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Advanced Wind Turbine Program Next Generation Turbine Development Project

June 17, 1997 — April 30, 2005

GE Wind Energy, LLC
Tehachapi, California

Subcontract Report
NREL/SR-500-38752
May 2006

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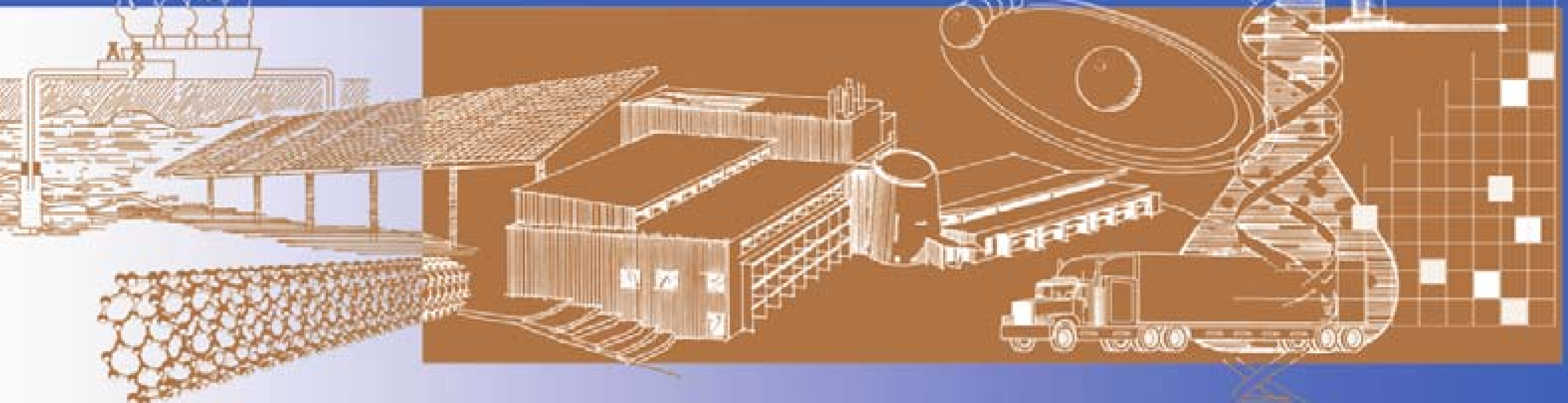

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NREL Technical Monitor: S. Schreck
Prepared under Subcontract No. ZAM-7-13320-26

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At the time of the submission to the U.S. Department of Energy of the proposal which resulted in award of NREL Subcontract Number ZAM-7-13320-26, the company now known as GE Wind Energy LLC was known as Zond Energy Systems, Inc. Zond was subsequently acquired by another company and eventually was sold to GE Power Systems Corp., a subsidiary of General Electric Corporation, and renamed GE Wind Energy. The present document uses only the names GE Wind Energy, or simply GE and refers to the company's products as the GE 750 or the GE 1.5, regardless of the time at which work being referenced was conducted. The titles of some subcontractor reports submitted during earlier stages of this contract and referenced herein contain the older name Zond or the interim names Enron Wind or Enron Wind Energy Systems Corporation (EWESC), and these are the only exceptions to the exclusive use of the GE name.

Acknowledgments

This report summarizes the results of nearly seven years of wind turbine research and development by GE Wind Energy and its predecessors. Such a comprehensive undertaking has involved the contributions of scores of GE employees and dozens of consultants, subcontractors, vendors, and suppliers. Unfortunately, their number is too great to list them all here, but GE is sincerely appreciative of all of their individual and collective efforts to accomplish the work presented in this report.

We have also received substantial technical and project management assistance from dozens of personnel of the U.S. Department of Energy, the National Renewable Energy Laboratory, and Sandia National Laboratories. Again, the contributions of all of these people are greatly appreciated. We would specifically like to express our gratitude to Alan Laxson, Brian Smith, Scott Schreck, Ed Cannon, and Paul Migliore, each of whom served as the NREL Project Manager during some portion of this project, and to Neil Wikstrom for his seven years as the Subcontract Administrator for this project.

Abstract

This document reports the technical results of the Next Generation Turbine Development Project conducted by GE Wind Energy LLC. This project is jointly funded by GE and the U.S. Department of Energy's National Renewable Energy Laboratory through Subcontract Number ZAM-7-13320-26.

The goal of the NGT Program is for DOE to assist the U.S. wind industry in exploring new concepts and applications of cutting-edge technology in pursuit of the specific objective of developing a wind turbine that can generate electricity at a levelized cost of energy of \$0.025/kWh at sites with an average wind speed of 15 mph (at 10 m height).

GE's NGT Project has consisted of three broad activities:

- Concept Studies
- Design, Fabrication, and Testing of the Proof of Concepts (POC) turbine
- Design, Fabrication, and Testing of the Engineering and Manufacturing Development (EMD) turbine.

GE Wind personnel, working with consultants, have completed investigations of a number of wind turbine system and component concepts. The purpose of these studies has been to determine the trade-off between cost and improvement in energy capture resulting from each of the concepts. These studies have focused on three broadly defined categories of concepts:

- Electromechanical systems
- Rotor and structural design
- Controls.

The electromechanical systems studies have focused upon a large number of configurations created by changing five key parameters:

1. generator synchronous speed and the corresponding required gearing
2. type of generator
3. number of generators
4. speed regulation, that is, variable speed or fixed speed
5. power conversion options.

The result of all of the electromechanical concept studies is that no concept produces a significant improvement in the COE delivered by wind power relative to the existing GE Wind turbine configuration. Only two concepts, 1) a medium-speed wound rotor induction generator operating in variable-speed, constant-frequency mode, and 2) a medium-speed wound rotor synchronous generator operating in fixed-speed mode show any improvement at all in cost of energy (COE), and even then, the estimated improvement is less than \$0.01/kWh.

The control strategies which have been investigated in the present study are:

- Coupling of Blade Pitch and Generator Torque Control
- Tracking of Peak Power Coefficient
- Tower Vibration Feedback Control

- Independent Blade Pitch for Asymmetric Load Control
- Load-Limiting Control
- Alternative Yaw Control Strategies
- Damage Monitoring and Feedback Control
- Adaptive Drive Train Damping.

All of these concepts except for the alternative yaw control strategies show potential for improving the cost of energy, either through reductions in loads or improvement of energy capture.

The rotor and structural concepts investigated include:

- Rotor and other turbine structural flexibility
- Concurrent aerodynamic and structural design optimization
- Carbon composite rotor blades
- Aeroelastic tailoring of rotor blades
- Variable diameter rotors.

Again, all of these concepts show potential for improving COE through either reduced component costs or improved energy capture.

The implications of the concept studies that show innovations most likely to produce near-term COE benefits at risk levels acceptable to the wind energy financial community are:

- Optimized low-solidity rotor blades
- Larger rotor enabled by sophisticated load-alleviating controls systems
- Advanced controls systems, including:
    - Independent blade pitch to effect asymmetric load control
    - Tower top accelerometer feedback for tower damping
    - Coupling of pitch and generator torque control
- Taller, more flexible towers.

Achieving the originally stated NGT goal of \$0.025/kWh at IEC Class II sites does not appear to be achievable in the near term for technologies with market-compatible risk levels. GE has identified through its NGT concept studies, high-risk concepts that can provide additional reductions in COE between 10%–25%, thereby making the \$0.025/kWh goal achievable. The most important of these factors are the variable diameter rotor and aeroelastic tailoring of rotor blades. Additionally, damage identification and feedback control may offer benefits to COE in the 5%–10% range. However, these three ideas entail risk perceived by the market as excessive, and they have not been pursued to the hardware stage in the NGT Program.

As the first of the concept studies were being completed, GE began designing the POC turbine. Installation of the turbine was completed at the GE wind farm in Tehachapi, California, in April, 2000. This turbine is rated at 1.5 MW and features a three-bladed, upwind rotor of 70.5-m diameter driving a six-pole wound rotor asynchronous generator through a three-stage planetary gearbox. The POC turbine employs several innovations identified in the concept studies:

- Flexible, low-solidity rotor blades employing high-lift, thicker airfoils
- Coupling of pitch and torque control
- Taller, soft hybrid steel/concrete towers.

In addition to innovations resulting from the concept studies, the POC turbine also employs a water-cooled generator.

Certification testing of the POC turbine for noise, loads, power performance, and power quality has been completed under the supervision of personnel of the National Renewable Energy Laboratory. Power performance exceeds the predicted results and the noise results indicate that the POC1.5/70.5 configuration is the quietest turbine on the market among units of similar size.

The EMD turbine was installed in Tehachapi, California, in April 2002. The EMD turbine is an evolution from the POC turbine incorporating the following improvements in its design:

- A larger 77-m diameter rotor
- 37-m blades with tip curvature
- Tower-top accelerometer feedback for tower damping
- Independent blade pitch control for asymmetric load control.

Certification testing of the EMD turbine for noise, loads, and power performance has been completed under the supervision of personnel at the National Renewable Energy Laboratory. Additional load testing to verify the performance of the asymmetric load control system has also been performed by GE personnel.

Through the application of the above-mentioned near-term concepts in the EMD turbine, GE has been able to shave the cost of energy from wind power by 13% relative to the baseline configuration, without even adjusting for inflation.

GE Wind Energy has commercialized the POC turbine as the GE1.5s. As of December 31, 2004, GE had manufactured and deployed 1160 turbines of this configuration in the United States alone. GE Wind has commercialized the EMD turbine as the GE1.5sle. As of December 31, 2004, over 1000 1.5sle-turbines were on order for delivery during 2005.

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List of Symbols and Abbreviations

ALC	Asymmetric Load Controller
CSI	Current Source Inverter
DOE	Department of Energy
EMD	Engineering & Manufacturing Development
GE	General Electric Corp.
GL	Germanischer Lloyd
GW	Gigawatts (power)
GWh	Gigawatt·hours (energy)
Hz	Hertz (cycles per second)
IGBT	Integrated Gate Bipolar Transistor
kg	Kilograms (mass)
kV	Kilovolts
kVA	Kilovolt-amps
kW	Kilowatts (power)
kWh	Kilowatt·hours (energy)
MW	Megawatts (power)
MWh	Megawatt·hours (energy)
NGT	Next Generation Turbine
NREL	National Renewable Energy Laboratory
NWTC	National Wind Technology Center
POC	Proof of Concepts
PWM	Pulse-width modulated
SCR	Silicon Controlled Rectifier
VAC	Volts, Alternating Current
VAR	Volt-amps, reactive (reactive power)
VSCF	Variable Speed Constant Frequency
VSI	Voltage source inverter

1 Introduction and Project Overview

This document summarizes work completed by GE Wind Energy LLC on the Next Generation Turbine Development Project. This project is jointly funded by GE and the U.S. Department of Energy's National Renewable Energy Laboratory (NREL) through Subcontract Number ZAM-7-13320-26.

1.1 Project Background

The U.S. Department of Energy began sponsoring the Turbine Research Program (then called the Advanced Wind Turbine Program) in 1990. The first phase of this program – Conceptual Design Studies – was completed in 1992 and identified incremental improvements and advanced configurations that could improve the competitiveness of wind energy.

Near Term Product Development, the second phase of the program, provided funding to several U.S. wind energy companies to design, fabricate, and test prototype turbines designed to produce electricity for $0.05/kWh or less at 5.8 m/s (13 mph) sites. Among these were two projects directed by GE. The first resulted in the development of the Z-40, a 550-kW turbine, of which GE Wind (then operating as Zond) installed 93 in China, Greece, Ireland, Korea, Mexico, and the U.S. The second project, the Near Term Research and Testing program, resulted in significant value engineering improvements to GE's 750-kW series of turbines. This project resulted in a substantial reduction in COE delivered by wind power.

The third phase of the Turbine Research Program is the Next Generation Turbine Development Project (NGT). The goal of the NGT Program is for DOE to assist the U.S. wind industry in exploring new concepts and applications in cutting-edge technology to develop a wind turbine that generates electricity at a levelized COE of $0.025/kWh at sites with an average wind speed of 15 mph (at 10-m height).

Through the NGT program, GE submitted a proposal to NREL in May 1995, for a project to develop an advanced-technology turbine capable of achieving the $0.025/kWh objective. GE received notice of award in May 1996. The NGT Project has proceeded in two stages. The first stage, initiated when GE and NREL signed a letter subcontract in April 1997, involved concept definition studies, which were intended to develop reliable performance and cost estimates for GE's proposed systems, along with a preliminary work plan, budget, and schedule for the second stage of the project, Prototype Development. The latter began in June 1998, when GE and NREL signed Subcontract Number ZAM-7-13320-26, authorizing full funding of GE's NGT Project. This Subcontract has been jointly funded by GE and NREL through a cost-sharing arrangement. Technical work under this contract was completed in April 2004, at which time personnel from GE, NREL, and the Sandia National Laboratories conducted the Project Final Design Review.

1.2 Scope of Work

The Statement of Work (SOW) governing this Project requires GE to design, fabricate, install, and test two prototype turbines:

- Proof of Concepts (POC) turbine
- Engineering and Manufacturing Development (EMD) turbine

The SOW also stipulates that GE seek certification of the POC turbine. The scope of work for this project requires GE to select a wind turbine system for development that will achieve a combination of improved performance, increased reliability, and decreased cost, such that the COE objective of the project is met.

The SOW stipulates that GE investigate through a series of comprehensive studies a number of concepts, the best of which would be incorporated into the POC turbine. The baseline turbine (GE's 750-kW turbine) would serve as a foundation for these studies as well as the starting point for development of the POC. The EMD turbine would represent a technological evolution from the POC, incorporating new innovations resulting from continuing concept studies.

Following signing of the subcontract, GE sponsored a kickoff meeting in July 1998, at its facility in Tehachapi, California. Present at this meeting were key GE engineering personnel and technical consultants. The purpose of the meeting was to identify a number of technologies and concepts that might be considered good candidates for further study in an effort to significantly reduce the cost of electricity generated by wind power. Concepts included both near-term technologies that might constitute incremental improvements to existing GE products, as well as advanced concepts requiring several years to develop.

On the basis of this meeting, GE identified a number of potentially beneficial concepts and signed several subcontracts with consultants to assist the company in completing studies of these ideas. The initial concept studies portion of the NGT project continued through December 1999. Some of the concepts were incorporated into the POC turbine, while other innovations required further study and have been incorporated into the EMD turbine. The concept studies are discussed in detail in Section 2 of this report.

As the first of the concept studies were being completed, GE began the design of the POC turbine, installation of which was completed at the GE wind farm in Tehachapi, California, in April 2000. This turbine is rated at 1.5 MW and features a three-bladed, upwind rotor of 70.5-m diameter driving a six-pole wound rotor asynchronous generator through a three-stage planetary gearbox. The turbine is installed on a free-standing 63-m tubular steel tower. Details of the POC turbine design, fabrication, installation, and testing are reported in Section 3.

As the initial phase of concept studies was ending and the installation of the POC turbine was underway, the design of the EMD turbine began in early 2000. As noted above, this effort began with continuing concept studies and then progressed to preliminary, detail, and final design efforts. A critical design review was conducted at NREL in March 2001. The EMD was installed in two phases. In the first phase, GE installed the 77-m rotor on the EMD turbine in Tehachapi in April 2002. In the second phase, advanced controls systems were installed on the turbine in 2003. Details of the EMD design are reported in Section 4.

2 Concept Studies

GE approached the design of its next generation turbine from the premise that any turbine concept or technology that might yield improvements in COE should be considered for inclusion in the design. As a result, at the onset of the project, GE chose not to focus all engineering efforts on designing and developing a preconceived concept for an advanced technology wind turbine. The company instead chose to identify a number of innovative concepts and technologies that might be expected to produce improvements in COE. At project commencement, GE expected to spend approximately 6 to 8 months conducting studies of these concepts and then, based upon the results of these studies, to select the most promising, feasible, and mutually compatible technologies to incorporate into the preliminary design of a POC turbine. As research and development progressed during the NGT project, new concepts were continually proposed and concept studies continued in parallel with the development of the POC and EMD turbines. As some of the best concepts were refined, they were introduced into the POC and EMD turbine designs, as explained in Sections 3 and 4 below.

GE Wind personnel and consultants have completed investigations of several wind turbine system and component concepts. The purpose of these studies has been to determine the trade-off between cost and improvement in energy capture resulting from each of the concepts. These studies have focused on three broadly defined categories of concepts:

- electromechanical systems
- controls systems
- Rotor and structural design.

Each of these categories of concept studies is discussed in Sections 2.1 through 2.3.

The concept studies were initially conducted at a rated power of 750 kW so that the baseline Z48/750 turbine could be used as a basis for comparison. As is noted in Section 3, the POC turbine was developed at 1.5 MW. As a result, some of the concept studies conducted subsequent to the initiation of the POC design were also conducted at 1.5 MW so that their results could be compared to the POC turbine.

Because of market advantages and GE experience, it was decided from the outset that the NGT design should favor the three-bladed, upwind, variable speed, and pitch-regulated configuration, although some alternatives to this were briefly examined. Additional guiding principles throughout the concept studies have been:

- Use of readily available (commercial off-the-shelf or COTS) components
- Modularity (i.e., distributed design)
- Simplicity and ease of operations and maintenance
- Manufacturability
- Transportability
- Scalability.

2.1 Electromechanical Systems Concepts

This category encompasses concepts relating to the drive train and all of the mechanical, structural, and electrical systems and components that support the drive train. This includes everything located at the tower top (except for the rotor), primary mechanical components (such as the gearbox), and electrical systems (such as the generator, power inverter, bedplate, and nacelle). These components are grouped together because they are so highly integrated. It is impossible to discuss fundamental changes to the gearing without considering the impact on the generator, and it is further impossible to discuss major changes to the generator without further considering how this impacts the power inverter.

The studies have identified wind turbine drive train concepts that differ fundamentally from the baseline configuration in terms of either:

a) The electromechanical arrangement they employ to convert the low-speed rotary motion of the wind turbine rotor into high-quality, three-phase, 60 Hz electrical power, or
b) The infrastructure they use to support the drive train.

Concept studies have been conducted using the GE 750-kW and 1.5-MW turbines as bases for comparison. The drive trains of the two turbines differ primarily in that the 750-kW uses an integrated design in which the gearbox casing serves as the main support frame and yaw deck to which the generator and the nacelle attach directly, while the 1.5-MW features a distributed design in which a main frame bedplate serves as the yaw deck to which all of the component–main bearing, gearbox, generator, and nacelle– are attached.

The salient features common to the drive trains of both the baseline turbine and the 1.5-MW turbine and from which all proposed changes have been measured are:

- Two- or three-stage gearbox, providing output-to-input speed ratios of approximately 40:1 to 100:1
- Doubly-fed (wound rotor) induction generator with 4 or 6 poles on stator and rotor, designed to generate 60 Hz at either 1200 rpm or 1800 rpm or 50 Hz at 1000 rpm or 1500 rpm
- Variable speed operation, with the wound rotor generator operating over a speed range of approximately 25%–30% sub-synchronous slip to 25%–30% super-synchronous slip
- Partial power conversion of only the generator rotor-supplied (or fed) current.

This arrangement is employed on all of the wind turbine models manufactured by GE.

The use of the wound rotor generator allows the stator to be electrically connected directly to the main electrical grid operating at either 50 Hz or 60 Hz, depending upon the location in the world. As the entire system operates at variable speeds, the generator rotor produces variable frequency current that is converted to 60 Hz power by the power converter. When the generator is operated at speeds above the synchronous speed (e.g., 1200 rpm for a 6-pole generator in the U.S.) the rotor feeds real power to the converter. When operated at speeds less than synchronous speed, the converter must feed real power to the rotor.

The advantage of this system is that from 70%–80% of the power is produced by the stator, which is connected directly to the electrical grid without need of power conversion. Only the 20%–30% of the power derived from the rotor requires power conversion. This arrangement allows for variable speed operation of the wind turbine rotor, thereby maximizing energy capture, while allowing the use of a power converter sized for only 20%–30% of the wind turbine rated power. Partial power conversion improves the system efficiency (i.e., energy production) while significantly reducing the cost of the power electronics. Furthermore, 4- and 6-pole induction generators and motors are widely manufactured and relatively inexpensive.

In moving away from the baseline drive train configuration, the designer can work with five key parameters:

- Generator synchronous speed and the corresponding required gearing
- Type of generator
- Number of generators
- Speed regulation (variable speed or fixed speed)
- Power conversion options.

Table 1 summarizes the configuration options associated with each parameter and the anticipated cost and performance trade-offs that have motivated the studies of these options. A total of 216 permutations of the configurations shown in Table 1 exist. However, many of the permutations obviously can be eliminated. For example:

- The direct-drive option only works with one generator. The use of multiple generators requires multiple output shafts from a gearbox.
- Only the wound rotor induction generator can utilize partial power conversion.
- Fixed speed operation requires no power conversion.

Table 2 summarizes the 36-drive train configurations examined in the present study. Every configuration in Table 2 does not need to be examined separately, because several of the design parameters can be changed independently. For example:

- Changing from variable speed to fixed speed operation will impact the aerodynamic energy capture similarly regardless of the drive train configuration.
- Changing from variable speed to fixed speed operation will eliminate the power converter regardless of whether the generator is direct-drive, medium speed, or conventional speed.
- Use of multiple generators should not significantly impact the cost of the power converter, as the cost of the latter is mostly a function of the total power rating.
- Use of multiple generators should not significantly impact the energy capture.

As a result of these assumptions, the field of candidate configurations for study have been narrowed to 11 categories, as defined in Table 2. These 11 categories of drive train configurations are each discussed in the following subsections 2.1.1 through 2.1.3.

A summary of the COE impacts on proposed design changes is summarized in Section 2.1.4.

Table 1. Summary of the Key Drive Train Design Parameters and Configurations Options

Parameter	Configuration Options	Trade-offs	
		Performance	**Cost**
Generator Synchronous Speed	• Low-Speed (Direct-Drive Generator) • Medium-Speed (Single Stage Gearbox) • Conventional (1200 or 1800 for 60 Hz, 1000 or 1500 for 50 Hz)	• gearboxes represent a significant source of mechanical loss • gearboxes represent a significant source of maintenance requirements	• multi-stage gearboxes represent a significant fraction of the cost of a wind turbine • unconventional generators can be expected to be more expensive.
Type of Generator	• Wound rotor induction generator • Squirrel cage induction generator • Wound rotor synchronous generator • Permanent magnet synchronous gen.	• Squirrel cage machines and synchronous machines either require full power conversion or must be operated fixed speed. Both options will reduce energy capture.	• Squirrel cage and wound rotor synchronous machines are widely manufactured and easily manufactured, making them very low-cost options. • Permanent magnet machines can be very compact and hence relatively low cost
Speed Regulation	• Variable Speed • Fixed Speed	• Fixed speed operation generally results in a loss of several percent of energy capture	• Fixed speed operation eliminates the need for expensive power converters
Number of Generators	• Single Generator • Multiple Generators	• Multiple generators might allow a better coupling of the aerodynamic (rotor) power –vs– wind speed curve and the generator power –vs– shaft speed curve, improving efficiency	• Use of more and smaller generators might provide a small manufacturing cost savings due to economies of scale
Power Conversion	• Full Power Conversion • Partial Power Conversion • No Power Conversion	• Rectifying and inverting more power will result in greater power losses	• The more power for which the converter is sized, the more expensive it will be.

Table 2. Summary of Drivetrain Configurations Investigated

Configuration	Generator Synchronous Speed	Generator Type	Number of Generators	Speed Regulation	Power Conversion
1.1. Baseline	Conventional	Wound-Rotor Induction	1	Variable	Partial
1.2	Conventional	Squirrel Cage Induction	1	Variable	Full
	Conventional	Wound-Rotor Synchronous	1	Variable	Full
	Conventional	PM Synchronous	1	Variable	Full
1.3	Conventional	Wound-Rotor Induction	1	Fixed	None
	Conventional	Squirrel Cage Induction	1	Fixed	None
	Conventional	Wound-Rotor Synchronous	1	Fixed	None
	Conventional	PM Synchronous	1	Fixed	None
1.4	Conventional	Wound-Rotor Induction	Multi	Variable	Partial
	Conventional	Squirrel Cage Induction	Multi	Variable	Full
	Conventional	Wound-Rotor Synchronous	Multi	Variable	Full
	Conventional	PM Synchronous	Multi	Variable	Full
	Conventional	Squirrel Cage Induction	Multi	Fixed	None
	Conventional	Wound-Rotor Synchronous	Multi	Fixed	None
	Conventional	PM Synchronous	Multi	Fixed	None
2.1	Medium Speed	Wound-Rotor Induction	1	Variable	Partial
2.2	Medium Speed	Squirrel Cage Induction	1	Variable	Full
	Medium Speed	Squirrel Cage Induction	1	Fixed	None
	Medium Speed	Wound-Rotor Synchronous	1	Variable	Full
	Medium Speed	Wound-Rotor Synchronous	1	Fixed	None
	Medium Speed	PM Synchronous	1	Variable	Full
	Medium Speed	PM Synchronous	1	Fixed	None
2.3	Medium Speed	Wound-Rotor Induction	Multi	Variable	Partial
	Medium Speed	Squirrel Cage Induction	Multi	Variable	Full
	Medium Speed	Wound-Rotor Synchronous	Multi	Variable	Full
	Medium Speed	PM Synchronous	Multi	Variable	Full
	Medium Speed	Squirrel Cage Induction	Multi	Fixed	None
	Medium Speed	Wound-Rotor Synchronous	Multi	Fixed	None
	Medium Speed	PM Synchronous	Multi	Fixed	None
3.1	Direct Drive	Squirrel Cage Induction	1	Fixed	None
	Direct Drive	Wound-Rotor Synchronous	1	Fixed	None
	Direct Drive	PM Synchronous	1	Fixed	None
3.2	Direct Drive	Wound-Rotor Induction	1	Variable	Partial
	Direct Drive	Squirrel Cage Induction	1	Variable	Full
3.3	Direct Drive	Wound-Rotor Synchronous	1	Variable	Full
3.4	Direct Drive	PM Synchronous	1	Variable	Full

2.1.1 Conventional-Speed Generators and Multi-stage Gearboxes

These options are most closely related to the baseline configuration. Possible changes are:

- Use alternative converter topologies.
- Use generators other than wound rotor induction machines.
- Operate at fixed shaft speed.
- Use more than one generator.

Each of these proposed changes is discussed further below.

2.1.1.1 <u>Alternative Converter Topologies</u>.

The baseline Z750 turbine uses an IGBT-based current source inverter with an active bridge rectifier (AR-CSI). The 6-pole-generator on the baseline turbine has a synchronous speed of 1,200 rpm at 60 Hz. In the Z50 configuration, it is operated from approximately 850–1300 rpm, which means it is operating both sub-synchronously and super-synchronously. As a result, the power converter must be designed to be bi-directional, that is, it must both feed and take power from the rotor.

Unidirectional Current Source Inverter

GE investigated using a unidirectional CSI with a passive (uncontrolled) diode rectifier (DR-CSI) by operating the wound rotor induction generator only sub-synchronously. This concept probably represents the smallest change to the baseline configuration of any concept studied in the project. The proposal was to operate such a generator from approximately 720–1100 rpm, remaining entirely sub-synchronous. Although operating at 720 rpm would represent higher slip and more rotor current than operating at 850 rpm, it would probably be more cost effective to only construct the converter to handle unidirectional power.

However, the study showed that the higher rotor currents would not only require larger rotor windings, but they would likely produce greater problems with generator cooling and overall losses. Although they did not conduct a detailed design, engineers at one of GE's generator suppliers briefly examined the concept and concluded that they would have to increase the frame by one, if not by two frame sizes, and that it could produce intractable cooling problems. Another GE-commissioned study concluded that the increase in rotor power would increase the generator volume, weight, and cost. Furthermore, this work indicated that the rotor power increase was also a more significant cost driver for the converter, and estimated that the cost of a unidirectional power converter for subsynchronous generator operation, assuming the same aerodynamic rotor speed range, would cost more than the bidirectional arrangement.

Results indicated this concept was not worth pursuing further.

Voltage Source Inverter Topologies

GE also examined voltage source inverter (VSI) topologies, including:

- VSI with passive diode bridge rectifier (DR-VSI)
- VSI with DC-to-DC Boost Converter (DR-B-VSI)
- VSI with Active Rectifier (AR-VSI)

The basic DR-VSI configuration, although an economical converter solution, is not the best choice from a system design standpoint. This is due to the difficulty of maintaining a constant AC line voltage over the generator speed range. In fact, the generator terminal voltage is a function of the generator speed; in the absence of DC bus voltage regulation, the relatively wide speed range would lead to a very large DC bus voltage fluctuation. This fluctuation of the DC bus voltage would force design changes of the system for full voltage at minimum speed. Allowing the DC bus voltage to grow as the speed increases would result in tremendous converter oversizing and poor machine utilization.

The DR-B-VSI solution is sometimes adopted in adjustable speed drives when AC voltage regulation is desired without pulse width modulation (PWM) harmonics and ripple. In general, the DC-to-DC converter is a buck converter or step down converter with a different arrangement of the devices. In the case of the wind power generator, the constant voltage side would be the output of the regulator, and the variable voltage side would be the input. This configuration is very flexible and can achieve good DC bus voltage regulation even if the machine terminal voltage is very low. It has two extra switches, and one extra inductor. Both the inductor and the diode carry the entire DC bus current, and contribute significantly to the total converter losses. All the active switches can be either IGBTs (preferred) or SCRs with commutation circuits. In addition to the PWM inverter, the passive three-phase bridge, and the DC link inductance loss, this topology has the diode and switch loss of the boost converter; these extra losses are expected to add an additional 0.5% loss at full load.

The AR-VSI technology should be readily available, even if the diode bridge front end version is more common. IGBTs are the power devices of choice in this case. This AC-to-AC converter can provide input and output power factor control and obtain the best machine utilization. The converter losses are somewhat higher than the other configurations but the generator efficiency can be improved because of a more flexible and accurate machine control.

The conclusion of the study was that the baseline configuration AR-CSI is probably the least expensive of the topologies appropriate for use with the wind turbines. As noted above, the DR-CSI unit is inappropriate for bi-directional operation, and the DR-VSI unit is inappropriate for use with wind turbines. This leaves only the AR-VSI or DR-B-VSI units, both of which demonstrate higher losses and cost more than the baseline converter topology. As a result, these studies did not present any superior alternatives to the baseline system.

2.1.1.2 High-Speed, Variable-Speed Operation of Alternative Generators

The option of using generators other than wound rotor induction machines (squirrel cage induction generators, wound rotor, or permanent magnet synchronous generators) in a variable speed mode was briefly considered at the outset of the concept studies because of the assumed higher cost associated with manufacturing the wound rotor machines. The use of any of the machines other than the wound rotor induction generator, however, involves the need for full power conversion.

In terms of cost, the choice of generator represents a simple trade-off between the cost of the generator and the cost of a power converter sized for full power conversion versus one sized for partial power conversion. At the time these studies were conducted in 1998 and 1999, it was estimated that the power converter for operating a single squirrel cage induction generator over the same speed range as the GE 750-kW operated generator would cost nearly as much as the cost of the wound rotor machine and partial power converter together. Furthermore, investigations showed that the best expected price which could be expected for a commercial off-the-shelf 750-kW squirrel cage generator was approximately 75% that of the wound rotor machine, and it was not clear that these machines would have the service factor sufficient to allow their use in wind turbine applications. The squirrel cage generator and full-power converter package would cost approximately 35%–-40% more than the baseline configuration hardware.

Similar prices were identified for 750-kW wound rotor synchronous generators. No permanent magnet synchronous generators rated at 750 kW were available commercially, but GE and its consultants agreed at the time that it could be expected that such a machine custom-built could cost as much or more than a conventional wound rotor synchronous generator. Furthermore, studies estimated the cost for a full power converter for a PM synchronous generator at approximately 120%–140% of the cost of the baseline generator/converter combination. Again, the combined cost of these two components would be well more than that of the baseline configuration.

In terms of energy capture, all four types of machines can be fabricated with high electrical and mechanical efficiencies, so the impact of the generator choice on energy capture is probably negligible. The typical efficiency associated with the power converter is approximately 98%–99%. Therefore, if all of the current is converted instead of rather than at most 20% of the current, this change represents an additional loss of approximately 0.5%–1.5% of energy capture. Such a relatively small loss might be justifiable if significant cost savings result, but given the probable increase in cost resulting from these configuration changes, this energy loss only exacerbates the problem.

Furthermore, Dynamic VAR capability is frequently demanded by customers of GE wind turbines and is an important marketing consideration. Cost of energy being equal, options that provide dynamic VAR control are preferable. A squirrel cage induction generator cannot provide dynamic VAR control unless it utilizes a converter that can supply VARs to the stator. This results in a double cost penalty for these machines: the rating of the converter is higher because of the reactive power to be supplied, and the cost per kVA is substantially higher because of the active rectifier.

2.1.1.3 Fixed-Speed Operation of Conventional Generators

Some commercially available wind turbines employ wound rotor induction generators operating at fixed speeds. The motivation for this is unclear, as GE identified no significant advantage to using a wound rotor machine versus a squirrel cage machine for fixed speed operation, other than possibly an improvement in generator efficiency. As a result, GE did not give significant consideration to this option.

GE, instead, examined the option of connecting squirrel cage or synchronous machines directly to the grid and eliminating the need for power converters. Squirrel cage asynchronous generators and wound rotor synchronous generators can be purchased for approximately 60%–70% of the cost of a wound rotor asynchronous machine. The converter cost is eliminated. The cost of a 4-pole or 6-pole permanent magnet synchronous machine was not thoroughly investigated by GE, as it was decided that the machine would be more expensive than a wound rotor synchronous generator. On the performance side, however, some energy capture is lost from fixed speed operation. This energy capture was estimated by GE personnel at approximately 3%. The impact on COE is negligible, as the reduction in cost is offset by a reduction in energy capture. As a result, GE decided not to pursue these fixed speed options further. These fixed-speed options also lack the ability to provide dynamic VAR control offered by the baseline system.

2.1.1.4 Multiple Conventional-Speed Generators

GE investigated the use of a two-stage hybrid planetary/helical gearbox driving multiple generators. Figure 1 illustrates the proposed drive train layout. Although originally proposed with wound rotor asynchronous generators in mind, the concept would apply equally well to squirrel cage machines or synchronous generators. The motivation to consider this option is the possibility that reduced generator frame size and increased part count could result in reduced generator cost.

Use of multiple generators might also provide for improved energy capture, as the wind speed, shaft speed, aerodynamic rotor power, and generator power can be more optimally matched with multiple generators than with only one generator. Two generators have been used in the past, most notably in the U.S. by Kenetech.

The perceived cost advantages of this concept did not materialize upon closer inspection. For the volumes being purchased by an OEM customer such as GE, the cost per kilowatt rated for wound rotor induction generators is nearly constant as generator rating changes. The size at which wind turbines are now being manufactured means that few economies of scale were found in cutting the generator rating in half or by a third. Examination of the volume costs of off-the-shelf squirrel cage generators showed similar trends. Furthermore, the benefits to the energy capture were believed to be generally small. As a result, the use of multiple generators does not seem to

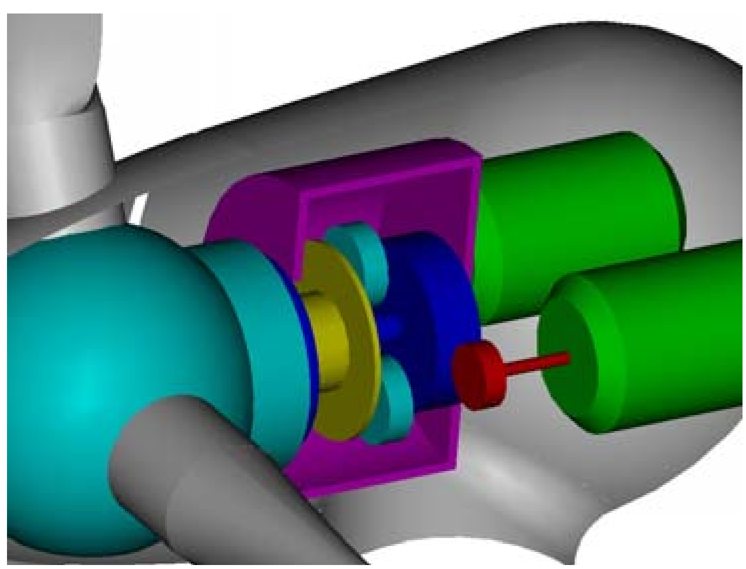


Figure 1. Multiple Generator Configuration

offer compelling advantages to COE. The conclusion, therefore, is that short of some other significant justification for using multiple generators (e.g., a significant change to the main frame which lends itself to multiple generators), GE could not justify moving in this direction on the basis of COE considerations.

2.1.2 Medium-Speed Generators and Single-Stage Gearboxes

A more significant departure from the baseline is represented by the use of a medium-speed generator coupled to a single-stage gearbox. This concept was originally proposed as an alternative to the direct-drive generator concepts discussed in Section 2.1.3. The direct-drive generators tend to be extremely large to the point of seeming almost impractical. On the other hand, the conventional arrangement of multi-stage gearbox and high-speed generator leads to a configuration in which the gearbox is substantially larger than the generator. Therefore,, it was speculated that a compromise between the two approaches employing a single gear stage and a medium-speed generator might provide for a more optimal matching of those two components. In fact, it was proposed that they could be integrated into a geared generator contained in one housing.

The concept was originally proposed in conjunction with an innovative new tubular nacelle concept in which the single gear stage and the generator could be integrated into the tubular nacelle, eliminating the gearbox and generator casings. This concept is discussed further in Section 2.1.3.3, but for the purposes of the present analysis, a more appropriate evaluation of this concept would result from considering stand-alone gearbox and generator components or a stand-alone geared generator.

2.1.2.1 <u>Medium-Speed Wound Rotor Induction Generator</u>

GE developed a design for an integrated, geared, medium-speed, wound rotor induction generator (Figure 2). Previous work on the direct-drive generator as reported in Section 2.1.3.2 showed it was not possible to design an induction generator operating at 20–35 rpm, the stator of which could be directly connected to the electric grid at 60 Hz, allowing for only partial power conversion. As a result, no great advantage exists for the induction design for direct-drive applications. For medium-speed applications, however, interest remains in demonstrating a wound-rotor induction machine that could be directly connected. As with the conventional-speed generators, such a machine would offer the advantages of variable speed operation with only partial power conversion, but with reduced gearing.

Several alternative generator/power electronics configurations were identified, including 32–, 40-, and 50–pole machines operating both sub- and supersynchronously as well as subsynchronously only. The study showed that the bidirectional systems (40zB and 50zB) could be constructed to cost no more than the conventional speed system. In both cases, the converters cost slightly more than the conventional converter, while the generators cost slightly less than the conventional system. The latter result was surprising. However, one consequence of the increase in pole count was a loss of approximately 2%–2.5% of system efficiency. This represents a significant loss in energy capture detracting strongly from any cost savings that might result from a reduction in gearing.

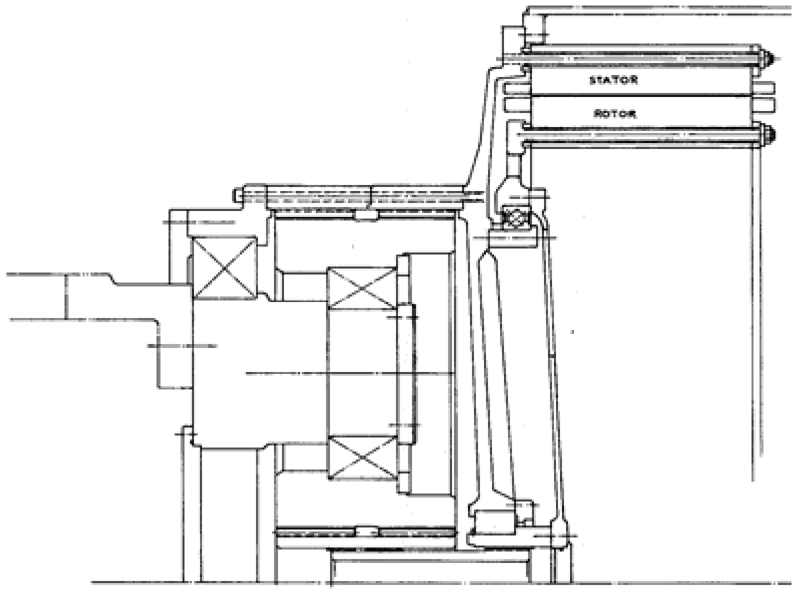


Figure 2. Concept layout of Single Stage Gearbox, Medium Speed Generator System

The initial concept planned for the unit to be integrated into a tubular nacelle, as described in Section 2.1.3.3. Therefore, some of the projected support structure cost savings resulted from that integration. Nevertheless, data shows that the stand-alone medium-speed generator should weigh and cost no more than a conventional generator, while it can be expected that simply integrating the single-stage gearbox with the medium-speed generator would result in a significant weight and cost savings for the gearbox, translating into a cost savings of nearly 2.5% of the installed turbine cost.

No estimate was made of the improvement in gearbox efficiency resulting from elimination of the second and third stages of the conventional gearbox, although it can be expected that anywhere from 1%–2% loss is associated with each stage. Therefore, it would not be unreasonable to assume that the loss of efficiency associated with the medium-speed generator is offset by a gain in efficiency on the gearing. The impact on the energy capture could be expected to be minimal.

Therefore, this concept could potentially result in some small reduction in COE. However, a relatively high amount of risk is believed to be associated with the concept, given that no medium-speed wound rotor induction generators are known to exist, and it would represent a substantially new technology.

2.1.2.2 Medium-Speed Squirrel Cage and Synchronous Generators

GE did not expend substantial effort in studying medium-speed squirrel cage induction, wound rotor synchronous, or permanent magnet synchronous generators. Based upon the results reported in Section 2.1.2.1, it is possible to design a medium-speed wound rotor induction generator similar in weight and cost to a conventional-speed machine. The NGT design team concluded that the same would likely be true for squirrel cage and synchronous machines. It can be expected that a small savings in generator cost would result.

Furthermore, the costs estimated for the power electronics for conventional-speed versions of these machines were very similar to the costs estimated for the power electronics for direct-drive versions of these machines. As a result, the NGT team further concluded that the cost for the power electronics for variable-speed operation of medium-speed squirrel cage and synchronous generators would also be similar to the costs for the direct-drive and conventional-speed versions. As noted in Section 2.1.1.2, these converter costs are substantially greater than power electronics costs for the wound rotor induction generator and more than offset any savings in generator cost that result from using squirrel cage or synchronous designs. As a result, it is safe to conclude that these configurations have no advantage over the wound rotor induction machine for variable speed systems.

As for fixed speed operation, the same conclusions reached in Section 2.1.1.3 apply to medium-speed machines as well. The cost savings from replacing the more expensive wound rotor induction generator with a synchronous machine and eliminating the power electronics is more than offset by the loss in energy capture. Furthermore, fixed speed machines are incapable of providing critical VAR support.

2.1.2.3 Multiple Medium-Speed Generators

The concept of using multiple stand-alone geared, medium-speed generators was not extensively investigated. However, drawing analogies from the results of Sections 2.1.1.4, 2.1.2.1, and 2.1.2.2, it is safe to conclude that this concept would hold no particular cost or performance advantage over using one large geared, medium-speed generator.

2.1.3 Direct-Drive Generators

Interest in direct-drive generators derives from the ability to completely eliminate the gearbox. The latter component is among the most expensive components in a wind turbine, and wind turbine gearboxes, in general, have been prone to require extensive maintenance industry wide.

2.1.3.1 Fixed-Speed Direct-Drive Machines

It would seem desirable to directly connect an electrical generator to the existing 60-Hz power grid. Although, it would have to operate as a fixed speed machine, sacrificing a portion of the input wind energy, power and frequency conversion with accompanying losses and costs could be avoided. But studies showed that direct connection forces unwieldy structure sizes for the shaft speed ranges that are typical for large horizontal axis wind turbines. A 30-rpm machine at 60-Hz drive must have a pole count of 240. For large pole count AC machines, the required physical length of a pole pitch at the stator air gap surface–lengths required by slot area and shape considerations–forces very large values of air gap diameters, in excess of 7 m. Machines of this size would clearly be unwieldy and very expensive to support and stiffen.

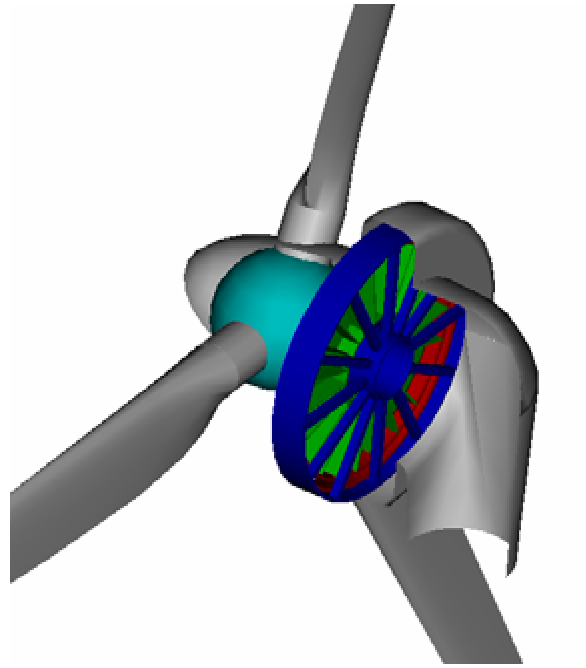


Figure 3. Direct-Drive Generator Wind Turbine System

The best machine GE designed was a wound field synchronous generator. Two aspects of this machine design are unacceptable: the excessively large air gap diameter necessary to obtain a reasonable pole pitch length and the low rated electrical efficiency of 84.5%. By replacing the field windings in the rotor with permanent magnets, one could increase the efficiency to very high levels, but the machine size would remain unchanged because of the number of poles and the 60 Hz nominal operating frequency. Most of the machine loss is in rotor resistance heating due to operation at the stator unity power factor, forcing high rotor currents necessary to push magnetic flux across the machine air gap at each of the 240 poles.

It is has been discovered that it is simply impossible to design a 30-rpm-direct-drive asynchronous machine that could be directly connected to the grid at 60 Hz. Iterations on air gap diameter and pole pitch length did not converge to a generator with reasonable power factor or efficiency.

As a result, fixed-speed operation of direct-drive machines without use of power conversion was impractical. The only means of solving this problem is to reduce the number of poles, which means that at the rated power shaft speeds typical of large wind turbines (18-35 rpm), the generator will produce power at frequencies much less than 60 Hz. This will require full power conversion. Once the converter is added, there is no longer an advantage to operating as a fixed speed machine. For the same reason, it is not possible to design a wound rotor asynchronous machine with a stator that can be directly connected to the grid at 60 Hz, thereby allowing variable speed operation with only partial power conversion. This method of operation, which is a key advantage of the baseline configuration, is simply not available to direct-drive generators.

2.1.3.2 Variable Speed Direct-Drive Machines

Both wound rotor and squirrel cage asynchronous machines were examined. One of the requirements imposed was that the variable-speed machines be able to provide dynamic VAR control. The advantages of this capability are discussed in Section 2.1.1.1. This requirement has an extremely negative effect on the squirrel cage machines. The squirrel cage induction generator cannot be designed for unity power factor operation. As a result, it must utilize a converter that can supply VARs to the stator. This results in a double cost penalty for these machines: the rating of the converter is higher because of the reactive power to be supplied, and the cost per kVA is higher because of the active rectifier.

The study showed that a direct-drive wound rotor generator can be expected to cost less than the combined cost of the gearbox and generator for the baseline turbine. However, the need for full power conversion increases the cost of the power electronics substantially, such that the estimated cost for the direct-drive generator and power electronics is significantly greater than the cost for the gearbox, generator, and converter for the baseline turbine.

Results indicated the wound rotor synchronous machine actually made the best candidate for a direct-drive generator. The rotor converter in the wound field synchronous machine concept is required to supply DC current to the rotor field windings. The wound field synchronous machine is a machine with a stator construction similar to an induction machine but with a rotor containing DC field windings that must be fed from an external supply. As a result, the wound rotor synchronous and asynchronous generators can be very comparable in construction and cost, while the power electronics for the synchronous machine will be less expensive. The total cost for a direct-drive wound rotor synchronous generator and power electronics is estimated to be slightly higher than the equivalent components in the baseline turbine.

Two different topologies for permanent magnet (PM) synchronous machines were examined:

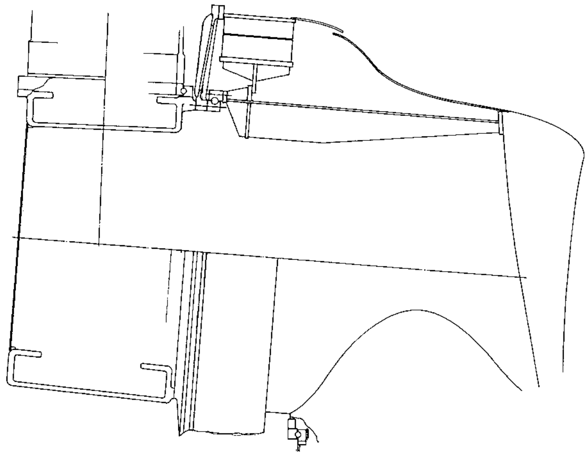
- Surface mount PM synchronous generator: This is an AC machine (sometimes, paradoxically, referred to as a brushless DC machine) with square-like current and voltage wave shapes rather than the sinusoidal wave shapes of the squirrel cage induction, doubly-fed induction, and wound field synchronous machines. The machine excitation is due to rotor magnets. Since there is no resistance heating loss for the machine excitation, the PM machine has the potential for very high operating efficiency. There is, however, a very real cost for the rare earth PM material.

- Doubly-salient PM synchronous generator is a PM synchronous machine type with the excitation permanent magnets mounted on the stator structure rather than in or on the rotor. The rotor of the DSPM machine is a very simple and rugged construction of stacked electrical steel laminations with protruding (salient) "poles". The stator windings of the DSPM machine are wound on salient poles of stator-laminated steel. As the rotor poles move across the faces of the stator poles, the stator PM excitation flux is switched from stator phase to stator phase producing pulsating flux variations within the stator phase windings and thus an AC excitation voltage.

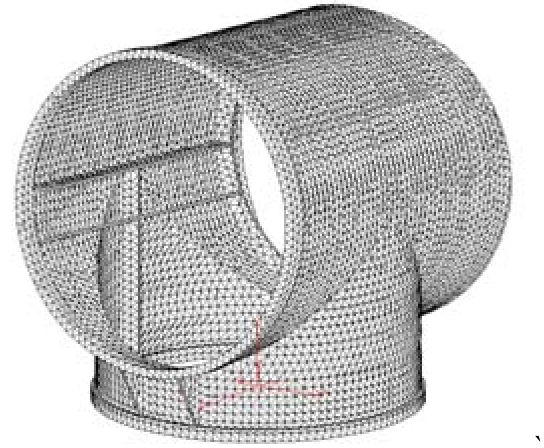
Extensive trade-off studies of stack length, active material weight, and rated efficiency as functions of air gap and air-gap diameter were conducted for the PM machines. As with the squirrel cage induction generator, the stator converter must supply reactive power or VARs to the DSPM machine. This requires that the machine-side portion of the stator converter must be an active rectifier, usually an IGBT bridge. The typical DSPM machine is not controlled through a standard three- phase power converter but requires a dedicated converter topology. As a result, the direct-drive PM synchronous configuration showed no advantages in COE for variable-speed operation.

2.1.3.3 Tubular Nacelle (External Monocoque)

GE studied the tubular nacelle structure shown in Figure 4, wherein a direct-drive generator would be supported directly by the nacelle structure itself. The nacelle could be constructed of sheets of steel welded together. Using this approach, substantial savings in the overall main frame cost would result for a direct-drive configuration. Although this concept was initially proposed in conjunction with the direct-drive generator concept, GE's concept evolved to consider it as a possible replacement for the bedplate main frame regardless of the drive train arrangement.



(a) Large Bearing Support



(b) Tubular Nacelle

Figure 4. Direct-Drive Support System

The motivation for using welded steel sheet is that the latter could be cut, rolled, and welded quite inexpensively, either in a factory setting or even in a field assembly facility. Minimal finish machining would be required, and the entire structure should save substantial money. Significant main frame weight reductions were predicted.

Extensive structural analysis of the tubular nacelle concept was conducted. This analysis was based around replacing the bedplate for the 1.5 MW POC turbine with the welded steel tubular nacelle. Internal structure was added in order to add rigidity to the cylindrical skin structure and to distribute the loads to the skin and the rest of the support structure in a reasonable fashion. An optimization of the tubular nacelle design was conducted in an attempt to minimize the amount of material used while maintaining structural integrity. Welded steel does not have particularly high fatigue strength, only 4.5 ksi, or 31 MPa. Figure 5 below illustrates the stress distribution for the final "optimized" design. Even this design, exhibiting areas of excessive fatigue (gray areas in Figure 5), weighs over twice what the existing bedplate weighs. As a result, this configuration is expected to cost more than the GE 1.5 MW bedplate and fiberglass nacelle. Further optimization of the design might be expected to shave more cost,, but not enough to provide a significant benefit. As shown in Figure 5, the distribution of material does not match the distribution of stress. A great amount of material above the centerline and behind the tower is very lightly stressed. This design was an inefficient structure for this application.

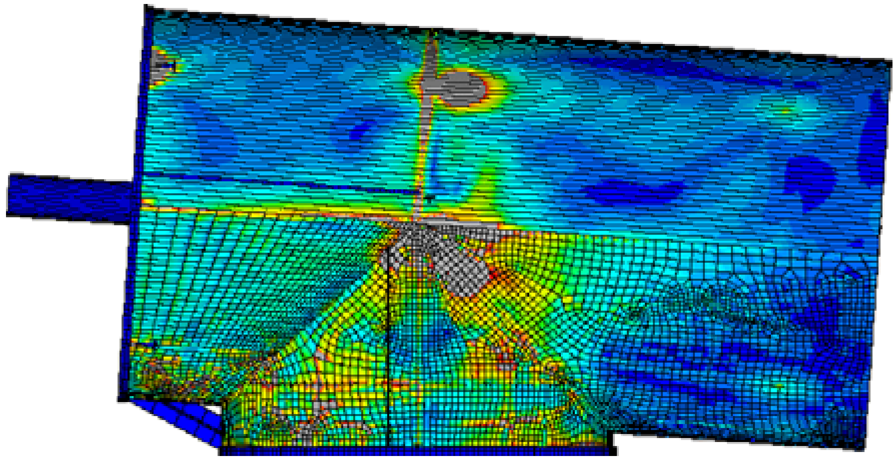


Figure 5. Fatigue Damage Equivalent von Mises Stress Distribution for "Optimized" Cylindrical Steel Nacelle

2.1.4 Summary of Drive Train Concepts

The result of all of the electromechanical concept studies is that no concept produces a significant improvement in the COE delivered by wind power. Only two concepts–the medium-speed wound rotor induction generator operating in VSCF mode and the medium-speed wound rotor synchronous generator operating in fixed-speed mode–showed any improvement at all in COE, and even then the estimated improvement was less than \$0.01/kWh, probably within the uncertainty range of these analyses. The remainder of the concepts produced no improvement or increase in COE.

As a result, GE decided that none of the concepts investigated showed potential for improving COE. The only one that has piqued some enduring interest is the medium-speed wound-rotor induction generator. This concept clearly allows for the elimination of all but the first stage of the gearbox, thereby guaranteeing some cost savings. However, the costs saved by eliminating the second and third stages of the gearbox are not overwhelming and may not justify the risk associated with development of a 40-pole generator.

Significant improvements in COE must be identified elsewhere, as noted in the subsequent sections.

2.2 Controls Concepts

Modern wind turbines such as those manufactured by GE feature electronic control systems that are relatively sophisticated compared to systems employed by wind turbine manufacturers only a decade ago. The GE baseline 750i turbine features both variable speed operation of the generator and active pitching of the blade for power regulation. Both systems are controlled by sophisticated power electronics.

The control system monitors the generator speed and torque. At speeds below rated, the pitch is generally maintained at fine pitch, and the generator torque is adjusted by controlling the current in the rotor winding in order to maintain turbine operation on the optimal speed-torque curve for maximum energy capture. At speeds above rated, the blade pitch is regulated to try to maintain constant shaft speed, while the torque is maintained at rated torque. This maintains constant power operation.

The baseline 750-kW turbine featured a collective pitch system through which the pitch settings of all three blades were substantially equal and controlled simultaneously. The collective pitch was effected mechanically through revolute joints on each blade connected to a hydraulically actuated cam. As part of GE's Near Term Research and Testing (NTRT) program jointly funded with NREL, this hydraulic/mechanical system was replaced on the Value Engineering (VE) turbine by a system of separate electromechanical drives on each blade. The pitch bearings on the VE turbine are geared on the inside. The pitch angle of each blade is controlled by a geared AC motor driving a pinion meshed with the bearing gear. At present, this system still features collective pitch, in that all three blades are maintained at substantially the same pitch setting at all times.

Despite the relative sophistication of this system, a number of innovative controls and monitoring concepts have been proposed during the NGT program, offering the potential for:

- Improving energy capture
- Reducing extreme or cyclic loadseducing fatigue damage accumulation, reactive maintenance, component failures, or other factors that result in reduced turbine life or operations and maintenance expenses.

The control strategies investigated in the present study are:

- Coupling blade pitch and generator torque control
- Tracking peak power coefficient
- Tracking tower vibration feedback control
- Tracking independent blade pitch control for asymmetric load control
- Tracking load-limiting control
- Tracking alternative yaw control strategies
- Tracking damage monitoring and feedback control
- Tracking drive train damping control.

Among the more intriguing benefits of advanced control concepts is that the cost of implementing them is frequently minimal compared to the overall turbine cost. For some of the control schemes, the cost consists only of reprogramming software. For some of the other schemes, implementation might require a slight upgrade of computational hardware or the addition of a few sensors. Therefore, the trade-off becomes less of a cost-benefit analysis than a comparison of benefits in some areas (e.g., load reduction) versus penalties in other areas (e.g., reduction in energy capture).

2.3 Rotor and Structural Design Concepts

The rotor arguably represents the most important component of a wind turbine. It is the component responsible for extracting energy from the wind, but it is also the component responsible for extracting loads from the wind and transmitting them to the rest of the turbine. Furthermore, the energy in the wind is unsteady and stochastic, and the rotor also becomes the principal conduit of dynamic loads experienced by the turbine. These loads induce fatigue and drive the design of most wind turbine components.

Therefore, a significant portion of GE's concept studies have focused on rotor design concepts that can significantly improve energy capture with a less than proportionate increase in dynamic loads and turbine cost. The challenge inherent in achieving this goal is that trying to extract more energy from the wind generally entails extracting more of the dynamic loads.

The most obvious and simplest design change that can affect energy capture is increasing the rotor size for a given drive train. There is no wind industry standard for the diameter of a rotor to be coupled with a generator of a given size. In recent years, however, all of the manufacturers of utility-scale turbines of roughly 500–1,650 kW rating have offered them with rotors corresponding to a loading of between 380 and 450 W/m^2. For reference purposes, this corresponds to rotors with a diameter between 38 and 41 m for a 500-kW turbine, between 46 and 50 m for a 750 kW turbine, and between 65 and 71 m for a 1,500-kW turbine.

Early in the concept studies, GE personnel began examining the possibility of reducing the rotor loading (i.e., increasing the diameter) to as low as 320 W/m^2, corresponding to a 55-m rotor on the baseline 750-kW turbine or a 77-m rotor on a 1.5-MW turbine. Such a design change is not trivial, however, for several reasons:

- For a pitch-regulated, variable speed wind turbine, increasing the rotor diameter will only increase the energy capture at wind speeds below rated. Therefore, a 20% increase in rotor swept area may result in a 20% increase in power production at wind speeds below rated; but it will result in less than a 20% increase in annual energy production, probably nearer a 8%–12% increase, depending upon the wind characteristics at the site.

- Acoustic emissions considerations may require that the rated shaft speed be reduced proportionately to the rotor diameter increase, in order to avoid increasing the rotor tip speed, which strongly affects aerodynamic noise. This will cause a proportionate increase in the low-speed shaft torque with concomitant effects on several major components, including the main shaft, gearbox, and bedplate.

- Increasing the rotor diameter increases the loads more than the energy capture. The power production is obviously proportional to the swept area, which increases as the square of the rotor diameter. If the shaft speed-vs-wind speed schedule remains unchanged up to rated power, however, then the tip speed increases proportionately with the rotor diameter, and the aerodynamic pressure at the tip increases roughly as the square of the tip speed. Assuming a roughly constant-chord section is added to the tip, the area over which the pressure acts is also proportional to the diameter, so the overall loading increases as the cube of the rotor diameter. If the entire blade is scaled, including the chord length, the loading scales as the fourth power of the diameter. The thrust loading and in-plane loading induce strong moments on the tower base and drive the design of the tower. Furthermore, the moment arm from the aerodynamic loading to the blade root is also proportional to the increase in diameter, so the increase in blade root bending moment scales as the fourth to fifth power of the rotor diameter. These loads drive the design of the blades, hub, and drive train mechanical and structural components. If the rated power of the turbine does not change, the mean loads on the system are not increased. However, every wind gust that strikes the turbine when it is operating at rated power is converted to a dynamic load on the turbine without producing any additional power. This can result in a substantial increase in fatigue on mechanical and structural components. Therefore, unless the rotor diameter increase is affected in an intelligent manner, it is possible that increasing the rotor diameter will impact the cost of fatigue-driven components in such a way that the turbine cost is increased more than the energy capture.

- Average wind speed and turbulence intensity strongly affect the importance of increased rotor diameter. For sites with strong winds or high turbulence, the dynamic loads experienced at and above rated wind speed can become a bigger design driver than additional energy capture below rated. The use of larger rotors is largely a consideration for wind farm sites with relatively low mean speeds and relatively low turbulence, where the turbine operates less often at and above rated wind speeds and is less likely to experience the strong wind gusts which induce fatigue damage.

The subsequent parts of this section address several alternative concepts investigated by GE for significantly increasing energy capture without proportionately increasing the loads or turbine cost. GE also investigated several alternative concepts that attempt to reduce rotor cost without impacting energy capture. These concepts include:

- Rotor and other turbine structural flexibility
- Concurrent aerodynamic and structural design optimization
- Carbon composite rotor blades
- Aeroelastic tailoring of rotor blades
- Rotor blade aerodynamic boundary layer control
- Variable diameter rotor.

2.3.1 Rotor and Structural Flexibility

The main aim in exploring the implications of rotor structural flexibility is that, while some steady state loads cannot be altered, dynamic loads on the wind turbine system are often design drivers and may be reduced with the energy content of the input wind loading in various frequency ranges partly dissipated by aerodynamic damping of blade motions. Hence, reduced loading is passed further into the wind turbine system. The other aspect of structural flexibility that is specific to a rotor system, as opposed a static cantilevered beam, is the effect of centrifugal force in providing relief of blade out-of-plane bending moments, both steady and dynamic. Reducing the stiffness of a tubular tower may also allow for a reduction in tower material, and introducing damped compliance into the machine carrier yaw and pitch (i.e., nodding) degrees of freedom may similarly reduce the absorption by the bedplate, yaw deck, or tower top of dynamic loading imparted by the rotor. It is likely that tower and blade flexibility may be cheaper and easier to engineer than yaw deck compliance and may even be a source of mass and cost reduction of blades and tower rather than an added cost to be paid for by other benefits. All concepts are investigated here.

The reduction in loads that might result from increased structural flexibility could result in benefits to COE either as benefits to component designs or through increased energy capture by enabling the use of a larger rotor with loads no higher than would be produced by a smaller, stiffer rotor.

2.3.1.1 *Upwind Rotor Blade and Tower Flexibility*

Two different studies were commissioned by GE to investigate the impact of reducing the wind turbine rotor blade stiffness. Both studies were conducted using the baseline 750i turbine as the study control. The studies were conducted in slightly different ways, as explained below.

In the first study, a model of the baseline 750i turbine using the baseline 48-m rotor was developed. A new "softer" blade was modeled with stiffness equal to 50% of the baseline value and mass reduced to roughly 70% of the baseline. Though softer, the blade is still stiff (above 2p). In addition to the baseline tower, a so-called "soft-soft" tubular tower with a first natural frequency less than the turbine main shaft maximum rotational rate was also modeled. Two towers and two rotor configurations yield four configurations for study.

Figure 6 shows the impact on several key loads resulting from the softening of the tower and rotor blades. All load increases and decreases are measured relative to the baseline configuration. The softening of the tower alone, Configuration 3, reduces all loads except tower top thrust when m=10 and blade edgewise bending moments for both m=4 and 10. Softening of the rotor alone, Configuration 2, reduces all loads except for a slight increase in blade flapwise bending moment and a fairly substantial increase in tower top thrust.

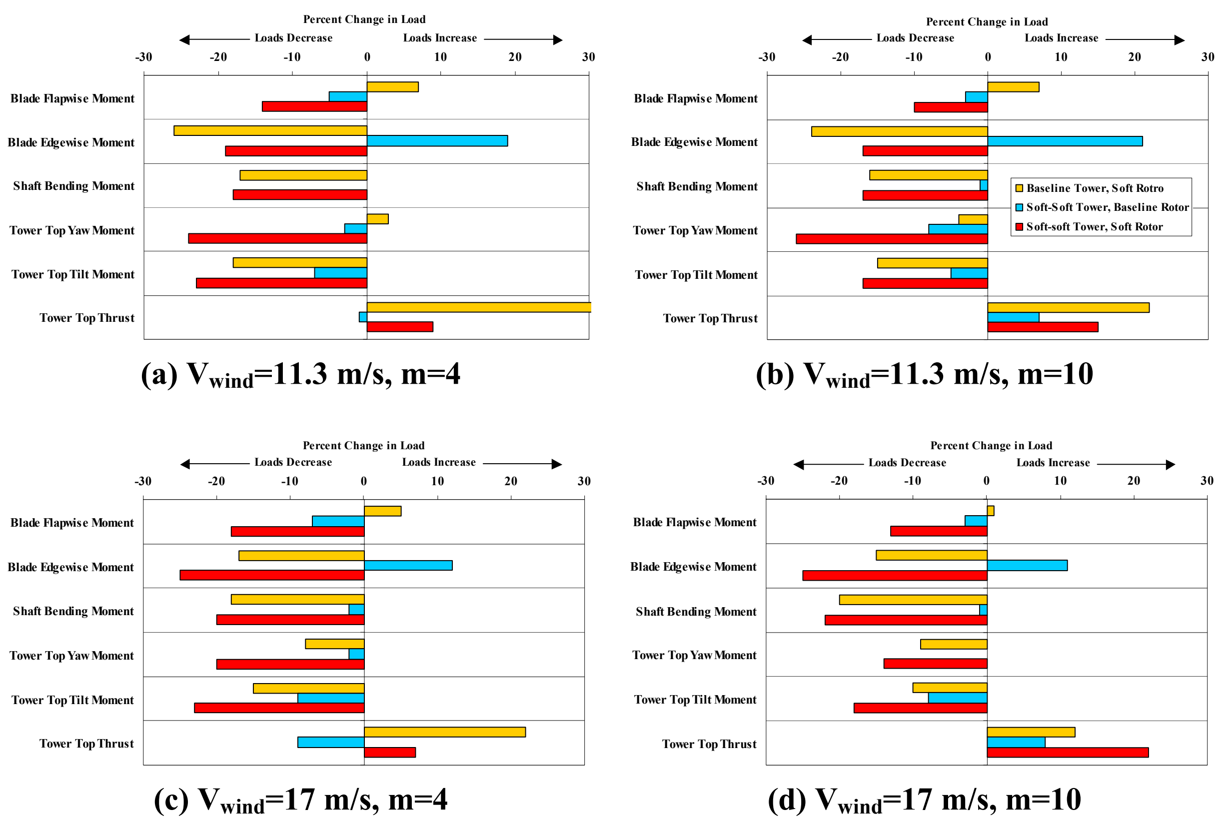


Figure 6. Effect on Fatigue from Reducing Rotor Blade and Tower Stiffness

Simulations of the Baseline 750i Turbine

Finally, softening both the tower and the rotor blades, Configuration 4, results in a fairly substantial reduction of all loads except for tower top thrust. It is predicted to have an 18% reduction in damage equivalent flap moment, 25% reduction in edge moment, 20% reduction in yaw moment, 23% reduction in tilt moment, and 20% reduction in hub moment (M_y) in 17 m/s winds. The reduction in edge moment corresponds approximately to the reduction in blade weight. There does not appear to be any reduction or amplification due to the reduced natural frequency.

The only predicted increase is in tower-top thrust (7%). This is attributed to tower motion interaction with the pitch control system and predicted that this increase could be reduced or eliminated using nacelle acceleration feedback to the controller. It is interesting to note that the thrust (m=4) and tower clearance are worse for configuration 2 than for configuration 4. The soft-soft tower provides some mitigation of the negative aspects of the soft blade. Use of the soft blade with the baseline tower (configuration 2) is not recommended. This configuration is predicted to have less benefit and a greater reduction in tower clearance than configuration 4.

The blade tip/tower clearance is reduced by up to 25% for Configuration 4. This is the largest challenge of the softer configurations. This problem can be reduced with little apparent negative consequence by further increasing the precone angle. The energy yield is predicted to decrease by 4%. Configuration 2 (with the same precone angle as configuration 4) has less energy loss at 11.3 m/s than configuration 4. This means that not all of the energy loss is due to the larger precone angle of the softer blade. Based upon projected area alone, the expected energy loss due to a 2° increase in preconing would be about 0.3%. Therefore, most of the loss is attributed to the poorer speed regulation with the softer systems. Further "tuning" of the control system can eliminate much of this loss.

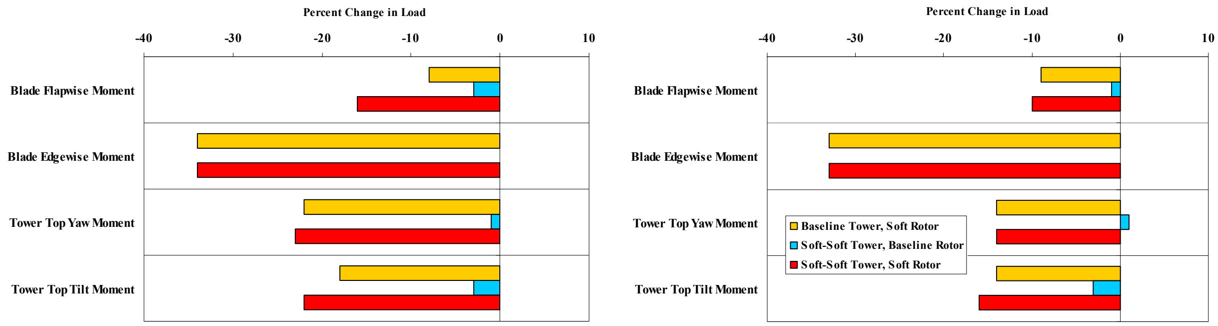
This study demonstrated clear benefits in terms of reducing blade mass and loads in general, but the study did not quantify in terms of impact on COE any of these benefits or costs such as energy loss. The second study was similar, but in addition to examining the impact of rotor flexibility on loads, this study also addressed several practical considerations in more detail:

- The relationship between mass and stiffness
- Tower clearance with more flexible rotor blades
- Impact on COE
- Engineering implementation issues, including scaling concerns.

In this study, the blade stiffness was halved, and the blade mass was reduced approximately 30%. A lifetime fatigue analysis and an extreme load calculation were performed. The extreme loads analyses yields approximately equal maximum blade deflection. The extreme blade deflection for the baseline design is 1.5 m. For the softened blade it is about 3.0 m. The principal challenge of reducing blade stiffness with an upwind rotor configuration is to provide a safe tower clearance to accommodate extreme blade deflections. The main options for increasing tower clearance are:

- Increased rotor overhang
- Increased rotor tilt
- Increased blade coning
- Introduction of blade curvature (out-of-plane).

The introduction of a moderate amount of tilt (up to 10°) will add cyclic loads that are small in comparison to the stochastic loads induced by turbulence. Simulations have been carried out with 8° of shaft uptilt and 3° of forward precone. This configuration provided for a maximum tip deflection substantially equal to that of the baseline configuration. The results are summarized in Figure 7, which can be compared directly to the results shown in Figure 6, which were obtained for similar cases with slightly different values of uptilt and precone. Most of the fatigue reduction is quite comparable, but the increases seen in the first study in a few components of fatigue have been eliminated by careful tuning of the controller to avoid tower/blade resonance. Overall, the results mentioned above are confirmed, in as much as substantial reductions of critical components of fatigue can be realized through relaxing the rotor blade stiffness.



(a) m=4 **(b) m=10**

Figure 7. Effect on Fatigue from Reducing Rotor Blade and Tower Stiffness Simulations of the Baseline 750i Turbine

The cumulative effects of softening both the tower and the rotor do not differ substantially from softening simply the rotor with the exception of possible benefits to blade flapwise bending. This is a somewhat different result than what was obtained in the first study, which showed substantial improvements in softening the tower in conjunction with a softer rotor, and is presumably attributable to the differences between the controllers used in the two simulations. Softening the tower in conjunction with softening the rotor, however, does not detract the benefits of rotor softness. Furthermore, softening the tower allows for a substantial reduction of tower mass, a critical conclusion which served as one of the design drivers for the towers for the POC and EMD turbines.

The COE impacts of implementing a more flexible rotor were also examined. The reduction in system fatigue loads that results from increased blade flexibility may be taken as a benefit to component design. Alternatively, a longer blade may be considered in order to increase energy capture while using structural flexibility to keep within the design loads of the baseline machine. The greatest reduction in COE using the baseline rotor diameter is about 3%. An extended, tilted rotor (50.77 m) is the most economic option, producing a 5% reduction in COE.

Four main conclusions can be drawn:

- Extending the blade to recover energy loss from tilting and/or coning options is always preferable for COE.
- Providing increased blade tip clearance solely by increasing overhang is not economical.
- Providing increased blade tip clearance by means of increased rotor tilt and compensating for energy loss by extending the blade is the most economical solution.
- Exploiting fatigue load reduction by increasing rotor size is more economical than taking benefit to components at standard rotor size.

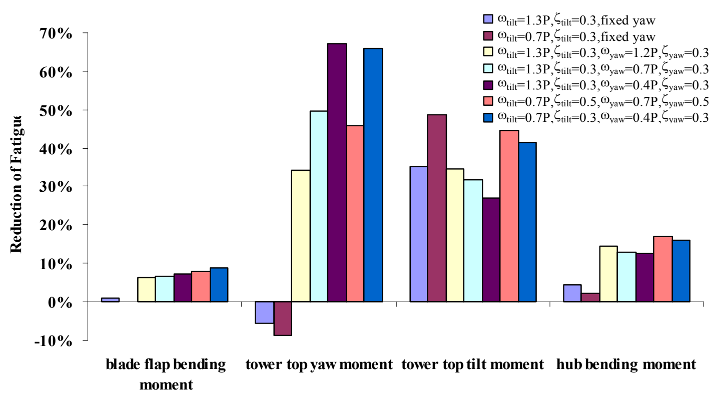
The latter is of critical importance in evaluating candidate designs for the POC and EMD turbines, as discussed in Sections 3 and 4. The conclusion that increased rotor flexibility should be exploited by increasing rotor size was instrumental in driving the preliminary design of the POC turbine towards a larger, more flexible rotor.

2.3.1.2 *Machine Carrier Yaw and Nodding Compliance*

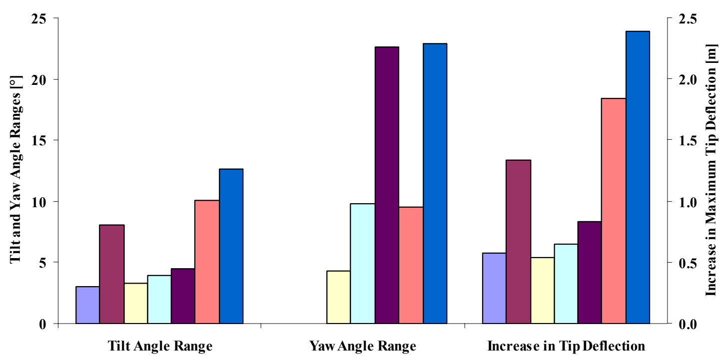
GE commissioned two studies to investigate the impact on loads from introducing compliance about the drive train tilt (or nodding) and yaw axes. Both studies were conducted using the baseline 750i turbine as the study control. The studies were conducted in slightly different ways.

Figure 8 illustrates the results of introducing yaw and nodding compliance as determined by the first study for different values of natural frequency and damping for each degree of freedom. Flap bending moment, tower top yawing moment, and hub (shaft) bending moment are all most strongly affected by the addition of yaw compliance, with the nodding, fixed yaw cases providing little or no load relief. Tower top yaw moments are further reduced by increased softening of the yaw degree of freedom (i.e., reduced natural frequency), but blade flap bending moments and hub bending moments are not otherwise appreciably affected by the magnitude of yaw compliance. The magnitude of tilt compliance has little effect on these three components of fatigue. The tower top tilt moment is obviously most strongly affected by the softness of the tilt degree of freedom, with increased softening of the yaw degree of freedom actually offsetting some of the loads improvement.

Configurations with yaw frequency of at least 0.7P are able to keep the yaw angle range within 10°, which was a stated goal. The Configurations with tilt natural frequency of 1.3P were able to keep the tilt angle within the desired range of ±2.5° and minimized the increase in maximum tip deflection. The stiffness is considerably higher than the value originally sought for the best load isolation. A larger yaw damper also helps to limit tilt motion. For a given tilt natural frequency, the tilt motion increases as the yaw natural frequency drops. The simulations do show an interaction between tilt and yaw motion. After examining all of these simulation results, it was concluded that the stiffest case (i.e., ω_{tilt}=1.3P,ω_{yaw}=1.2P showed the best combination of load reduction with minimal yaw and tilt motion. This case is seen as the yellow bars in Figure 8. Fatigue is reduced by 34% for yaw moment, 35% for tilt moment, 6% for blade flap moment, and 14% for low-speed shaft bending. The tower clearance is reduced by 0.54m.



(a) Reduction of Fatigue



(b) Tilt &Yaw Angles & Tip Deflection

Figure 8. Impact of Yaw and Nodding Compliance on Fatigue & Motion

As a progression from this study, a more detailed analysis of tower top flexibility was completed. This study introduced a drive train nodding degree of freedom to the flexible blade configurations studied above.

From Figure 9, it is evident that introducing nodding freedom into the tower top has only a small and mixed effect on most of the components for fatigue on the baseline configuration as well as the other three combinations of rotor and tower flexibility. The only component of fatigue that strongly benefits from the introduction of nodding compliance is the tower top tilt moment. Energy capture was negligibly affected by the introduction of nacelle nodding. While the maximum blade tip deflection was increased significantly by introduction of nacelle nodding, the blade tip-to-tower clearance was not strongly affected.

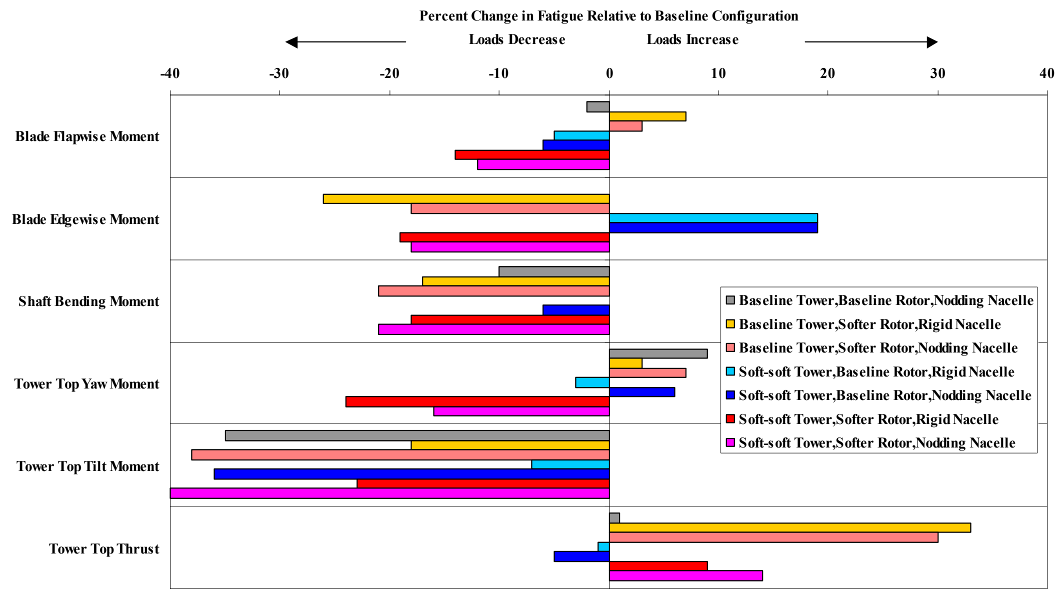


Figure 9. Impact of Nodding Nacelle on Fatigue
As Determined through Simulations, m=4, V_{avg}=11.3 m/s

The second study also built upon the earlier second study of rotor and tower flexibility to add nodding and yaw compliance. This study introduces a drive train nodding degree of freedom and a yawing compliance degree of freedom. Two different configurations of nodding and compliance were introduced:

- **Case a:** The nodding hinge is introduced at the center of gravity (c.g.) of the baseline nacelle plus rotor. The hinge stiffness is set to provide a natural frequency in nodding of ω_{tilt}=1.3P and a damping ratio of ζ_{tilt}=30%. The yaw compliance is set to provide a natural frequency of ω_{yaw}=1.2P and a damping ratio of ζ_{yaw}=30%.
- **Case b:** The nacelle center of gravity is adjusted to lie on the tower centerline. A natural frequency of ω_{tilt}= ω_{yaw}=0.7P and a damping ratio of ζ_{tilt}= ζ_{yaw}=30% is applied to both nodding and yaw rotations.

The results illustrated in Figure 10 are quite similar to those shown in Figure 8 and Figure 9; namely, the impact on fatigue from adding tower top compliance to any of the other four configurations is actually quite minor except for the tower top tilt and yaw components. The most significant reductions in blade flapwise bending fatigue come from addition of rotor and tower softness, with the addition of tower top compliance that add little additional benefits. The most significant reductions in blade edgewise bending fatigue derive from the softening of the rotor, with the addition of tower top compliance actually increasing fatigue for the baseline rotor and offsetting some benefits of the soft rotor.

While it is evident that the softest tower top system provides the greatest fatigue load alleviation there, the results show that the differential benefit in fatigue load reduction of Case b compared to the stiffer hinge/yaw system of Case a is not great. The angular range of nodding motion of

Case b is about twice that of Case a. It may not be worth engineering a system to cope with the large motions of Case b when most of the benefit can be provided with the Case a system. This view is reinforced when the effect of nodding rotations on tower clearance is considered. The results show that the Case a configurations are marginal for tower clearance (probably inadequate for lifetime extreme tip deflections), and that the Case b hinge is unrealistic for an upwind machine configuration.

Overall, there is some cumulative benefit in having a soft tower, soft blade, and soft tower top connection but it is much less than simply additive, and it seems necessary to have a soft blade to prevent increase of blade loading because of tower head flexibility. The benefits from reductions in tower top tilt and yaw fatigue are questionable, as the only wind system component driven by these components of fatigue is the bedplate-tower interface, which will increase in cost and complexity anyway, in order to introduce the compliance. A brief engineering analysis of the mechanical requirements for implementing the tower top compliance was conducted. The conclusions show the Case a system added to the baseline 750i turbine featuring a soft-soft tower and a softer rotor and would cost approximately the same as the yaw drive system, which is the only clear cost savings implemented from yaw or tilt compliance. There is no benefit to blades, and unless a substantial benefit to tower costs, nacelle structure, or drive train can be established, it will be hard to justify the costs.

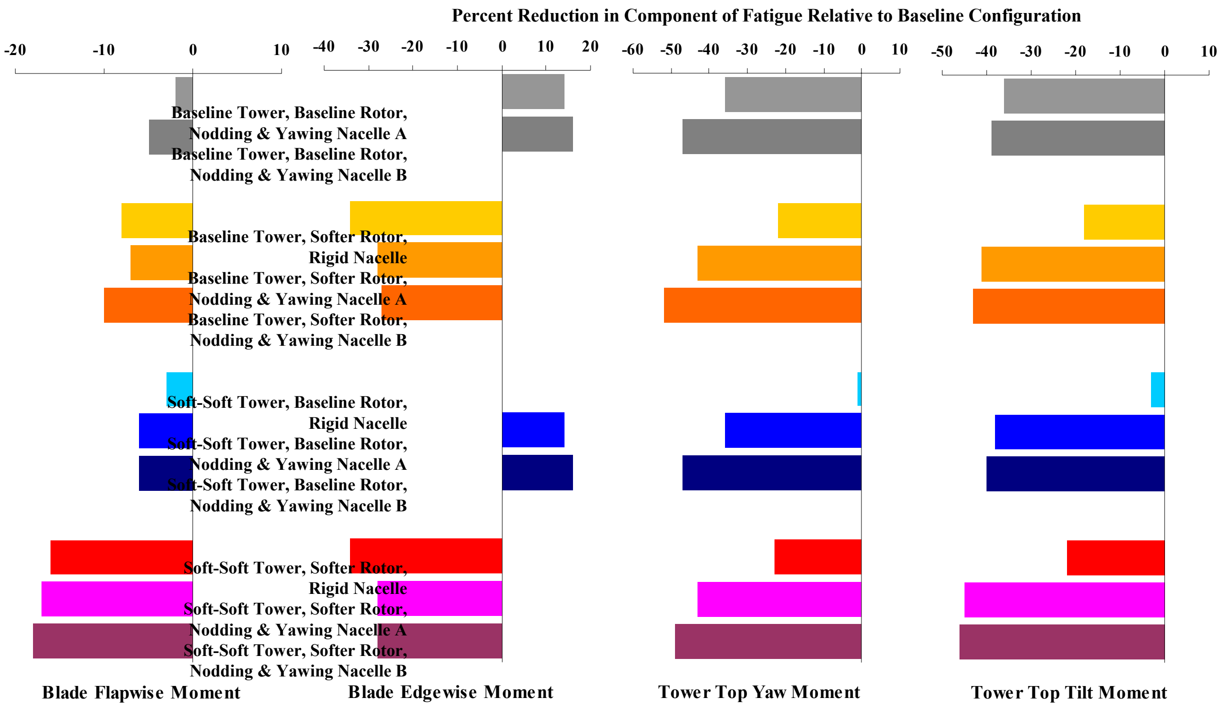


Figure 10. Impact of Rotor, Tower, and Tower Top Flexibility on Fatigueas Determined through Simulations

2.3.1.3 *Flexible Downwind Rotor Concepts*

Wind turbines featuring downwind rotors with flexible blades have been designed, built, and tested by a number of researchers and turbine manufacturers in the past, with varying degrees of success. Some were technically unsuccessful, and none have proved commercially successful. The reasons are varied. Some were probably overly ambitious to the degree of flexibility attempted. Some may have been affected by the number of technical innovations introduced simultaneously in the turbine design, which challenged the ability of engineering modeling and analysis at the time. Inadequate modeling led to unforeseen dynamic problems during testing and proved insurmountable.

From this experience, the utility-scale wind industry has, during the course of nearly a quarter-century of progress, converged on the three-bladed, upwind rotor configuration. Every utility-scale wind turbine currently manufactured either in the U.S. or in Europe, features this design. As noted at the beginning of Section 2, GE believes there are market advantages to the three-bladed upwind rotor configuration and decided, from the outset of the NGT project, that the NGT design should favor this concept.

Nevertheless, a downwind rotor featuring flexible rotor blades represents one means of overcoming the most significant challenge to implementing more flexible rotor blades in an upwind rotor configuration–accommodating blade tip tower clearance requirements. In order to halve the stiffness of the blade in an upwind rotor arrangement, approximately 2.5% of energy capture is sacrificed by incorporating forward coning sufficient to ensure adequate tip tower clearance, while approximately 1.75% is sacrificed through use of uptilt. These represent fairly significant losses that offset a substantial portion of the COE savings resulting from the use of more flexible rotor blades.

GE decided to commission a study of the downwind rotor concept to determine whether introducing flexible rotor blades downwind could produce savings in COE sufficient to justify further pursuit of this concept. The study was quite broad, examining a number of variations on the downwind flexible rotor concept, including:

- Degree of rotor blade stiffness
- Number of blades
- Hub type (i.e., rigid, teetered, or hinged/flapping)
- Tower height and type (free standing vs. guyed)
- Rotor solidity and tip speed.

Various rotor configurations were examined. The 3-bladed, upwind, rigid hub configuration was used as a baseline and the other configurations were compared to that case. The "hinged" hub configuration modeled blades independently hinged flapwise at their roots. The "soft" blade had a flapwise stiffness of 0.5 and a mass of 0.67 of the corresponding regular blade properties. The same blade was used on both the two- and the three-bladed rotors. In order to make the rotors equivalent and to have the same performance curve, the rotor speed of the three-bladed version was reduced from 42 rpm to 35.5 rpm.

To obtain information on both the peak loads and the fatigue loads, the study concentrated on two operating conditions: normal operation in a "standard" turbulence with a mean wind speed of 18 m/s, and the extreme coherent gust with direction change (ECD) at rated power. The simulations of 10-minute records of normal operation were carried out four times, each with a different random seed.

Design rules were used to estimate the effect of different blade designs on weight and cost. The performance and annual energy production of a number of different blade and rotor designs were also investigated. The most promising way in which the rotor might be redesigned to reduce the COE, was to sweep an area as large as possible with as little blade as possible, while minimizing the impact on drive train torque. This implies that the blade must move faster, which leads to a higher rotor shaft speed that also reduces the shaft torque and the gearbox cost. All of the modified rotors had larger diameters than the baseline and also had lower solidities. The maximum tip speed was close to 90 m/s, considerably higher than for most current rotors. The relative effects of changes in loads and rotor configuration on blade weight and cost were examined.

The impact on COE is summarized in Figure 11. The minimum COE for each configuration shown in Figure 11 does not necessarily occur at the same rotor size or turbine rating for each configuration, and so the COE values shown reflect variously sized turbines. The order in which the results are presented in Figure 11 were selected to reflect the level of perceived technical risk associated with the respective configurations. Some of the configurations use technologies that are speculative, such as the hinged blades. Those architectures have been placed at the end of the list.

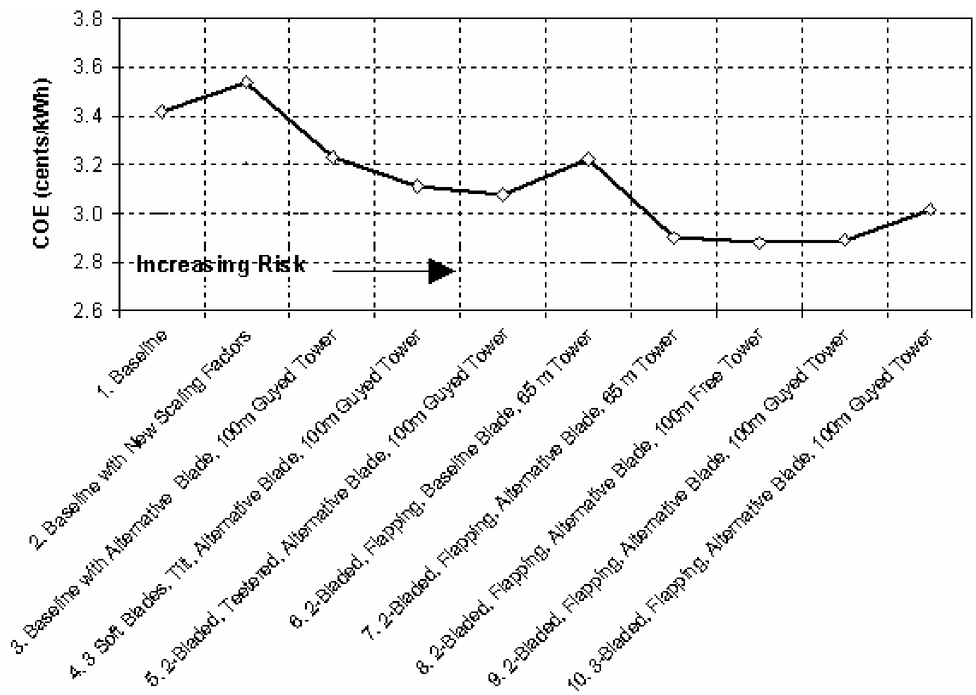


Figure 11. Summary of Downwind Configurations

In the end, GE concluded that a slightly more conventional version of Option 4, using an upwind configuration and conventional free-standing tower, could achieve a COE nearly as good as that reported for Option 4 with less risk. Only Options 7 through 9 offer noticeable improvements in COE, and those improvements (on the order of 10% relative to Option 4) do not justify the risk entailed. Similar savings can be found elsewhere with less risk.

2.3.2 Concurrent Aerodynamic and Structural Design Optimization

The design of a wind turbine rotor blade is generally accepted to be a compromise between aerodynamic performance and structural integrity. Aerodynamic considerations relate to maximizing energy capture and drive the designer towards the use of relatively thin sections with high lift-to-drag ratios. Satisfying structural requirements relating to static strength, fatigue resistance, buckling, and tip deflection (i.e., avoiding tower strikes) while minimizing material usage suggest the use of thick, structurally efficient sections. It is frequently possible to introduce thicker sections inboard for structural considerations without seriously impacting energy capture. It may be similarly possible to relax structural considerations outboard in order to maximize energy capture. This is especially true if more blade flexibility is permitted, or even desired, as a result of the concept studies discussed in Section 2.3.1. Furthermore, entirely new approaches to blade optimization may be possible, and were accomplished with alternative materials such as carbon fiber reinforced plastic.

Developing an optimal compromise between these criteria is a serious design undertaking and one which GE approached rigorously in the development of the POC rotor blades, as discussed further in Section 3. To provide guidance in the POC design, GE commissioned two different studies of methods for optimizing rotor blade design considering aerodynamic and structural requirements, concurrently.

The first study focused on developing not only an aerodynamically optimum blade but also "planform envelopes" of greater and lesser solidity than the optimum planform, which show the variations in planform allowable within a certain specified energy sacrifice. The results showed that a structural designer can contemplate a much wider range of potential designs than had been previously thought possible and may therefore be able to achieve a greater degree of structural optimization and blade cost saving.

The second study focused on the influence of airfoil section thickness on the performance and mass of rotor blades. While this study did not touch on many of the issues addressed by the first study, it did provide a more realistic assessment of the ability to simultaneously optimize aerodynamic and structural considerations dependent on airfoil section thickness. The study involved development of a roughly designed but customized family of airfoils with thickness-to-chord ratios of 16, 18, 21, 24, 27, and 30 percent. The design objectives for these airfoils were:

- Design lift coefficient between 1.1 and 1.2 at the Reynolds numbers typical of the baseline turbine rotor blades
- Limited upper surface laminar flow in order to minimize sensitivity to leading edge surface roughness. This criterion is in sharp contrast to the NREL families of airfoils, which are specifically designed to produce extensive laminar flow.

Aerodynamically optimum rotor blades employing these airfoils in various distributions from hub to tip were then designed. The objective for the blade designs was to maximize gross annual energy production (GAEP), which was based on a Rayleigh wind speed distribution with an average wind speed of 8.5 m/s (IEC Class II wind regime). The first step in the blade design process was to identify the optimum lift distribution to be prescribed. To quantify the effect of the lift distribution on gross annual energy production and rotor thrust loads, a sensitivity study was performed using the baseline rotor design. In this sensitivity study, constant distributions of C_l were prescribed. The results showed a variation of at most 0.1% in GAEP, meaning that there is a relatively wide range of C_l yielding similar performance in terms of energy output and thrust loads.

Several key conclusions of this study are:

- Significant improvements over the baseline design are possible.
- Use of higher thickness-to-chord ratio section does not automatically result in excessive energy loss.
- Excessive thickening of the tip should be avoided.
- If one expects surface-roughened conditions, then the blades should be optimized using the rough-surface (i.e., fixed transition) aerodynamic characteristics.

These results were encouraging that it might be possible to concurrently optimize the rotor aerodynamic and structural performance by thickening inboard stations while leaving outboard stations thinner. With this in mind, GE studied the benefit or detriment to blade mass using the same thickness and chord distributions. They assumed the baseline turbine rotor blade tip deflection limitation and also a relaxed deflection limitation (i.e., twice the baseline deflection). The results of the structural analysis were compiled along with the aerodynamic analysis to reflect the combined impact on COE resulting from both the aerodynamic and structural changes.

The final conclusion is that all of the changes have a relatively minor impact on overall COE.. Changes are in the ±1% range for the baseline tip deflection and under 2% reduction for the relaxed tip deflection. The best improvement to COE can be realized by using softer blades derived from increasing the rotor diameter and energy capture rather than from trying to realize significant savings in blade cost. Section 3 describes the strong influence the conclusions of this section had on the design of the POC turbine.

2.3.3 Aeroelastic Tailoring of Rotor Blades

GE was motivated to determine whether composite rotor blades can be designed and cost-effectively manufactured in such a way that they provide an aeroelastic response tailored to ameliorate fatigue-inducing transient loads and concomitant extreme blade deflections. Traditionally, aeroelastic response–that is, the tendency of a structure to exhibit coupling between aerodynamic loading and structural behavior was something the designer, whether aerospace, civil, or mechanical, had to accept with design considerations. In the past three decades, the advent of composite materials have allowed for tailoring of aeroelastic properties through deliberate design of desired flap-twist coupling. Off-axis (e.g., 20°) orientation of reinforcement fibers (e.g., glass, carbon, Kevlar, etc.,) along a supporting spar will cause the spar to twist under sufficient bending strain. This can be used to tailor the bend-twist coupling of an aircraft wing or a wind turbine blade.

GE's pitch-regulated wind turbines maintain constant power above rated wind speed by constantly pitching the blades in response to wind speed gustiness. When the baseline turbine is operating at wind speeds just less than or at rated wind speed. it is most vulnerable to rapidly rising wind gusts in terms of their ability to induce damaging transient loads and blade deflections. The blade pitch is at a fine setting to optimize the power factor, and so any gust that hits the turbine is seen as a sudden and significant increase in angle of attack along the entire blade span. Since for optimal power factor, the blade design and pitch are set so each station is operating near the angle of attack for maximum lift-to-drag ratio, any increase in angle of attack results in increased lift. The change in lift combined with the change in inflow angle results in a sudden increase in both in-plane and out-of-plane loading on the blade. Such transient loads induced by wind gusts combined with the response of the turbine are the most significant contributions to structural fatigue on wind turbines.

Presently, the pitch control system responds to increased rotor torque by pitching the blades to smaller angles of attack, thereby reducing the blade loading and torque. As the wind subsides, the pitch is returned to normal. Nevertheless, the pitch system can respond only so quickly, particularly when it must transition from Region 2 to Region 3 of the power curve. As a result, the wind turbine will still realize significant transient increases in loads before the pitch system can counteract them.

An aeroelastically tailored blade could reduce such transient loads. As the gust would induce flapwise deflections, the tailored bend-twist coupling of the blade would passively induce a twisting of the blade towards reduced angles of attack, thereby relieving the loads and reducing pitch activity. This will prevent the loads and hence the deflection from rising to the levels they would reach if the blade were not twist-bend coupled. Although it might be desirable to have an entirely passively controlled blade, GE has never believed this was achievable, nor that it was the goal of research for the concept. The reason is because of a simple but unavoidable quandary–the blade can only reduce loads by twisting into the wind, and it will only twist more if it flaps more, and it only flaps more if it is subjected to higher loading. It is simply not possible to shed all transient loads passively. But trimming the amplitude of transient loads and reducing pitch activity are both extremely desirable goals for twist-bend coupling.

Introduction of twist-bend coupling is not without cost, however. First, twist-bend coupling can cost some energy capture if not implemented carefully. If a turbine is operating at a mean wind speed near rated, the blades will be subject to some relatively high mean loading that will produce an average flapwise deflection of the blades and will in turn induce some coupled twist distribution along the blade span. If a blade originally designed without twist-bend coupling,(with a chord length and twist distribution optimized for energy capture) is altered to introduce twist-bend coupling (without changing the unloaded twist distribution), then at rated wind speed, the blade's twist distribution will no longer be optimal due to flap-induced twisting. This will cost energy capture. Energy minimization can be partially demonstrated by designing the blade to achieve its optimal twist distribution at a wind speed at or near rated, such that the blade operates off the optimal twist distribution at predominately lower wind speeds.

Second, introducing diagonal plies into the blade construction might be challenging and more expensive. The blades of the baseline turbine feature unidirectional and ±45° bias plies. The latter are substantially more expensive than the former, but introducing material such as ±20° plies for twist-bend coupling could be even more expensive.

Third, introducing off-axis plies could soften the blade. While this could be desirable, as discussed in Section 2.3.1.1, if one cannot accommodate additional blade deflection, then some means of increasing the blade stiffness will have to be introduced, most likely through additional material and cost.

As the concept studies progressed, it also became apparent that in-plane sweep of the blade could be employed to achieve aeroelastic coupling. While this presents some challenges to manufacturing and shipping, it avoids some of the problems associated with the use of oriented fibers.

Studies of aeroelastic coupling continued over a period of nearly six years on the NGT Project. Many of the early studies served as foundation for more sophisticated studies to follow. The results of all of these investigations of aeroelastic tailoring were assimilated in one final comparison of comparably coupled blades that achieve coupling through sweep or oriented fibers, or a combination of both. This final investigation attempted to develop blades with practical levels of sweep or coupling.

The study focused initially on the design and analysis of the structure of four new carbon/glass hybrid blades with varying levels of twist-flap coupling:

- Carbon Spar Baseline: The GE37a all-glass blade and the glass spar cap replaced with one constructed of unidirectional carbon fibers of appropriate thickness to ensure adequate tower clearance and acceptable axial strain values in the carbon fibers
- Coupled Blade 1: The Carbon Spar baseline with the very modest coupling added through oriented fibers in the skin
- Swept/Coupled Blade: Coupled Blade 1 with in-plane sweep
- Coupled blade 2: Coupled Blade 1 with additional oriented fibers in the spar cap.

The operation of the EMD turbine employing a set of each of these blades was dynamically simulated. Different pitch control systems were modeled: Version B for the baseline collective pitch controller, and the Enhanced Controller A2, which includes independent blade pitch control. The operation of the EMD turbine employing all the various blades and controller combinations was simulated for a variety of IEC DLCs, as well as fatigue.

The conclusions to be drawn from the results are:

- Coupling (via oriented fibers and/or sweep) combines with the enhanced controller to produce reductions in nearly all key design-driver loads.

- Oriented fibers and sweep are complementary.
- Oriented fibers and sweep produce very similar impacts on loads and fatigue for comparable levels of twist-flap coupling.
- Twist-flap coupling and the enhanced controller algorithms are complementary.

An analysis of the impact on COE to be expected from aeroelastic coupling was conducted. When presented with a concept that offers loads reductions in a wind turbine system, two different means exist for translating those reductions into reductions in COE:

1. Increase the rotor diameter until loads are returned to baseline levels, thereby realizing an improvement in the energy capture
2. Redesign the components affected by the reduced loads to realize cost savings.

The basic principles of the first approach are:

- Increase the rotor diameter until key driver loads/fatigue are returned to baseline levels, taking account of:
  - Influence of rotor diameter on loads
  - Influence of blade mass on loads (see below)
- Account for impact on energy capture, including:
  - increased energy capture due to larger rotor diameter
  - reduction in energy capture due to coupling
- Account for increased blade mass/cost required to maintain adequate blade tip-tower clearance, taking account of any initial changes in the minimum tip clearance due to changes in blade stiffness, coupling, sweep, etc.
- Account for any increases in component costs to accommodate any loads/fatigue that are allowed to increase above baseline levels
- Determine net impact on COE.

The results of the COE analysis show that reductions of approximately 6% can be expected from the moderate aeroelastic tailoring. The enhanced controller operating alone shows a potential for approximately 2% reduction in COE, while combining it with the uncoupled carbon blade shows a potential for approximately 3% reduction in COE. These conclusions are based upon substantial scaling of the rotor diameters. The fundamental conclusion of this study is that aeroelastic tailoring allows for a significant increase in the size of the rotor placed on a given drive train.

The reductions in fatigue and ultimate loads may also allow a wind turbine to be certified to operate in a higher wind class than that for which it is presently certified. Reductions in COE in the range of 9%–11% appear achievable by increasing the rotor size while increasing the wind class.

Incorporation of sweep requires development of a brand new blade mold, the cost of which was beyond the scope of the NGT project. Similarly, incorporation of off-axis carbon fibers requires a substantial materials testing program that was beyond the scope of the NGT program.. Aeroelastic tailoring was therefore not incorporated into the EMD turbine.

2.3.4 Cast Blade Roots

A substantial portion of fiberglass blade weight and cost is contained in the root and transition sections, which contribute negligibly to energy capture. Less expensive steel or cast iron pieces could be inserted between the hub and the blades, which in turn could be shortened. In other words, the fiberglass blade root would be replaced by a steel or cast iron blade root. The goal would be to reduce the COE by reducing the turbine cost by a greater percentage than the energy capture. The use of steel blade roots could also possibly reduce the cost of transporting large blades or even enable the use of larger rotor diameters without resulting in significant transport problems.

As part of a structural and cost analysis of steel or cast iron blade roots, the following studies were conducted:

- Impact studies of the impact on the aerodynamic performance and energy capture of the 34-m blades for the GE 1.5 MW wind turbine resulting from adopting cylindrical blade root (extension lengths 6 m, 8 m, and 9 m) and assuming that:
  - Blade T-bolts similarly employed now would be used with a relatively long radial transition from the round root to the aerodynamic section would be required to avoid blade buckling
  - A new means of clamping the blade to the bearing could be created, such that the blade would feature no structural transition to a round section, and the blade could be fitted with a radially short cosmetic piece to provide rapid transition from the round blade root (extender) to the aerodynamic sections
- Structural analyses of four different blade root concepts in an effort to determine the materials requirements:
  - Cast iron blade roots
  - Rolled and welded sheet steel blade roots
  - Cast iron Y-shaped hub
  - Rolled and welded sheet steel Y-shaped hub
- Cost-benefit analysis of these concepts.

Five different blade geometries were produced, two each (rapid cosmetic transition and long structural transition) for the 3- and 6-m blade roots, and one only (rapid transition) for the 9-m blade roots (or their equivalent Y-shaped hubs). Figure 12 summarizes the expected annual energy capture to be derived from each of the six blades for the defined IEC Class II wind. The expected energy losses are small with the 6-m roots, but rather significant with the 8- and 9-m roots. On the basis of the structural analysis, the only configuration that makes any sense is one with cast iron cylinders 2 m in diameter, with inboard flanges connected to the hub, and outboard flanges connected to the blade. The design is driven by fatigue considerations, and the fatigue strength of weldments is so much less than that of ductile cast iron that a second configuration employing a cylinder constructed of welded steel sheet could not be optimized to a realistic weight. The same concept applied even greater to a Y-shaped hub constructed of welded steel sheet.

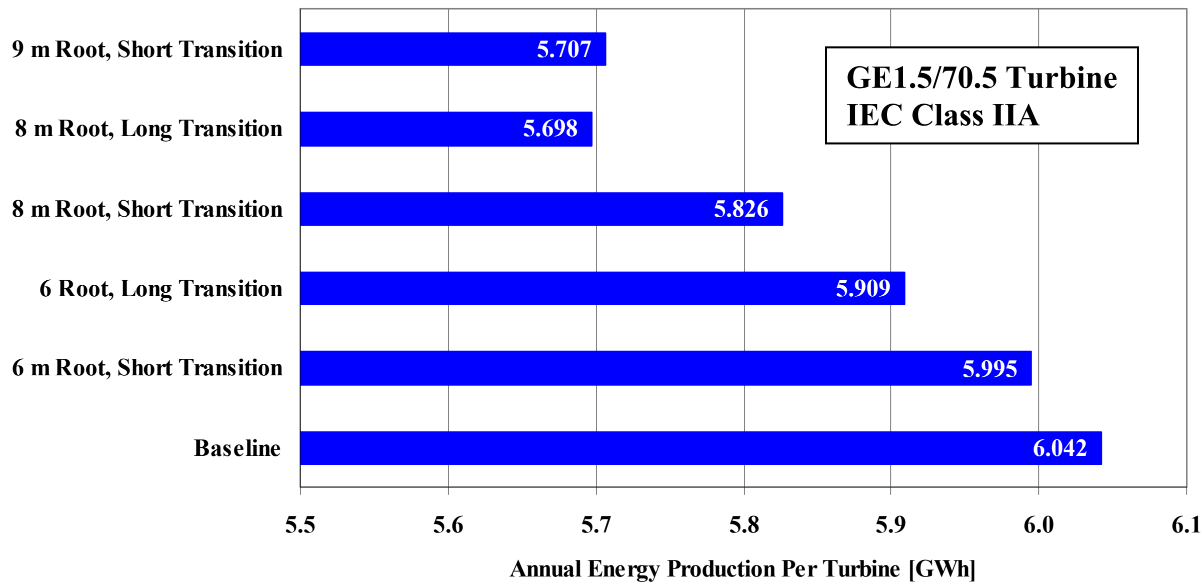


Figure 12. Impact on Energy Production from Blade Root Inserts

The conclusions drawn from this study are:

- Extensions of the cylindrical blade root beyond 6 m from the current blade root (i.e., beyond 7.25 m radial location) lead to unacceptable loss of energy production and outweighs any cost savings.
- Any blade root lengthening concept is dependent upon the use of radially short transitions from the round sections to the aerodynamic sections. The use of radially long transitions in the blade structure itself is not acceptable.
- Cast iron blade roots 6 m in length might offer a slight improvement in COE, but the present study shows savings of no more than 1%, or less than $0.05/kWh.
- The use of cast iron blade roots of 6 m in length could increase the total rotor weight by as much as 10,000 lb, or nearly 20% of the current rotor weight.
- A large cast iron Y-shaped hub does not appear to offer any advantages over separate cast iron blade roots and creates severe logistical (transportation) problems.
- Welded structures appear to be entirely inappropriate for hub and blade applications given the relatively low fatigue endurance limits of welded steel and the relatively high fatigue to which the blades and hubs are subjected.

The concept might have advantages, however, not from a cost perspective but from the perspective of transportation. As the blades on larger, future turbines grow in length, it may become necessary to adopt two-piece blades for transportation reasons. The cast iron blade root presents an alternative with minimal or no negative impact on performance.

2.3.5 Boundary Layer Control

GE commissioned a review of the current state of the art aerodynamic boundary layer control via blowing and suction and an evaluation of the potential for applying these concepts to a wind turbine blade. The study identified four motivations for employing boundary layer control in fluid dynamic systems:

1. Boundary Layer Control (BLC) via blowing or suction to eliminate leading edge laminar separation
2. Laminar Flow Control (LFC) using suction to delay or avoid transition of the laminar boundary layer to turbulent
3. BLC to control the development of the turbulent boundary layer
4. BLC to delay or avoid turbulent boundary layer separation using suction or blowing.

Only Items 2 and 4 have potential for wind turbine applications. GE's study includes an exhaustive review of research on laminar flow control (LFC and boundary layer control (BLC, including the drag reductions and lift improvements possible, as well as the requirements for magnitudes of blowing and suction to achieve desired results. The study also examined practical methods for implementing suction and blowing, derived equations for calculating the power required for flow control, and estimated the cost of implementing a flow control system into a wind turbine blade.

These results were used to conduct power performance and cost modeling of four possible configurations for introducing suction into a wind turbine rotor blade. All of the system modeling uses the GE37a blade as a baseline. The key conclusions drawn from this study are that the maximum benefit expected from applying suction for flow control on the GE37a blade is near a 2% reduction in the COE.

2.4 Variable Diameter Rotor

The essence of the variable diameter concept is to have a rotor that may augment its diameter and increase energy capture in the frequently occurring moderate wind speeds below rated wind speed where most energy is available. At the same time, the diameter can reduce in higher wind speed and avoid loads which would otherwise penalize a rotor of increased diameter. The cost reducing principle applicable to the present concept is to introduce *adaptive* characteristics responding to the value (i.e., energy) and avoiding the penalty (i.e., loads).

Two differing implementations of the variable diameter concept were identified. The first concept, subsequently referred to as S-VADER (where “S” refers to sliding), employs an outer section of blade that can retract “telescopically” into the main blade section. The second concept, referred to as C-VADER, changes the swept area of the rotor by coning the blades. GE subsequently decided that only the S-VADER concept merited further consideration.

The S-VADER concept employs a telescoping rotor blade configuration, in which an outer, extensible blade section fits inside of a main blade (see Figure 13). This rotor concept could feasibly be incorporated into the existing upwind, pitch-regulated configuration. The main blade would bolt via a pitch bearing to the hub, and the pitch of the blade would be controlled in the same manner as presently. The outer blade would be extended and retracted by a second mechanical system in response to gross changes in wind speed (see Figure 14), while pitch control would still be employed to regulate the power.

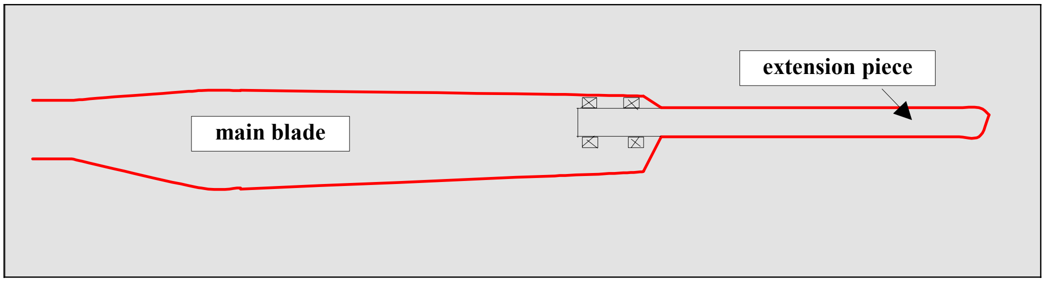


Figure 13. Illustration of S-VADER Blade Concept

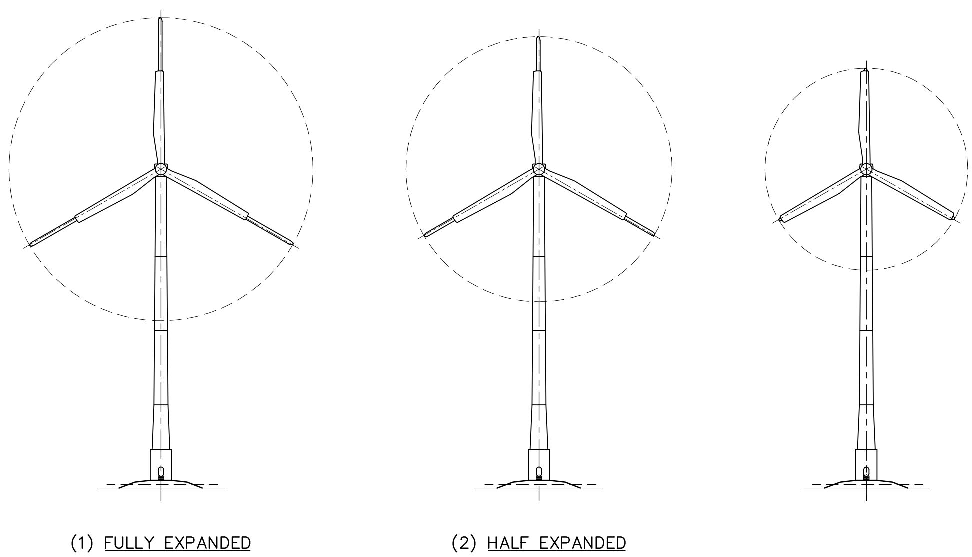


Figure 14. Illustration of S-VADER Operation

The key is to engineer a system without having an aerodynamically inefficient rotor. The requirement that the outer blade fit within the inner blade, combined with the need for bearings and mechanical drive systems will necessitate the use of an outer blade with a relatively short chord length. This requires airfoils of much higher design lift coefficient on the outer blade section.
A rotor optimized for a specified speed schedule has a unique lift distribution. This is a determinate independent of airfoil selection. The lift is proportional to the product of airfoil design lift coefficient and blade chord. Therefore, any airfoils with adequate general performance characteristics will realize optimal blade performance with a chord distribution that suits the design lift coefficients of the airfoils. For a variable diameter rotor, it was initially unclear what type of airfoils and what chord widths might be preferable. Two basic criteria were established:

1. The main blade airfoils must be wide enough to accommodate the extension piece and the bearing system.

2. The variable diameter system must be operated within boundaries set by a conventional reference wind turbine of the same rated electrical output. Specifically, this defines limits on power, torque, thrust, and tip speed.

Perhaps the most important outcome of the COE investigations are an appropriate choice of design parameters. The concept shows a COE benefit of just a little over 12%, using steady state power curve comparisons and prototype rotor costs. Using turbulent wind power curves, increasing tower height to optimum for the VADER and assuming that in production, VADER added rotor cost can be further reduced, leads to a predicted COE reduction of approximately 18%. The study further suggested that improved control could realize about 3.5% more.

The original goal of the NGT Program was a reduction of COE to $0.025/kWh at the NREL Reference Site 1. GE estimates that this COE would require a reduction in COE of 30% from the baseline values. Therefore, it is concluded that with a little optimization, the VADER concept could move GE two-thirds or more towards the goal. Combined with other improvements identified elsewhere in this report, it is believed that the S-VADER concept could achieve the stated NGT goal.

2.5 Summary of Concept studies

The implications of the concept studies show the best chance for providing near-term reductions in windpower COE at risk levels that are acceptable to the wind energy financial community are:

- Optimized low-solidity rotor blades
- Larger rotor enabled by sophisticated load-alleviating controls systems
- Advanced controls systems, including:
- Independent blade pitch to effect asymmetric load control
- Tower top accelerometer feedback for tower damping
- Coupling of pitch and generator torque control
- Taller, more flexible towers.

All of these features were incorporated into either or both of the POC and EMD turbines, as noted in the following sections of the report. Although at the outset of the project GE had defined the 750-kW machine as the baseline, as a result of the concept studies, it was decided to develop the POC at the 1.5-MW scale. New 34-m and 37-m low-solidity blades were designed for these machines using custom-designed high-lift, thick airfoils. The controller of the POC machine was modified to include coupling of pitch and torque control. The tower of the POC turbine at 70 m is taller than the 50-m-tower of the baseline machine, and the 70-m tower is much softer. Installation constraints at the Tehachapi, California, test site prevented employing an even taller 80-m-tower, which would have been desired. The EMD machine incorporates asymmetric load control and tower top damping control.

The electromechanical concept studies showed two concepts–the medium-speed wound rotor induction generator operating in VSCF mode and the medium-speed wound rotor synchronous generator operating in fixed-speed mode–that might produce small improvements in COE, but less than $0.01/kWh. These were not felt to justify moving away from proven drive train technology with significant commercial advantages for GE Wind.

Of all the concepts examined, only 2 seem to offer potential for substantial (10%-25%) reductions in the COE: aeroelastic tailoring (via either oriented fibers or in-plane sweep) and the variable diameter rotor (S-VADER). Damage identification feedback shows potential for 5%-8% reductions in COE. Consideration of the financial and marketing risks led GE not to prototype hardware incorporating these concepts as part of the NGT Project.

3 Proof of Concepts Turbine

The POC turbine was erected at GE's facility in Tehachapi, California, in April 2000. It is rated at 1.5 MW and features a three-bladed, upwind rotor of 70.5-m diameter driving a six-pole wound-rotor asynchronous generator through a three-stage planetary gearbox. The turbine is installed on a free-standing 63-m tubular steel tower on a 5-m concrete pedestal foundation.

3.1 Design of the POC1.5/70.5

Relative to the GE1.5/65 or the baseline 750-kW turbine, the innovations incorporated into the POC turbine are:

- flexible, low-solidity, longer (34-m) rotor blades employing high-lift, thicker airfoils
- coupling of blade pitch and generator torque control
- taller (70-m), soft hybrid steel/concrete tower
- a water-cooled generator.

Table 3 summarizes the POC turbine configuration as it was finally designed, built, and tested. Figure 16 shows a schematic of the turbine as installed. The POC turbine's innovative features were subsequently adopted for the GE1.5/70.5 production machine.

Loads analysis of the POC turbine in the form of its production version, the GE1.5/70.5, have been performed. A comparison of the results of those loads analyses with the design loads for the original GE1.5/65 turbine led GE to conclude that the GE 34-m blades could be installed on the existing GE1.5/65 platform to form the POC machine without any changes to the remaining GE1.5 hardware. GE has also generated the predicted performance curves summarized in Figure 15. The annual energy production data presented in Table 3 is based upon this predicted performance. The following subsections detail the design and testing of several of the components or component systems of the turbine, including the tower and foundation, the drive train, controls systems, and the rotor blades.

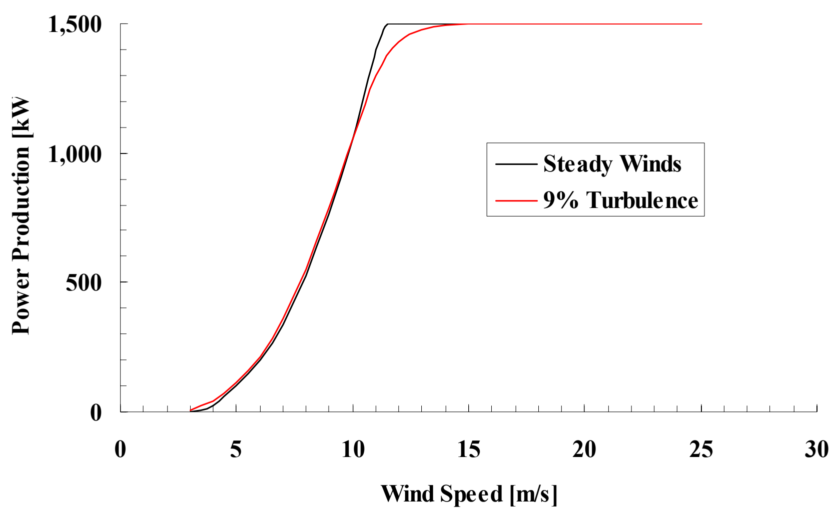


Figure 15. Predicted Power Performance for the POC1.5/70.5 Turbine

Table 3. Proof of Concepts Turbine Configuration Description

General Configuration	
Type	Grid Connected
Rotation Axis	Horizontal
Orientation	Upwind
Number of Blades	3
Rotor Hub Type	Rigid
Rotor Diameter	70.5 m
Hub Height	71.0 m
Performance (Sea level 10 m)	
Rated Electrical Power	1,500 kW
Rated Wind Speed	11.4 m/s
Cut-in Wind Speed	3.0 m/s
Cut-out Wind Speed	25.0 m/s
Extreme Wind Speed	59.5 m/s
Gross Annual Energy Production	
at NREL Reference Site 1	4.5 GWh/yr
at NREL Reference Site 2	5.6 GWh/yr
Rotor	
Swept Area	3,902 m^2
Rated Rotational Speed	20.0 rpm
Cut-in Rotational Speed	11.8 rpm
Coning Angle (forward precone)	1.5°
Shaft Tilt Angle	4.0°
Blade Pitch Principle	Independent Blade Pitch
Blade Pitch Actuation	Independent Electric Drive
Blade Pitch Angle	Fully Variable
Hub Material	Ductile Cast Iron
Direction of Rotation	Clockwise
Power Regulation	Variable Speed/Pitch
Over-speed Control	Variable Pitch
Blades	
Make	GE34a
Material	Glass Fiber-Reinforced Epoxy
Length	34.0 m
Airfoil Types	Proprietary GE

Table 3. Proof of Concepts Turbine Configuration Description (Continued)

Drive train	
Gearbox Type	3-Stage, Planetary/Helical
Gearbox Rated Power	1,660 kW
Gearbox Ratio	1:72
Shaft Brake Type	Single Disk Active
Shaft Brake Location	High-Speed Shaft
Generator	
Generator Type	Doubly Fed Asynchronous
Number of Poles	6
Synchronous Speed	1,200 rpm
Rated Generator Speed	1,440 rpm
Minimum Power Generator Speed	850 rpm
Speed Range	800-1,600 rpm
Turbine Line Voltage	575 VAC
Rotor Rated Current	585 A
Turbine Generating Frequency	60 Hz
Generator Insulation Class	F
Cooling Systems	Water-to-Air Cooled
Yaw System	
Wind Direction Sensor	Wind Vane
Yaw Control Method	Planetary Drive
Yaw Bearing Type	Friction
Control System	
Make, Controller	GE Winteligence
Power Conversion Strategy	Doubly Fed
Power Logic/Device Type	PWM/IGBT
Controller Logic Systems	Microprocessor-based
SCADA	GE VisuPro
Tower	
Type	Tapered Tube
Material	Certified Steel
Height	63.1 m
Foundation	
Type	Pad/Pedestal
Material	Concrete
Height	5.0 m

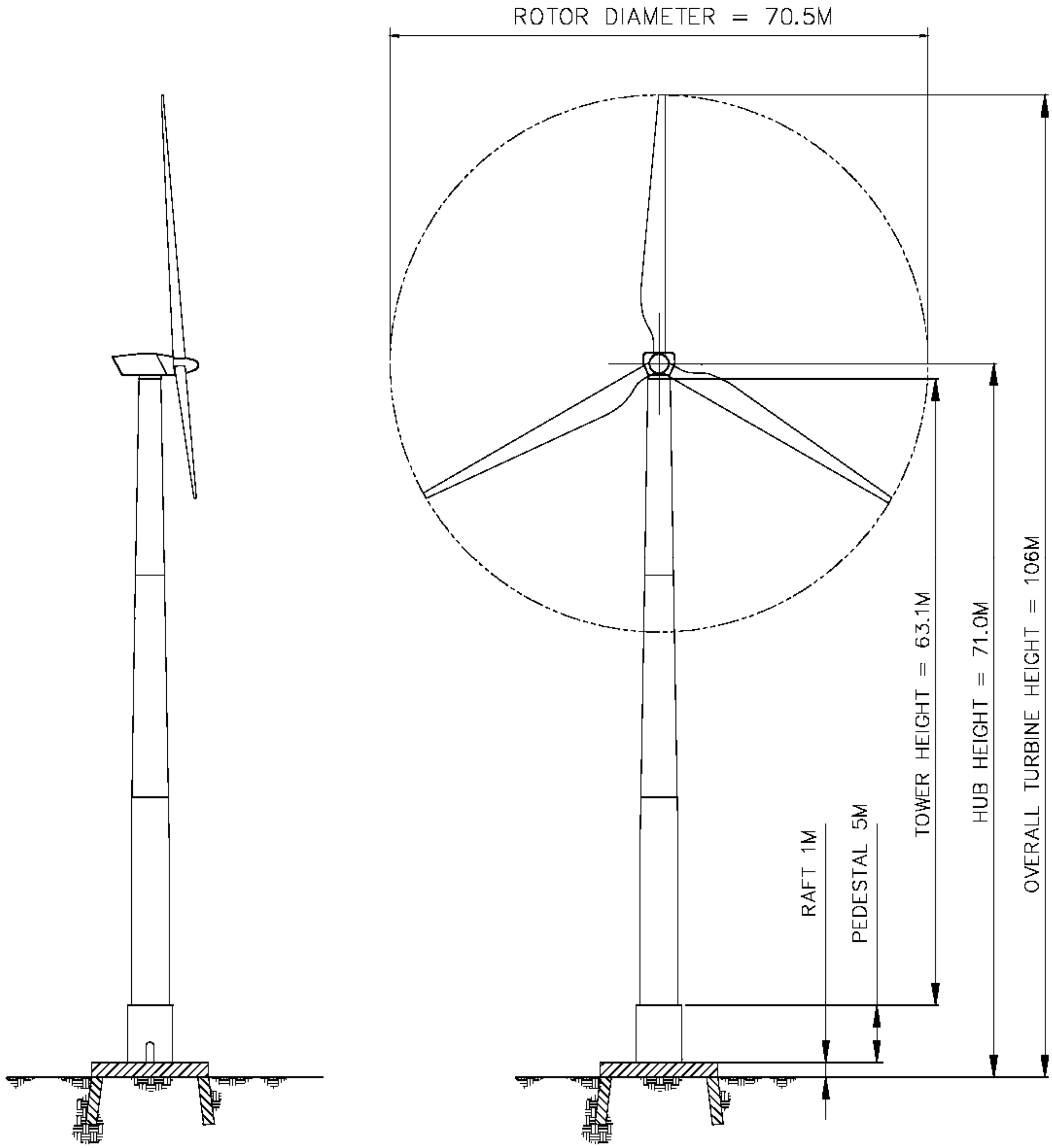


Figure 16. Schematic of the Proof of Concepts Turbine

3.1.1 Tower and Foundation Design

A hub height of 70 m was achieved with a new 68-m tower designed using a new hybrid approach utilizing a 5-m concrete pedestal and a 63-m steel tower to achieve an 80-m hub height. Several advantages of the hybrid approach are significant. First, using concrete at the base minimizes the diameter of the steel tower that needs to be shipped. Shortening the height of the steel section also helps resolve buckling considerations. Finally, access doors can be installed in the concrete much more easily than in the welded steel construction. Studies showed, however, that for the POC design, using more than 5 m of concrete was not cost effective. These studies indicated, however, that for a larger turbine, a taller concrete pedestal might be cost effective. Figure 17 shows the tower being installed at the test site in Tehachapi, California.

Figure 17. POC Tower in Tehachapi, California

3.1.2 Drive Train Design and Testing

Figure 18 illustrates the drive train layout. The drive train is substantially similar to that of the GE1.5/65 turbine with the addition of the water-cooled generator (Figure 19). Figure 20 illustrates additional components of the drive train, including the main shaft, gearbox, the high-speed shaft, and the yaw drive. The POC 1.5-MW drive train–from the main shaft to the generator–was delivered to the National Wind Technology Center (NWTC) at NREL in Golden, Colorado, for comprehensive testing at their large-scale dynamometer facility. Figure 21 illustrates the test setup at NWTC. The objectives of this testing were:

- Verify proper load transfer at the gear mesh using contact patterns test.
- Measure the efficiency of the gearbox and generator systems at different power levels using thermal loss measurements.

- Measure thermal characteristics of the drive train after stabilization at rated power.
- Run a gearbox endurance test equivalent to 20 years turbine operation.
- Verify sufficient lubrication in idling situations without grid (oil supply pump not running) with different oil temperatures.

The unit was tested from November 2002 until February 2003 for a total of 1,600 hours of test time. The target test torque load was approximately 130% of the design load to accelerate test time to equal gear tooth fatigue of 20 years. A teardown inspection of the gearbox was performed following the testing. The conclusions of the inspection were that after a test of fatigue damage equal to the design life of 20 years, the transmission was in serviceable condition. There were no components with breakage or extraordinary wear.

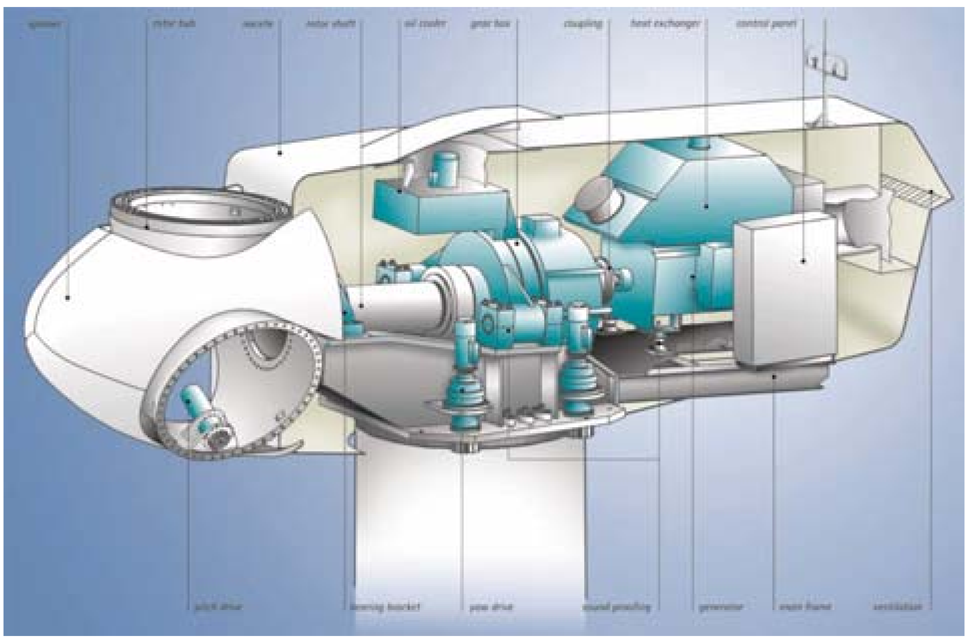
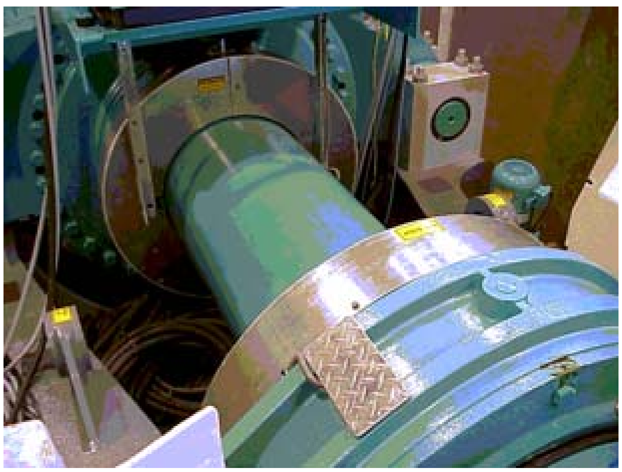


Figure 18. Drivetrain Layout of NGT Proof of Concept 1.5 MW Turbine

3.1.3 Controls System Design

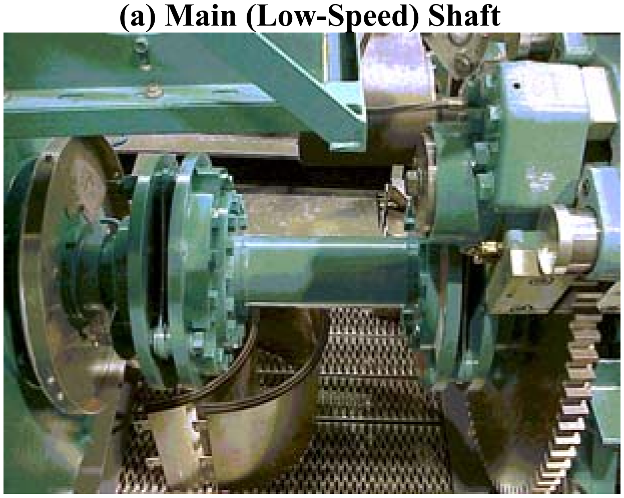
The coupling of pitch and torque control was identified as a promising concept in the concept studies phase of the project and is incorporated in the POC turbine. The POC main control cabinet is illustrated in Figure 22.

Figure 19. POC Water-Cooled Loher 1.5-MW Generator

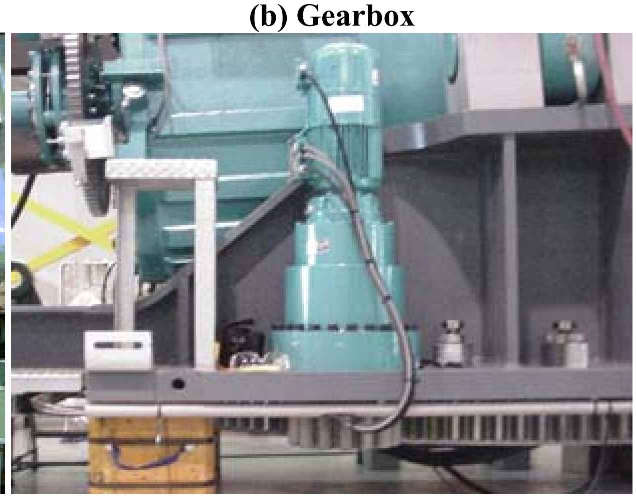


(a) Main (Low-Speed) Shaft

(b) Gearbox



(c) High-Speed Shaft



(d) Yaw Drive

Figure 20. POC Drive Train Components

Figure 21. POC Drive Train on the NWTC Dynamometer Test Stand

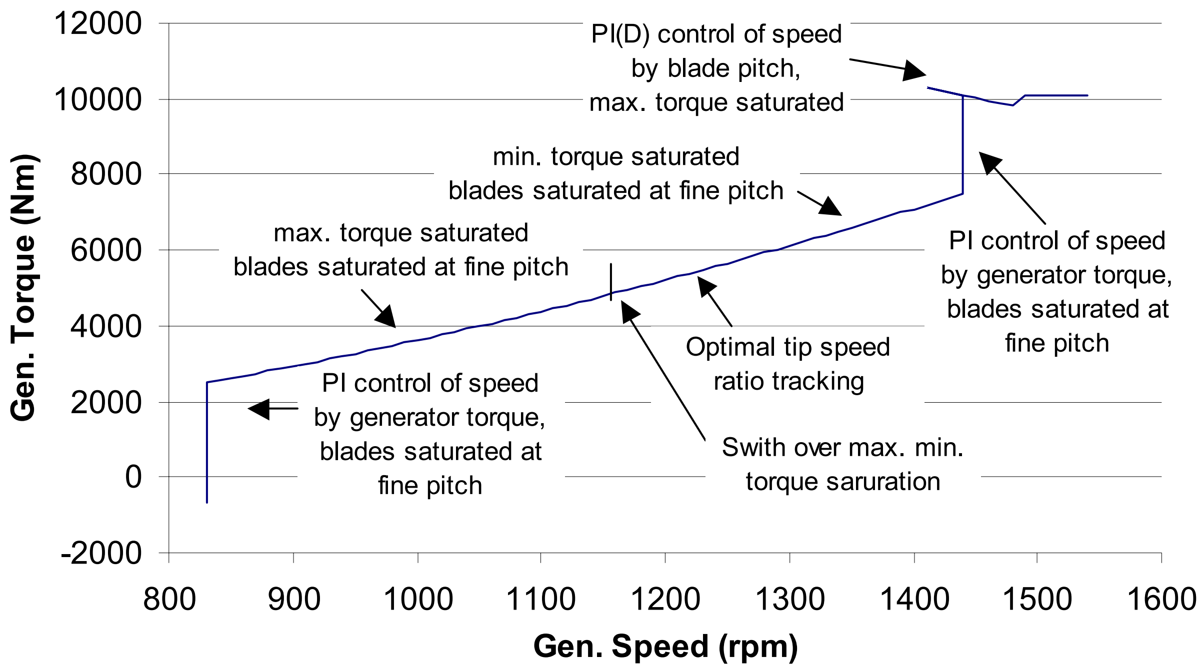


Figure 22. POC Main Control Cabinet

3.1.4 POC 34-m Rotor Blade Design and Testing

The POC1.5/70.5 configuration uses 34-m blades (GE34a), as shown in Figure 23. The GE34a blade employs a family of airfoils custom designed as part of the NGT project for the POC turbine. GE personnel and consultants designed a family of airfoils with the characteristics summarized in Table 4. In addition to the quantitative drivers shown in the table, GE also required that:

- The lift and drag characteristics of the airfoils with leading edge surface roughness should be optimized, as opposed to "clean" performance.
- Given all other design constraints, the designs should seek to maximize the lift-to-drag ratio of each airfoil section.
- The airfoils should be designed as a family such that they can be physically blended into a smooth rotor blade surface.
- The maximum lift coefficients should be insensitive to leading edge surface roughness.

The six airfoils produced by this effort (14%, 18%, 24%, 27%, 30%, and 45% thick sections) are shown in Figure 24. These airfoils have been patented in the U.S. (USPTO Number 6,503,058), Europe (EPO Number 1152148A1), and internationally (WIPO Number WO 01/83983A1).

Models of three of the NGT Airfoils (the 18%, 24%, and 30% thick sections) were actually built and tested. The models were designed and fabricated by subcontractors in response to requirements established by GE. The testing was performed in the Wichita State University (WSU) 7 x 10-foot Low-Speed Wind Tunnel (LSWT). Each model was nominally 84 inches long with a chord of 36 inches, and was mounted vertically in a floor-to-ceiling arrangement in the LSWT test section.

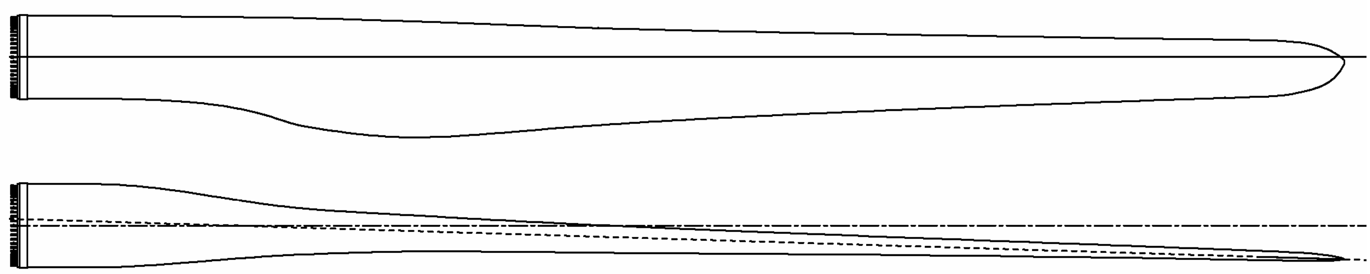


Figure 23. POC 34-m Blade Planform

Table 4. Summary of Required Airfoil Characteristics

Airfoil	t/c	Design C_l (C_l at L/D_{max})	Re_c at Rated	Re_c at Cut-in	Blade Location r/R
1	14%	1.25	8 million	4 million	90%
2	18%	1.25	8 million	4 million	75%
3	24%	1.25	8 million	4 million	55%
4	27%	1.35	6 million	3 million	40%
5	30%	1.50	4 million	2 million	25%
6	45%	Maximized	3 million	2 million	15%

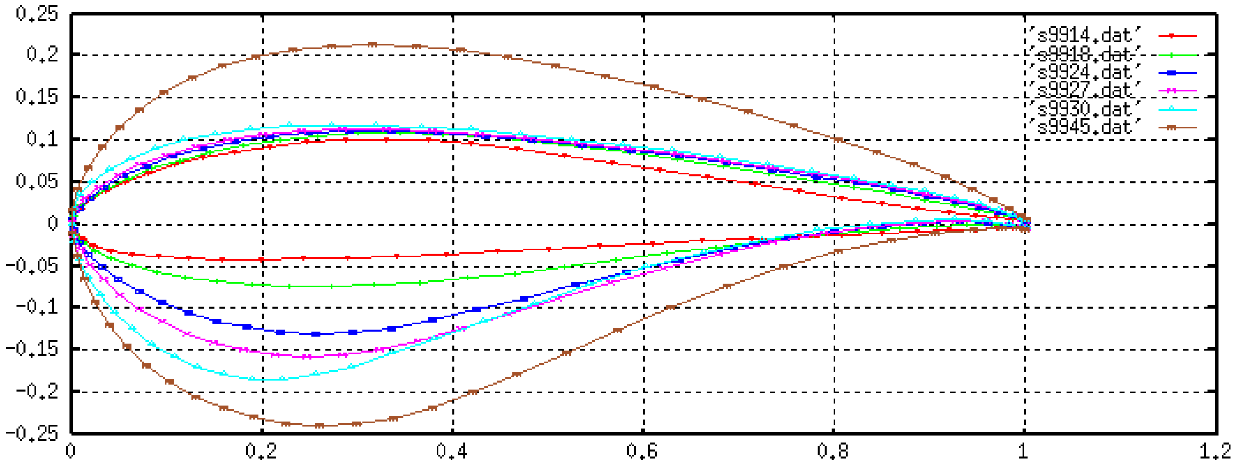


Figure 24. NGT POC Airfoil Family

For each model, the two-dimensional airfoil properties (lift, drag, and pitching moment coefficients) were measured at a pre-determined schedule of tunnel speeds and angles of attack. Aerodynamic forces were measured indirectly using a combination of surface static-pressure taps and wake dynamic pressure surveys. In addition, the LSWT floor and ceiling balances were used to measure and record the direct aerodynamic forces on the model.

The majority of the test technical objectives were met. One requirement for the testing was to reach a chord Reynolds number of 4 million, and values of 3.8, 3.7, and 3.5 million were attained for the 18%, 24%, and 30% airfoils, respectively. This was deemed acceptable. Two-dimensional force and moment measurements were made to a high level of confidence for all three airfoils in both smooth and rough surface conditions.

Figure 25. 18% Airfoil Model in the WSU Wind Tunnel Test

The test data lift curve slopes agree quite well with thin airfoil theory. A review of the predicted lift data showed that the predicted lift curve slopes were approximately 10% higher than the value from thin airfoil theory. It has, therefore, been concluded that the wind tunnel lift data are of high confidence, and that the calculations over-predicted the lift curve slope. Comparisons of measurements with predictions showed:

- Minimum drag values and width of drag bucket showed good agreement with predictions.
- Measured $C_{L,max}$ was between 5% and15% lower than predicted by XFOIL (expected result).
- Stall for all airfoils was sharper than predicted, with the 24% and 30% thick airfoils showing particularly sharp stall.

A finite element model of the blade structure was built in ANSYS and used to analyze the blade's structural integrity with respect to:

- Tip deflection
- Buckling stability
- Extreme strain
- Laminate Fatigue
- Root attachment failure due to fatigue, extreme static loads, and flange liftoff
- Natural frequency requirements.

Structural analyses showed that tip deflection constraints were the overriding design driver. The target tip deflection was 65% of tower clearance. Forward coning was added to the blade at the 5.5-m station in order to provide an additional 0.7-m of tip offset and reduce spar cap material usage. Coning from 5.5 m keeps most mass on the pitch axis. Nevertheless, tip deflection requirements still required an increase in blade mass. Full-scale tests indicated that 59.7% of tower clearance was being consumed, versus the 70% maximum allowed by Germanischer-Lloyd. The maximum strain on the blade is well below the strain allowable. Fatigue damage remains everywhere under unity. Buckling analyses indicated that no core material was needed in the spar cap. The POC blade design departs from the approach used by the baseline 750-kW turbine's 24-m blades, which employ balsa in the spar cap for buckling resistance. By narrowing and thickening the spar cap of the POC blade, the use of balsa was avoided for equal cost. The GE34a uses a T-bolt root attachment developed in the 750-kW NTRT program. The blade uses 54 M30 T-bolts.

The prototype set of blades is shown in Figure 26.

Figure 26. Prototype Set of the GE34a Blades at Tecsis

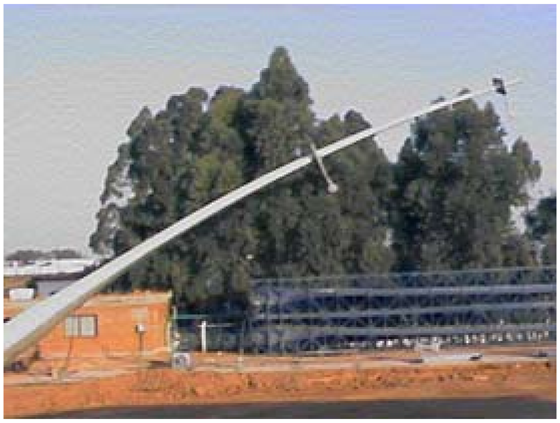
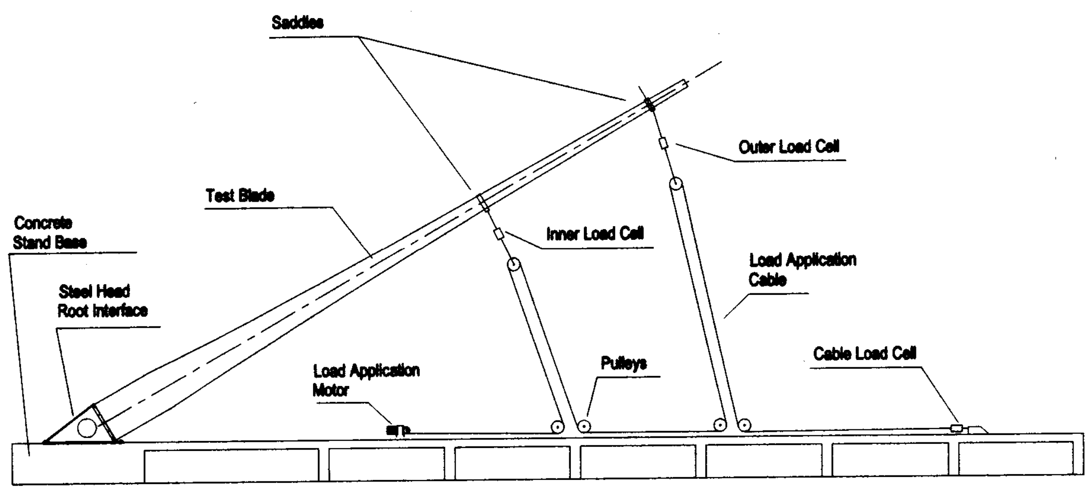
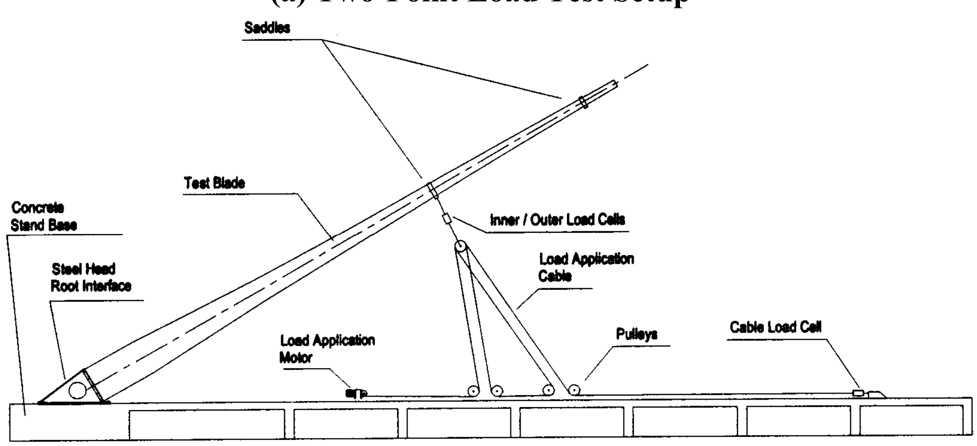


Figure 27. Static Blade Tests

One of the prototype blades was initially subjected to a flapwise load test while still at the fabrication facility, as shown in Figure 27. The tests conducted were a two-point static load test, a single-point static load test, and a natural frequency measurement, all of which were conducted under the supervision of GE Wind Energy and Germanischer Lloyd personnel. Figure 28 illustrates the arrangements of the two-point and single-point tests. The magnitudes of the loads which were applied were provided by GE Wind Energy personnel to Tecsis test engineers. These tests confirmed the structural integrity of the blade before it was first fielded. The blade did not fail under 100% loading. Maximum values of strain that were measured were approximately 0.4%.



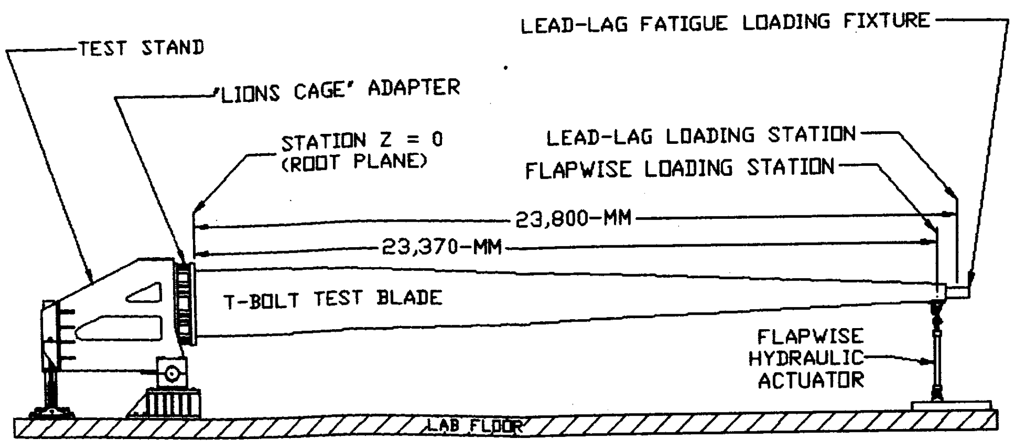
(a) Two-Point Load Test Setup



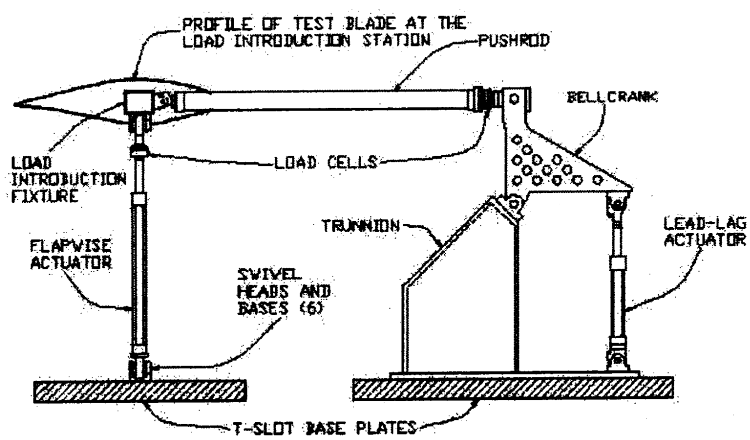
(b) Single-Point Load Test Setup

Figure 28. Static Blade Test Setup

A blade certification testing program was conducted at the National Wind Technology Center (NWTC) in Golden, Colorado, under the supervision of personnel of the National Renewable Energy Laboratory. Fatigue testing of a GE34a blade began in August 2001, and was completed January 10, 2002. The same blade was used as had been subjected to the static testing described above. The objective of the fatigue test was originally to conduct a limited life fatigue test in order to give basic assurance that no major catastrophic fatigue failures are likely. The target number of test cycles was 300,000 for a ten percent life. This was based on observations that past fatigue testing has revealed problems early in the test duration. Later, this objective was expanded to include a full 20-year life of damage using a one-million-cycle load duration. The test operated for 1.9 million cycles.



(a) Blade Setup



(b) Two-Axis Hydraulic Actuator

Figure 29. NWTC Fatigue Test Setup

Loads were provided to NWTC by GE Wind to represent a 3-million-cycle design life. The testing began with the 3 million cycle loads. Beginning at approximately 928,000 cycles, however, the test loads were increased to accelerate the test to achieve 100% damage in 1 million cycles. Table 5 summarizes the operationally equivalent fatigue damage resulting from the fatigue test at various spanwise locations on the blade. These results were determined via finite element analysis of the blade. These results indicate that the critical test regions of the blade were tested to at least 20 years of equivalent loading, the design life of the blade, without significant loss of global properties. In summary, NWTC concluded from the fatigue tests that the blade sustained the full load spectrum without significant loss of properties, including stiffness and ability to carry load.

Table 5. Test Equivalent Damage

Spanwise Station z [m]	Test Life (Years)		Notes
	Flap	Edge	
6.4	16.4	2.0	
9.2	25.9	5.6	
12.8	37.3	21.7	Critical Flap Station
16.9	20.2	37.6	Critical Edge Station
20.9	0.1	5.0	

3.1.5 POC Hub Analysis

Figure 30 shows the POC hub on the floor of GE's manufacturing plant. During the NGT program, concern arose that much better fatigue properties for the GGG40.3 ductile cast iron used in the hub were required. A subcontract was issued to conduct an exhaustive materials characterization effort for this material. Analyses of the hub static stresses and fatigue concluded that, with a redesign of one small part, the hub does not appear to have any fatigue-driven problems. Therefore, it was concluded that the hub is acceptable in fatigue for greater than 20 years.

3.2 Installation and Testing of the Proof of Concepts Turbine

The POC turbine was erected at the GE facility in Tehachapi, California. Installation was completed in April 2000. The turbine was installed on a free-standing 63-m tubular steel tower on a 5-m concrete pedestal foundation. The rotor installation is illustrated in Figure 31, and the completed POC turbine is shown in Figure 32.

Certification testing of the POC turbine for noise, loads, power performance, and power quality has been completed under the supervision of personnel of the National Renewable Energy Laboratory.

Figure 30. POC Hub

Figure 31. POC Turbine Rotor Being Installed

Figure 32. POC Turbine Installed in Tehachapi, California

3.2.1 *Power Performance Testing of the POC Turbine*

Power performance testing was conducted on the POC turbine between February 9, 2001 and March 21, 2001. A total of 265 hours of data with wind out of the acceptable wind directions (295° to 335°) and with the turbine available was collected during that time. Figure 33 compares the measured and theoretical power curves.

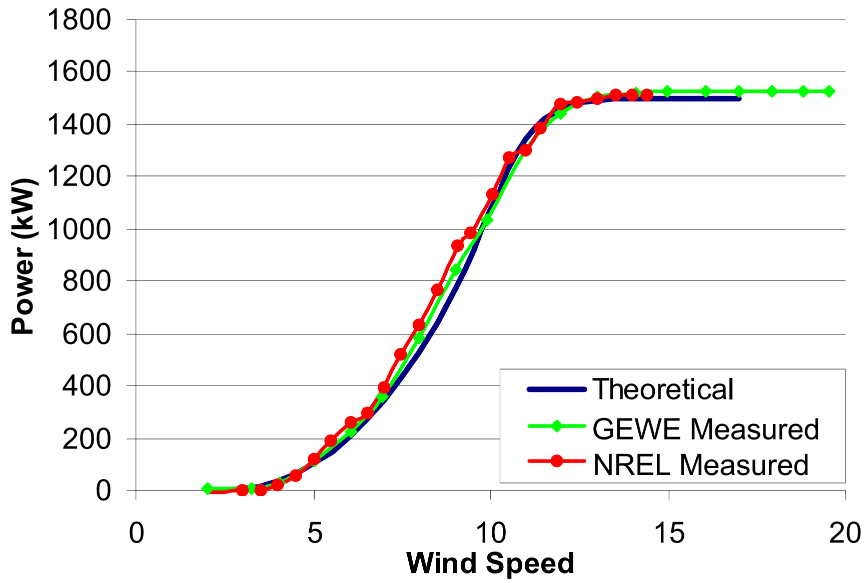


Figure 33. Comparison of Theoretical and Measured POC Power Curves

3.2.2 *Power Quality Testing of the POC Turbine*

Power Quality testing of the POC turbine was conducted between May 31, 2001 and June 25, 2001. A total of 973 hours of data were collected while the turbine was available, but only 315 hours of data was recorded during 10-minute intervals during which the turbulence intensity was between 8% and 16%, as required by the IEC 61400-21 Power Quality Standard. The testing consisted of a long-term phase and a short-term phase, conducted between 31 May and 9 June. The tests conducted as part of the power quality testing were:

- Maximum measured power (60-second and 0- and 2-second), reactive power demand
- Turbine start and stop tests
- Voltage fluctuations (flicker)
- Current imbalance measurements
- Voltage and current harmonics.

The results of the testing indicated no power quality problems for the POC turbine.

3.2.3 *Mechanical Loads Testing of the POC Turbine*

Mechanical loads testing was conducted on a production GE1.5/70 turbine installed in a wind farm at Desert Sky, Texas. Loads were determined to be acceptable.

3.2.4 Acoustic Noise Testing of the POC Turbine

Turbine and background acoustic noise data were collected between July 6, 2000 and July 8, 2000. The A-weighted sound power level as 8 m/s wind speed was calculated using two methods for determining wind speed. The first method uses wind speed as measured on a temporary 10-m meteorological tower upwind of the turbine. The second method derives the wind speed from measurements of turbine power and a power curve obtained from power performance testing. Additional acoustic testing included wind speed sensitivity, directivity, one-third octave spectra, and tonal analysis. Figure 34 shows a comparison of the POC noise level of approximately 98 dBA to levels historically associated with wind turbines of various sizes. It is significant to note that the POC turbine exhibits noise emissions substantially below those of turbines of similar size. Figure 35 shows that this is true in spite of the POC's relatively higher rotor tip speed.

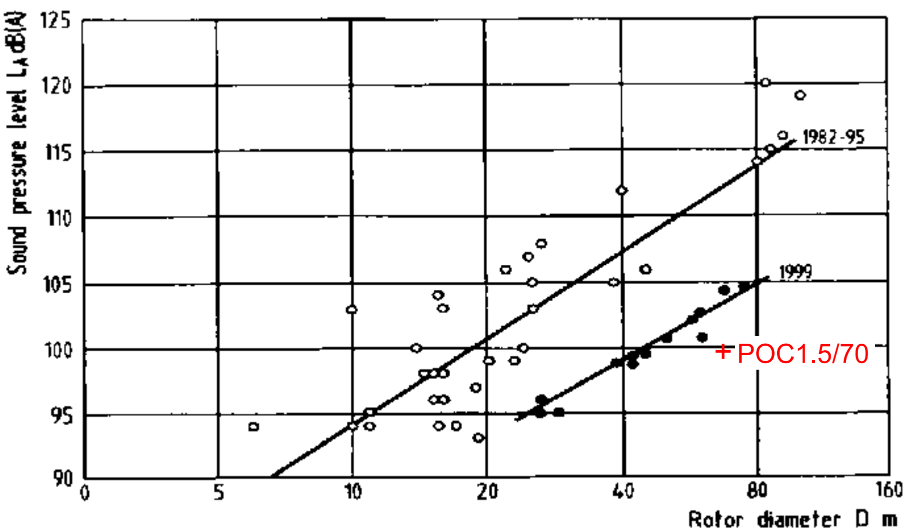


Figure 34. Comparison of the POC Noise Levels to Historical Levels

3.3 POC Certification

In December 2000, Germanischer Lloyd (GL) granted GE wind certification of design compliance to IEC 61400-1 Class IIA for the production derivative of the POC turbine, the GE1.5/70.5.

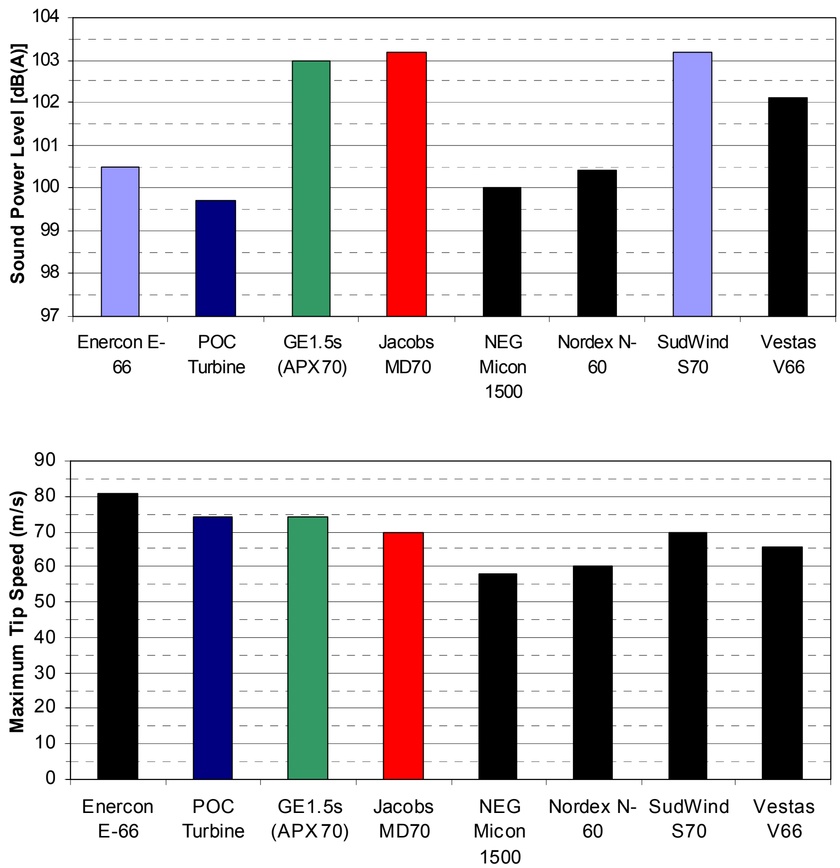


Figure 35. Comparison of Noise Levels and Rotor Tip Speeds

4 Engineering and Manufacturing Development Turbine, EMD1.5/77

The Engineering and Manufacturing Development (EMD) turbine is an evolution from the POC turbine incorporating the following improvements in its design:

- A larger 77-m-diameter rotor
- 37-m blades with tip curvature
- Tower-top accelerometer feedback for tower damping
- Independent blade pitch control for asymmetric load control.

These improvements were implemented in two stages. The POC configuration was converted to the EMD configuration officially in April 2002, with the installation of the 37-m blades. Following the power performance, loads, and noise testing of that configuration, the EMD was further upgraded with the addition of the asymmetric load controller (ALC) and associated turbine sensor system. The system in its final configuration was then, in June 2003, subjected to additional loads testing.

Table 6 provides a summary of the EMD turbine configuration.

4.1 IEC Class S High Energy Capture Rotor Configuration and Loads

The development of the high energy capture 77-m rotor was motivated by two considerations. First, earlier design efforts conducted as part of the development of the POC turbine showed that the rotor diameter for the GE1.5 turbine could be increased to at least 77 m and possibly higher, without a significant increase in loads if the design took advantage of the fatigue-reducing benefits of increased rotor flexibility.

The second motivation for the 77-m rotor involved consideration of the operating conditions. Analysis of wind speed histories from several commercial wind farms in the U.S. led to the conclusion that one could expect 50-year gusts in level great plains areas at the 65–80 m hub heights to be less than 52.5 m/s. The analysis also shows that both IEC Class 2A and 2B overpredict the level of turbulence intensity at all wind speeds compared to every wind farm analyzed. The value of k=2 for the Weibull coefficient that is typically used to define frequency distributions of wind speed generally overpredicts the frequency of very high winds relative to what is occurring at typical wind farms.

Since the distribution of very high winds and the turbulence intensity are factors that most define extreme wind and fatigue, study results indicate using the IEC standard wind classes will result in the design of a turbine that is overly conservative for the conditions typical at the sites where wind farms are being installed in the U.S. Use of a larger Weibull factor than 2.0 and turbulence intensities lower than IEC Class 2B would allow the use of a much larger rotor and result in the proper projection of significantly improved energy capture and reduced COE.

Therefore, GE decided to define a special S-Class environment for purposes of designing the 77-m rotor that was felt to be typical of real operating environments:

- A 52.5 m/s 50-year 3-second-average extreme gust
- 9 m/s annual average wind speed
- A Weibull distribution of wind speeds with coefficient k=2.3
- IEC Class B turbulence intensity
- 1.2 kg/m^3 annual average air density.

Table 6. Engineering & Manufacturing Development Turbine Configuration

General Configuration	
Type	Grid Connected
Rotation Axis	Horizontal
Orientation	Upwind
Number of Blades	3
Rotor Hub Type	Rigid
Rotor Diameter	77.0 m
Hub Height	71.0 m
Performance (Sea level 10 m)	
Rated Electrical Power	1,500 kW
Rated Wind Speed	13.0 m/s
Cut-in Wind Speed	3.0 m/s
Cut-out Wind Speed	25.0 m/s
Extreme Wind Speed	m/s
Gross Annual Energy Production	
at NREL Reference Site 1	4.8 GWh/yr
at NREL Reference Site 2	5.9 GWh/yr
Rotor	
Swept Area	4,657 m^2
Rated Rotational Speed	18.3 rpm
Cut-in Rotational Speed	10.9 rpm
Coning Angle (forward precone)	1.3°
Shaft Tilt Angle	4.0°
Blade Pitch Principle	Independent Blade Pitch
Blade Pitch Actuation	Independent Electric Drive
Blade Pitch Angle	Fully Variable
Hub Material	Ductile Cast Iron
Direction of Rotation	Clockwise
Power Regulation	Variable Speed/Pitch
Over-speed Control	Variable Pitch
Blades	
Make	GE37a
Material	Glass Fiber-Reinforced Epoxy
Length	37.25 m
Airfoil Types	Proprietary GE Wind

Table 6. Engineering & Manufacturing Development Turbine Configuration (Continued)

Drive train	
Gearbox Type	3-Stage, Planetary/Helical
Gearbox Rated Power	1,660 kW
Gearbox Ratio	1:78.246
Shaft Brake Type	Single Disk Active
Shaft Brake Location	High-Speed Shaft
Generator	
Generator Type	Doubly Fed Asynchronous
Number of Poles	6
Synchronous Speed	1,200 rpm
Rated Generator Speed	1,432 rpm
Minimum Power Generator Speed	850 rpm
Maximum Generator Speed	1,600 rpm
Turbine Line Voltage	575 VAC
Turbine Generating Frequency	60 Hz
Generator Insulation Class	F
Cooling Systems	Water-to-Air Cooled
Yaw System	
Wind Direction Sensor	Wind Vane
Yaw Control Method	Planetary Drive
Yaw Bearing Type	Friction
Control System	
Power Conversion Strategy	Doubly Fed
Power Logic/Device Type	PWM/IGBT
Controller Logic Systems	Microprocessor-based
SCADA	GE VisuPro
Tower	
Type	Tapered Tube
Material	Certified Steel
Height	63.1 m
Foundation	
Type	Pad/Pedestal
Material	Concrete
Height	5.0 m

GE briefly considered the costs and benefits of not only increasing the rotor diameter, but also increasing the turbine rating to 1.7 MW. However, the analysis showed that the costs associated with increasing the turbine rating more than offset the benefits in terms of increased energy capture, resulting in an increase in the COE. This is true for both Rayleigh and non-Rayleigh wind speed distributions, as shown in Figure 36.

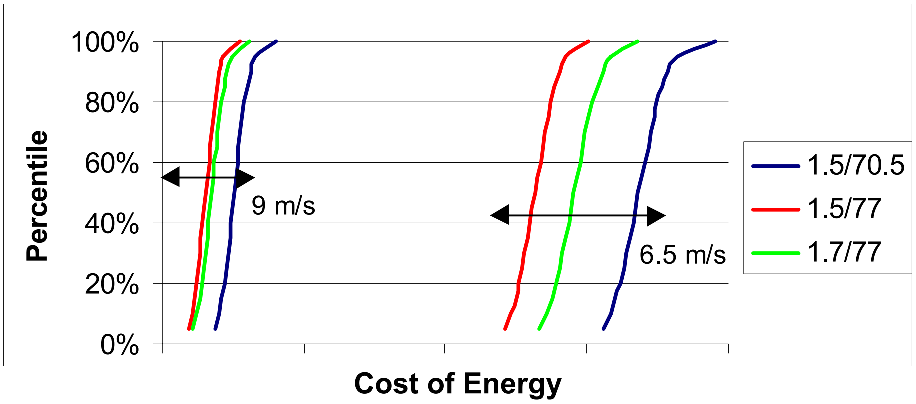


Figure 36. Benefits to COE from Increasing Rotor Size and Turbine Rating

Four different sets of loads have been generated for the EMD 1.5/77 Turbine:

1. Loads generated using DIBT WZ2 conditions (similar to the IEC S-Class) with the baseline controller and heavy blades reflective of the prototype blade weight
2. Repeat number 1 for lighter blades reflective of product improvements
3. Loads using IEC Class III-A conditions, an intermediate blade weight, and the baseline controller
4. Loads using IEC Class III-A conditions, heavier blades, and the ALC.

The key conclusion to be drawn from the loads data is that the EMD1.5/77 turbine can operate in either the IEC Class S defined above or in IEC Class 3A with loads less or on par with the IEC Class 2A loads for which the POC1.5/70.5 machine was certified. Finally, the ALC promises to produce some substantial reductions in several key components of both ultimate loads and fatigue.

As a consequence of these loads predictions, none of the components of the POC turbine would need to be redesigned in order to accommodate loads changes effected by the conversion to the EMD configuration. In the following sections, only detailed design information about the changed components–the gearbox, rotor blades, and the control system–will be presented.

Figure 37 summarizes the predicted power for the EMD1.5/77 turbine, as determined via simulations.

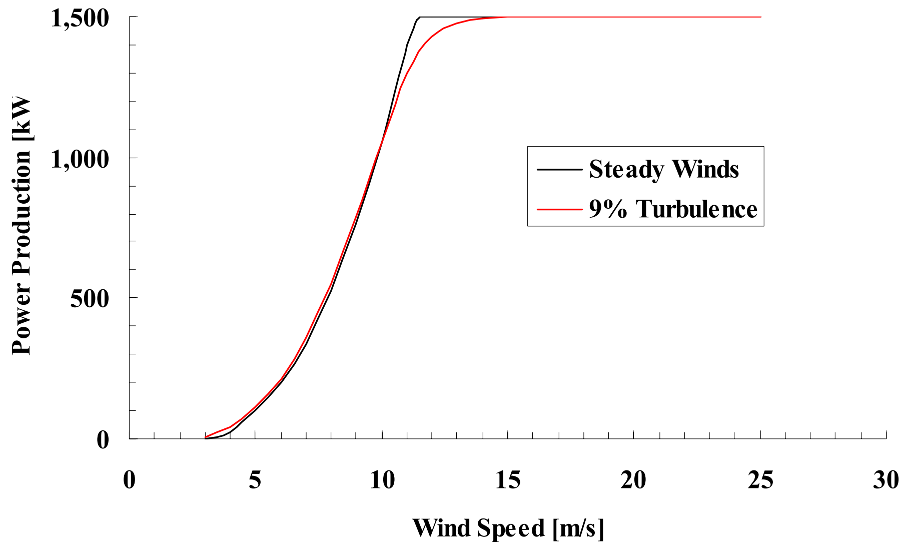


Figure 37. Predicted Power Performance for the EMD1.5/77 Turbine

4.2 EMD1.5/77 Component Designs

4.2.1 EMD Gearbox Modification

To maintain the rotor tip speed at the baseline value of 73.8 m/s when operating at rated power, the gear ratio was increased to 78.25:1 rpm. The gearbox supplier analyzed the gearbox and determined that the change could be effected by modifying the third stage only. The first stage gear can accommodate the increased ultimate loads.

4.2.2 EMD 37-m Rotor Blade Design and Testing

Several approaches to designing the 37-m blades were considered, including adding material to either or both of the root and the tip of the GE34a blade. GE finally decided to choose the approach motivated by energy capture and add all of the 3.25 m of new blade to the tip. The contour is exactly the same as that of the GE34a blade out to the 30.95-m spanwise location. Details of the blade geometry are shown in Figure 38. The GE37a blade employs another innovative feature: blade curvature. During the concept studies (see Section 2.3) it was noted that one alternative to blade coning is the use of blade curvature, that is, gradually curving the blade out of the rotor plane. For the GE37a blade, such curvature is only employed at the tip of the blade in order to provide a small amount of additional tower clearance. This is needed because of the extra length of the blade. The blade still employs 1.3° of forward precone in the root.

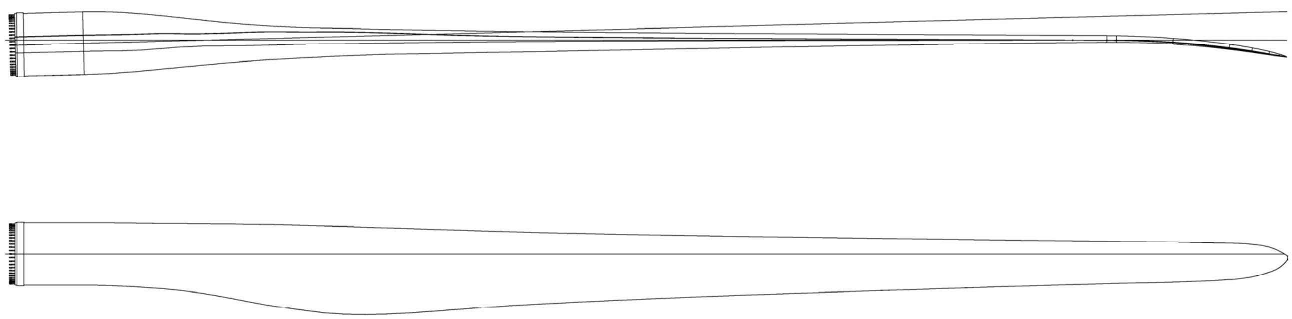


Figure 38. EMD 37-m (GE37a) Blade

The laminate schedule of the spar cap of the GE37a blade consists of substantially more layers than that of the GE34a in order to ensure proper tower clearance. Structural analyses show that the peak strains are well within the allowables. Moreover, the GE37a blades have sufficient margin with respect to both buckling and fatigue as a consequence of meeting deflection requirements. Finally, the T-bolts are acceptable based upon on a comparison of the rotated blade root fatigue damage equivalent loads (m=4), which are only 4% greater in the 9 m/s average wind S-Class with the 37a blades than with the GE34a blades in IEC Class 2A. Margins on the POC T-bolts were greater than 4%. As a result, reanalysis of the T-bolts was not required.

In the end, GE has been able to achieve a 19% increase in rotor swept area with only a 3% increase in blade mass.

The prototype GE37a blade underwent ultimate load tests in December 2001 and survived to greater than 100% of the S-Class extreme flap and edge loads, including the 1.35 safety factor required by GL. The National Wind Technology Center has conducted comprehensive fatigue testing of a GE37a blade in connection with the NGT Program.

4.2.3 EMD Controls Systems

The EMD turbine initially operated with the baseline POC controller. This portion of the EMD development was referred to as the EMD Phase 1. The subsequent phases consisted of modifications to the EMD control system to implement an ALC with independent blade pitch actuation and tower top damping. The changes were implemented in several phases as follows:

- EMD Phase II: develop and refine the software and hardware necessary for the ALC
- EMD Phase III: Asymmetric loads control testing
- EMD Phase IV: ALC refinement.

At the conclusion of the NGT Program, Phases II and III were complete and Phase IV was ongoing. This section documents the efforts completed in Phase II, while Section 4.3.4 summarizes the results of Phase III. Phase IV will be completed outside of the NGT program and is not documented in this report.

The control objective using the ALC is to maximize energy capture while minimizing load subject to the following constraints:

- Rotational speed constraints
- Maximum power output
- Maximum loads
- Limitations on actuation force and rate

The control schemes implemented in the EMD controller include:

- Independent blade pitch for reduction of asymmetric loads due to wind shear, yaw misalignment, and turbulence
- Tower-top accelerometer feedback driving collective blade pitch for damping fore-aft tower oscillations due to pitch changes and turbulence

The two control approaches are coded as separate tasks in the controller, which communicates to the main task through the SVI Interface.

4.2.3.1 <u>Tower-top Accelerometer Feedback</u>

The tower-top accelerometer feedback was identified during the concept studies as a promising means of reducing transient loads that were actually being induced by the pitch system itself.

Development of the concept required more time, however, than would permit incorporation into the POC machine. Therefore, it has been incorporated at the EMD stage.

As Phase 2 of the development, a wide variety of tower top accelerometers were subjected to exhaustive testing. Accelerometers from several different vendors were evaluated with respect to the performance, reliability, and quality of the manufacturer. GE selected a modified version of the unit that was already being employed for diagnostic monitoring in the GE1.5 turbines. Figure 39 illustrates this unit and its location on the EMD turbine.

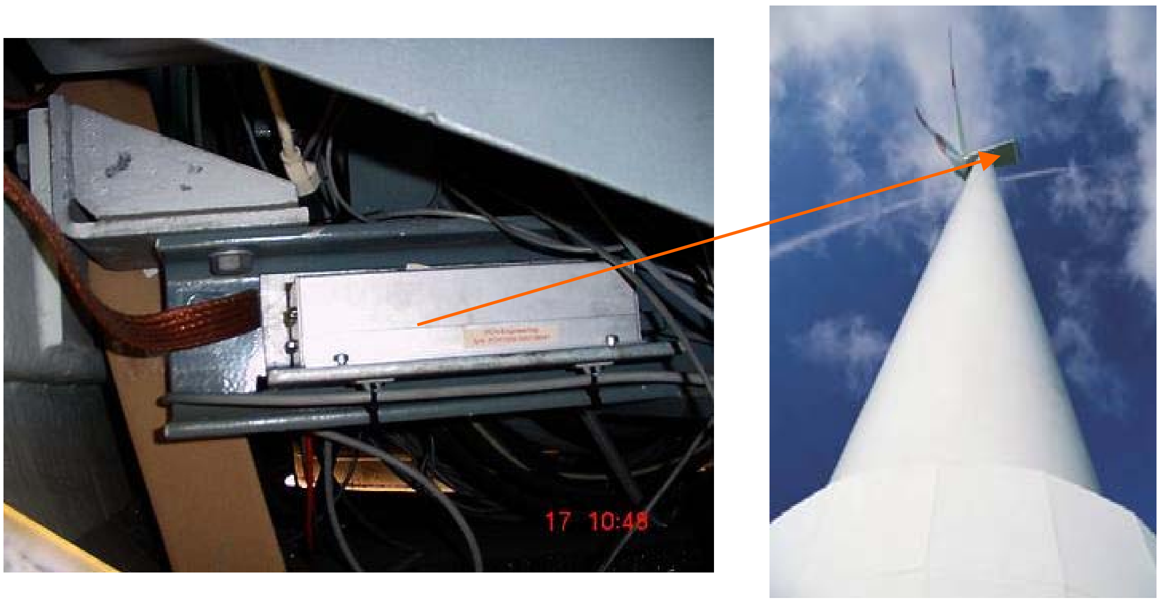


Figure 39. EMD Tower-Top Accelerometer Package

Fore-aft acceleration is measured from the unit via an output that is wired directly to the controller analog input. The control algorithm is PI from the filtered signal to collective pitch increment, as illustrated in Figure 40. Error traps are programmed for mean value drift, mean value significantly different from zero, non-changing signal, and large change in signal in one time step. The system was tested in emulation with serial communication to the controller before being implemented on the operating turbine.

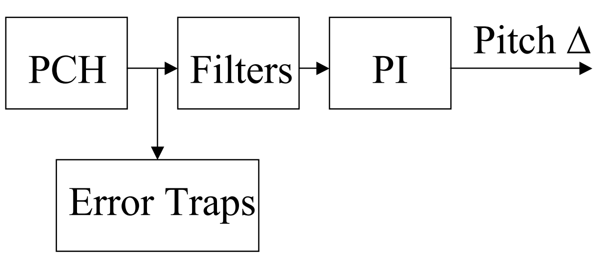


Figure 40. Flowchart of the EMD Tower Top Accelerometer Feedback

4.2.3.2 Rotor Feedback Control

Results of the concept studies showed that independent blade pitch for asymmetric load control could produce significant reductions in key loads with little or no impact on the energy capture. As noted above in Section 4.1, EMD loads analysis supported significant loads reductions from using the ALC.

The POC turbine featured independent blade pitch actuation, insomuch as the pitch of each rotor blade was controlled by an independently functioning electric drive. This concept was inherited from both the value-engineered version of the baseline 750-kW turbine and the original version of the GE1.5 turbine. However, like those turbines, the control system of the POC turbine was designed to effect synchronized actuation of the blades. Absent a control fault or a failure of one of the pitch drives, all three blades should ideally be at the same pitch setting at all times.

The EMD control system implements the functionality to control the pitch setting of each blade independently. As first noted during the concept studies, independent blade pitch control can be used to ameliorate asymmetric rotor loads that cannot be otherwise mitigated using collective or synchronized pitching of the blades. Implementation of asymmetric load control requires two sets of measurements:

1. Blade or rotor loads
2. Blade azimuthal position.

The primary tasks in Phase II were:

- Evaluate, install, and document candidate input sensors for the ALC.
- Select the best sensors from the suite of candidate test sensors for ALC.
- Design and test communications (hardware/software) interface between the candidate control sensors and the PLC.
- Test and verify sensor signal data acquisition in the PLC by passing acquired signals through PLC to the ADAS DAQ system.
- Implement and test PLC control code (software) changes for ALC system.
- Perform system dynamics study to determine controller parameters:
    - ALC algorithm coding on PLC
    - ALC simulation and emulation.
- Develop and verify ALC system safety features including:
    - Signal plausibility and error-trapping routines to prevent spurious input signals from damaging the turbine through unstable ALC feedback
    - Implementation of adjustable ALC pitch modulation limiter
    - System safety verification prior to field testing via ALC simulation and emulation
    - algorithm functionality and sensitivity of gain settings.

The first step in the development of the system consisted of an exhaustive survey of measurement approaches and appropriate sensors. An engineering evaluation of the sensors was conducted to assess sensor performance and reliability. This evaluation included numerous tests. First an industry review of vendors was conducted to screen sensors meeting Critical To Quality requirements. A series of screening tests were performed on the selected sensor and target material in order to determine significant performance factors. Three factors were investigated: power supply rejection, target temperature and sensor temperature. Power supply rejection and target temperature were found to have very little influence on the sensor output and were not characterized. Sensor temperature was found to be a significant factor and was incorporated into the characterizations tests.

Bench tests verified PLC control module operation, and confirmed hardware/software interface and wiring schematics. Bench tests also confirmed program routines for flagging software safety traps, such as when a sensor signal is out of range. Software safety traps included signal failure, signal range errors, and plausibility errors. Laboratory bench tests also included testing pitch control algorithms under various turbulent wind conditions using software to simulate and emulate turbine operation. The objective of simulations and PLC emulation was to test the algorithms to ensure they were working properly and to ensure safety when testing ALC operating on the prototype turbine. Thus, bench scale laboratory tests were used to confirm proper wiring and interfacing between the sensors and the PLC software/hardware and to predict the behavior of the ALC system before "real-world" testing. These tasks included:

- PCH raw signal output tests
- PCH system self-test being initiated by PLC
- PCH system self-test failure relay detection by PLC
- Sensor operation and interface to PLC
- Software subroutine timing measurements
- Test software error traps for sensor input signals (upon failure, write flag code to text file and shut down ALC control functions).

4.3 Installation and Testing of the EMD Turbine

The first stage of the conversion of the prototype 1.5-MW machine in Tehachapi, California, from the POC configuration to the EMD configuration was completed in April 2002, with the installation of the 37-m rotor blades. The turbine remained installed on the free-standing 63-m tubular steel tower on a 5-m concrete pedestal foundation. The completed EMD turbine is shown in Figure 41.

Certification testing of the EMD turbine for noise, loads, and power performance have all been completed under the supervision of personnel of the National Renewable Energy Laboratory.

Figure 41. EMD Turbine Installed in Tehachapi, California

4.3.1 Power Performance Testing of the EMD Turbine

Power performance testing was conducted on the EMD turbine between May 4, 2002 and June 17, 2002. A total of 495 hours of data with wind out of the acceptable wind directions (295° to 335°) and with the turbine available was collected during that time. Figure 42 compares the measured and theoretical power curves.

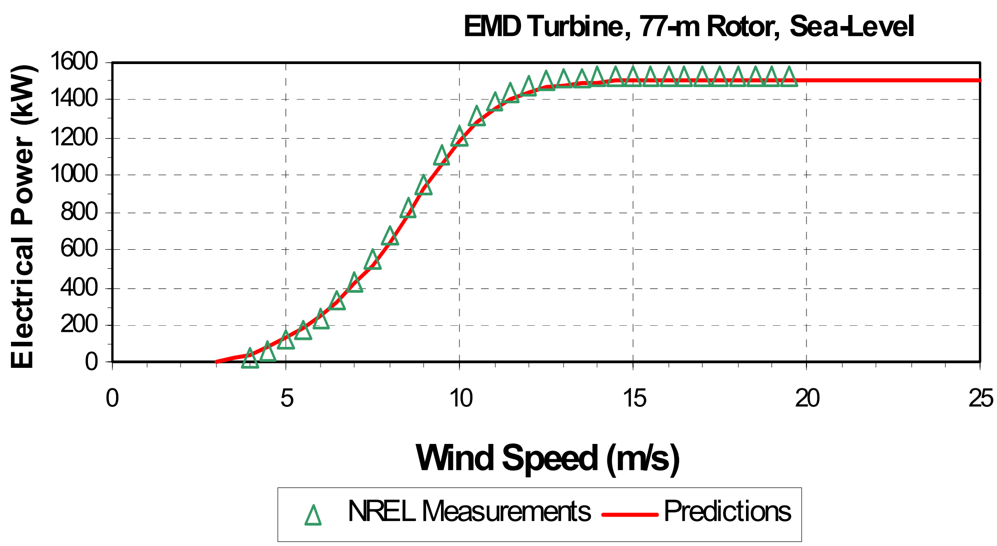


Figure 42. Comparison of Theoretical and Measured EMD Power Curves

4.3.2 Mechanical Loads Testing of the EMD Turbine

Mechanical loads testing was conducted on the EMD turbine between June 11, 2002 and September 17, 2002. A total of 77 hours of valid data were collected during normal operation of the turbine. The objective of the loads testing was to determine the relationships between wind conditions and loads on the POC turbine under all normal and emergency operating conditions. Loads were found to be acceptable.

GE conducted an internal study of the comparison between the measured mechanical loads and those predicted by the dynamic simulations. The results show that the dynamic simulations do an acceptable job of predicting the blade flapwise fatigue and the shaft torsion and bending fatigue. They also generally do a good job of predicting the mean shaft torque.

The figures also show that the simulations tend to slightly underpredict the blade edgewise fatigue and tower bending fatigue. This discrepancy is due, in part, to discrepancies in the calculation of the rotating axes in the dynamic simulations versus the measured data. In general, however, the agreement between predicted and measured data seems quite reasonable and inspires confidence in the use of the simulations.

4.3.3 Acoustic Noise Testing of the EMD Turbine

Turbine and background acoustic noise data were collected between May 2, 2002 and May 4, 2002. The A-weighted sound power level at 8 m/s wind speed was calculated using two methods for determining wind speed. The first method uses wind speed as measured on a temporary 10-m meteorological tower upwind of the turbine. The second method derives the wind speed from measurements of turbine power and a power curve obtained from power performance testing. In the case of the present testing, the two methods yielded very similar results of, respectively, 103.0 dBA and 103.3 dBA sound power levels. Additional acoustic testing included wind speed sensitivity, directivity, one-third octave spectra, and tonal analysis. Figure 43 shows a comparison of the EMD noise levels to levels historically associated with wind turbines of various sizes. It is significant to note that the EMD turbine exhibits noise emissions below the average for turbines of similar size.

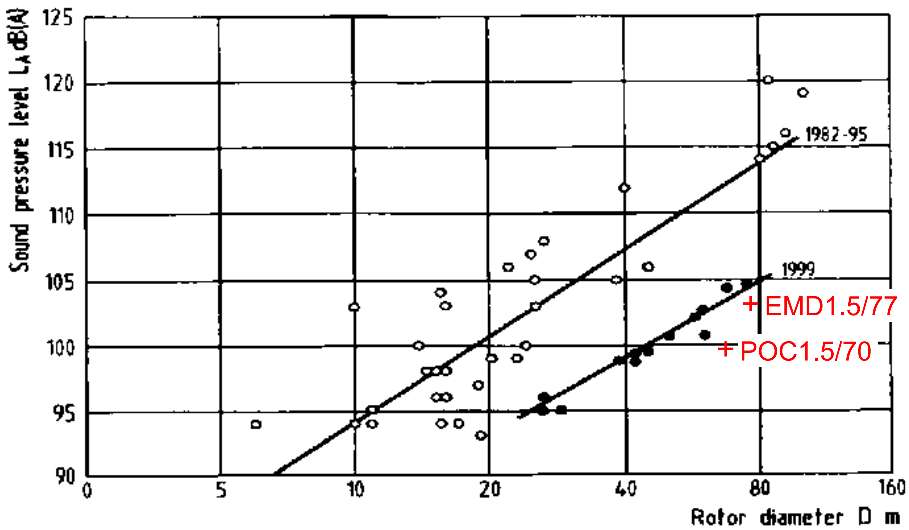


Figure 43. Comparison of the EMD Noise Levels to Historical Levels

4.3.4 Testing of Asymmetric Load Control System

Following implementation of the ALC system into the EMD, tests were conducted to document the differences in blade, main shaft, and tower loads ascribed to using ALC versus normal control operation on the GE EMD1.5 wind turbine at various wind speeds. The five main objectives of Phase III testing are:

1. Validate computer simulations of fatigue loads during ALC operation of the 1.5 MW EMD turbine.
2. Compile 10-minute datasets in sufficient quantity and with sufficient consistency to be used in scatter plots comparing ALC operation to ordinary operation.
3. Provide fatigue loads reduction information that can ultimately be used to estimate the effect on overall COE.
4. Gain knowledge of the ability of the turbine's pitch system to handle the increased activity that arises from ALC operation.
5. Verify the proper operation of the sensors used as inputs to the ALC algorithm.

This test was conducted on the EMD 1.5 wind turbine during the month of June 2003. The turbine generally functioned appropriately with the ALC control active. A total of 221 ten-minute sets of valid data were collected. Of those, 73 sets were collected during ALC operation of the turbine, and 148 sets were collected during normal or ORD operation. The data sets were obtained by manually switching between normal operation and ALC operation. The procedure was generally to switch operation between collection of 10-minute data sets in an attempt to obtain similar wind statistics for comparison.

The results show that blade flap bending fatigue is reduced by approximately 12% over the range of wind speeds where the ALC is active. If we assume an IEC class II wind distribution, this results in approximately a 12% reduction in total fatigue. The maximum, minimum, and maximum range values for flap bending seen in the data are also reduced for ALC operation. Blade edge bending DELs were not affected by the control method. There is also a reduction in the fixed frame mainshaft bending loads as well as the main shaft torsion fatigue. The reduction of the blade fluctuating loads may contribute to the reduction in torsion fatigue. The results confirm the motivation for the ALC concept.

5 Commercialization of GE NGT Technology

GE Wind Energy commercialized the NGT Proof of Concepts (POC) turbine as the GE1.5s. GE Wind commercialized the NGT Engineering and Manufacturing Development (EMD) turbine as the GE1.5sle. As of December 31, 2003, GE had manufactured and deployed 902 GE1.5s turbines in the U.S. alone. As of April 2004, when technical work on the NGT project was completed, an additional several hundred turbines of both configurations were on order for delivery during 2004 and 2005.

GE Wind has been the beneficiary of several R&D contracts from the U.S. Department of Energy during the past decade. Figure 44 shows how USDOE funds were leveraged by GE internal R&D funding. Over the course of the last decade, GE matched with internal resources nearly $6 for every $1 of USDOE funding. Figure 45 shows the overall return on that investment to the U.S.: $98 million worth of R&D, of which DOE contributed 15%, has translated into $1.9 billion worth of wind turbine hardware manufactured and deployed in the U.S., with nearly four-fifths domestic content.

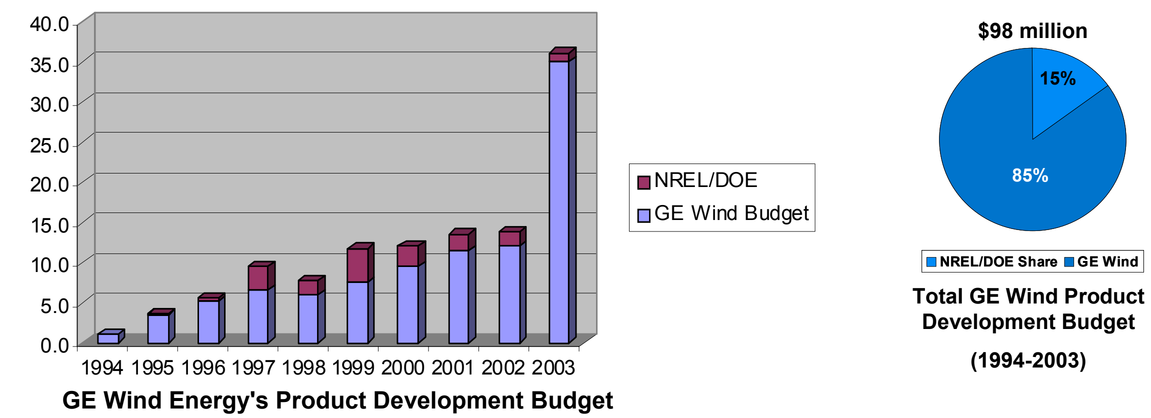


Figure 44. Leverage of USDOE R&D Funding by GE Wind Energy

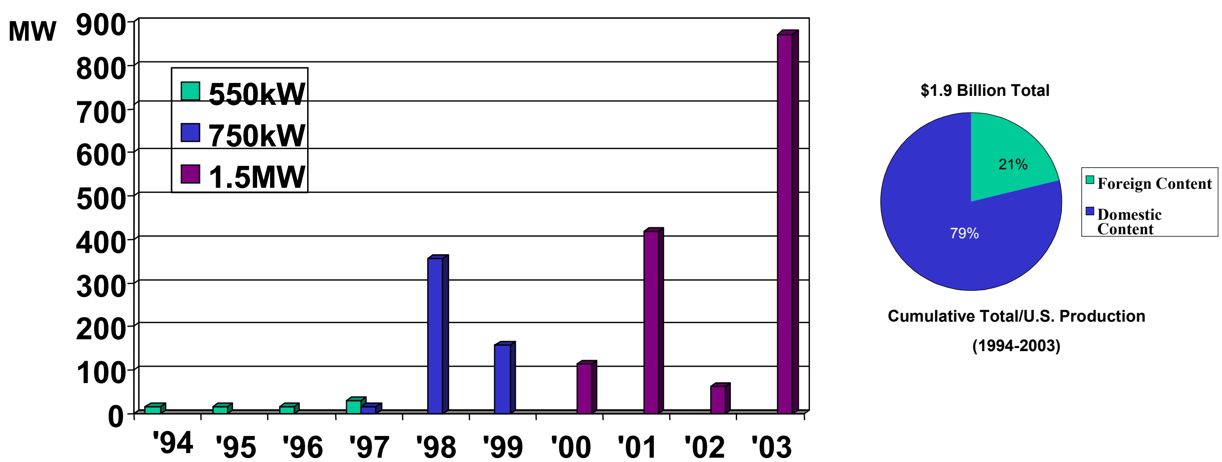


Figure 45. Return on Investment of NREL Funds in GE Wind

6 Conclusions

Conclusions of the seven-year effort spent on this project include:

- GE NGT Concept Studies identified the following concepts as those with the best chance for providing near-term reductions in the COE from windpower at risk levels acceptable to the wind energy financial community:
  - Optimized low-solidity rotor blades
  - Larger rotor enabled by sophisticated load-alleviating controls systems
  - Advanced controls systems, including:
    - Independent blade pitch to effect asymmetric load control
    - Tower top accelerometer feedback for tower damping
    - Coupling of pitch and generator torque control
    - Taller, more flexible tower.
- Through the application of these concepts in the EMD turbine, GE has been able to shave the COE from windpower by 13% relative to the baseline configuration, without adjusting for inflation
- Achieving the originally stated NGT goal of $0.025/kWh at IEC Class II sites does not appear to be achievable with technologies with near-term market-compatible risk levels
- GE has identified through its NGT Concept Studies, high-risk concepts that can provide additional reductions in COE between 10% and 25%, making the $0.025/kWh goal achievable. The most important of these are:
  - Variable diameter rotor
  - Aeroelastic tailoring of rotor blades.

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14. ABSTRACT *(Maximum 200 Words)*
This document reports the technical results of the Next Generation Turbine Development Project conducted by GE Wind Energy LLC. This project is jointly funded by GE and the U.S. Department of Energy's National Renewable Energy Laboratory.The goal of this project is for DOE to assist the U.S. wind industry in exploring new concepts and applications of cutting-edge technology in pursuit of the specific objective of developing a wind turbine that can generate electricity at a levelized cost of energy of $0.025/kWh at sites with an average wind speed of 15 mph (at 10 m height).

15. SUBJECT TERMS
wind energy; wind turbine; next generation turbine; cost of energy benefits

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