like machine noise (in general) and may or may not be important for a given wind turbine design. Krishnappa and Hammell (1982) have reported on the noise measurements carried out on the Indal 50 kW VAWT gearbox. For this particular turbine design (single stage spur gear transmission), the gearbox noise is important. The results of their study are discussed later.

Aerodynamic noise has been found to be often the greater problem. This noise is generated by fluctuating aerodynamic lift (and hence pressure) on the turbine blades resulting from steady and unsteady air loads. The unsteady loading is responsible for most of the objectionable aeroacoustic noise. Hemphill, Kelley and McKenna (1982) state that:

"The Darrieus-type VAWT has the potential to suffer from unsteady loads from discrete vortex shedding from the central supporting tower or torsion column, imbedded turbulence in the flow, from vortices shed from the upwind blade which translate downwind to be intercepted by a downwind blade, or from any combination. Also there is a slowly-varying unsteady load imposed on the blades inherent in the Darrieus design."

They go on to state that:

"Random small-scale turbulence will produce unsteady loading noise from blades which is largely incoherent in the acoustic far field. A vortex element intercepted by the turbine blades may, however, produce quite coherent noise in the far field which has special implications for structures that may respond to the concentrated acoustic energy and magnify any annoyance."

Wind Turbine Acoustic Emissions Design Requirements

Few wind turbine noise design requirements and standards have apparently been published. Those that exist were released very recently. It is therefore unlikely that turbine noise was seriously considered in the design of the Darrieus wind turbines for which noise measurement data have been published.

The following requirements have been published:

1. ECN Regulations for the Type-Certification of Wind Turbines: Technical Criteria. February 1991.

The emission-relevant source strength $L_{WR}(A)$ of the test specimen must be smaller than the value calculated for the rotor dia in the formula below. This formula gives the maximum permissible source-strength (in dB(A)) as a function of the rotor dia D (in m):

$$L_{WR}(A) = 22\log D + 69$$

2. Pacific Gas and Electric draft procurement specifications. 1991.

The overall sound pressure level produced by the wind power plant must not exceed 45 dB(A) at a distance of 910 m from the nearest wind turbine at any position around the plant. This must be achieved for a wind speed of 8 m/s at 10 m above ground level.

6.1.2 Noise Measurements

Acoustic (or noise) surveys have been conducted and reported for the NRC/Hydro Quebec 24m, the Indal 6400, the DOE 100 kW (17m) and the Eole 64m turbines. FloWind Corporation is also believed to have carried out noise measurements but the data have not been made public.

Indal 6400/500kW VAWT - Southern California Edison

Ambient noise surveys were carried out in 1984 and 1985 (Wehrey, Yinger and Handschin 1987, and Wehrey, Heath, Yinger and Handschin 1985) at SCE's Wind Energy Test Center near Palm Spring's California. The 1985 data are of the most interest since the turbine operated continuously for the 24 hour test period at average power outputs of between 210 and 370 kW (or 0.35 to 0.62 kW/m^2). The average wind speeds during the 24 hour period were between 13 and 18 m/s.

The test site layout and the locations of the upwind and downwind monitoring stations are shown in Figure 6.1.2a. The wind direction during the test period was from the west. A power transmission line running north and south greatly influenced the measurements at station 600W. Figure 6.1.2b shows the locations of all of the monitoring positions.

The Indal 50 kW turbine operated during the survey except for a period of 6.5 hours. The Wenco 100 kW turbine operated for only three hours during the survey.

The general conclusions of the survey, based on analyses of the data, were:

1. The Indal 6400/500kW turbine influenced the noise environment on the surrounding desert up to about 245 m in all directions on the day of the survey. The rate of decay of noise level with distance from the wind turbine followed approximately the 6 dB per distance doubling expected, beginning at 60 m from the turbine (2.5 rotor diameters).

2. The noise environment at the fenceline of the ESI wind farm was dominated by the operation of the turbines installed there. See Figure 6.1.2a.

3. Noise levels downwind from the Indal 6400/500kW turbine were found to be higher than upwind. At distances of 61 m and greater from the wind turbine, downwind levels were consistently 4 to 5 dB higher than those upwind. See Figure 6.1.2c. The exception is station 600W and this was determined to be due to the wind noise produced by high power lines directly overhead.

4. No significant tonal energy was evident in the infrasonic (0 to 50 Hz) spectra regardless of distance from the VAWT. However, examination of the 0 to 1000 Hz spectra shows a tone at 305 Hz and its harmonics at 610 and 915 Hz (see Figure 6.1.2d). These tones were strong at locations 200W (upwind) and 200E to 1000E (downwind) and are produced by the gear meshing (spur gear twin pinion system). 305 Hz is within one percent of the gear meshing frequency.

NRC/Hydro Quebec 24m Magdalen Islands Turbine and DOE 100 kW Turbine

The results of noise measurements on these two turbines have been reported by Hemphill, Kelley and McKenna (1982) and Kelley, Hemphill and Sengupta (1981). The measurements were made in 1980 and 1981.

Measurements of the NRC/Hydro Quebec turbine were made downwind at 1.5, 3, 6 and 9 rotor diameters and at 45° and 90° off the wind axis. Measurements of the DOE 100 kW turbine were made downwind at

0.5, 1.5, 3, 6, and 9 rotor dias and at 45° and 90° off the wind axis. The turbulence intensities at the NRC/Hydro Quebec site were typically 2 to 10% while at the DOE 100 kW site they were typically 8 to 13%. Unfortunately a summary of the sound pressure levels measured at the various distances were not presented so no direct comparison with the results of the Indal 6400 turbine tests are possible.

Their main findings and conclusions were:

1. The directivity pattern of A-weighted sound pressure levels was found to be of a cardioid shape centered on the wind axis, indicative of dipole radiation sources. The spreading rate of the sound was found to be less than would be expected for normal geometric spreading within three rotor diameters particularly along the axis perpendicular to the wind vector. For the DOE 100 kW turbine the A-weighted sound pressure dropped by about 3 dB downwind of the turbine between 1.5 rotor diameters and 3.0 rotor diameters (25.5 to 51 m). This indicates that at this distance the rotor is not a theoretical single source and the decay rate has not reached the higher value of 6 dB per distance doubling that is normally measured in the far field (Shepherd and Hubbard 1985).

2. The acoustic spectrum was dominated by the sound pressure corresponding to the blade passage frequency, with levels much higher than are usually found on similarly rated HAWTs.

3. The amplitude of the blade passage frequency sound pressure level was found to increase monotonically with increasing wind speed.

4. Unsteady loading noise was believed to be caused by the unsteady wake from the torsion column.

5. The DOE 100 kW (17m) turbine was found to exhibit the potential for impulsive noise radiation while the NRC/Hydro Quebec 24m turbine did not indicate a similar tendency within the limited data set available. The low turbulence regime and perhaps the machine design itself are possible reasons for this. Local atmospheric conditions, specifically the structure of the inflow turbulence was shown to be related to the increase in coherency in the DOE 100 kW VAWT acoustic emissions.

6. A joint probability analysis technique indicated that the DOE 100 kW turbine had the potential for occasionally reaching impulsive levels similar to those associated with MOD-1. The frequency of occurrence was expected to be highly site dependent.

7. Interpretation of the data places the Darrieus wind turbines between the up and downwind HAWTs in the potential for severity of acoustic noise generation which may lead to community perception and annoyance.

8. Large VAWTs placed in sensitive environments have the potential to cause unacceptable low frequency acoustic radiation.

Projet Eole 64m Turbine

Measurements of the noise emitted by the Eole wind turbine were carried out in October 1989 (Dube 1990). The background noise levels measured before and after the turbine installation at different measurement sites were compared. Noise levels were also measured at equal distances upwind, downwind and perpendicular to the wind direction. Figure 6.1.2e shows the far measurement sites.

The main results were that, at the turbine site, the Eole turbine operation increased the background noise levels by about 8 dBA but did not increase the background noise levels beyond a 300 m radius from the turbine. Between 200 and 300 m from the turbine, the increase in the noise levels was not found to be significant. In this zone, the noise produced by other sources, such as rural traffic or human activities, was found to offset the noise level increase due to the turbine. The wind speed was about 12 m/s and the turbine was producing about 1.4 MW (or about 0.35 kW/m^2) at a rotor speed of 13.35 rpm.

Only two measurements could be made downwind of the turbine, at 87 m (about 1.36 rotor diameters) and 174 m (about 2.72 rotor diameters). A drop in the A-weighted sound pressure level of about 3 dBA was recorded. This is the same result reported for the 17m turbine at about the same distances from the turbine in terms of rotor dias.

Figure 6.1.2f shows the change in noise level at 87m from the turbine at various azimuth angles when the wind turbine was started up and operated. The wind was from the west (260°) at 12 m/s, about 5 m/s above the cut in wind speed. This figure shows that at this distance background noise levels increased by about 8 dBA with the turbine operating at 13.35 rpm.

Indal Technologies 50 kW Turbine - Gearbox Noise

Noise and vibration measurements were carried out on the gear box and power house panels of an Indal Technologies 50 kW vertical axis wind turbine (Krishnappa and Hammell 1982). This turbine had a vertical shaft single mesh spur gear and pinion gear box with a gear ratio of 15:1. The gears were found to be generating excessive noise. The vibrations produced by the gears were transmitted to the panels of the power house (or power module) through the gear casing. The side panels became the sources of noise generation. See Figure 6.1.2g.

Due to the absence of wind, the noise and vibration tests were carried out by energizing the generator/motor. This was believed to be the equivalent of low power output operation.

It was found that the sound power radiated from the power house panels was much higher than the levels radiated from the gear box casing and that the noise produced by the power house panels was the result of transmitted vibrations from the gear box casing. The noise and vibration signatures of the spur gears showed the presence of strong tones at the gear mesh frequency and its harmonics.

Several recommendations were made to reduce the noise, including inserting damping material between the gear box casing and the top of the power house module, modifying the structure of the walls, lowering the pressure angles of the gears and using helical gears instead of spur gears.

6.1.3 Conclusions

1. Published measured noise data for Darrieus wind turbines is quite limited. Test procedures and objectives differed considerably. There is no evidence that the published data are based

on measurements carried out in accordance with any standard published test procedure for wind turbines. Hence comparisons are very difficult.

2. There is some indication that, under certain inflow turbulence conditions, specific Darrieus turbine designs could exhibit very low frequency impulsive noise radiation similar to the MOD-1 experience.
3. The acoustic spectrum is dominated by the sound pressure corresponding to the blade passage frequency. The amplitude of the blade passage frequency sound pressure level was found to increase monotonically with increasing wind speed.

4. The rate of noise decay in the far field is about 6 dB per distance doubling as reported for HAWTs.

5. The suggestion that the Darrieus turbine is between up and downwind HAWTs in the potential for severity of noise generation appears reasonable. However there appear to be no data directly comparing noise measurements for VAWTs and HAWTs.

6. There is no evidence in the literature that the design of any Darrieus turbine has included noise design criteria. The Indal 50 kW gearbox and power module noise problem is indicative of this.

6.2 Electromagnetic Interference

Published electromagnetic interference data for Darrieus wind turbines appear to be confined to television interference. All of the turbines tested had aluminum blades.

6.2.1 Indal 6400/500kW Turbine - Southern California Edison

Southern California Edison carried out TV interference measurements in January, 1986 (Wehrey, Yinger and Handschin, 1987). The equipment used for the tests is shown schematically in Figure 6.2.1a.

The video portion of the UHF broadcast station at 639.25 MHz was used as the reference. The station broadcasts from Edom Hill, which is in the area of the WETC as shown in Figure 6.2.1b. Measurements were made at five locations ranging from about 69 to 183 m from the wind turbine.

The main findings were:

1. Few TV reception problems were observed as long as a good directional antenna was used and it was directed to the TV broadcast station.

2. When the antenna was directed towards the rotating metal blades, some pulsed brightness of the picture occurred and a slight loss of synchronization.

6.2.2 Sandia 17m Research Turbine

Kelley, Hemphill and Sengupta (1981) and Sengupta, Senior and Ferris (1983) have reported on television interference studies carried out on the Sandia 17m research turbine. The RF sources were commercial signals from six channels (VHF and UHF) whose transmitters were all located on Sandia Crest, about 21 km from the VAWT. Figure 6.2.2a is a schematic of the measurement system and the layout. At each of the eight measurement sites, ambient field strength and television interference were measured. Sites 2 and 5 are of particular interest because site 5 is in the forward scattering region (approximately 180 degrees to the reference direction) and site 2 is in the backward scattering region (approximately 0 degrees).

The general findings were as follows, when the antenna beam was directed at the transmitters (ie. properly oriented):

1. In the forward direction (180°) threshold level video distortion was observed on UHF Channels 23 and 48 at a distance of 33 m from the turbine. However, the baseline ambient signal on channel 23 was so weak that the quality of reception was unacceptable even when the turbine was parked.

2. In the backward directions (0°) there was no video distortion on any channel at distance of about 37 m.

3. At the sites in the 90° direction 21 and 23 m from the wind turbine, unacceptable video distortion occurred on Channel 23.

4. At two sites 23 m from the wind turbine and in the 45° and 135° directions, there was interference above acceptable levels on Channels 13 (VHF) and 23 (UHF) respectively.

Other findings were:

1. With the antenna beam directed at the wind turbine (ie. improperly directed for backward region sites), interference ranging from slight to violent was observed on some or all of the channels at all eight sites.

2. The modulated waveforms appeared to be site and frequency (channel number) dependent and more complicated than those produced by a HAWT. They varied from almost sinusoidal to pulsed in nature, with a dominant component repeating at twice the rotation frequency of the blade.

3. Based on a simple theoretical model and the measured data, it is found that the equivalent scattering area of the Sandia 17m Darrieus turbine on low VHF channels is of the same order as that for a HAWT having similar power rating. The Darrieus turbine appears to produce less interference than a HAWT on all other channels.

6.2.3 Conclusions

1. The findings from both studies are consistent and can be summarized as showing that the Darrieus VAWT does not generally interfere with TV reception within a radius of 200 m except when the antenna is directed at the wind turbine and the antenna is in the backward scattering region (between the transmitter and the turbine). The antenna is clearly improperly directed in this case.

2. In both the Indal 6400 and Sandia 17m tests, monitoring was carried out relatively close to the wind turbine, within a zone in which it is unlikely that any dwellings would be located (due to noise). The results can therefore be considered as characterizing almost the worst case conditions for single wind turbine installations. There appear to be no reports in the literature on measurements of television interference due to an array of vertical axis wind turbines.

3. The rapid growth of cable TV installations and satellite dishes, at least in Canada and the U.S.A., may obviate further research in this area.

6.3 Visual Impact

6.3.1 FloWind Blade Painting

FloWind Corporation is known to have painted the blades of its 17m and 19m turbines (installed in California) in response to either requests or orders from the authorities that granted the installation permits. The blades were painted a dull grey or light brown color in order to eliminate "blade flashing" resulting from light reflection and to better blend the turbines into the background colors of the surrounding terrain.

6.3.2 Public Reaction Survey - Southern California Edison

Southern California Edison Company (Wehrey, Yinger and Handschin 1987) conducted a survey to assess public reaction to the Indal 6400 turbine (termed the DAF-500 WT) installed at the SCE wind turbine test site near Palm Springs, California. A questionnaire was placed in a holder on a sign at the wind energy test site. 117 responses were received and tabulated over a 32 week period starting in July, 1985. This survey appears to be the only one of its type ever undertaken for vertical axis wind turbines.

Respondents were asked if they preferred the DAF-500 WT (Indal 6400) more or less than the DAF-50, Howden or WENCO designs. The latter two were horizontal axis wind turbines. All four turbines were installed at the same site. Two thirds of the respondents observed the turbine in operation.

Between 62% and 75% of the respondents found the Indal 6400 turbine more acceptable than the other three turbine designs and close to three-fourths felt that fewer large Indal 6400 turbines would be preferred over many, smaller machines. The majority of respondents felt that the 6400 turbine was acceptable for appearance, noise and its impact on animals and plants, but did not know how it would affect television reception. The vast majority of miscellaneous comments were positive.

6.3.3 Newfoundland Light and Power Report

Newfoundland Light and Power (1991) reported on a survey conducted in the United States about 1977 to assess public response to wind turbine designs via slides. The results indicated a strong preference for "Dutch-style" wind turbines as compared to other HAWTs, Darrieus and giromill turbines. Since this study was carried out before any wind farm developments had occurred in the United States, its validity is questionable.

Newfoundland Light and Power also reported that lattice towers are the most visually accepted at a distance but that conical towers (presumably tubular towers) are thought to be more visually acceptable. Most vertical axis turbines constructed to date would be classified as having "guyed conical towers".

6.3.4 Conclusions

The Southern California Edison public reaction survey appears to be the only documented study pertaining to the observation of actual horizontal axis and vertical axis turbines. Although the results of the survey favor the Darrieus vertical axis wind turbine design over HAWT designs in terms of visual impact, the results may be of limited value since public reaction is now probably most influenced by the impact of wind farms, rather than individual turbines.

6.4 Land Use Impact

All of the existing commercial wind farm installations and most individual installations of vertical axis wind turbines consist of guy cable supported turbines.
Observations have been made by Schienbein (unpublished) on the California wind farms and other installations concerning land disturbance for foundations, roadways, power transmission lines and transformers and domestic animal behavior. On one windfarm in the Altamont Pass, HAWT and VAWT clusters are intermingled. The following observations have been made and conclusions reached.

1. Guy cable support reduces the size of the foundation required for the VAWT stub tower or base structure. Therefore less excavation is required for a cable supported VAWT when compared to a cantilever tower supported HAWT of equivalent size. However, about the same amount of land is cleared for a "pad" around the turbine in both cases to accommodate ongoing maintenance access.

2. Cantilever supported VAWTs should exhibit about the same foundation requirements as cantilever supported HAWTs. In both cases, the dimensions of the foundation are in large part determined by the type of tower design chosen.

3. Only a very small area is disturbed for each guy anchor installation (usually a poured concrete "deadman" type) and anchors are inspected by personnel on foot, not in vehicles. Therefore the land area near the anchors can be restored and remain relatively undisturbed. VAWT guy anchors have not been replaced in the FloWind wind farms, except in a very few cases (probably less than six altogether out of about 1500) where they have been damaged due to a turbine rotor mishap. In some cases guy anchors have been fenced, presumably to protect the guy anchors (and lower portions of the guy cable assemblies) and the farm equipment operator from injury. Some measures, short of fencing, would probably have to be taken in a wind farm of guy cable supported turbines to prevent collisions and injury when agricultural equipment is operated around the turbine guy anchors.

3\. Road access requirements are virtually identical for both VAWTs and HAWTs. The width and path of the roads is generally determined by construction requirements, that is, the requirements imposed by having to move mobile cranes to the individual turbine installation sites.

4\. Runoff water drainage patterns are affected by the network of roads in all wind farms. There is no reason to suppose that the effects will be better or worse for VAWTs versus HAWTs. The effects are dependent on the location and size of the roads and pads and the measures taken to mitigate them in the design of the wind farm.

5. Domestic animals, such as cattle, have no difficulty accepting VAWTS and HAWTs within their grazing territories. Cows are often observed resting and grazing under operating wind turbines of both types. Cattle and other domestic animals do not appear to damage the guy anchor assemblies.

6.5 Impacts on the Natural Environment

6.5.1 Animal Habitat

Animal habitat in a wind farm is disturbed mainly by the installation requirements of the wind turbines (including the foundations and leveled pad areas), other wind farm structures, transmission lines, transformers and substations, roads, emissions (such as oil leakage), construction debris and cleared areas (possibly compacted), fences and human activity, mainly measured by vehicle movements. The impact of HAWTs and VAWTs in these areas are very similar.

Disturbance of habitat due to the turbine pads, turbine foundations, transmission lines, transformers and substations, wind farm structures (other than the turbines), fences and access roads should be about the same for a wind farm constructed using HAWTs or Darrieus VAWTs. (See also Section 6.4.) Vehicle movements after construction depend mainly on scheduled and unscheduled maintenance events, which will be about the same for mature HAWT or VAWT designs. Vehicle movements are greatly reduced in wind farms where reliable remote monitoring systems are used and where reliable turbines are installed.

Soil contamination due to fluid leakage (such as bearing and gearbox lubricants) or careless transport and transfer of liquids is as likely for HAWTs as for Darrieus VAWTs. HAWTs may pose more of a threat due to the fact that the power transmission and brake systems are mounted at the top of the tower. For example, the rupture of a HAWT brake fluid line could result in wider dispersal of the fluid than would occur for a Darrieus turbine,
where the brake system is located near ground level.

Additional construction related habitat damage depends on construction practices and not on the type of wind turbine installed. Considerable damage can result from careless movement and storage of construction materials and turbine parts.

It has been suggested that noise and ground vibration due to wind turbine operation does adversely affect animal habitat. There appear to be no published studies comparing HAWTs and Darrieus VAWTs on this subject.

Turbine structures and power transmission lines do affect birds of prey and migratory patterns. This is discussed in section 6.5.2. Land disturbance and human activity reduce the habitat and availability of prey.

6.5.2 Bird Mortality

California Energy Commission Study

Orloff (1991) has summarized the results of the first year of a study to evaluate the extent and significance of the impact of wind turbines on bird life and to recommend mitigation measures. This two year study is focused on the Altamont Pass and Solano wind resource areas and dealt mostly with HAWTs.

Of 183 dead birds found within the sample sites, 65% were raptors (birds of prey). It was found that 84% of the deaths could be attributed to collisions with turbines. (Electrocution and collisions with wires accounted for most of the other deaths.) Most of the collision deaths were the result of amputation injuries.

The analysis of the death rate and its causes at each of five different turbine types was inconclusive. However, about 61% of all raptors were observed to be flying within 52 m of the ground. This is below the maximum blade height of most turbines installed in California and is also well below the rotor top height for very large Darrieus turbines such as the Indal 6400 (rotor top height of about 40 m).

No birds were seen flying into the wind turbines, so that it could not be determined how the birds put themselves at risk. However, most of the dead birds were found downwind of the turbines and this suggests that they were flying with the wind when they hit the blades. Many of the birds were observed flying close to the blades and seemed unaware of the danger. Familiarity may be an important factor contributing to the death rate.

Bird Mortality Study - Southern California Edison

A post construction monitoring program was initiated by Southern California Edison Company (Wehrey, Yinger and Handschin, 1987) at its Wind Energy Test Center (WETC) to determine the magnitude of actual wind turbine collisions during peak bird migration periods. Large numbers of migratory birds move through the San Gorgonio Wind Resource area during the spring and fall. Most of the bird movements occur at night. Up to 10 percent of the total population was thought to be at risk based upon actual radar measurements of altitude distribution.

SCE's study was carried out over 67 days (September 16 through October 12, 1984 and April 8 through may 17, 1985). No bird fatalities were found during that period.

The value of the study results are hampered by the fact that none of the three turbines were operated during the spring 1985 portion of the study period. No explanation was provided in the report for this site shutdown.

SCE concluded that some bird collisions with the WETC and the Indal 6400 turbine may have occurred but that overall bird collisions were minimal. The fact that 38 percent of the nocturnal bird migration occurred at or below the height of the Indal 6400 and that one dead bird was found before the study suggests that bird deaths were likely. These fatalities may not have been recorded because of the high rate of carcass removal by scavengers, observer bias and inactivity of all three of the turbines during the spring study period.

The authors concluded that, considering these facts, the results of this study should not be used to estimate the amount of bird mortality from an array of wind turbines.

6.5.3 Soil and Vegetation

It can be expected that the impact of wind farm development and operation on soil and vegetation will be virtually identical for wind farms of the same number of machines and machine size, be they vertical axis or horizontal axis turbines.

Soil and vegetation impacts depend mainly on the way in which the turbines are distributed, the access requirements for construction and maintenance of the wind turbines and the power collection system, and the construction practices. (See also section 6.4.)

6.5.4 Conclusions

1. There is no evidence to suggest that Darrieus vertical axis wind turbines affect the natural environment more adversely than do HAWTs.

2. The impact of the cables of guy supported Darrieus rotors on the natural environment is generally insignificant. However, the cables probably do add to the dangers facing birds within a wind farm.

6.6 Darrieus Wind Turbine Wakes

6.6.1 Introduction

Understanding the characteristics of the wake downwind of a wind turbine is important in order to determine the turbine array that will optimize energy production and the rotor fatigue damage accumulation resulting from the interaction with the wake of upwind turbines. The characteristics of the wake may require increased structural design requirements for downwind turbines.

The wake of an upwind turbine decreases the energy production of a downwind turbine because of the momentum deficit. Furthermore the performance of downwind turbines may be reduced by the gradients in the mean flow, altered turbulence structure and discrete vorticity introduced by the upwind turbines.

The energy deficit experienced by a downwind wind turbine in an array depends not only on its distance downstream of the upwind turbine but also on the incident turbulence, the tip speed ratio (mid-rotor blade speed divided by ambient wind speed) of the upwind turbine, the effects of the wakes of other upwind turbines, the effects of adjacent turbines and the annual wind speed distribution. (Energy deficit is defined as the annual energy lost by a turbine operating within an array when compared to the energy captured by the same turbine operating outside of the array.)
In order to design turbines to be part of arrays, the wake of the individual turbine must first be understood and this has been the thrust of a number of wind tunnel and full-scale field test programs.

6.6.2 Full Scale Field Test Results for Individual Turbines

Full scale field measurements of individual turbines have been carried out for the Indal Series 6400 turbine (Wehrey, Heath, Yinger, and Handschin 1985, and Wehrey, Yinger and Handschin 1987), the Indal 50 kW turbine (Gyatt and Zambrano 1982), the FloWind 17m turbine (Baker, Walker and Katen 1984, and Liu, Buck, Germain, Hinchee, Solt, LeRoy and Srnsky 1987), the Sandia 17m research turbine

(Akins 1983), a 5.3 m turbine (Vermeulen, Builtjes, Dekker and Bueren 1979) and a 4 m turbine (Brons, Jensen, Jensen and Petersen 1979).

Wake Velocities for Single Turbines

Akins (1983) reported that the center line wake velocity ratio (wake velocity divided by free stream velocity) for the Sandia 17m research turbine decreased with increasing tip speed ratio (see Figure 6.6.2a). This is consistent with the fact that the rotor drag coefficient increases with tip speed ratio as shown in Figure 6.6.2b (Templin 1974). Liu, Buck et al (1987) suggested that based on their measurements, the maximum velocity deficit, V_d, tends to decrease with increasing wind speed, S, according to:

$$V_d = S^{-n}$$

where n equals one for low speeds and decreases with increasing speed. (Velocity deficit is the difference between the free stream wind velocity and the wake velocity.)

Akins' velocity deficit results are in very good agreement with those from the measurements for the FloWind 17m turbine (Liu, Buck et al 1987) shown in Figure 6.6.2c. For example, Figure 6.6.2c suggests that at tip speed ratio of about 4, the maximum velocity deficit was about 0.20 at 5.5 diameters downwind. Akins' data (Figure 6.6.2a) suggest that for the same location and tip speed ratio the velocity deficit was also about 0.2. Some caution must be exercised in interpreting the results of these two test programs since wake decay is strongly dependent on incident turbulence. This is shown very clearly in Figure 6.6.2d for the FloWind 17m turbine tests reported by Baker, Walker and Katen (1984). The downstream extent and width of the wake are greatly reduced as the incident turbulence increases.

The test results for the Indal 6400/500kW turbine reported by Wehrey, Yinger and Handschin (1987) do not agree with the results of the preceding studies. Furthermore no clear trend as reported by the other two studies was evident. Indeed, the measured velocity deficit tended to decrease with increasing tip speed ratio at distances of 5 diameters and greater while for distances of 1.5 and 3 diameters the velocity deficit increased with tip speed ratio as reported by the other two studies. No explanation has been offered for the lack of agreement.

The vertical spreading of the wake at a tip speed ratio of 4.25 is suggested by Figure 6.6.2e (Akins 1983) which shows the velocity ratios measured at the rotor centerline height and at two other positions above and below the centerline height.

Akins claimed that his data demonstrated that a fully developed wake existed as close as 1.5 rotor diameters downstream of the turbine. The velocity deficit in the far wake region appeared to decay with distance downwind as a function of the -1 power of the downwind distance. This conclusion is supported by the results for the FloWind 17m turbine as shown in Figure 6.6.2f (Baker, Walker and Katen 1984).

Baker, Walker and Katen (1984) compared the velocity deficit of the FloWind 17m turbine to the MOD-2 for similar incident turbulence and atmospheric stability conditions as shown in Figure 6.6.2f. When downwind distance is normalized by the rotor diameter, the centerline velocity deficit as a function of downwind distance is seen to be virtually identical for the two turbines, with the deficit falling to 10% about 10 diameters downstream in both cases.

Wake Turbulence

Akins (1983) reported that the wake turbulence intensity (standard deviation of wind velocity divided by the mean velocity) at 1.5 diameters downwind was approximately twice the free stream value. At the 5.5 diameters position the turbulence had returned to the free stream value. This suggests that the standard deviation of wind speed had actually decreased to less than the freestream value at 5.5 diameters, which seems to be an improbable result. However, Akins may have been referring here to the actual standard deviation and not to the turbulence intensity. In addition, Akins did not report the value of the tip speed ratio for these measurements.

Test data reported by Schienbein (1980) suggest a ratio of turbulence intensity in the wake to that in the free stream of about 1.5 to 2 at 1.6 rotor diameters downstream for the Indal 50 kW turbine. The tip speed ratio was not given.

Gyatt and Zambrano (1982) reported that for the Indal 50 kW VAWT, in a different test program at a tip speed ratio of about 4.5, the wake turbulence intensity level at two diameters downstream was about 1.87 times that of the free stream level. However, the velocity ratio was 0.59 so this indicates that the standard deviation was only about 1.1 times that in the freestream. At five diameters, Gyatt and Zambrano reported a wake turbulence intensity of 1.36 times that of the free stream level with a velocity ratio of 0.80. This implies that the standard deviation of wind speed in the wake was 1.09 times that of the free stream.

Baker, Walker and Katen's results (measurements are for downwind distances greater than 3 diameters) show a wake turbulence intensity of about 1.5 to 1.6 times that of the free stream at 5.5 diameters for three different ranges of freestream turbulence. See Figure 6.6.2g. This appears to closely agree with the results reported by Gyatt and Zambrano for the Indal 50 kW turbine. It appears that the corresponding turbine thrust coefficient was about 0.7. At this thrust coefficient, the centerline velocity deficit at 5.5 diameters downstream would have been about 0.3, and therefore the standard deviation of wind speed on the wake centerline appears to have been about 1.05 to 1.10 times that in the free stream, which generally agrees with Akins' result.

Baker et al (1984) concluded that there was probably only a small increase in the vertical and cross-wind components of turbulent kinetic energy as a direct result of the motion of the turbine blades. They further concluded that the large increase in the turbulent kinetic energy observed at times in the along-wind component was probably due more to movement of the wake back and forth across the anemometer causing changes in wind speed than an actual increase in the level of turbulence.

6.6.3 Full Scale Field Test Results for An Array of Turbines

Full scale field test results for an array of FloWind 17m Darrieus turbines have been reported by Liu et al (1987). Nine test turbines were selected within an already existing array. The turbines in the array were spaced at three rotor diameters crosswind and eight rotor diameters row to row. The nine test turbines (three in each of three rows) had exhibited similar performance based on actual production data for the 10 month period preceding the test program. The turbine layout is shown in Figure 6.6.3a. Note that turbines T122, T123 and T124 are the upwind turbines. The fixed anemometer towers (both upwind and downwind) were installed along the line connecting turbines T102 and T123 as shown in Figure 6.6.3a.

The main findings of the study were:

1. The turbine power measurements turned out to be the most straightforward means of gauging the effects of wakes of upwind turbines. Velocity deficit data were difficult to use because the data were derived from fixed sensors that could not provide sufficient areal velocity deficit information.

2. The power loss or deficit depended on both the wind speed and direction, as expected.

3. For the spacing of this array, the primary wake-producing turbine had the dominant effect on the performance of the downwind turbine. The turbines immediately adjacent to the primary turbine (T123) had only secondary effects, whereas the turbines farther away have insignificant effects. This can also be seen in Figure 6.6.3b, for example, where the wake deficits at five rotor dias downwind of T123 are shown for the case where only T123 is on and for the case where T122, T123 and T124 are on. There is little difference.

4. The power deficit reached a maximum when the predominant wind was in the direction that aligned the wake of the upwind primary turbine with the downwind turbine under consideration.

5. The power loss maximized at wind speeds around 12 m/s, whereas the power deficit tended to decrease with increasing wind speed (Figure 6.6.3c).

6. The annual energy deficit for turbine T102 (based on the 1986 actual wind-speed distribution data and the measured power deficit) was estimated to be 10%, when an average availability of 95% was assumed for the wind turbines in the array. (Turbine 102 is immediately downwind of turbine T123 as shown in Figure 7.6.3a).

6.6.4 Wind Tunnel Tests

Darrieus rotor wind tunnel flow field measurements have been reported by Giles (1977), Boschloo (1977), Vermeulen (1979), Michos, Bergeles and Athanassiadis (1984) and Base, Phillips, Robertson and Nowak (1981). In addition, wind tunnel array studies have been carried out using screens to simulate the turbine wakes (Base et al 1981).

Wind tunnel test results appear to be of limited value in providing design data due to the generally low incident turbulence values (compared to the natural wind), the constraints imposed by the tunnel walls and the low test Reynolds numbers and resulting inferior energy conversion performance of the model turbines. (Akins has pointed out, for example, that the downwind extent of the wake represents an upper limit when the incident turbulence approaches zero.)

For example, Michos, Bergeles and Athanassiadis carried out a wind tunnel test of a Darrieus VAWT having a rotor diameter of 1.64 m (H/D of 0.982) at blade Reynolds numbers of only about 150,000. The maximum measured power coefficient reached 0.30 compared to about 0.40 to 0.50 that can be attained by full scale Darrieus turbines operating at blade Reynolds numbers of one to four million. Free stream turbulence intensities were only 0.5 to 1%, about 10 to 20 times less than that which would be experienced by a wind turbine in the natural wind. In addition, the wind tunnel test section was 2.5 m high and 3.49 m wide. This results in significant blockage of the wake development.

Nevertheless the results reported by Michos, Bergeles and Athanassiadis confirm many of the general trends found through full-scale field tests. Their results showed that:

1. The flow in the wake is strongly retarded compared to the undisturbed flow upstream of the rotor.

2. The velocity defect increases with increasing velocity (or tip speed) ratio.

3. Minimum velocities in the wake occur at a distance of about one rotor diameter. Further downstream the rate of flow recovery strongly depends on the velocity ratio.

4. Turbulence intensities increase sharply in the wake.

Wind tunnel tests are particularly useful in permitting detailed mapping of the wake velocity profile and the turbulence variation throughout the wake. This is extremely difficult and costly to do in the field on full-scale turbines. For example, Michos, Bergeles and Athanassiadis found that the velocity profile became more symmetric as the velocity ratio increased because the flow around the blades is attached and the blade loading is more uniform. (See Figure 6.6.4a.) This supports, for example, the general trend of the shape of the profiles predicted by Strickland and Goldman (1980). Michos, Bergeles and Athanassiadis also found that the turbulence intensity decreased with increasing velocity ratio and attributed this effect also to the fact that the flow around the blades becomes primarily attached as the tip speed ratio increases (ie. the blades experience less stalled flow during one rotor revolution as the tip speed ratio increases).

6.6.5 Theoretical Models

Strickland and Goldman (1980) have presented preliminary results with regard to utilization of a vortex model (VDART) for predicting the wake structures behind a single turbine. Figure 6.6.5a shows a comparison of the VDART predictions and the full scale experimental data of Vermeulen, Builtjes, Dekker and Bueren (1979) for the velocity deficit. For the two cases shown (1.1 and 3 rotor diameters downstream) the agreement is quite good.

Builtjes and Smit (1978) applied Lissaman's array wake model to two parallel rows of vertical axis wind turbines (essentially circular VAWTs). Taylor's model (1980) was applied to the same array. The results of the calculations are in close agreement as shown in Figure 6.6.5b. Unfortunately neither model has been verified by full scale field measurements.

7.0 DESIGN ALTERNATIVES

The overall cost effectiveness of a wind energy conversion system is based on six main factors.

1. Wind turbine manufactured cost.
2. Energy capture.
3. Installation cost.
4. Site preparation cost.
5. Maintenance cost.
6. Financing cost.

All but the last of these can be affected by the designer and must be considered when appraising design alternatives. This section discusses the important alternatives and parameters in the design of a Darrieus rotor and identifies those options which show the most promise for future developments.

7.1 Concepts

Over the years innumerable concepts have been proposed for schemes to capture wind energy. These have involved axial-flow machines (such as HAWTs) and cross-flow rotors (such as Darrieus rotors) as well as aerodynamic drag devices. Most of these have not even reached a prototype or test stage. A number of the more innovative Darrieus rotor concepts are listed in Table 7.1.1 together with some comments. Sketches of some of the conepts can be found in Figures 5.1a,b

Table 7.1.1 Innovative Concepts for Darrieus Rotors

concept	**advantages**	**disadvantages**	**comments**
2 blades in L shape	smooth loads, torque	balance needed	Lungstrom 1986
double blade	stable in high winds	aero inefficiencies, strut losses, added mass	Lungstrom 1986
spiral blades	smooths load,torque	unstable in survival wind	Lungstrom 1986
inclined shaft/tripod	minimizes foundations	lower energy capture	Lungstrom 1986
compound rotors	fewer foundations added height	added mass/swept area maintenance	DAF Indal 1984
hinged blades	lessens torque ripple lower out-of-plane forces	starting/stopping impacts hinge detail	Adecon see below
Y-shape blades	blades are straight	high blade bending	
Delta-shape	blades are straight	high blade bending	
Sail blades	all components in tension	membrane fatigue,	Feldman 1989

stowing mechanism

7.2 Past Selection

Table 7.2.1 lists the major items and alternatives which have been incorporated in the past design of rotors. This table should be viewed together with Table 4.1 which lists the major features of most of the Darrieus rotors that have been built. Section 4 of this report discusses the rationale for the selection of a number of design alternatives in the past.

The most common combination has been a two-bladed rotor using aluminum extrusions for the blades, a tubular steel central column, full struts and an aspect ratio of between 1.3 and 1.5. However there have been a number of exceptions which have had varied success and which are worthy of closer attention. A few of these are discussed below.

7.2.1 Three-Bladed Rotors

Section 4.3 discusses the advantages and disadvantages of two vs. three blades. Three bladed rotors were tried early in the development of the Darrieus but gave way to the two-bladed rotor. This is probably an example of the laws of structural- and aero-dynamics combined with overall economics. The rotor with the lowest solidity will usually capture the most energy for the least installed mass and cost. However, structural considerations favour blades with larger chord since the elastic modulus (controlling stress for a given bending moment) increases with the square of the chord. The logical outcome of this would lead to a one-bladed machine. However this confronts the designer with balance problems which the industry has not yet seriously addressed.

The additional complexity of erecting a three-bladed rotor, has also favoured the two-bladed rotor. The only circumstances which might lead to a cost-effective three-bladed rotor is the demonstration that the former has structural dynamics which are considerably more benign than those for the latter. Such an exercise has probably never been carried out because until recently the design tools have not been available.

7.2.2 Hinged Blades

The most noticeable feature of the Adecon 19m machine was the replacement of the guy cable system by a rigid external frame. However, one of the other innovative features of that machine, which was possible more significant, were the hinged blades. This feature, which is conceptually similar to the design of some helicopter rotors, is discussed in Adecon (1986, 1987) and Malcolm (1988b).

The hinges used by the Adecon 19m were located at the roots of the blades, oriented vertically and offset from the axis of rotation of the rotor. This allowed the blades to swing freely out of the plane of the (two-bladed) rotor. Under operational centrifugal and mean aerodynamic forces an equilibrium is reached with the blade advanced a certain amount and the torque is transferred into the central column by the blade pulling eccentrically on the offset hinge.

The principal advantage of such a configuration is that all out-of-plane actions of the blade are attenuated. This applies to the 2P torque ripple and the 1P blade out-of-plane "butterfly" behaviour. Experience and analysis confirmed the principal features of this concept and the subsequent reduction in fatigue stresses. However, fatigue due to in-plane motion of the blades was not attenuated, and it proved difficult to control the blades during braking, especially after a partial start. Nevertheless, the proponents should be given much credit for imaginative problem solving and for tenacity.

Malcolm (1988b) showed that the in-plane blade bending was not attenuated and that the resulting stresses became critical for blade fatigue life.

Table 7.2.1. Design Alternatives Selected for Past Designs

Parameter	**Values/Choices**				
number of blades	1		2*		3
aspect ratio	2.0		1.5*		1.0
blade material/construction	aluminum* extrusion		sheet steel	steel core	
struts	no struts		full struts*		no struts
central column	uniform tube*		tapered tube		steel truss
brakes	low speed*	high speed	electrical	aerodynamic	
support	guys*		cantilever		rigid frame

Note: "*" indicates the most common configuration

7.2.3 Cantilever Rotors

The only cantilever Darrieus rotor of any size and which has been documented is the 24 m machine "Pionier I", which was funded and operated through the Netherland's Energy Research Foundation. Machielse et al. (1986a,b) and Machielse & DeGroot (1986) give information on the structural response of this machine. The machine tested was clearly a research prototype and no comments were made on the potential economics of production versions. See also Section 2.2.4.

While a cantilever Darrieus has the advantage of fewer foundations and no potential guy-blade interference, it also has a number of disadvantages.

1. The bearings restraining the rotating parts have to be of large diameter.

2. The bending moments in the rotating and/or non-rotating column are greater than for a guyed rotor.

3. It is necessary for one tube to fit within another for at least part of the rotor height. This is wasteful.

4. To have the same natural frequencies as an equivalent guyed structure, a cantilever system must be stiffer (and therefore more costly).

7.2.4 Sail Blades

The first known vertical axis wind turbines (used in the Middle East) were made with a number of vertically rigged cloth sails. This concept has received more recent attention by Feldman (1989) who has claimed significant cost advantages of such a system.

Feldman proposed to use two parallel cables between which the membrane would be supported and which would develop an airfoil profile by the action of aerodynamic pressures. The overall profile was defined by the centrifugal forces on two spacer masses on each blade. The blades would be deployed or withdrawn by winching the cables in or out.

No known prototype work has been carried out on this concept and the problems have not been evaluated. These problems would include fatigue on the sail membrane, reliability of the deploying mechanism, and transfer of the torque from the cables to the central column.

7.2.5 Double Blades

Ljungstrom (1989) proposed a series of rotors which incorporated double blades connected by a number of spacers. This concept has the advantage that the in-plane stiffness and strength of the combination is many times greater than a single blade and survival wind stability can be achieved with relatively small blades.

Disadvantages are that the second blade does not contribute to the performance as if it were a single blade; the parasitic loss at the intersections of the blades and the spacers can be considerable; and the blades could be costly to manufacture.

7.3 Future Design Alternatives

The cost-effectiveness of Darrieus rotor wind turbines is often first judged by comparison with horizontal axis wind turbines. In this comparison the Darrieus rotor suffers the disadvantage that it has not benefited from development beyond what might be labelled a first generation machine. One objective of this study is to identify alternatives and improvements that could enhance the Darrieus rotor design. Those improvements might come from the following areas.

1. Improved airfoils or combinations of airfoils to improve overall aerodynamic efficiency.

2. Airfoil modifiers such as vortex generators, and boundary layer control by blowing.

3. Increased height above ground to make use of vertical wind shear.

4. Alternative support schemes to attenuate fatigue stresses.

5. Simplified drive train and rotor support using the advantages of ground level mounting; optimizing the drive train with the rotor shape and speed.

6. Inclusion of aerodynamic braking.

7. Variable speed operation.

Whether new concepts and configurations are introduced or not, the same principles of reduction in cost of energy will apply. These are listed at the start of Section 7.0. For the designer these can be further expanded.

1. The swept area must be maximized while limiting the peak torque and power.

2. The total mass must be minimized while maintaining structural integrity.

3. Lower mass (and cost) can be obtained by minimizing solidity and increasing rotor speed.

7.3.1 Airfoils

The peak and overall efficiencies of HAWTs have increased considerably over the past decade. For example peak aerodynamic efficiencies of 0.49 can be reached and in a 8.04 m/s mean (Rayleigh) wind speed annual production of 1500 kWh/m^2 of swept area can be extracted (electrically). This has been achieved by improved airfoils, variable speed or multi-speed operation and more efficient drive trains.

These levels of performance cannot, at present, be matched by the Darrieus rotor although the gap is not great. The following modifications might improve aerodynamic efficiency.

> -Airfoil profiles that reduce drag. These might be improvements on the attempt at laminar flow blades designed at Ohio State University (Hoffman & Gregorek 1989) and used on the Sandia 34m Test Bed. See Section 2.1.3.2.
>
> -Changing the chord and/or airfoil section along the blade span. This was done only in a stepwise manner on the 34-m Test Bed but was planned for the CENEMESA 23 m rotor. This change depends largely on manufacturing technology. See Section 2.1.3.2.
>
> -Offsetting the blade (discussed in Section 2.1.3.6). This is equivalent to changing the pitching of the blade and was investigated on one of the earlier Sandia test machines (Klimas & Worstell, 1981). The concept shows some promise and should be more thoroughly explored.

Nearly all blade configurations on Darrieus rotors have the disadvantage of not being able to twist the blade to tune the lift and drag to the angle of attack. In addition it is not easy to incorporate pitchable tips or ailerons to control peak power output. These are aspects which the Darrieus design must overcome by alternative concepts or by lower capital cost.

7.3.2 Height

The additional energy to be captured by increasing the height of the rotor can be considerable in wind regimes where there is vertical wind shear. Recent design of HAWTs has tried to capitalize on this method of improving performance. At least one study of Darrieus rotor design has included this parameter (DAF Indal 1984c). However, in the design of HAWTs, the addition of height, without undue increase in total cost, is considered a primary design objective.

There are two principal ways in which the rotor may be raised - by increasing the supporting structure or by extending the central (rotating) column downwards. The DAF Indal study concluded that both methods were effectively neutral to the cost of energy in a regime with a wind shear exponent of 0.16. The extension to the central column could lead to larger column diameters if the column stiffness was critical to the design.
These conclusions might change if, for example, the design criteria for the column changed and the cost penalty associated with extending the column downwards were to decrease; or if the penalty for longer guy cables were to lessen.

7.3.3 Support System

One method of reducing fatigue stresses has been to introduce vertical hinges at the blade roots (see above). Another approach is to lower the stiffness of the supporting guy cables so that the fundamental natural frequency is lower than the 1P excitation . This possibility has been investigated by Malcolm (1992) and LavalinTech/R. Lynette (1992) who compared equivalent rotors with normal (stiff) and soft support systems. They predicted that changing to a soft system would halve both the in-plane and out-of-plane fatigue stresses in the blades. Figure 7.3.1 shows the fanplot (natural frequency vs. rotor speed) for one "stiff" and one "soft" rotor. Table 7.3.1 is a summary of some of their major results.

The rms of the cyclic stresses is commonly used to compare fatigue loading. It is apparent from Table 7.3.1 that the two stiff rotors (L24 and GL2B) have fatigue stresses approximately 2.0 times their soft counterparts (GL2A and GL2B). Nearly the same can be said of the upper restraint forces.

Table 7.3.1. Summary of Blade Forces and Stress and Upper Restraint Forces, V=20 m/s (Malcolm 1992)

	Stiff + Struts Al. Blade L24	Soft No Struts Fibre Blade GL2A	Stiff, No Struts Fibre Blade GL2B	Soft, + Struts Fibre Blade GL3B
Maximum bending moments (N.m)				
OP, in-plane	380	1400	1400	340
OP, out-of-plane	8300	17000	17000	6200
1P, in-plane	1200	1500	5800	1000
1P, out-of-plane	11200	21000	55000	6700
3P, in-plane	2100	5000	2900	1200
3P, out-of-plane	13600	11000	8000	2300
Maximum rms cyclic stress (MPa)				
Trailing/leading edge	11.5	5.8	13.5	4.8
Inner/outer faces	7.0	4.7	8.7	3.8

Upper restraint forces (kN)

1P, in-plane	49.2	20.1	100.2	20.1
1P, out-of-plane	11.8	27.9	55.4	26.4
3P, in-plane	3.44	3.52	22.3	3.25
3P, out-of-plane	4.57	1.10	5.66	1.13

7.3.4 Drive Train

Although the drive train of a wind turbine is not so visible as the rotor and does not so clearly characterize the machine, it does represent about 50% of the manufactured cost. It is also affected by the type of rotor. Moreover its design can greatly affect the energy capture and overall cost effectiveness of the wind turbine. Some consideration must, therefore, be given to the available options and future possibilities. Design of the drive train is also discussed in Sections 4.11, 4.12 and 4.13.

The most common configuration has been a helical gearbox (or speed increaser) feeding into an induction motor/generator. There has usually been one or more low speed couplings and one high speed coupling. There have been many variations of this arrangement, such as parallel or right angle gearbox, 1200 or 1800 rpm generator, and length of low speed shaft. Some of the more fundamental alternatives are listed below.

Direct Drive

The cost of the gearbox is the major drive train cost and its elimination has tempted numerous designers. For small high speed rotors a direct electrical drive is possible without requiring a large number of poles in the motor to generate at 50 or 60 Hz. For large machines, the number of electrical poles required and the implied diameter becomes prohibitively large.

However, a direct drive was selected for Projet Eole although a rectifier-inverter system was required to convert the power to 60 Hz. This variable frequency and speed system was of considerable advantage for a prototype system but it did contribute to the high cost of that machine.

Much effort is being expended at present to develop an inexpensive electical sytem to operate at varible speed which might operate with or without a speed increaser.

Belt Drive

Adecon Energy Systems (1990a) has proposed the replacement of a conventional gearbox with a belt drive. This system, which is still in the design and development phase, would have one large driving wheel and one or more driven pulleys and would use a kevlar belt. Such a system is well suited to a ground-mounted drive train and promises to reduce the cost of the speed increaser by several times. However there may be much testing to do before good reliability can be assured. Problems may lie with the amount of energy loss, and the sensitivity to dust and dirt.

Specialized Gearbox

Rather than using available commercial components some manufacturers have designed special speed increasers. Principal among these has been Indal Technologies who combined a single bull gear with a brake disc. This system has been described in Section 2.3.1.4.

The drive train might have been perfected if given required resources, but problems were found with the pinion wear, with retaining the hydraulic fluid, and with the strength of the journal bearing rings. The possible lesson from the Indal 6400 was that specialized design and testing of drive train components is just as major an undertaking as development of the rotor itself.

Combined Gearbox and Rotor Support

It is possible to upgrade the low speed bearing of a gearbox so that it can support the direct weight of the rotor. This has been done in the VAWTPOWER machines and in several Adecon wind turbines. The advantage of such a system is that no separate support structure and bearing is required and two couplings and a connecting shaft are eliminated.

Some disadvantages are that the gearbox must be specially made, some provision must be made to support the rotor while the gearbox is removed, the rotor column must be extended downwards to maintain height,and the problem of braking must still be addressed. However, the arrangement does represent simplicity and should be seriously considered for any new design.

7.3.5 Aerodynamic Braking

Most Darrieus rotor wind turbines have used a mechanical brake on the low speed shaft (see Table 4.1). It is estimated that the cost of such an arrangement can represent up to 15% of the cost of the entire wind turbine and there is much incentive to consider alternatives.

Mechanical brakes have been used on the high speed shaft (such as on the Adecon 19m). However, the braking torque must then be passed through the gearbox, and that component should be upgraded accordingly, largely negating the lower cost of the braking system.

Electrical braking can be used in machines equipped with variable speed but the costs of such a system are not yet commercially attractive. Some Darrieus rotors have used some kind of aerodynamic spoiler to control the machine in the event of an overspeed. The NRC/Hydro Quebec 24m machine on the Magdalen Islands, several earlier DAF Indal machines, and Projet Eole included spoilers at mid-rotor. They have been largely abandoned because it was considered more cost effective to spend the effort on a reliable mechanical brake (which was required and used for normal operation). The effectiveness of spoilers was discussed in Section 2.1.3.3.

Efforts to design a reliable aerodynamic braking system might be well worthwhile, especially if it can be used for normal braking and thereby reduce the mechanical braking requirements and cost (paralleling the configuration of many HAWTs). Adecon Energy Systems have developed a "paddle-wheel" air brake which is hinged at a point on the horzontal struts. The arm is released in an overspeed condition, swings out and generates considerable drag. The system reliability is not known.

Another system was developed and patented at Sandia National Laboratories (Sullivan 1985). It consisted of a blade spoiler which could slide along the blade and which normally rested at the lower root. Upon release, centrifugal forces moved the spoiler up and out along the blade.
Sandia also investigated the braking or power control obtainable through pumped spoiling (see Section 2.1.3.5). While the conclusion of their research was that pumped spoiling was not effective enough to act as a braking system, it did offer the potential to control peak power and torque. The further development of this technique, does, therefore hold some promise.

7.3.6 Variable Speed

The ability to operate at variable or multiple speeds can increase the energy capture by between 5% and 10% (largely depending on the wind regime). This is an attractive possibility if it can be achieved economically.

Variable speed has been included in at least two Darrieus rotor wind turbines - Projet Eole, and the Sandia/USDA 34m Test Bed. Both machines used an electronic rectifier-inverter system, but neither would claim that it was a cost-effective measure. However, in both cases the variable speed feature enabled the speed to be tuned for research purposes or to select speed(s) most suited to the structureal dynamics and windspeed.

A number of HAWTs have been adapted to 2-speed operation by using motor/generator with double sets of windings. Another method is through the use of multi-speed gearboxes. Such systems are equally applicable to the Darrieus rotor and could be equally cost effective. One FloWind 19m drive was adapted in this way (Schienbein 1987)

7.3.7 Summary

This section has reviewed possible alternative configurations and features. Some have already been used or tested while others have been studied or merely been the subject of speculation. Table 7.4.1 lists those features which are considered to be the most technically and economically viable.

There are several steps that can be taken to lower the capital costs per swept area. The most promising of these are soft guy cables and pultruded figreglass blades, together with a combined gearbox/lower bearing, high speed brakes backed up by pumped spoiling for power control.

The other aspect that must be addressed is the efficiency with which energy is extracted from the swept area. For example, comparison of the energy extracted per m^2 by the Lavalin 24m design and the most recent HAWT designs in an 8.04 m/s mid-rotor mean Rayleigh windspeed, shows a 18% disadvantage. This gap has to be narrowed or bridged. This can be partially achieved by better stall-controlled airfoils, vortex generators, and offset blades. Tapered chord blades can improve energy capture by a further 5-10% but their manufacture may not be cost effective.

Table 7.4.1 Viable Improvements

feature	**comments**	**dev. simplicity** 0 to +5	**cost reduction** -5 to +5	**energy increase** -5 to +5	**total** -10 to 15
soft guy cables	very promising, should be tested	3	5	0	8
truss column	combine with soft guys	4	4	0	8
pultruded fibreglass blades	promising; elastic bending?	3	3	0	6
hinged blades	partially successful reliability?	3	3	0	6
blade offset	useful, simple	5	0	1	6
combined gearbox/ bearing	already used, promising	4	2	0	6
belt drive	applicable to VAWT; R&D needed.	1	5	-1	5
vortex generators	already used; need testing with new blades	4	-1	2	5
multiple discrete speed generator	already tested	4	-1	1	4
tapered chord	manufacture?	2	-2	3	3
pumped spoiling	power control, more development	1	-2	1	0
aerodynamic spoilers	reliability?	1	-2	0	-2
variable speed	5% energy increase, cost?	1	-3	1	-2
direct drive	used on Eole, expensive	2	-3	0	-1

Note:
1. The LavalinTech L24 is used as a basis for evaluation.
2. Least difficult to develop = 5
   Most difficult to develop = 0
3. Greatest reduction in manufacturing cost = 5
   Zero reduction in manufacturing cost = 0
   Greatest increase in manufacturing cost =-5
4. Greatest energy output increase = 5
   Zero energy increase = 0
   Greatest energy output decrease =-5

8.0 TURBINE COSTS

8.1 Introduction

8.1.1 Comments on Actual and Estimated Costs

Real manufacturing and installed cost data for wind turbines, both HAWTs and VAWTs, are not readily disclosed in the wind power industry. This is mainly due to the fact that costs depend heavily on the size of the order (project), the location of the project, the terms of the financing, the terms of the power purchase agreement and the timing of the project development.

Although some cost estimates and actual cost data are available for research turbines and proposed designs, great caution must be exercised in making comparisons and in extrapolating these data to volume commercial manufacturing costs and total installed costs of wind farms. This is because there is no well established relationship between Darrieus wind turbine prototype costs and volume production costs. Only one company, FloWind Corporation, has manufactured and installed more than 100 Darrieus turbines of any model in one year. FloWind's maximum production rate reached only 209 units/year in 1984 for their 17m commercial turbine.

Cost estimates made by manufacturers or those linked closely to the manufacturing environment may carry more authority. Estimates made by FloWind Corporation, Indal Technologies, Alcoa, Adecon Energy Systems, Bechtel Group Inc. and Lavalin Engineers would fall into that category. The Bechtel Group estimates are particularly valuable as they appear to be based on the most comprehensive wind turbine and wind farm cost study in the public domain.

Caution must also be exercised in interpreting costs because many of the estimated and actual costs date from the period 1980 to 1985.

Costs presented as dollars per installed kW are difficult to interpret and to compare. Most estimates and comparisons in the wind power industry today are done on the basis of the ratio of factory cost or installed cost (to the developer) to the annual energy capture for the turbine at the proposed installation site, after the output is adjusted for availability, array losses, electrical losses and so on. This removes rated output from the comparison and the misinterpretations that can arise due to the arbitrary rating of wind turbines. (The ratio of annual production to turbine installed generating capacity can be an important measure of cost effectiveness for the designers.) This is particularly important for Darrieus turbines, where it has been found that installed generating capacity (which was often used as the rated power of the turbine) was not well matched to peak output due to a poor understanding of the stall behavior of this class of turbines.

Furthermore, commercial prototype and production Darrieus turbines designed in the late 1970s and early 1980s appear to have been designed to operate in very aggressive wind regimes (ie. high average annual energy capture per unit of rotor swept area) as evidenced by the very low ratios of rotor swept area to installed generator capacity. For example, the FloWind 19m turbine has a ratio of about 1 m^2 per kW. By comparison the Bonus 120 turbine (19.4m diameter), designed to operate in far more modest wind regimes, has a ratio of about 2.5 m^2 per kW. FloWind's proposed 23m/300kW turbine has a ratio of about 1.5m^2 per kW, demonstrating an attempt to optimize the installed capacity of its machines.

8.1.2 Manufacturers

Table 8.1.2 shows the known commercial manufacturers of Darrieus curved blade wind turbines and the estimated number of turbines manufactured. Included in the table are all turbines that are believed to have been offered for sale in a concerted manner. In other sections of this report, some of these turbines (like the Indal Technologies 50 kW/11.2m unit) are categorized as prototype machines. This is based on the assumption that, for cases where the total number of turbines manufactured was very small and the production was spread out over several years, the design had probably not been "frozen", as would be expected for commercial production. Only the FloWind 17m and 19m turbines and the VAWTPOWER 185 (19m) turbines would likely meet today's definition of commercial mass produced wind turbines.

Table 8.1.2 Commercial Wind Turbine Manufacturers

Manufacturer	Country	Turbine Models	No. Manuf.
FloWind Corporation	USA	17m & 19m	521
Indal Technologies	Canada	11.2m & 25m	17 (est.)
Adecon	Canada	19m	20 (est.)
VAWTPOWER	USA	185 (19m)	40 (est.)
Alcoa	USA	12.8m and 25m	10 (est.)

8.2 Quantity Production Costs

8.2.1 Bechtel Group Inc. (EPRI Funded) Study Results

Bechtel Group Inc. (Bechtel 1986) studied the costs and performance of complete wind power plants (or wind farms) based on the Indal 6400/500kW and FloWind 19m wind turbine designs. The goal of the study was to furnish the electric utility industry with objective, well documented and consistent cost estimates for VAWT designs for several wind power plant configurations and two production scenarios. Their work was exhaustive and the results can only be summarized and the main results discussed here.

Bechtel's cost estimates were presented in mid-1984 dollars and the authors indicated that the estimates were intended to represent mature designs for the two turbine models chosen.

Four estimates were prepared for combinations of the two turbine designs and large and small cluster sizes based on existing (1984) production methods at a rate of 100 to 500 units per year. Another set of four estimates was prepared based on a high production rate of 400 to 1000 units per year and a five year production run.

Table 8.2.1 shows the turbine costs to the utility (price ex works), the estimated annual energy capture for the San Gorgonio Pass (California) wind distribution used and the ratio of the two for the two production scenarios described in the report. The turbine prices (ex works) were extracted from the tables of "facilities investment" costs by using the amounts shown in "Cost Account 30" which was reported as the purchase price of the turbine, including "design, engineering, fabrication and shop assembly". The study used performance adjustment factors of 0.823 for FloWind and 0.831 for Indal to adjust the theoretical perfect energy capture predictions to account for array losses, control system efficiency (start/stop logic), electrical losses and turbine availability. It is interesting to note that these are about the same factors used today for wind farm development feasibility studies, based on about eight to ten years of operating experience in California.

Table 8.2.1 Turbine Cost and Production Based on the Bechtel Study

Turbine	Existing Method			High Production Method		
	Cost	**Energy**	**$/kWh**	**Cost**	**Energy**	**$/kWh**
FloWind 17 (500 unit cluster)	$99,046	218,000 kWh	0.45	$89,594	218,000 kWh	0.41
FloWind 17 (20 unit cluster)	105,000	218,000	0.48	89,600	218,000	0.41
Indal 6400 (250 unit cluster)	315,900	639,100	0.49	286,068	639,100	0.45
Indal 6400 (10 unit cluster)	341,600	639,100	0.54	286,100	639,100	0.45

Although there appears to be little to choose between the two turbines, based on the results shown in the table, Bechtel found that the levelized cost of energy was about 20% less for the Indal turbine. This was based on Bechtel's financial model for investor-owned utility financed wind power plants (constant mid-1984 dollars). Bechtel found the capacity factors were equivalent for the two turbines. This reflects the similarity in the performance of the turbines when reduced to nondimensional terms.

Bechtel also found that higher volume production methods decreased the cost of energy by only 10%. This may suggest that these particular VAWT designs did not lend themselves to significant savings in labor, materials or handling costs when produced in quantities of up to 1000 units per year (Bechtel's high volume production method assumption). It might also be concluded that a volume of 1000 wind turbines per year is still not sufficient to take advantage of production line cost savings found in the manufacture of many consumer goods, for example. This problem may well be common to all wind turbine manufacturers. If true, the steady increase in the size of commercial HAWTs must be the result of having to consider other factors in the cost relationship, such as limited land availability and noise.

FloWind's commentary on the report, dated 85/09, included a number of recommendations where the assumptions would need to be updated. Of particular interest is FloWind's claim that the FloWind 19m produces 30% more energy than the 17m turbine at a 10% increase in installed cost.

8.2.2 FloWind Corporation

High volume production of curved blade Darrieus vertical axis wind turbines has only been attempted by FloWind Corporation. FloWind manufactured 521 turbines during the period 1982 through 1985, with 512 being installed in two California wind farms during a period of only two years (83/12 to 85/12). Table 8.2.2a shows the breakdown of the turbines installed in the wind farms. The remaining nine turbines were installed as single units at various sites in the United States and Antigua. In addition, three FloWind designed 19m turbines were manufactured and installed in Spain under a licensing agreement with CENEMESA of Spain (1987-88).

From Table 8.2.2a, it is seen that the maximum number of turbines installed by FloWind in one year was 250 (1984). Assuming that all of these turbines were manufactured in that year, FloWind's maximum rate of manufacture of a given turbine model reached only a peak of 209 turbines per year (1984). Similarly the maximum rate of production for the 19m turbine appears to have been 160 turbines per year in 1985.

Table 8.2.2a FloWind Wind Farm Turbines

Turbine Model	Altamont Pass		Tehachapi				
	1983	1984	1985	1983	1984	1985	Total
17m	40	88	20	0	121	40	309
19m	0	21	0	0	20	160	201
25m	0	0	0	0	0	2	2
All	40	109	20	0	141	202	512

19m Turbine Actual Costs

FloWind[4] reported that the manufacturing cost of the 19m turbines (rated at 250 kW) produced in late 1985 was $88,000 per unit FOB factory (ex works). FloWind also reported that the average per unit installation cost of 143 of the 19m turbines installed in California in 1985 was $40,000. FloWind stated that these costs of manufacturing and installation (in California) were very competitive with the costs of HAWTs of similar scale in 1985.

23m Advanced Turbine Estimated Costs and Selling Price

FloWind[5] reported that a 23m/300kW turbine of advanced design could decrease the cost of energy by at least 30% over the 19m turbine and would compete with HAWTs like the Vestas V-27 and Mitsubishi 28 m turbines. Wind Power Monthly reported factory prices of $180,000 and $190,000 respectively for these turbines in 1989.

FloWind suggested that its advanced 23m/300kW turbine could be manufactured for a cost of $102,000 per unit in 1989 and deliver a ratio of 1987 factory cost to annual energy capture of $0.19/kWh for a 7.15 m/s Rayleigh wind site. (The FloWind study was done in 1987 and a rate of inflation of 5% per year was used between 1987 and 1989.)

Schienbein (1991b) later reported FloWind's estimated selling price for the advanced 23m turbine (F-23) as $185,000 (ex works) in 1991 in a production quantity of only 100 turbines. Table 8.2.2b is taken from Schienbein's presentation materials and shows that the proposed F-23 turbine would compete with the five currently available commercial HAWTs for which turbine price data were available. This is based on the ratio of calculated ex works price to annual kWh of delivered energy.

[4] FloWind Corporation, Brochure, October 1989.
[5] FloWind Corporation, Brochure, October 1989.

Table 8.2.2b FloWind F-23 Versus HAWT Turbine Prices

	Horizontal Axis Wind Turbine					FloWind F-23
	H1	H2	H3	H4	H5	
Rating (kW)	300	225	400	300	450	300
Rotor Dia. (m)	32	27	34	29	36	23
Swept Area (m^2)	804	573	908	661	1018	458
Output/Area[(1)]	1070	1026	1072	1117 771[(3)]	1023	933[(2)]
Turbine Price[(4)]	350	262	459 314	261-	462	185
Price/kWh	0.41	0.45	0.47	0.35- 0.43	0.44	0.41- 0.49

Notes:

1. kWh/m^2, 7.15 m/s annual average wind speed at 40 m, density 1.08 kg/m^3.
2. No Shear
3. With 1/7 power law wind shear exponent assumed.
4. $1000

34m Turbine

FloWind has provided some data regarding a conceptual 34 m diameter turbine called the F-34 (Schienbein 1991b). For a "not optimized" F-34, FloWind reported that the energy capture per unit of rotor area was 96% of that for the conceptual 23 m, or F-23 two speed, turbine in the wind regime at the "California Site" and 94% for a 7.15 m/s site (no shear). The rotor swept area for the F-34 was given as 955 m^2, or about 2.09 times that of the F-23 turbine.

Wind Stats (Summer 1991) reported that FloWind had set a cost target of $400,000 for the 34m turbine. If this is assumed to be a selling price ex works, the not optimized F-34 appears to be about 9% less cost effective ex works than the not optimized F-23. No data are available to compare the costs of energy on an installed cost basis. Nevertheless it might be concluded, on the basis of these limited data, that the target price for the F-34 is too high, if the F-34 is to be a significant improvement over the F-23.

Cost Comparisons for FloWind Turbines

The preceding FloWind data and the data provided by the Bechtel study can be combined to investigate the trends in the size, costs and energy production of the FloWind 17m, 19m and proposed 23m wind turbines.

Several assumptions need to be made:

1. The FloWind 19m cost to the developer in 1985 was 10% greater than that of the 17m turbine, based on FloWind's commentary in the Bechtel study and assuming that the installation and manufacturing costs both increased by 10%.

2. The FloWind 19m cost to the developer (ex works) in 1985 was $121,275, based on Bechtel's FloWind 17m price of $105,000 (mid-1984) for existing production methods in a 20 unit cluster and an inflation rate of 5% per year as used by FloWind. This yields a manufacturers' gross margin of $33,275 per turbine or about 38% of the manufacturing cost reported by FloWind. This is a reasonable margin.

3. The FloWind proposed 23m/300kW price ex works was quoted as being $185,000 (early 1991) for a quantity of 100 units. This can be converted to a late 1985 price by applying an inflation rate of 5% per year for five years, resulting in a price of $144,952 in late 1985. The F-34 target of $400,000 can be converted to a late 1985 price in the same way.

4. Energy production estimates for the FloWind 23/300 kW turbine are assumed to be identical to those reported by Schienbein and Im (1989) for the FloWind high energy rotor (23m) VAWT. These show that the high energy rotor will produce 70% more energy in a Rayleigh distribution at 6.26 m/s with no shear and about 59% more energy for 7.15 m/s. than the 19m rotor.

Dodd (1990) showed the 19m FloWind turbine generating 3.4% more energy per m^2 than the 17m turbine in a 6.3 m/s Rayleigh site with no shear. Dodd also reported estimated production for a Sandia 34 m turbine as being 38.6% higher than that of the 17m turbine in terms of kWh/m^2.

Using the data from Dodd for the 19m and 34m turbines versus the 17m turbine and the Schienbein and Im data for the 23m/300kW versus the 19m turbine, a table of costs (in late 1985 dollars) and energy production for the four turbines (approximate 6.2 m/s Rayleigh site) has been constructed as shown in Table 8.2.2c.

Table 8.2.2c Estimated Prices and Energy Production for FloWind Turbines

Turbine	**Area**	**Price**	**Rel. $**	**Rel. kWh**	**Rel. $/kWh**
17m	241	$110,250	1	1	1
19m	316	$121,275	1.1	1.35	0.81
23m/300kW	458	$144,952	1.31	2.3	0.57
34m	955	$313,410(1)	2.84	5.49	0.52

(1) FloWind's reported target price converted to a price in late 1985.

This table shows almost 30% improvement in the cost of energy (price ex works divided by annual production in kWh) for this wind regime in going from the 19m turbine to the 23m/300kW. By comparison, Malcolm (1987d) showed a decrease in the cost of energy of about 22% for a tapered chord high energy rotor compared to the baseline Indal 6400 (25m) rotor for a site where the average wind speed was 7.4 m/s at mid rotor with K=1.68. Although these studies are not entirely comparable, annual energy increases for high energy rotors when compared to baseline rotors will tend to decrease with average annual wind speed. The 59% increase reported by Schienbein and Im (1989) for the 7.15 m/s Rayleigh distribution yields a relative cost of energy for the 23m/300kW rotor of about 0.61 or a reduction in the cost of energy compared to the FloWind 19m rotor of about 25%. This is reasonably close to Malcolm's result of 22% for a similar average annual wind speed.

A 9% decrease in the cost of energy is shown in Table 8.2.2c going from the 23m/300kW to the 34m turbine for this wind regime. This compares to a 9% increase based on the performance estimates reported by FloWind (Schienbein 1991b). This is probably attributable to the fact that Dodd's estimates were done for a variable speed rotor while the estimates for the FloWind F-34 were for a fixed speed of 34 rpm (the 23m/300kW turbine estimates assumed two speed operation at 35 and 52 rpm). Dodd reported that the variable speed rotor produced 15% more energy than the same rotor operating at a fixed speed of 35 rpm for the 6.3 m/s site. This nearly accounts for the approximately 18% difference noted above.

8.2.3 Polymarin

Polymarin of Holland reported in 1984 ("Holland and Wind Energy -Efforts for a Worldwide Application", Consultancy Services Wind Energy Developing Countries (CWD), 1984) that the price of its Pionier I turbine (15m cantilever design) was Dfl. 325,000. They also reported that they had developed a scaled-up 21m model that would have an estimated output twice that of the Pionier I while the price would be approximately the same.

8.2.4 Indal Technologies Inc. (Formerly DAF Indal)

Schienbein and Malcolm (1983) reported on the economics of the 50 kW and 500 kW turbines. The price of the 50 kW turbine for remote applications was given as CDN $220,000 and for windfarms CDN $200,000. The 500 kW prices were given as CDN $750,000 for remote applications and CDN $700,000 for windfarms.

The authors noted that both turbines were already cost effective at remote sites, based on electrical energy prices in Northwest Territories, Canada averaging CDN $0.249 per kWh. It was further stated that, where the average annual wind speed exceeded 8 m/s with a Rayleigh distribution, the levelized cost of energy for the 500 kW turbine was less than CDN $0.16 per kWh. The authors stated that "levelized costs of energy of less than 19 cents/kWh (Canadian) are attractive for wind farm applications in California". The Bechtel study determined that the cost of the Indal 500 kW turbine to the developer would be only US $341,600 (mid-1984), even using existing production methods and for a small (10 unit) installation. The difference between CDN $700,000 and US $341,600 cannot be completely explained by the then prevailing currency exchange rate. Indal did not comment on this difference in the written commentary (85/8) that was printed in the Bechtel report.

Schienbein and Malcolm (1983) stated that "although the isolated applications market is the most favourable immediate market for the 500 kW VAWT in terms of cost of energy difference, adequate technological development and field support to meet the unique demands of this limited market cannot be achieved unless the wind farm market is penetrated and large volume production and sales are established". This is probably a fair statement of the situation in 1992, as well.

8.2.5 Adecon Energy Systems

The manufactured cost for the Adecon 19m turbine was reported as $60,000 "in the box" in 1986 (personal communication by W. McEachern). Of this, the drive train accounted for $24,000 (gearbox, brake and motor). The relative cost breakdown is shown in Table 8.3.1. This is by far the lowest Darrieus curved blade turbine cost reported on a $/kW basis.

8.2.6 LavalinTech

LavalinTech estimated their cost to manufacture the L-24 turbine at about US $165,000 (response to questionnaire). This cost may only be valid for the manufacture of two or three prototype units. This is quite consistent with FloWind's price of $185,000 ex works for the very similar 23m/300kW turbine (early 1991).

8.2.7 Dornier Deutsche Aerospace

Dornier (response to questionnaire) provided cost breakdown data for the proposed VAWT rated at 500 kW and with a maximum rotor dia of 33.3 m. These data are shown in Table 8.3a along with other data. Dornier reported the absolute cost level to be above that of comparable HAWTs.

8.2.8 Touryan, Strickland and Berg

Touryan, Strickland and Berg (1987) reviewed VAWT technology and stated that conservative estimates for the cost per unit energy for VAWTs were about $0.50/annual kWh which when amortized over a 10 year period would translate into $0.06-0.09/kWh, depending on interest rates and inflation figures. They stated that a reasonable levelized cost goal would be about $0.06/kWh in 1990 and that turnkey costs would have to be $500 to $800/kW. Presumably this was to suggest that these cost goals would make these turbines competitive.

8.3 Relative Cost Breakdown by Subassembly

Touryan, Strickland and Berg (1987), Malcolm (1991), Dornier (1992) and McEachern (response to questionnaire) have reported on the relative cost breakdown of vertical axis wind turbines by subassembly.   Table 8.3a shows a comparison of the data. The Touryan et al data are for a FloWind 19m turbine (extruded aluminum blades), Malcolm's data are for a second generation 24m design using constant chord NLF airfoil extruded aluminum blades, the Dornier data are for a proposed 33 m turbine with continuously tapered wood composite blades and McEachern's data are for the Adecon 19m turbine which used an external support frame (rather than guy cables) and very light extruded aluminum blades.

Table 8.3a VAWT Subassembly Cost Breakdown

Components	Cost, % of Total			
	FloWind 19m	**Eole-D 33m**	**Lavalin L-24**	**Adecon 19m**
Rotor	37	42	40	17
(Blades, % of rotor)	(43%)	-	(38%)	(50%)
Rotor Anchors	-	-	10	-
Guy Cable	5	-	5	-
Drive Train	18	14	32	40
Electrical System	-	30	-	-
Controller	19	-	5	-
Base Structure	21	-	8	-
Support Structure	-	-	-	17
Support Struct./Found.	-	14	-	-
Total	100	100	100	Incomplete

The rotor costs (as a percent of the total cost) are very nearly equal, except for the Adecon turbine. This probably reflects the fact that, with the exception of Adecon, the rotors are relatively heavy and stiff designs using tubular steel columns (masts). The Adecon external frame support design appears to have had the effect of reducing the relative cost of the rotor. This is in part because the external frame eliminates the rotor column thrust loads. Hence the rotor column cost decreases. This is indirectly shown by the increased fraction of the rotor cost attributable to the blades.

The reasons for the differences in the remaining subassembly costs are difficult to determine because, with the exception of Malcolm (1991), detailed breakdowns for each subassembly were not provided and cost category definitions differed. For example, the electrical system category for the Dornier 33m probably includes the controller, and the base structure category for the FloWind 19m probably includes the foundations.

Malcolm's more detailed breakdown for the L-24 turbine is shown in Table 8.3b.

Table 8.3b Approximate Cost Breakdown for the L-24 Turbine

Rotor		
Blades and Connections	15%	
Column and Weldments	15	
Anchors		10
Struts		5
Bearings and Housings		5
Guys		5
Rotor Total		55%

Drive Train	
Gearbox	14
Brakes	10
Structure and Foundations	8
Generator	4
Low Speed Drive/Couplings	4
Controls and Switchgear	5
Drive Train Total	45%

8.4 HAWT Versus VAWT Costs

Table 8.4 compares the projected prices (ex works) of three new and advanced VAWTs to that of an advanced light-weight HAWT. To date none of these turbines have been built. Most of the data for the L24 and H6 turbines were provided through correspondence in response to the questionnaire. Cost data for the GL20 turbine (dynamically soft rotor) were obtained from Malcolm (1992). A factor of 1.4 was used to adjust the production costs to prices ex works. Note that the estimated price for the L24 turbine is for a quantity of only four turbines. The remaining prices are for production quantities of at least 50 units.

The data in Table 8.4 and Table 8.2.2b show that the new VAWTs would compete with the five currently available commercial HAWTS (H1 through H5). However, the commercially available HAWTs and the new VAWTs do not meet the cost of energy standard set by the advanced HAWT (H6). Turbine GL20 (dynamically soft and light rotor) appears to be the most competitive VAWT design. (The L24 and F-23 turbines are very similar designs with dynamically stiff and heavy rotors like that of the Flowind 19m turbine.) All three advanced VAWTs use the NLF blade sections.

Table 8.4 Comparison of HAWT and VAWT Cost of Energy

	Advanced HAWT	Advanced VAWTs		
	H6	GL20	F-23	L24
Rating	275 kW	250	300	250
Rotor Diameter	26 m	20	23	24
Swept Area	540 m^2	471	458	485
Output/Area(1)	1022 kWh/m^2	873	933	860
Turbine Price(2)	$154,000	146,000	185,000	238,000(3)
Price/kWh	$0.28/kWh	0.36	0.41	0.57

Notes:

1. 7.15 m/s annual average wind speed at 40 m, density 1.08 kg/m^3, no shear.
2. Ex works (factory) price for a production quantity of at least 50 units.
3. Quantity of four units.

Malcolm (1990c) compared the costs of a proposed 24m VAWT to the cost of a 23m Aerotech HAWT in some detail and concluded that the Darrieus turbine should be more cost effective than the HAWT. Of particular interest is the contention that the combined cost of the blades and column (or tower in the case of the HAWT) are considerably less for the VAWT. Furthermore he stated that the costs of the pitching and yawing mechanism for the HAWT would be about equal to the costs of the guy cables and horizontal struts for the VAWT. It was concluded that the cost of the gearbox would be less for the 24m VAWT due to the higher rotor speed (50 rpm versus 43 rpm for the Aerotech).

8.5 Conclusions

1. The use of the natural laminar flow blade sections can reduce the cost of energy of a Darrieus VAWT by about 20 to 30% when compared to the first generation designs such as the FloWind 19m turbine.

2. Advanced Darrieus type vertical axis wind turbines are competitive with commercially available horizontal axis wind turbines.

3. The dynamically soft and light-weight advanced VAWT (GL20) appears to be the most competitive near term design. An advanced dynamically soft VAWT should be developed and tested.

4. The advanced light-weight HAWT (H6) appears to be about 20% more cost effective than the advanced VAWT (GL20) and about 30% to 35% more cost effective than the commercially available HAWTs. However, it must be noted that the advanced VAWTs and the advanced HAWT have not yet been constructed or tested.

5. There is no well established relationship between Darrieus wind turbine prototype costs and volume production costs because only one company, FloWind Corporation, has manufactured and installed more than 100 Darrieus turbines of any model in one year.

6. Commercial prototype and production Darrieus turbines installed in the late 1970s and early 1980s appear to have been designed to operate in very aggressive wind regimes (ie. high average annual energy capture per unit of rotor swept area) as evidenced by the very low ratios of rotor swept area to installed generator capacity. The most recent VAWT designs show a significant increase in the ratio of rotor swept area to installed generator capacity.

7. The isolated applications market could be the most favorable immediate market for Darrieus VAWTs because of the inherent simplicity of the design and the ease of maintenance. However, the technological development and field support required to meet the unique demands of this limited market may not be achieved until the wind farm market is penetrated.