DR- 2044

MASTER

COO-2617-75/1

Distribution Category UC-60

**FEASIBILITY INVESTIGATION OF THE GIROMILL
FOR GENERATION OF ELECTRICAL POWER**

Midterm report, April–November 1975

By
R. V. Brulle

NOTICE

PORTIONS OF THIS REPORT ARE ILLEGIBLE. It has been reproduced from the best available copy to permit the broadest possible availability.

Date Published—November 1975

McDonnell Aircraft Company
St. Louis, Missouri (USA)

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

UNITED STATES ENERGY RESEARCH & DEVELOPMENT ADMINISTRATION
OFFICE OF PUBLIC AFFAIRS ● TECHNICAL INFORMATION CENTER

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

This report has been reproduced directly from the best available copy.

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161

Price: Printed Copy	\$6.75 (domestic)
	\$9.25 (foreign)
Microfiche	\$2.25 (domestic)
	\$3.75 (foreign)

**FEASIBILITY INVESTIGATION OF THE
GIROMILL
FOR GENERATION OF ELECTRICAL
POWER**

Mid-Term Report for the
Period April - November 1975

R.V. Brulle

MCDONNELL AIRCRAFT COMPANY
Box 516
St. Louis, Missouri 63166

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Date Published - 1 November 1975

PREPARED FOR THE UNITED STATES
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
DIVISION OF SOLAR ENERGY

Under Contract No. E(11-1)-2617

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED *flg*

FOREWORD

This mid term study report was prepared for the United States Energy Research and Development Administration (ERDA) by the McDonnell Aircraft Company (MCAIR), a division of the McDonnell Douglas Corporation, P. O. Box 516, St. Louis, Missouri 63166. The study was performed under ERDA contract E(11-1)-2617. This mid term report covers the period from 1 April to 15 October 1975.

The study is under the direction of Robert V. Brulle. Contributing personnel are: William E. Simon, aerodynamics and performance; Thomas V. Hinkle, structural strength analysis; Rudy N. Yurkovich, structural dynamics; Bruno Fajfar, control system; Fred R. Cole, design; Robert A. Juergens, electronics; and James H. Carlson, costs.

TABLE OF CONTENTS

<u>Section</u>	<u>Page</u>
FOREWORD	i
SYMBOLS AND DEFINITIONS	vii
SUMMARY	ix
1. INTRODUCTION	1
2. STUDY GROUND RULES	3
2.1 Study Purpose	3
2.2 Design Philosophy	3
2.3 Selected Configurations	3
2.4 Giromill System Ground Rules	4
2.4.1 Site Ground Rules	4
2.4.2 System Ground Rules	4
2.4.3 Costing Ground Rules	7
3. AERODYNAMICS AND PERFORMANCE	8
3.1 Cyclogiro Vortex Theory	8
3.2 Cyclogiro Computer Program Idiosyncrasies	17
3.3 Aerodynamic Analysis	19
3.4 Performance Design Charts	23
3.5 Blade Rock Angle Analysis	32
3.6 Rotor Start Up Analysis	45
3.7 Operational Power Envelopes	48
4. GIROMILL SYSTEM DESIGN	55
4.1 Giromill Design Concepts	55
4.2 Rotor Blads	57
4.3 Generator Selection	60
4.4 Generator Drive System	60
4.5 Giromill General Arrangement Drawings	61
5. STRUCTURAL/STRENGTH DESIGN	65
5.1 Structural Arrangement	65
5.2 Structural Loads Environment	68
5.2.1 Operating Loads	68
5.2.2 Gust Loads	69
5.2.3 High Winds	69
5.3 Structural Design Criteria	69
5.4 Structural Design	74

TABLE OF CONTENTS (Cont.)

<u>Section</u>	<u>Page</u>
5.4.1 Blade Design	75
5.4.2 Blade Supports and Tower Design	77
5.5 Weight Trends	77
5.6 Structural Dynamics	80
5.6.1 Structural Dynamics Analysis Overview	80
5.6.2 Structural Dynamics Results	84
5.6.3 Conclusions	89
6. CONTROL SYSTEM	94
6.1 Functional Requirements	94
6.1.1 Start Up Operation	94
6.1.2 Normal Operation	97
6.1.3 Power Interrupt Condition	97
6.2 Hardware Requirements	98
6.2.1 Sensors	98
6.2.2 Blade Actuators	99
6.2.3 Power Supplies and Switching	102
6.2.4 Computing and Control Elements	102
6.3 Control System Integration	103
7. ENERGY OUTPUT EVALUATION	106
7.1 High Wind Speed Areas	106
7.2 Wind Duration Curves	108
7.3 System Operating Envelope	108
8. COST ANALYSIS	115
8.1 Basic Cost Elements	116
8.2 Production Costs for Giromill Subsystems	117
8.3 Total Giromill System Cost	129
9. CONCLUSIONS	138
10. REFERENCES	139

LIST OF ILLUSTRATIONS

<u>Figure</u>	<u>Page</u>
1. The Giromill	2
2. Summary of Study Configurations	5
3. Wind Velocity Duration Curves	6
4. Cyclogiro Rotor Angle Definitions	9
5. Cyclogiro Vortex Pattern	11
6. Flowfield Through the Giromill Rotor	14
7. Typical Giromill Characteristics	15
8. Cyclogiro Vortex Theory Check with Experimental Data	16
9. Cyclogiro Vortex Theory Effective Angle of Attack Input Profiles.	18
10. Airfoil Section Aerodynamics	21
11. Blade Surface Effect on Rotor Performance	22
12. Blade Angle of Attack Effect on Rotor Performance	22
13. Gust Effect on Angle of Attack	24
14. Lift Coefficient at High Angle of Attack	25
15. Drag Coefficient for High Angle of Attack	26
16. Effect of Rotor Aspect Ratio on Rotor Performance	27
17. Effect of Solidity on Rotor Performance	27
18. Effect of Wind Speed on Rotor Performance	29
19. Effect of the Number of Blades on Rotor Performance	29
20. Effect of Rotor Eccentricity on Rotor Performance	30
21. Effect of Rotor Flip Region on Rotor Performance	30
22. Design Chart Equations	31
23. Giromill Rotor Capture Area Required for Specific Power Output .	33
24. Giromill Sizing Chart - 120 kW Rated Power	34
25. Giromill Sizing Chart - 500 kW Rated Power	35
26. Giromill Sizing Chart - 1500 kW Rated Power	36
27. Blade Chord Variation with Rotor Solidity (σ) and Rotor Diameter - 2-Bladed Rotor	37
28. Blade Chord Variation with Rotor Solidity (σ) and Rotor Diameter - 3-Bladed Rotor	38
29. Blade Chord Variation with Rotor Solidity (σ) and Rotor Diameter - 4-Bladed Rotor	39
30. Optimum Rotor Advance Ratio for Maximum Power and Efficiency . .	40

LIST OF ILLUSTRATIONS (Cont.)

Figure	Page
31. Rotor Advance Ratio J , RPM, Diameter and V_w Relationship Nomogram	41
32. Blade Effective Angle of Attack to Maintain Constant Power and RPM	42
33. Blade Rocking Angle Profile to Maintain Constant Power and RPM .	43
34. Minimum V_R to Limit Blade Rock Angle to $\pm 50^\circ$	44
35. Giromill Start Up Blade Modulation Scheme	46
36. Starting Torque	47
37. Giromill Starting Conditions	49
38. Giromill Rotor Characterisitics	51
39. Operating Angle of Attack Profile	52
40. Giromill Efficiency for Low Wind Velocities	53
41. Giromill Power Curve	54
42. Giromill Concepts	56
43. Blade Concepts	58
44. 120 kW Giromill Baseline Blade Design	59
45. Baseline Generator Drive	62
46. General Arrangement 120 kW Giromill	63
47. General Arrangement 500 and 1500 kW Giromill	64
48. Giromill Structure	66
49. Typical Rock Angle History	70
50. Typical Design Loads - Configuration 2	71
51. Expected Blade Loads	76
52. Total Expected Tower Shear - Normal Operating Condition	78
53. Maximum Expected Tower Bending Moments	79
54. Giromill Substructure Weight 120 kW System	81
55. Giromill Structure Weight	82
56. Blade Flutter Velocity Restraint Stiffness	85
57. Tower and Blade Support Arm Dynamics	87
58. Tower and Blade Support Arm Dynamics	88
59. Giromill Component Resonant Frequencies	90
60. Giromill Component Resonant Frequencies	91
61. Giromill Component Resonant Frequencies	92
62. Giromill Component Resonant Frequencies	93

LIST OF ILLUSTRATIONS (Cont.)

<u>Figure</u>	<u>Page</u>
63. Giromill Control System Functional Diagram	95
64. Blade Rocking Angle Control	96
65. Blade Actuator Block Diagram	101
66. Control System DC Power Requirements	103
67. Giromill Control System Block Diagram	104
68. Mean Annual Wind Speeds (MPH)	107
69. Wind Speed Duration Curves for Two Values of Mean Wind Speed . . .	109
70. Estimation of Annual Energy Output	110
71. Comparison of Giromill Power Output/Wind Speed Relationship . . .	112
72. Specific Output Relationship	113
73. Calculation of Annual Giromill Energy Output	114
74. Cost of Capital Analysis for Typical Capital Structure	118
75. Cost Relationship for 100 kW Synchronous Generator	121
76. Cost Relationship for 500 kW Synchronous Generator	122
77. Cost Relationships for 100 kW and 500 kW Speed Increases	123
78. Cost Relationship for Flexible Coupling	124
79. Cost Relationship for Large Diameter Combination Bearing	125
80. Cost Relationship for Small Diameter Combination Bearings	126
81. Mechanical Power Transmission Components and Characteristics . . .	127
82. Estimated Costs for Rotor Blade Actuators	128
83. Preliminary Cost Estimating Relationships for 120 kW and 500 kW Giromill	130
84. Production Cost Elements for 120 kW Giromill Configurations . . .	131
85. Cost of Energy Produced for 120 kW Giromill Configurations	132
86. Production Cost Elements for 500 kW Giromill Configurations . . .	134
87. Cost of Energy Produced for 500 kW Giromill Configurations	135
88. Giromill Energy Cost Sensitivity for 120 kW and 500 kW Systems . .	136
89. Giromill Energy Cost Sensitivity for 120 kW and 500 kW Systems . .	137
90. Range of System Variables for Minimizing Energy Costs	133

SYMBOLS AND DEFINITIONS

A	Rotor capture area = $b \times D$
AR_R	Rotor aspect ratio = b/D
b	Rotor blade span
c	Rotor blade chord
c_d	Rotor blade section drag coefficient
c_l	Rotor blade section lift coefficient
C_D	Rotor blade total drag coefficient
C_L	Rotor blade total lift coefficient
CG	Center of gravity
cm	Centimeter
D	Rotor diameter
E	Total energy output
F_N	Total blade force normal to blade chord
F_X, F_Z	Tower shear loads
J	Rotor advance ratio = $\frac{V_w}{V_p}$
kg	Kilogram
M	Bending moment
m	Meters
N_B	Number of rotor blades
P_R	Power output at rated wind speed
R	Rotor radius
R	Blade radial force
t/c	Thickness ratio of airfoil section
T	Torque
T	Blade tangent force
V_R	Rated wind velocity
V_w	Wind velocity
V_{CI}	Cut-in wind velocity
V_{CO}	Cut-out wind velocity
V_p	Rotor peripheral velocity = ωR
W	Weight
α_e	Blade effective angle of attack
α_i	Induced angle of attack
β	Blade flap eccentricity

θ	Specific energy output
θ_R	Blade rock angle
λ	Blade speed ratio = $1/J$
$*\sigma$	Solidity = $N_B c / 2\pi R$
ψ	Blade phase angle
ω	Rotor rotational velocity

* Solidity now based on cylindrical area. A change to a solidity based on capture area ($\sigma = N_B c / 2R$) to be implemented shortly.

SUMMARY

The first 6 months of effort under the current ERDA contract has concentrated on the parametric feasibility analysis of Giromill configurations. It started with the establishment of a set of study ground rules chosen to facilitate comparison of the Giromill results with those obtained by GE and Kaman for conventional windmills. Some of the more principal ground rules are shown below.

STUDY GROUND RULES

- o Analyze 3 Giromill Systems
120, 500, and 1500 kw
- o Three Bladed Configurations
- o Constant Rotor RPM with Synchronous Generator
- o Withstand 60 MPH Wind Gusts Operating
- o Withstand 120 MPH Winds with Blades Feathered
- o Design Life:
 - Static Components - 50 years
 - Dynamic Components - 30 years

The cyclogiro computer program, obtained from Prof. H. C. Larsen of the United States Air Force Institute of Technology, was modified to incorporate computation of blade loads for the normal operating and gust loading conditions. The program was also changed to allow computation of the effects of smoothing the blade rock angles in the region where they experienced large oscillations due to passing through a vortex shed by the previous blade. Using this program the various effects of rotor geometric parameters were investigated; the significant results are presented in the next chart.

PERFORMANCE RESULTS

- o Optimum Operating Point - $\alpha = 9^\circ$ - End of Lift
Coefficient Linear Range
- o Power Output is Independent of Rotor Aspect Ratio
(AR_R)
- o RPM Varies Inversely with Rotor Diameter for Given
Power - High AR_R , Low RPM
- Low AR_R , High RPM
- o Maximum Rotor Efficiency is Independent of Number of
Blades and Solidity
- o Optimum RPM Varies Inversely with Solidity
- Low Solidity, High RPM
- High Solidity, Low RPM
- o Rotor Power is Proportional to Capture Area
- o For Given Power and Solidity RPM Varies Directly
with Rated Wind Speed
- High Rated Wind, High RPM
- Low Rated Wind, Low RPM
- o Operating Blade Air Loads on Similar Configurations
Scale Directly with Wind Velocity Squared and
Capture Area

Giromill configuration design concepts were explored. A baseline concept was adopted having an upper structural triangular tower extending through the lower support tower and supported by two main rotor bearings. The baseline blade concept consisted of a rolled sheet metal leading edge, a brake formed spar, beaded trailing edge skins and pressed sheet metal ribs. Other features of the baseline system are shown in the next chart.

Twenty-one different Giromill systems covering a power range of 120, 500 and 1500 kW were then synthesized. These were structurally analyzed and sized. Several of these systems were also evaluated from a structural dynamics viewpoint to assure structural sufficiency. The resulting structural weight of the various systems was then computed for use in the cost analysis.

An automatic electronic control concept built around existing equipment and employing state of the art techniques was developed. The control system provides the signals for starting and synchronizing the Giromill, keep it running normally, and shuts down the system when events, exceeding the capabilities of the Giromill are encountered. A simple method of computing the required blade rock angle was developed for normal operation. The rock angle

that gives a blade angle of attack of zero under idealized, no induced effects, is first computed. This rock angle is successively modified for: (a) the angle of attack to produce the desired power and rpm, and (b) the induced angle of attack effects. A feedback of the power generation condition provides an incremental correction to angle of attack and induced effects to keep the Giromill running smoothly. This control is functionally illustrated below.

Preliminary cost estimates for generating electrical power from the Giromill systems were completed. Cost estimating relationships of the major items of equipment were formulated. Vendor data, vendor extrapolations and McDonnell Douglas Producibility personnel were employed for this task.

Production costs based on existing manufacturing and construction techniques were used. Also estimated were the recurring costs such as cost of capital and operation and support costs. The resulting Giromill costs were then related to the energy produced to establish an energy cost for the system. A comparison of the energy cost of the best 120 and 500 kW Giromill (analysis of the 1500 kW Giromill has not been completed) with the GE and Kaman costs are shown below. It should be noted that the Giromill costs do not reflect a large amount of optimization to reduce costs. Even so they are certainly cost competitive.

The cost analysis has uncovered the important cost drivers. For example, the main rotor bearings for the present design are very costly. Several design changes are now being evaluated that reduce the bearing cost considerably. Optimizations of this sort will be conducted during the second half of the study in addition to a continued system analysis to arrive at a more optimum Giromill cost. In addition, a next phase test program will be prepared to investigate the critical Giromill feasibility areas.

1. INTRODUCTION

Wind energy has been used for centuries as a source of power on a small scale. To become a significant contribution to the nation's energy needs, wind energy extraction devices will have to be developed on a scale that would entail a very large capital investment. The most effective system must be identified before such a commitment can be undertaken.

One such device that offers considerable promise is the Giromill, shown in Figure 1. It consists of a number of vertical blades rotating around a central tower. The blade angles of attack are individually modulated to achieve high conversion efficiency regardless of the wind direction. Compared to conventional windmills, tower and blade construction are considerably simplified.

The current study concentrates on identifying the potential advantages and problem areas of the Giromill, for comparison to more conventional systems being studied elsewhere. By theoretical analysis and design studies, the various geometric factors will be established, structural concepts developed, existing subsystems identified, cost evaluated and a test program developed. This report presents the results of the first 6 months of this 1 year study.

Figure 1
The Giromill

2. STUDY GROUND RULES

2.1 Study Purpose

The purpose of this present study is to theoretically determine the feasibility and cost effectiveness of the Giromill. Can it, because of its high efficiency and simplified construction, be made more cost effective than conventional windmills? What items are the major cost drivers and why? The answers to these questions are required so that the standing of the Giromill within the stable of wind energy conversion systems can be determined.

2.2 Design Philosophy

The design techniques employed are all state of art. No exotic design features were used even though they may appear to enhance the Giromill. Where possible, off-the-shelf equipment was specified. The General Electric and Kaman windmill studies presently in progress were used as a guide for defining the design techniques (and ground rules) used in our study. The Giromill designs shown are therefore considered to be producible at this time.

A consistent design philosophy was adhered to during the parametric portion of the study just completed and documented in this mid term report. The design techniques of the major cost drivers will be evaluated and optimized during the next portion of the study.

2.3 Selected Configurations

Three categories of Giromill systems are being parametrically analyzed in this feasibility study. These categories are: the 120 kW system, the 500 kW system, and the 1500 kW system. Seven configurations were defined for the 120 kW system and nine configurations were defined for the 500 kW system with systematic variations in rotor aspect ratio, rotor solidity, and rated velocity. Preliminary analyses had indicated that the lowest cost configuration would probably have a minimum number of blades. Configurations having 2, 3 and 4 blades had been considered. This very preliminary analysis pointed out that a 3 bladed configuration had several advantages such as:

- (1) result in minimum difficulty in the self starting procedure, (2) assure that positive torque is being produced at all points on the rotor orbit, and (3) result in minimum vibration loads due to symmetry of the polar moment of inertia of the rotor about its axis.

On the basis of these preliminary analyses a 3 bladed configuration was selected for the 120 and 500 kW systems to be analyzed in this study. This preliminary analysis also indicated that the 1500 kW system, because of its greater loads, may be more cost effective with more blades. To investigate this factor it was decided to vary the number of blades on the 1500 kW systems. However, to keep the study within bounds only five 1500 kW systems were defined varying only solidity and number of blades. Figure 2 presents a summary of the geometric characteristics of these selected study configurations.

2.4 Giromill System Ground Rules

The ground rules for the Giromill system were defined to be as consistent as possible with those established for the horizontal axis systems described in References 1 and 2. These fall into three categories: site, system, and cost. The following sections describe the ground rules in each category.

2.4.1 Site Ground Rules - The site criteria are primarily concerned with the wind conditions that the Giromill system is subjected to. These wind conditions are:

- (a) The mean wind velocity ($\bar{V}$) at the site is 5.4 m/sec (12 mph) for the 120 and the 500 kW systems and 8.1 m/sec (18 mph) for the 1500 kW system.
- (b) The velocity duration curves for the above mean velocities are presented in Figure 3.
- (c) The rated velocity (V_R) for the Giromill is defined as the minimum velocity at which rated power is generated.
- (d) The mean velocity and the rated velocity are defined at an altitude of 9 m (30 ft). The Giromill operational wind velocity is assumed to be at an altitude of 30 m (100 ft) and is defined using the seventh power velocity distribution law.
- (e) A minimum 15 m (50 ft) ground clearance is used.

2.4.2 System Ground Rules

- (a) The design life of the static components is fifty years. The design life for the dynamic components is thirty years including periodic maintenance and replacement if cost effective.
- (b) The rotor blades must withstand winds that gust up to 60 mph. The blade load condition for the gust is determined for the blade orientation consistent with the rated velocity.

CONFIGURATION NUMBER	RATED POWER kW	RATED VELOCITY mps (mph)	NUMBER OF BLADES	ROTOR SOLIDITY	ROTOR ASPECT RATIO	ROTOR DIAMETER meters (feet)	ROTOR SPAN meters (feet)	ROTOR BLADE CHORD meters (feet)
1	120	8.05 (18)	3	.05050	1.073	20.4 (67)	21.9 (71.8)	1.08 (3.5)
2	120	8.05 (18)	3	.06313	1.073	20.4 (67)	21.9 (71.8)	1.35 (4.4)
3	120	8.05 (18)	3	.07575	1.073	20.4 (67)	21.9 (71.8)	1.62 (5.3)
4	120	8.05 (18)	3	.06313	0.746	24.5 (80.3)	18.3 (59.9)	1.62 (5.3)
5	120	8.05 (18)	3	.06313	0.477	30.6(100.4)	14.6 (47.9)	2.02 (6.6)
6	120	7.15 (16)	3	.06313	1.073	24.3 (80)	26.1 (85.6)	1.61 (5.3)
7	120	8.94 (20)	3	.06313	1.073	17.4 (57)	18.7 (61.3)	1.15 (3.8)
8	500	8.05 (18)	3	.02525	1.073	43 (141)	46.2(151.5)	1.14 (3.7)
9	500	8.05 (18)	3	.03787	1.073	43 (141)	46.2(151.5)	1.71 (5.6)
10	500	8.05 (18)	3	.0505	1.073	43 (141)	46.2(151.5)	2.26 (7.4)
11	500	8.05 (18)	3	.02525	0.477	64.6(211.8)	30.8(101)	1.71 (5.6)
12	500	8.05 (18)	3	.03787	0.614	56.9(186.6)	34.9(114.6)	2.26 (7.4)
13	500	7.15 (16)	3	.03787	1.073	51.3(168.2)	55 (180.5)	2.04 (6.7)
14	500	8.94 (20)	3	.03787	1.073	36.7(120.5)	39.4(129.3)	1.46 (4.8)
15	500	8.05 (18)	3	.07575	1.073	43 (141.2)	46.2(151.5)	3.41(11.2)
16	500	8.05 (18)	3	.03787	0.85	48.3(158.6)	40.8(134.9)	1.92 (6.29)
17	1500	11.6 (26)	3	.0505	1.073	43 (141.2)	46.2(151.5)	2.26 (7.4)
18	1500	11.6 (26)	3	.07575	1.073	43 (141.2)	46.2(151.5)	3.41(11.2)
19	1500	11.6 (26)	3	.101	1.073	43 (141.2)	46.2(151.5)	(4.54)(14.9)
20	1500	11.6 (26)	4	.07575	1.073	43 (141.2)	46.2(151.5)	(2.56)(8.4)
21	1500	11.6 (26)	5	.07575	1.073	43 (141.2)	46.2(151.5)	2.04 (6.7)

Figure 2
Summary of Study Configurations

Figure 3
Wind Velocity Duration Curves

- (c) The rotor blades must withstand the loading condition that is developed when the blades are suddenly flipped to a 90 degree rock angle caused by a control failure.
- (d) The rotor blades are designed to withstand a two percent rotor overspeed at rated velocity conditions.
- (e) The tower structure must withstand winds up to 120 mph with the rotor blades in a "weather vane" position.
- (f) The structure must withstand 2.54 cm (1 inch) diameter hail stones.
- (g) The Giromill is designed for constant rpm throughout its operational range.
- (h) The highest efficiency rpm at rated velocity is used.
- (i) The rotor blade modulation scheme is designed to provide a constant magnitude angle of attack in both the plus and minus senses.
- (j) The rotor blade pivot and CG are at the twenty-five percent chord point.
- (k) The basic generator is a constant rpm synchronous generator with an assumed efficiency of 95%.
- (l) The rpm speed increaser is a gear type with an assumed efficiency of 96%.
- (m) The Giromill power rating is given as the output of the generator.

It should be noted that the efficiencies indicated under item (l) and (m) are consistent with the efficiencies of References 1 and 2. The cost sensitivities for increasing rotor capture area for variation of generator and speed increases will be developed for the Giromill system.

2.4.3 Costing Ground Rules

- (a) Constant 1975 dollars are used.
- (b) Giromill system costs include manufacturing costs, capital costs, site acquisition costs, and maintenance costs.
- (c) Costs are based on a production of one hundred units.
- (d) Costs are based on a three year production schedule.

3. AERODYNAMICS AND PERFORMANCE

The fundamental problem in determining the Giromill performance is finding the magnitude and direction of the local velocity at all points on the orbit so that the blade forces can be computed. Consider a single rotor blade traveling on the orbit starting at a blade phase angle $\psi = 0^\circ$ (see Figure 4). Between $\psi = 0$ and ψ near 90° , the blade is modulated to maintain a constant negative angle of attack. In the $\psi = 90^\circ$ region, the blade is flipped to a positive angle of attack. This blade flipping nominally occurs over a 30° blade phase angle travel and results in a vortex being shed from the blades which travels downstream at the local velocity. From $\psi = 90^\circ$ to the $\psi = 270^\circ$ region, the rotor blade angle of attack is maintained at a constant positive value. In the $\psi = 270^\circ$ region, the blade is flipped back from a positive angle of attack to a negative angle of attack resulting in another vortex being shed. This vortex and the previous vortex are connected by the free vortices emanating from the tips of the blade. The process is repeated as the blade travels from the $\psi = 270^\circ$ region to the $\psi = 90^\circ$ region. These vortices from each blade create substantial induced velocities and induced angles of attack at all points on the orbit. The resulting flow field through the rotor system does not lend itself to a simple analysis in finding the local resulting velocity magnitude and direction. However, a cyclogiro aircraft theory and computer program were available that were subsequently modified for computing the Giromill performance. This theory and program are described in Section 3.1.

With the computations of the velocity magnitude and direction along with the aerodynamic lift and drag characteristics of the rotor blade, the resulting forces on the blade are computed in both the operational mode and under wind gusting conditions. Based on these results the blade design and strength requirements are determined and the performance of the rotor and the resulting power output are determined. In addition, the required blade modulation to maintain the prescribed angle of attack is also obtained. Also, the effect on rotor performance for variations in various geometric parameters are computed utilizing the performance computer program.

3.1 Cyclogiro Vortex Theory

A cyclogiro vortex theory and computer program was developed by Prof. Harold C. Larsen of the U.S. Air Force Institute of Technology. The theory and program will be published in USAF Institute of Technology Research

NOTE: SOME ANGLES SHOWN NEGATIVE
TO BE CONSISTANT WITH NORMAL
OPERATING CONDITIONS SELECTED
FOR GIROMILL OPERATION

Figure 4
Cyclogiro Rotor Angle Definitions

Technical Report 72-2. This section briefly explains this theory and computer program.

The cyclogiro rotor, embodied in the Giromill, has been theoretically studied and experimentally tested by many investigators. The experimental results did not verify the theoretical results, blamed primarily on the non steady aerodynamic forces created by the rotor. To correct these deficiencies, a cyclogiro vortex theory model was utilized as Prandtl had done for wing theory and Goldstein-Theodorsen for propeller theory. Complete integrals were written for the flow motion and solved using a digital computer. The theory accounts for the induced velocities due to the forces produced by the blades, and the vortex wake shed by the blades. The spacing and orientation of the wake is determined from continuity and momentum considerations of flow through the cyclogiro rotor capture area by an iteration scheme which computes the local induced velocity at each point on the blade orbit due to the wake, and corrects the local angle of attack at each blade. A solution is obtained when the momentum flux through the capture area induced by the rotor and wake agrees with momentum flux calculated from the forces on the blade. This is accomplished by adjusting the strength of the concentrated vorticity in the wake and the rotor.

The cyclogiro vortex system can be satisfactorily approximated by the configuration illustrated in Figure 5. The blades are not shown, but are represented by bound vortices such as $BV-BV'$. The blades rotate in a circle about the axis of rotation. In order for the blades to produce a useful force, they must alternately change from positive to negative and back from negative to positive angle of attack each time the blade passes the end of the "capture area" and reaches the "blade flip point." The angle β is referred to as the eccentricity.

Each time the angle of attack changes there must be a starting and stopping vortex, or vortex ring, to satisfy the Helmholtz vortex law. This poses problems, some of which were answered by Prandtl. Vorticity is continually being generated in the boundary layer on the surface of a body immersed in fluid, and is continuously shed into the flow as "free" vortices which "float" or "swim" in the fluid. Their actual motion depends on the mutual interference among all the vorticity in the field. The von Karman Vortex Street is the classical two dimensional picture of this continuous generation

Figure 5
Cyclogiro Vortex Pattern

of a trailing vortex array or system. In the three dimensional system, this array is a series of free vortex rings downstream from the point of generation.

In the cyclogiro, this point may be taken as the flip point, since at that point, the direction of the circulation must be reversed. This means that as the bound vortex is reversed in sense, a free vortex must be generated at the trailing edge of the blade that has twice the magnitude and opposite rotation of the bound vortex. This means that, in the spanwise direction, free vortices appear periodically at the front and rear flip points, and are "washed" downstream by the local flow velocity. These vortices must be joined to the bound vortex which generates them by a continuous path so that a series or array of free vortex rings is formed as illustrated in Figure 5. The exact strength and location depends upon the number of blades, whether odd or even, the free stream velocity, and the blade forces being developed.

These rings probably ultimately form closed curves unless broken up by free stream turbulence. However, near the rotor they are essentially rectangular as shown. For simplicity, they have been taken as rectangular since the Biot-Savart law can be applied easily to straight line segments.

If the geometry (location, orientation, spacing, etc.) is known for the array, as well as the strength of each wing, the induced velocity at any point in the flow associated with each straight line segment of the array can be computed by summing the effects of each segment. Unfortunately, the array extends for very great distances, theoretically to infinity. But, since the induced velocity decreases inversely as the normal distance from a straight line segment to the point in question, the induced velocity dies out rapidly at points far removed from the vortex. It was found that line segments located more than 5 rotor diameters from any point on the rotor orbit contributed less than 0.1% of the total induced velocity, depending in part on the orientation of the segment. The spanwise segments were most influential. Thus, a ring array extending 5 diameters downstream from the rotor was taken as a boundary.

The vortex wing array is free to swim or float in the fluid. Thus, it is carried downstream with the local stream velocity. If the local stream velocity could be determined, the location of the array could be determined as well as the spacing of the array. This was accomplished by considering continuity and momentum in the horizontal and vertical direction, and devising an iteration scheme to converge on the final position.

The strength of the bound vortex is related to the force being developed. The total change of momentum in the horizontal and vertical direction is equal to the force creating that change of momentum. In the convergence procedure, a uniform flow is assumed in each direction to find a resultant uniform velocity change. Then, using this value of induced velocity around the orbit as a start and the proper value of circulation for the lift being generated, the first correction to the velocity on the orbit is calculated from the ring array, using the spacing based on the uniform flow assumed. This is iterated until continuity and momentum are satisfied by integrating conditions across the capture area. It is repeated until the integrated value is less than 1% in error from values computed from the momentum theory. This fixes the array spacing and the velocity induced on the orbit. Once the correct value of the induced velocity is known, the correct value of the angle of attack can be determined, and the forces on the blade computed. It is necessary to include the mutual interference effects of the bound vortices and the end joining vortices, as this has a significant effect.

Once the force on the blade is known (magnitude and direction) at an orbit point, it can be resolved into lift, drag, and torque components. These are then integrated around the orbit to obtain mean values for one revolution. These can be reduced to coefficient form for general application.

To illustrate the results obtained from this theory, Figure 6 shows the flow field through the rotor of a Giromill at the peak power development condition. It is seen that the flow is affected far from the rotor and has a low velocity in the wake resulting from the energy extraction. Note also the increase of velocity outside the wake which is demanded by flow continuity.

Figure 7 illustrates the energy extraction efficiencies, torque and power at an effective angle of attack of 9° as a function of the blade speed ratio λ . The torque curve is quite flat and peaks slightly before the power curve.

Figure 8 shows the results of applying the theory to a cyclogiro rotor that was tested by the NACA in 1935, and reported in NACA TN 528. The cyclogiro rotor tested was 4 bladed and had a diameter and span of 8 ft. The blades were driven in a sinusoidal blade rocking motion by an eccentric cam. The theoretical program cannot be applied directly to this condition as explained in the next section. The theoretical results plotted in Figure 8 were obtained by

Figure 6
Flowfield Through the Giromill Rotor

Figure 7
Typical Giromill Characteristics

- CURVES EXTRACTED FROM NACA TN 528, FIGURE 5.
- SYMBOLS ARE CYCLOGIRO THEORY CHECK POINTS

Figure 8
Cyclogiro Vortex Theory Check with Experimental Data

approximating the blade rocking with a sinusoidal angle of attack modulation, and then correcting the induced angle of attack effects to provide the proper rocking angle. Evaluation of these results and comparisons with conventional windmill results appears to validate the theoretical approach. Further verification must await more definitive test results.

3.2 Cyclogiro Computer Program Idiosyncrasies

The computer program implementing the cyclogiro vortex theory explained in the previous section was utilized to obtain the Giromill performance. During the running of this program several drawbacks of the theory and program were uncovered.

The program is now limited to an effective angle of attack α_e input. This can be in the form of a sinusoidal wave, or as a constant $\pm$ value with the blade flip from a positive to a negative (and vice versa) occurring as the blade passes through the specified eccentricity β (and $\beta + 180^\circ$). The program then computes the blade rocking angle θ_R , measured from the rotor tangent to the blade chord. These relationships are shown in Figure 4 and 9. The θ_R thus computed results in the blade having the input α_e accounting for the induced effects of the vortex pattern. When the blade passes near a vortex, θ_R is required to oscillate rapidly to maintain α_e . These rapid oscillations of θ_R will be smoothed in an actual Giromill. The program was modified so that a hand smoothing of the oscillatory θ_R can be input and the rotor performance then recomputed. The computation assumes that the total velocity vector is unchanged and that the changes of θ_R due to the smoothing changes only the angle of attack - not any of the induced velocities. The program should be modified so that the change in induced effects can also be accounted for. The effects of this θ_R smoothing on the Giromill system is covered in Section 5.2.1.

Another drawback of the program is its inability to converge to a solution at high blade speed ratios. This happens shortly after the peak power has been reached. The reason for this solution degeneration has not been firmly established, but it appears to be related to the assumed vortex pattern in the wake. An increase in blade speed ratio λ is accompanied by a rapid decrease in the wake vortex ring spacing, which causes the program to use more vorticies in the solution. The result is that the rotor exhibits a flow field state somewhat akin to the ring vortex state of a propeller. Prof. Larsen has analyzed this phenomenon and has reached the conclusion that this flow field state can actually exist. He feels, however, that a more accurate representation

Figure 9

Cyclogiro Vortex Theory Effective Angle of Attack Input Profiles

of how the vortex pattern rolls up in the wake would help to alleviate this problem and allow calculation at higher blade speed ratios. For the performance calculations at the high blade speed ratios, estimates of the Giromill power curve based on windmill type characteristics were made. This is discussed in Section 3.7.

3.3 Aerodynamic Analysis

One of the items required for utilizing the performance computer program is the lift and the drag of the rotor blades. In this case, the rotor blade can be considered a wing traveling a circular orbit. Since the blades must operate in the positive and the negative angle of attack range, the airfoil section used is symmetric. Cambering the airfoil can be a help in increasing the torque forces on half the orbit. However, whether it would offset the deleterious effect on the other half has not been investigated. For similar reasons, the rotor blades are untwisted. The aspect ratio of a rotor blade taken as a wing is very high (on the order of 15) and it does not appear that tapering the blades would be an improvement. Therefore, the rotor blades are rectangular, untwisted, and uncambered wings. This type of blade also lends itself to simple and low cost construction.

As explained in the previous section, the cyclogiro computer program computes the induced effects due to the rotor and wake vortex systems. These induced effects are three-dimensional, i.e., they vary along the diameter and span of the rotor. For expediency, in this feasibility analysis, it has been assumed that the span wise variation of induced effects is small and that the center of the blade value is representative of what occurs over the entire blade. This assumption appears valid in view of the large aspect ratio blades used. It also simplifies the analysis since it implies a constant blade loading.

The two-dimensional wing section lift and drag for a 64₂-015 airfoil with "standard roughness" surface condition from Reference 4 was used in this analysis. Standard roughness was obtained by spreading 0.28 mm (.011 inch) grain carborundum to cover from five to ten percent of the area from the leading edge to eight percent of the chord for a wing section having a 0.61 m (2 ft) chord. This roughness is more severe than that caused by usual manufacturing irregularities or deterioration in service. The 64₂-015 aerodynamics was used because it did have a relatively high maximum lift coefficient and did not have a sudden stall. Also, the aerodynamic center is near 27% chord,

allowing the blade CG and pivot point to be located at 25% chord and still leave enough stability to allow the blade to weathervane into the wind when declutched from its actuator. Figure 10 presents the aerodynamics for the rough airfoil section.

Figure 10 also presents the smooth aerodynamics for a NACA 0015 airfoil taken from References 5 and 6. Figure 11 shows the Giromill power output using these two different airfoils. Assuming that the only difference between airfoils is the smoothness effect, Figure 11 shows that the power output for the smooth case is about 5% higher than the rough surface condition case with about a 5% increase in rotor rpm. The nine degree angle of attack is the optimum (max power) angle of attack for the rough surface condition as shown in Figure 12. Similarly (although not shown) the smooth surface condition has an optimum angle of attack of twelve degrees. Figure 12 shows that there is only a small difference in optimum power output at angles of attack near optimum, but that there is a significant rotor rpm difference as indicated by the blade speed ratio shift. The impact of the rotor rpm must be determined by analyzing the cost of the Giromill system. High rotor rpm requires a less costly speed increase between the rotor and the generator. However, the high rotor rpm results in a higher centrifugal loading on the blades, thereby increasing the total system cost. For the purpose of this feasibility study, the aerodynamics of the 642-015 airfoil with a standard roughness surface condition was used. Furthermore, it was assumed that these aerodynamics were valid for section thickness ratios of 15 to 20%.

To complete the aerodynamic requirements for the performance computer program, it was necessary to have the lift and drag coefficients for all angles of attack up to one hundred and eighty degrees. This was necessary in order to analyze the gust forces on the rotor blades. The gust loads were determined as follows:

- (1) The blade orientation relative to the flight path (rock angle history) was assumed to be consistent with rated conditions.
- (2) The wind velocity was suddenly increased from rated velocity to 27 meters/sec (60 miles per hour) with no change in blade orientation.
- (3) At all points on the trajectory new velocity vectors were computed both in magnitude and direction.
- (4) With the high angle of attack lift and drag coefficient, new blade forces were computed.

Figure 10
Airfoil Section Aerodynamics

Figure 11
Blade Surface Effect on Rotor Performance

Figure 12
Blade Angle of Attack Effect on Rotor Performance

Figure 13 shows the effect of the gust velocity on the angle of attack of the rotor blades.

There is a decided lack of experimental data for the aerodynamics of wing sections at large angles of attack. Figures 14 and 15 present the data we used in our analysis.

3.4 Performance Design Charts

Initially a study was conducted to determine the effect of various rotor parameters on the Giromill performance. These parameters include rotor aspect ratio, AR_R , rotor solidity, σ , operational wind velocity, V_w , number of blades, N_B , eccentricity, β , and flip region blade phase angle.

The first four parameters were investigated using the smooth surface aerodynamics of Figure 10 with an effective angle of attack of $\pm 12^\circ$. The rotor capture area was 1204 m^2 ($12,980 \text{ ft}^2$) and the operational wind speed was 6.7 mps (15 mph). For rotor aspect ratios of 0.25, 1.0, 1.298, 1.7, 2, and 4, the rotor performance was computed for a range of blade speed ratios. These results were plotted and the blade speed ratio for maximum power output was defined. Figure 16 presents a plot of the results. As can be seen from this plot, the maximum power output, efficiency, and blade speed ratio at maximum power are generally unaffected by the rotor aspect ratio.

To ascertain the effect of rotor solidity on performance, a similar procedure was used. For solidities of 0.01471, 0.02943, 0.05883, and 0.06313, the blade speed ratio λ for maximum power output was determined. These results are plotted in Figure 17 and show that the efficiency and maximum power output are not significantly affected by the rotor solidity. However, the blade speed ratio for maximum power is an almost linear function of the solidity. This implies, for a given Giromill capture area, that an increase in solidity will be accompanied by a corresponding decrease in maximum power rpm.

For the determination of the operational wind velocity effect, computations were made for velocities of 6.7 mps (15 mph), 8.9 mps (20 mph), 11.2 mps (25 mph) and 13.4 mps (30 mph). However, since a constant power output was desired for the determination of the velocity effect, it was necessary to change the capture area for each velocity. This was done by scaling the capture area as the inverse of the cube of the wind velocity. The capture area as a function of velocity is presented in Figure 18. Also presented in this

Figure 13
 Gust Effect on Angle of Attack

Figure 14
Lift Coefficient at High Angle of Attack

Figure 15
Drag Coefficient for High Angle of Attack

Figure 16
Effect of Rotor Aspect Ratio
on Rotor Performance

Figure 17
Effect of Solidity on Rotor Performance

Figure are the blade speed ratio for maximum power output and the rotor efficiency as a function of the operational wind velocity. As can be seen from these plots the optimum blade speed ratio and the efficiency are weak functions of the velocity.

Performance data were also computed for rotors with two, three, four and five blades for a constant solidity. These data are presented in Figure 19. Again, rotor performance is a weak function of the number of blades.

For determining the effects of rotor eccentricity and the blade phase angle flip region, the rough surface aerodynamics of Figure 10 with an angle of attack of nine degrees was used along with an operational velocity of 6.37 mps (14.3 mph) and a capture area Giromill of 1508 m^2 ($16,236 \text{ ft}^2$). Computations were made for eccentricities ranging from a minus eighty degrees to a minus one hundred degrees. Figure 20 presents the results of this analysis. It is evident that rotor efficiency and power output are not significantly affected by rotor eccentricity. Computations were made for variations in the blade flip region of 15, 30, and 45 degrees. These results are presented in Figure 21 and show that the power output and the efficiency decrease slightly as the blade flip region is increased.

It has been shown that rotor performance is relatively independent of rotor aspect ratio, rotor solidity, operational velocity, number of blades, eccentricity, and blade flip region. Also, that the power output is a direct function of the capture area and that rotor rpm (through the blade speed ratio) is a function of solidity as shown in Figure 17. Design charts were prepared relating the Giromill size (span and diameter), blade chord, rotor solidity, blade speed ratio and rotor rpm. These charts were used to synthesize the various Giromill configurations for cost analysis.

In addition to the above relations, several other assumptions had to be made as detailed below:

- (a) Rotor wind energy conversion efficiency is 54%.
- (b) Speed increaser and generator have a combined efficiency of 91%.
- (c) Rough surface aerodynamics are assumed.
- (d) Blade angle of attack is 9° .

These assumptions, with the previously established relations, allowed generation of a set of equations from which the Giromill design charts could be constructed. The equations are shown in Figure 22.

Figure 18
Effect of Wind Speed on Rotor Performance

Figure 19
Effect of the Number of Blades on
Rotor Performance

Figure 20
Effect of Rotor Eccentricity on Rotor
Performance

Figure 21
Effect of Rotor Flip Region on Rotor
Performance

$$A = \frac{P_{(kW)} 550}{.746 \rho / 2 V_w^3 \eta}$$

A = Rotor capture area (Span x Diameter)

$P_{(kW)}$ = Wind energy power to be extracted. This is the rated power divided by mechanical and electrical efficiency (.91).

550 and .746 constants relating to ft/lbs/sec and kW

ρ = Air density. Sea level value assumed.

V_w = Wind Speed

η = Giromill wind energy conversion efficiency (.54)

$$D = \sqrt{\frac{A}{AR_R}}$$

D = Rotor diameter

AR_R = Rotor aspect ratio (span/diameter)

$$b = \frac{A}{D}$$

b = Rotor span

$$C = \frac{\sigma \pi D}{N_B}$$

C = Blade chord

σ = Rotor solidity

N_B = Number of rotor blades

$$J = \frac{V_w}{V_p}$$

J = Rotor advance ratio

V_w = Wind speed

V_p = Rotor peripheral speed = $\frac{\text{rpm}}{60} \times \pi \times D$

Figure 22
Design Chart Equations

Figure 23 presents the rotor capture area as a function of the rated velocity for rated powers of 120, 500, and 1500 kW. Figures 24, 25, and 26 present the relationship between rotor span and rotor diameter as a function of the rotor aspect ratio and the rated velocity for rated powers of 120, 500, and 1500 kW systems. Figures 27 through 29 present the blade chord - rotor diameter relationship as a function of the blade solidity for two, three, and four bladed rotors.

Figure 30 presents the advance ratio (J) as a function of rotor solidity. These data were computed utilizing the performance computer program. For these calculations the rough surface aerodynamics of an effective angle of attack of 9° was used.

Figure 31 is a nomograph relating the blade speed ratio, the rotor diameter, the rotor rpm, and the rated velocity for the Giromill system. Utilizing the plots presented in Figures 23 through 31, the size of the rotor and the rotor blades along with the rotor rpm can be obtained for any 120, 500, or 1500 kW Giromill system.

3.5 Blade Rock Angle Analysis

An analysis of the blade rock angle to maintain constant power and rpm as the wind velocity (V_w) is increased beyond the rated wind velocity (V_R), and how this then relates to the cutoff wind speed for a specified blade rock angle limit, has been completed. This analysis shows that using a cutoff wind velocity (V_{CO}) of 13.4 m/sec (30 mph) and 20.1 m/sec (45 mph) for Giromills designed for a mean wind ($\bar{V}$) of 5.4 and 8.0 m/sec (12 and 18 mph) respectively, limits the operating blade rock angle to ± 50 deg. This greatly simplifies the blade control. Using these low cutoff wind velocities does not appreciably affect wind power production since these wind velocities are exceeded less than 100 hours in a year (see Figure 3)

Figure 32 shows a typical variation of effective angle of attack, α_e , to maintain constant power and rpm over a range of wind velocity ratios. The blade rock angles to achieve the α_e shown in Figure 32 are shown in Figure 33 at several velocity ratios. The most noticeable feature in this plot is the change in the character of the rock angle as the blade speed ratio, λ , approaches one. At that point the rock angle goes from a low pitch motion ($\lambda > 1$) to a high pitch motion. These two different types of motion make more difficult the control system task for commanding a rock angle profile. To

Figure 23

Giromill Rotor Capture Area Required for Specific Power Output

Figure 24
Giromill Sizing Chart - 120 kW Rated Power

Figure 25
 Giromill Sizing Chart - 500 kW Rated Power

Figure 26
 Giromill Sizing Chart - 1500 kW Rated Power

Figure 27

Blade Chord Variation with Rotor Solidity (σ)
and Rotor Diameter - 2-Bladed Rotor

Figure 28
Blade Chord Variation with Rotor Solidity (σ)
and Rotor Diameter - 3-Bladed Rotor

Figure 29

Blade Chord Variation with Rotor Solidity (σ)
and Rotor Diameter - 4-Bladed Rotor

Figure 30
Optimum Rotor Advance Ratio for Maximum Power and Efficiency

Figure 31

Rotor Advance Ratio J , RPM, Diameter
and V_w Relationship Nomogram

Figure 32
Blade Effective Angle of Attack to Maintain Constant Power and RPM

Figure 33
Blade Rocking Angle Profile to Maintain Constant Power and RPM

simplify this task it would be desirable to limit the rock angle to about $\pm 50^\circ$; keeping the blade modulation scheme in the low pitch region. Cross plotting, from Figure 33, the blade rock angle at $\psi = 60^\circ$ (a region of high blade rock angle values) versus velocity ratio V_w/V_R , shows that if the V_w/V_R is restricted to values less than 1.94 the blades will not have to be modulated greater than $\pm 50^\circ$. This relates to a minimum rated velocity $V_{R\text{Min}}$ if a maximum cut-off velocity V_{CO} is selected; i.e., $V_{CO}/V_R < 1.94$. Figure 34 shows the $V_{R\text{Min}}$ for the two mean winds used in this study and assuming the V_{CO} shown. These $V_{R\text{Min}}$ are less than what appears to be an optimum V_R for the Giromill. Hence, a V_{CO} of 13.4 mps (30 MPH) will be used for Giromills designed for a $\bar{V}$ of 5.4 mps (12 MPH), and a V_{CO} of 20.1 mps (45 MPH) used when a $\bar{V}$ of 8.0 mps (18 MPH) is selected.

Mean Wind Velocity $\bar{V}$ mps (MPH)	Cutoff Velocity (V_{CO}) mps (MPH)	Min. Allowable ($V_{R\text{Min}}$) mps (MPH)
5.4 (12)	13.4 (30)	6.19 (13.9)
8.0 (18)	20.1 (45)	9.3 (20.8)

Figure 34
Minimum V_R to Limit Blade Rock Angle to $\pm 50^\circ$

An analysis of the maximum blade flip rate was also performed. Blade flip actuator analysis indicated that limiting the rate to 60° deg/sec was a highly desirable goal. A limiting blade flip rate of 60° deg/sec means that between a $\lambda = 1.25$ and $\lambda = 1.9$, the blade cannot keep up with the optimum commanded value. This is illustrated in Figure 33. This does not cause any problems. Since this only occurs when the wind velocity is greater than required for rated power, a small α_e is needed. The off optimum blade rocking power variation can easily be compensated. For the case shown in Figure 33 the off optimum blade rocking actually increased the power output by 8 kW, and hence, only a small correction in α_e (and consequently θ_R) over the remainder of the phase angle travel would be required to get the power back to rated. Also, the large blade flipping occurs at a point when the blade is traveling back with the wind, and is experiencing a low dynamic pressure. Theoretically, at $\lambda = 1$ the blade dynamic pressure is 0 at $\psi = 90^\circ$.

A limiting blade flip rate of 60° deg/sec appears adequate to drive the Giromill blade under operating conditions.

3.6 Rotor Start Up Analysis

An analysis was conducted to investigate Giromill start up techniques and characteristics. It was determined that a step wise blade modulation scheme varying as a function of rpm will be required to assure that the Giromill will start.

Several schemes for starting the Giromill have been analyzed. The simplest way of starting would be to modulate the blades to the steady state rocking angle value for whatever wind is blowing at the time of start up. Preliminary analyses indicated that this scheme may work so a more detailed analysis was conducted. The results of this more detailed analysis were inconclusive. Positive starting torques were computed at all the discrete start up rpm values looked at, however some were quite low. Since a continuous and elaborate type of analysis would have to be done to confirm this technique, it was decided to drop it at this time and propose a slightly more complicated but positive starting scheme.

The fastest and most assured method of starting the Giromill would be to modulate the blades as a function of wind velocity and starting rpm to produce the maximum torque. This would require an extensive control system capability to implement. A simplification that modulates the blades to several discrete rocking angle profiles as a function of start up blade speed ratio λ appears feasible. This method was analyzed and the results show it will work.

Figure 35 presents a series of blade modulation curves needed to get an effective angle of attack, α_e , of 9° at the λ values shown. These curves were used as the blade modulation scheme for the range of start up λ values shown in the blocked off tabulation in the upper right of Figure 35. The resulting torque computations at several discrete start up λ values are shown in Figure 36. The range of applicability of the various modulation curves was not optimized for maximum total torque. Note that this scheme violates the criteria of maintaining the rock angle within $\pm 50^\circ$ as discussed in the previous section. This implies that a separate start up scheme mode will have to be implemented in the control system. See Section 6 for further discussion.

An additional calculation was made to determine the starting torque required due to bearing friction. This was checked using the Messinger friction torque equation:

Figure 35

Giromill Start Up Blade Modulation Scheme

Figure 36.
Starting Torque

$$\text{Torque} = \mu \frac{D}{2} (1.2 T + 1.8 F)$$

where: μ is the bearing friction coefficient, = 0.002 or 0.0015

D is the bearing diameter

T the bearing thrust load

F the bearing radial load

Since there are two rotor bearings, the equation must be applied to both and the torque added. Also, large bearings may have different characteristics, so a torque multiplying factor of 5 is recommended for the starting torque.

The forces and dimensions for a typical 120 kW Giromill are sketched in Figure 37. The upper bearing takes the weight of the Giromill rotor. The lower bearing reacts only the moment. Applying the forces shown in Figure 37 results in a bearing friction torque:

Upper bearing = 290 ft-lbs

Lower bearing = 32 ft-lbs

Total = 322 ft-lbs

Applying a multiplying factor of 5 gives a total friction starting torque required of 1610 ft-lbs. This is also shown in Figure 36, and indicates that the rotor torque produced by the blades is adequate. It should be pointed out that this analysis assumed a wind velocity of 15 mph. This analysis will be done again for the minimum wind cut-in speed.

The results of Figure 36 indicate that the Giromill will start up quite rapidly under the conditions shown. Assuming a rotor inertia of about 600,000 slug-ft², it would take less than a minute to reach the steady state rpm.

3.7 Operational Power Envelopes

The cyclogiro vortex theory computer program, that is used to compute the Giromill performance, does not converge to a valid solution at blade speed ratios, λ , much above the λ for maximum efficiency. Prof. H. C. Larsen, developer of this theory and computer program, attributes this to the close spacing of the assumed shed vortex array and their combined effect on the local flow. He feels that a modification of the theory to account for a roll up of the shed vortices would alleviate this problem. He is convinced however that the efficiency decrease trend indicated by the computer program at the high λ values is valid.

To present the entire operational power envelope requires the computation of the power output at wind velocities below the rated wind velocity. Since

UPPER BEARING LOADS

RADIAL $\sim F_U = 10,143 \text{ LBS}$

THRUST $\sim T = W = 32,000 \text{ LBS}$

FRICTION COEFFICIENT $\sim \mu = 0.002$

LOWER BEARING LOADS

RADIAL $\sim F_L = 5,143 \text{ LBS}$

THRUST $\sim T = 0$

FRICTION COEFFICIENT $\sim \mu = 0.0015$

Figure 37
Giromill Starting Conditions

the cyclogiro vortex theory computer program could not obtain these characteristics, an engineering extrapolation was accomplished. This section presents these results and extrapolation rationale.

Figure 38 shows a typical Giromill rotor efficiency plotted as a function of blade speed ratio. The vortex theory computer program computes the efficiency characteristics shown by the solid lines. For clarity, only the high angles of attack, α_e , are plotted to a speed ratio of 1.0; the lower α_e values would be similar. The vortex theory computer program provides valid results for increasing λ just past the peak efficiency. For λ 's beyond peak power and efficiency, the computer programs solution tends to flatten out and then breaks down and cannot converge to a solution. A conservative extrapolation of the efficiency was made by using a second order curve fit, tangent to the vortex theory computed curve, and intersecting the abscissa at a slightly increasing interval similar to that experienced by a propeller.

Figure 38 also shows the envelope of maximum efficiency along with a circle symbol at the α_e tangent points. Since a constant rotor rpm is being commanded, the α_e can be related to the wind ratio V_w/V_R as shown in Figure 39. This curve assumes that the operating λ is the maximum efficiency point for $\alpha_e = 9^\circ$. When the wind is below the rated wind, the angle of attack profile as shown on the left of $V_w/V_R = 1.0$ should be commanded. When the wind is higher than the rated wind, the angle of attack shown on the right of 1.0 should be commanded (this was taken from Section 3.5). Crossplotting the efficiency, from Figure 38, against the wind ratio from Figure 39, results in the curve shown in Figure 40. The final Giromill power curve is shown in Figure 41. The choice of cut in speed ratio of 0.6 is arbitrary but appears reasonable.

Admittedly the analysis lacks firm scientific justification; however, none is available at this time. The final resultant power curve is not much different than what one would expect from a conventional windmill, and hence lends some credence to the curve extrapolation. However, since the portion of the curve below $V_w/V_R = 1$ in Figure 39 is questionable, cost sensitivities to that portion of the curve will be determined.

Figure 38
Giromill Rotor Characteristics

Figure 39

Operating Angle of Attack Profile

Figure 40
Giromill Efficiency for Low Wind Velocities

Figure 41
Giromill Power Curve

4. GIROMILL SYSTEM DESIGN

The Giromill baseline design requirement was to create a concept that would be adaptable with minimum modifications to the complete kW range under investigation. The design was to make use of commercial type materials and construction techniques, that are familiar to the building and fabrication industries, to minimize fabrication and erection costs. Recent energy related studies at McDonnell Douglas have indicated that a steel truss or space frame would be the most economical construction method for tower type structures. This method was adopted as baseline for this study. Blade fabrication was assumed to use aluminum alloys to allow a low mass moment of inertia about their pivot. This moment of inertia dictates the size of the blade rocking angle actuator. Ground clearance of 15 meters was selected, this being the same height as used on horizontal wind generator studies conducted by Kaman and General Electric.

4.1 Giromill Design Concepts

Three basic Giromill designs were investigated for this study and are detailed below. The design concept illustrated in Figure 42a has received extensive distribution in publicity releases and various journals. The rotor is stabilized between an upper bearing positioned with guy wires and a lower bearing supported by a ground attached base tower. Use of guy wires to provide the upper support proved impractical. Cable sag and stretch allowed too much freedom of movement. Cable tension creates a downward force vector on the structure requiring added structural strength with the attendant weight and cost penalties. This force plus the added structural weight will also result in increased friction losses within the rotor bearings.

Figure 42b shows a fixed central tower concept. This design uses a stationary central tower with the blades and their support arms revolving around the tower on large diameter bearings. Problem areas with the design are the difficulty in transmitting torque from blade support arms to a power take off point. This would require either a central tower enclosing the stationary tower or extensive rotor structure joining the blade support arms into one rigid unit. This concept does not allow for a central shaft power output and complicates the drive system.

The design concept adopted as baseline is shown in Figure 42c. Three blades are supported on tubular radial arms attached to a triangular central tower. A diagonal tension member supports the main radial arm to which the

Figure 42
Giromill Concepts

rock actuator is attached. The other radial arms are stabilized through the blades. This configuration reduces rotational drag losses to a minimum. The triangular central tower is extended through the ground attached base tower to provide the maximum couple distance between rotor bearings. This will reduce the bearing loads and friction to a minimum. With this configuration, the generator and associated equipment can be ground mounted for easy accessibility. Location of wind direction and force sensors has not yet been determined. Giromill mounted units will probably experience local interference from the rotor but could probably be calibrated. Further investigation will determine suitable location. Foundation design will follow accepted civil engineering practices for the various types of soil conditions found at erection sites.

4.2 Rotor Blades

Aerodynamic studies showed that a wide range of blade spans and chords should be considered in determining baseline blade construction. All configurations have to incorporate the blade center of gravity at the blade pivot point (25% chord) to minimize inertia effects when following the blade rock angle program during operation. Primary objective of blade design was to use minimum number of components and absolute minimum cost.

Previous horizontal wind energy studies were investigated for economic blade construction techniques. Figure 43a illustrates a weld bonded, rolled channel concept developed during a previous study. This construction appears suited to the Giromill's constant chord, constant thickness untwisted shape, although it is apparent that some form of mass balance would be required. Since this method of construction is unproven, it was not selected as the baseline approach for this study.

A minimum component concept utilizing an extruded 'D' cell leading edge and sheet metal trailing edge as shown in Figure 43b was evaluated. Extrusion vendors contacted did not consider this large closed extrusion to be cost effective. Based on producibility and vendor inputs, this design evolved into the concept shown in Figure 43c. This concept uses a rolled sheet metal leading edge, a brake formed spar, beaded trailing edge skins and pressed sheet metal ribs. The center of gravity of this construction method can be controlled by varying the material thicknesses. A typical blade design for the baseline 120 kW Giromill is shown in Figure 44.

This design concept was selected as the Giromill baseline blade approach due to its adaptability to various blade sizes and its conventional construction

(a) ROLL FORM/WELD BOND

(b) EXTRUDED LEADING EDGE - SHEET TRAILING EDGE

(c) FINAL CONFIGURATION

Figure 43
Blade Concepts

DETAIL 1 MACHINED END FITTING
TYP 2 PLACE
SCALE 1/8

SYM AIRFOIL SECTION
6 BLADE ASSY REQD PER GIROMILL

DETAIL NO.	QTY PER ASSY	MATERIAL	AREA ELEMENT IN ² /IN	WT./LB
2011	20	2024-T3 x .127cm (.05) THICK	8.48	1.97
2009	2	7075-T651		2.94
2007	1	7075-T6 x .31cm (1.25) THICK		7.23
2005	1	7075-T6 x .63cm (2.5) THICK		
2003	2	7075-T6 x .127cm (.05) THICK		
2001	1	7075-T6 x .63cm (2.5) THICK		
DWN	J. K. Cole	18/7/75	MCDONNELL AIRCRAFT COMPANY SAINT LOUIS, MISSOURI	
CHK			MCDONNELL DOUGLAS CORPORATION	
SIZE			120 KW GIROMILL	
PL			BASELINE BLADE DESIGN	
SIZE			CODE	
IDENT NO.			76301	
SCALE	1/8	1/8	1/8	1/8
LB SHEET	1	1	1	1

BASELINE GIROMILL DATA
CAPTURE AREA = 446.8
RPM = 24
J = .37
AR = .06313
Vr = .1073
= 6.05 mps
(18 mph)

method requiring no state of the art advances.

4.3 Generator Selection

A baseline requirement for this study is to provide a generating system comparable to the electrical utilities small peak period generating stations. These stations use AC synchronous generators and achieve electrical synchronization with the utilities main plants by a start up technique that drives the generator as a motor until synchronous speed is achieved. Extensive use of the synchronous generator has resulted in a large data base and a wide range of 'off-the-shelf' components, hence the synchronous generator was selected as the baseline. Adoption of this system will require no state of the art advances and is the simplest method available to achieve synchronized AC electricity generation.

If the Giromill is used as an independent and isolated power source, DC or induction generators could be substituted. Energy output would be similar to the synchronous generator. It is not the intention of this study to determine variances in work output of these types of generators.

4.4 Generator Drive System

During Giromill start up, the generator will be declutched from the rotor. The generator will draw power from the utility grid system and be driven as a synchronous motor. When the Giromill rotor achieves rated rpm, the generator and rotor will be 'clutched' together. Aerodynamic control, rotor inertia, and generator power loading will combine to maintain this rpm over the determined power range.

This procedure allows the use of a fixed ratio rpm increaser required to drive a synchronous AC generator. Overall ratio of this rpm increaser is dependent on the Giromill configuration and the generator input rpm. Generators are commercially available with a range of fixed input speeds. Three methods of increasing the rotor rpm were considered and are detailed below.

Ring/Pinion Gear - The Giromill rotor revolves about a vertical pivot which dictates the axis of the low speed primary drive shaft. A method of increasing this shaft rpm output at the rotor was investigated. A ring gear was added to the upper bearing used for rotor support and used to drive a pinion gear attached to the primary shaft. The rpm increase available by this method was determined to be insufficient to drive the generator without an additional speed increaser mechanism. Cost of the ring/pinion gear exceeded the cost reduction possible in a conventional speed increaser (gear box) by the

lowering of the input/output ratio. Environmental protection and adequate lubrication of the ring gear is difficult to achieve.

These problems although solvable indicate that this method is not an economical approach.

Friction Drive - The large diameter of a Giromill's rotor structure could be used to provide a peripheral track for a friction drive wheel attached directly to the generator. Industry use of automotive type wheels and tires for conveyor drive systems indicated that a similar system might be adaptable for generator drive. Feasibility and limitations of this system are being determined. Substitution of the auto wheel with a railroad type steel wheel will also be investigated in order to extend component life. Maximum efficiency and life of this system may require a method of adjusting the radial contact force in relation to the transmitted torque. Friction drives although used with obvious success in automobiles and railroad systems have not proven suitable in most industrial applications. For this reason a friction drive, although attractive for larger type Giromills, would require testing and verification before use.

Conventional Drive System - The system adopted as baseline for this study is shown in Figure 45. Rotor rpm is increased by a gear box (speed increaser) and the generator attached directly to the speed increaser output shaft via a clutching device. The gear box is completely enclosed and positively lubricated, thus assuring maximum mechanical efficiency and life. Commercially available speed increasers and generators are designed for horizontal use and would require some modification to their lubrication systems when used in the Giromill applications. Components are basically 'off-the-shelf' for the 120 kW units. The larger Giromills (500 kW and 1500 kW) will follow this baseline system but will require custom designed speed increasers as off-the-shelf units capable of handling the large power levels at low rpm have not been found commercially available.

4.5 Giromill General Arrangement Drawings

Typical Giromill general arrangement drawings were completed based on the preceeding discussion. These typical arrangements were used to complete the structural and cost analyses which follow. Figures 46 and 47 present these typical arrangement drawings for the 120 and 500 kW configurations.

Figure 45
Baseline Generator Drive

McDONNELL DOUGLAS		McDONNELL AIRCRAFT COMPANY	
3 BLADE ROTOR	12'	GENERAL ARRANGEMENT	
18 0.05 m (18 in)	12'	120 KW GIROMILL	
ASPECT RATIO 1.075			
SOLIDITY .06515			
2	37		
RPM 24 RPM			
		CODE	
		IDENT NO.	
		76301	
		SCALE 1/8" = 1'	1/8" = 1'

FIGURE 46

SECTION A-A
TYPICAL FOR ALL BLADE SUPPORT ARMS

500-DESIGN BASELINES		OWN F. R. COLE		MCDONNELL AIRCRAFT COMPANY SAINT LOUIS, MISSOURI	
3 BLADE ROTOR		CHK		MCDONNELL DOUGLAS CORPORATION	
Vt 8.05 mps (18 mph)		STR		GENERAL ARRANGEMENT	
ASPECT RATIO .614		GP		500-1500 KW GIROMILL	
SOLIDITY .03727		FE			
T .26					
RPM 11.5					
				SIZE	CODE
					IDENT NO.
					76301
				SCALE 1/20 WT	18 SHEET

FIGURE 47

5. STRUCTURAL/STRENGTH DESIGN

The structural strength, stiffness, and dynamic response characteristics of the Giromill were evaluated to determine the feasibility of the Giromill concept and to identify structural items having major impact on total system weight and thus system costs. The structural designs developed in this study represent low cost configurations. Weight trends were generated which indicate the Giromill sizes most amenable to further optimization. These trends are based on the structural configuration, load conditions, and design criteria discussed in the following sections.

For analysis purposes, the Giromill structure was divided into substructures having elements of design commonality. Each was evaluated with regard to the criterion and load condition having the most impact on its design. Using this procedure, weight trends for each substructure were determined. Seven systems rated at 120 kW and eight rated at 500 kW were evaluated in this manner. Two of these, one 120 kW and one 500 kW system, were selected for a structural dynamics evaluation. Work on a higher power systems (1500 kW) and a more detailed analysis of a promising 120 kW or 500 kW systems are planned.

5.1 Structural Arrangement

Low-drag, three-bladed rotors and a clean straightforward design characterize the structural arrangement shown in Figure 48. Three blades were selected over two, four, etc., to preserve polar symmetry while minimizing the blade support structure required. The Giromill was studied as three substructures: 1) blades, 2) blade supports, and 3) tower.

Giromill blades exhibit a constant cross section and stiffness along their length. A typical cross section is included in Figure 48. Airfoil thickness is 20% of chord. The leading edge skin and single spar (main box) carry all spanwise loads. The trailing edge is corrugated chordwise and provides both a load path to the main box and torsional stiffness. Main ribs distribute loads at blade supports, and secondary ribs are spaced throughout the blade for minor load redistribution and stiffness. Blades are ballasted to locate the center of gravity at the centerline of blade rotation, 25% chord. Close tolerance mechanical fasteners are used for assembly.

Blade loads for operating conditions consist of airloads, which are radial and tangential with respect to the blade orbit, and centrifugal loads. Radial and tangential load supports are provided for each blade at three or

TYPICAL BLADE CROSS SECTION

TYPICAL BLADE SUPPORT

Figure 48
Giromill Structure

more places in the Giromill systems studied here. Blades are supported torsionally and vertically at the center support. In high winds, blades are declutched and allowed to weather vane into the wind.

Radial and tangential load supports for each blade consist of planar trusses (frames) each extending radially from the tower to an apex at the blade. Each frame consists of two columns joined by a blade attach fitting at the apex and cross-braced for increased in-plane strength and stiffness (Figure 48). Three blade supports are generally required to maintain acceptable blade stress levels in the 120 kW Giromill systems; four are required in the 500 kW systems. A tension brace extends diagonally from the top of the tower to the blade. It provides the necessary load path for vertical loads. Combination (thrust) bearings are used at all blade/support interfaces, so unbraced frames are supported vertically by the blade. Large radius rotors require additional out-of-plane frame bracing to preclude structural instability. Round tubular cross sections are typical of all blade support members, except streamlined tubing is recommended for the large radius rotors to further reduce air drag.

The Giromill tower is composed of an upper and lower section. The upper tower rotates as a part of the rotor and effectively forms a torque shaft connecting the torque-generating blades to the speed increaser and generator at the base of the tower. To reduce wind loading the upper tower is designed as an open lattice structure which has a triangular cross section. Corner posts are columns designed for the combination of dead weight and overall tower bending, which results from blade loads and wind loads on the structure. Cross bracing between the corner posts provides the necessary column stability.

The diagonals in each of the three tower faces are also designed as columns capable of sustaining 100% of the shear loads on the tower. Diagonals and corner posts considered together provide the flexural and torsional stiffnesses necessary to preclude strong resonances with the primary dynamic excitations.

Bulkheads housing large diameter bearings are located in the upper tower and in the lower tower at each end of the upper/lower tower overlap. The upper bulkheads are designed for lateral (tower shear) and vertical (structural weight) loads while the lower bulkheads are designed for lateral loads only. In addition to facilitating load transfer between the upper and lower tower sections, a long overlap increases the tower torsional stiffness by

reducing the length of the actual torque shaft connecting the rotor to the ground-based power generating equipment.

Tower weights were calculated on the basis that the size of the cross section is proportional to the height of the tower. For large cross sections the bearings may become major cost drivers. In this event, the cross section of the overlapping portion of the upper tower can be reduced in size to allow the use of smaller diameter bearings. The stiffness of this tower section must be maintained, however, and a weight (and cost) increase will result for the tower structure. Also, the lower bearing can be reduced in size if necessary by providing a necked down end fitting for the tower bottom.

The lower tower section is an open lattice structure designed to the same criteria as the upper tower but rectangular in cross section. The batter (slope of the corner posts) of this section can be adjusted to provide the necessary base rigidity. A steel grillage foundation or cast-in-place pier-with-bell anchors each cornerpost.

For all tower structure, the use of open section members, such as channel or I shapes, instead of tubular members improves access for maintainability and simplifies joint designs. Patterns of preloaded close-tolerance bolts are used for all tower joints. Splices of corner posts are minimized.

5.2 Structural Loads Environment

Giromill Loads Environment:

Three load conditions were used for the sizing of Giromill structure:

- 1) Normal operation under rated conditions
- 2) Operation under wind gust conditions
- 3) High winds

5.2.1 Operating Loads - As indicated in Section 3.3, the cyclogiro computer program computes the lift and drag on the rotor blades as they travel around the blade orbit. These forces are converted to forces tangent to and normal to the blade orbit. They are computed for every five degrees of the blade phase angle based on the geometric and operating conditions defined in Figure 2. One of the outputs of the computer program is the blade rock angle history for maintaining the prescribed angle of attack. Plots of these rock angle histories indicated that in certain areas of the blade orbit, the rotor blade was subjected to very high accelerations in order to maintain the rock angle requirements. These areas can be identified as the points where the vortex rings are released due to the blade flip, and when the blade traverses a vortex ring. For a three bladed rotor with a $\beta = -80^\circ$, vortices

are shed when a blade, initially a $\psi = 0$, reaches $\psi = 40^\circ, 100^\circ, 160^\circ, 220^\circ, 280^\circ$, and 340° . The blade travels through or near a shed vortex from another blade between $\psi = 240$ to $\psi = 260^\circ$. These are the areas where blade oscillations would cause large accelerations if it were required to follow the computed blade rock angle history. In order to reduce these accelerations, the rock angle histories were "smoothed by hand" and the radial and tangential forces were corrected for the smooth rock angles. Figure 49 presents a typical rock angle history. The computed values are indicated by the circled points and the solid line represents the smooth rock angle history. Figure 50 presents the tangential and the radial forces on the blades for the operational conditions consistent with the smooth rock angles of Figure 49. There is about a four kilowatt loss in power for the 120 kW system due to the smoothing of the rock angles.

5.2.2 Gust Loads - For computing the gust loads where the wind velocity was suddenly accelerated to 26.82 mps (60 mph), the rock angle history as shown in Figure 49 was used. For this history the total velocity vector (magnitude and direction) was determined at all points of the orbit for the operating condition. The components of the velocity vector in the direction of the wind were increased in magnitude by the difference of 26.82 mps (60 mph) and the operating velocity, V_w . New total velocity vectors (magnitude and direction) were determined at all points on the trajectory from which the rotor blade angle of attack history was computed. Using the blade aerodynamics as shown in Figures 14 and 15, new tangential and radial forces were computed. Figure 50 presents the gust forces on the rotor blade for the typical example presented in Figure 49. These gust loads were computed for all the configurations defined in Figure 2.

5.2.3 High Winds - As indicated in Section 2.4.2, the complete structure must withstand winds up to 53.64 mps (120 mph) with the rotor blades in a weathervane condition. Standard civil engineering practices were used to determine the tower loads and the allowable stresses. These standards are described in Reference 7. The resulting air loads on the Giromill components are shown later.

5.3 Structural Design Criteria

Design criteria used in this study are:

Service Life - Dynamic components, e.g., blades, shall endure the loads spectrum associated with a 30 year service life. Static components, e.g., lower tower, shall endure the loads spectrum associated with a 50 year service

Figure 49
Typical Rock Angle History

Figure 50
Typical Design Loads - Configuration 2

T - BLADE ORBIT TANGENT FORCE
R - BLADE ORBIT RADIAL FORCE

ψ DEG	NORMAL OPERATION				GUST LOAD			
	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS
0	3733	12144	847	2730	-1739	18314	-391	4117
5	3643	11455	819	2575	-200	18550	-45	4170
10	3490	10770	785	2421	-293	17162	-66	3858
15	3310	10120	744	2275	-218	15725	-49	3535
20	3105	9502	698	2136	-133	14310	-30	3217
25	3878	8919	647	2005	-22	12851	-5	2889
30	2656	8367	596	1881	12	11455	28	2575
35	2420	7838	544	1762	267	10071	60	2264
40	1939	7349	436	1652	236	8269	53	1859
45	1668	6864	375	1543	400	6868	90	1544
50	1388	6446	312	1449	-3812	3923	-857	882
55	1103	6108	248	1373	307	4119	69	926
60	823	5845	185	1314	164	2909	37	654
65	543	5650	122	1270	-890	1663	-200	374
70	262	5529	59	1243	-400	983	-90	221
75	-22	5480	-5	1232	-53	476	-12	107
80	-311	5512	-70	1239	-44	209	-10	47
85	-623	5641	-140	1268	0	4	0	1
90	-872	5294	-196	1190	18	-138	4	-31
95	-333	3536	-175	795	93	-538	21	-121
100	-71	-18	-16	-4	165	-1272	37	-286
105	1085	-4755	244	-1069	76	-2242	17	-504
110	1401	-8705	315	-1957	-285	-2727	-64	-613
115	1179	-9203	265	-2069	-365	-3029	-82	-681
120	1200	-9061	281	-2037	-351	-3923	-79	-882
125	1396	-9323	314	-2096	-435	-4995	-98	-1123
130	1472	-9675	331	-2175	-613	-6072	-138	-1365
135	1490	-9995	335	-2247	-849	-7175	-191	-1613

Figure 50 (Cont.)
Typical Design Loads - Configuration 2

ψ DEG	NORMAL OPERATION				GUST LOAD			
	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS
140	1485	-10240	334	-2302	-1156	-8242	-260	-1853
145	1481	-10440	333	-2347	-1450	-9386	-326	-2110
150	1490	-10618	335	-2387	-1757	-10512	-395	-2363
155	1534	-10792	345	-2426	-2015	-11681	-453	-2626
160	1121	-11143	252	-2505	-2829	-12340	-636	-2774
165	1205	-11366	271	-2555	-3073	-13577	-691	-3052
170	1267	-11606	285	-2609	-3354	-14778	-754	-3322
175	1298	-11828	292	-2659	-3518	-15974	-791	-3591
180	1312	-12033	295	-2705	-3830	-17113	-861	-3847
185	1321	12233	297	-2750	3896	18083	-876	-4065
190	1325	12424	298	-2793	4008	19217	-901	-4320
195	1334	12611	300	-2835	3856	20249	-867	-4552
200	1343	12789	302	-2875	3616	21299	-813	-4788
205	1361	12972	306	-2916	3429	22411	-771	-5038
210	1392	13198	313	-2967	2887	23572	-649	-5299
215	1401	13492	315	-3033	2651	24800	-596	-5575
220	1307	13697	294	-3079	2068	25885	-465	-5819
225	1223	13345	275	-3000	1436	26410	-323	-5937
230	1810	13025	407	-2928	1379	27660	310	-6218
235	2575	14889	579	-3347	4457	31832	1002	-7167
240	2326	16922	523	-3804	4853	35436	1091	-7966
245	1058	15013	238	-3375	6067	38252	1364	-8599
250	858	83585	193	1879	302	23221	68	-5220
255	8274	16010	186	-3599	1218	37611	274	-8455
260	2153	23056	484	-5183	4666	46593	1049	-10474
265	1161	21655	261	-4868	2838	49969	638	-11233
270	324	13625	73	-4187	7117	38657	16	-8690
275	-191	-11673	-43	-2624	-9919	-16886	-223	-3796
280	-204	-17794	-46	-4	-787	25302	177	5688

Figure 50 (Cont.)
Typical Design Loads - Configuration 2

ψ DEG	NORMAL OPERATION				GUST LOAD			
	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS	T NEWTONS	R NEWTONS	T POUNDS	R POUNDS
285	-422	12558	-95	2823	287	49813	647	11198
290	235	18060	53	4060	3500	41584	787	9348
295	916	19804	206	4452	2046	39355	460	8847
300	1468	19488	330	4381	2001	38158	450	8578
305	1944	19097	437	4293	1788	36713	402	8253
310	2362	18652	531	4193	1935	85200	435	7913
315	2726	18145	613	4079	1107	33523	249	7536
320	3033	17593	682	3955	1125	31988	253	7191
325	3287	16997	739	3821	1863	30418	419	6838
330	3492	16375	785	3681	733	28986	165	6516
335	3661	15721	823	3534	7206	27415	162	6163
340	3794	15040	853	3381	435	26014	98	5848
345	3874	14319	871	3219	346	24310	78	5465
350	3892	13585	875	3054	182	22949	41	5159
355	3857	12860	867	2891	1059	18345	238	4124
360	3768	12144	847	2730	-1739	18314	-391	4117

life.

Structural Dynamics - Dynamic resonance at rated operating conditions shall be avoided. Blade flutter and mechanical instabilities shall not occur under any operating conditions.

Ultimate Loads - Ultimate loads equal the factor of safety times the maximum expected loads. A factor of safety of two shall be used for blade design, and a factor of safety of three shall be used for blade support and tower design. The structure shall sustain ultimate loads without failure.

5.4 Structural Design

For a Giromill operating at rated power and under rated wind and rotor speeds, the variation in operating loads as the rotor makes one revolution was used to define the fatigue environment. In this feasibility study, the effects of resonance dwell during start up and shut down and the effects of operating at wind speeds other than rated speed were not considered. Also, the beneficial effects of occasional high load cycles from wind gusts were ignored. The relatively high load cycles associated with the wind gust condition occur too infrequently to cause fatigue damage.

Based on this variation in normal operating loads and the service life requirements, 10^8 to 10^9 load cycles can be expected in any structural member. For most engineering materials, this means operating at stress levels at or below the materials constant amplitude, fatigue endurance limit. The combination of infinite life, load cycle characteristics, and structural configuration defines this allowable stress (endurance limit) for the selected material. Because of the large number of load cycles which accrue in a year, the allowable stress for a service life of one year will be only slightly higher than that for a service life of 30 years.

A design ultimate load is equal to the maximum expected load times a factor of safety (F.S.). For blade design, a F.S. of 2 is consistent with the materials and manufacturing quality associated with aerospace structures. To maintain low operating stress levels in blade support and tower structure, a F.S. of 3 is used. This factor is characteristic of commercially constructed transmission towers and buildings. Structural stability considerations dictate allowable stress levels for design ultimate loads.

In this study, the high wind and gust wind loadings comprise ultimate load conditions. Blades are declutched and allowed to weather vane during high winds; however, wind loads in conjunction with a reasonable structural

solidity does become a design consideration for the tower.

Ultimate gust loads are accounted for in the blade design only. A gust which is sustained long enough to load the entire Giromill structure would precipitate either a blade declutching because the cut-off wind speed is reached or change the blade modulation to counter the increased wind speed. In either case, blade loads and thus total tower loads are reduced.

5.4.1 Blade Design - The design of Giromill blades is basically dictated by service life requirements. As such, the peak loads associated with normal operation under rated conditions become the design loads. These loads, along with gust condition loads, are presented in Figure 51 for two representative Giromill systems. Gust loads become important in low aspect ratio, large blade chord systems where large flat-plate areas of the leading edge are stability critical rather than fatigue critical.

The loads shown in Figure 51 act normal to the blade chord. Air and inertia loads resolved along the chord are small relative to the values shown and are not included in the blade analysis. The constant inertial loads plus the variable air loads produce the one per revolution load cycle. For the 120 kW system, Figure 51a, inertial loads are high relative to airloads. The lower rotor speed of the 500 kW system (Figure 51b) results in minor blade load reversal under normal operation. For both systems a load or stress ratio of zero (minimum load divided by maximum load) is used in determining an allowable stress for the fatigue loading.

The blade flip regions for both Giromill systems are centered on blade positions of 100° and 280° from the wind vector. Blade load gradients occur as a result of airload reversal when each blade flips through a rock angle difference of approximately 15°.

An aluminum alloy was selected for the blade material. In addition to the high structural efficiency of aluminum relative to other common materials, such as steel, aluminum is easily formed, naturally resistant to galvanic corrosion and the selected alloy, 2024-T6, exhibits good resistance to stress corrosion. All of these factors are considered important, for maintenance reasons, to the design of a long-life, closed structure.

The endurance limit of this aluminum, for a stress ratio of zero and a stress concentration factor representing spanwise patterns of mechanical fasteners, is 96.5 MN/m^2 (14000 psi).

Figure 51
Expected Blade Loads

5.4.2 Blade Supports and Tower Design - The blade supports and tower are the major weight items of the Giromill system. Both are designed to withstand ultimate loads associated with the high wind load condition. Because of the expected impact of these substructures on total system costs, materials and fabrication techniques used commercially for structures such as transmission towers are relied upon as representing the maximum economy possible in this type of construction. Consequently, commercial steels, such as A-36, A-441, and A-514, were selected for the blade support and tower substructures.

The factor of safety of 3 chosen for this analysis effects maximum expected stresses approximately equal to those specified by agencies governing the design of commercial structures, such as the AISC. Loads associated with high wind conditions exceed those of the normal operations condition resulting in low operating stresses.

For normal operations, the variation of tower shear loads with rotor position are shown in Figure 52 for a representative 120 kW and 500 kW system. Each curve represents the sum of all blade loads which have been resolved into the indicated tower coordinate. Because three-bladed rotors are being studied, tower loads repeat every 120° of rotor travel yielding three load cycles per revolution. A comparison of the two Giromill systems indicates that peak tower loads are proportional to the rated power of the system.

The relative magnitudes of tower bending moment for the high wind and normal operating conditions are presented in Figure 53 for the 120 kW system. The maximum resultant load from Figure 52 was used to calculate bending moments for the operating condition. At the 15.24m (50 ft) elevation, the ratio of bending moments is .344, operating to high wind. The design column allowable is 310 MN/m^2 (45000 psi) and for a F.S. of 3 the maximum expected stress occurring for the high wind condition is 103.3 MN/m^2 (15000 psi). Therefore, the expected operating stress level is approximately $.344 \times 103.3 = 35.6 \text{ MN/m}^2$ (5200 psi). The allowable for constant amplitude fatigue of mechanically-fastened, carbon steel joints is based on 10^9 load cycles (3 cycles per revolution, 25 rpm for 50 years) and a stress ratio conservatively assumed to be -1. A comparison of the operating stress with this allowable, 58.5 MN/m^2 (8500 psi), indicates a fatigue-safe design.

5.5 Weight Trends

Study results are summarized in Figures 54 and 55. For 120 kW systems, weight trends of each substructure and of the total Giromill structure are

Figure 52
Total Expected Tower Shear - Normal Operating Condition

Figure 53
Maximum Expected Tower Bending Moments

presented in Figure 54. High tower weights and large blade spans characterize large aspect ratios. High blade support weight and large blade chords characterize low aspect ratios. Blade weights comprise only a minor part of the total system weight, but blade size strongly influences the weight of the other substructures.

Increases in rated wind speed reduces the blade size required for a given power output. The reduced Giromill weight associated with a higher rated wind speed (9 m/sec, 20 mph) is indicated in Figure 54. Although operating at higher rated wind speed is a weight effective approach, it may not be cost effective because the Giromill will operate fewer hours each year.

A comparison is made between total structure weights for the 120 kW and 500 kW Giromill systems in Figure 55. Going from the low to the high power system at constant aspect ratio, a 420% increase in power is accompanied by a representative 350% increase in structure weight. On a kilowatt per pound basis this indicates higher efficiency for larger Giromill systems.

5.6 Structural Dynamics

5.6.1 Structural Dynamics Analysis Overview - The purpose of the structural dynamics investigation is to determine the resonant frequencies and associated mode shapes of vibration of the Giromill system, and to investigate any aeroelastic instabilities that may exist. Vibrations may be divided into two types, ordinary and self-excited. Ordinary vibrations are those in which a system of springs, dampers, and masses is forced to vibrate by some alternating external force which is independent of the motion. The amplitude of the vibration per unit of applied force may vary widely, depending on the characteristics of the system such as damping and the ratio of the applied frequency to the systems resonant frequency. Vibrations of this type, because of the associated load amplification, are important in determining the fatigue life of the system. In order to avoid a reduction in fatigue life, it is important that no resonances occur at or near the operating point (normal rotational speed) of the system.

Self-excited vibrations are those in which the alternating forces are sustained or controlled by the vibratory motion itself. When self-excited vibrations occur, the system is in an unstable condition and any small disturbance can cause a diverging oscillation. Types of self-excited vibration that can occur in the Giromill system are blade flutter, tower whirl, and coupling of tower bending and support arm horizontal bending. The latter

Figure 54
 Giromill Substructure Weight 120 kW System

Figure 55
 Giromill Structure Weight

vibration is similar to the ground resonance phenomenon in helicopters. Because of the unstable nature and potential destructive characteristics of self-excited vibrations, it is important to design the system such that these effects do not occur within the operating speed range.

With the Giromill, as with any rotating system, the sources of excitation for vibrations comes from the rotation of the system itself. These excitations can be divided into two types; those due to mass unbalance and those due to aerodynamic imbalance between the blades. The frequencies of these excitations will always be at multiples of the shaft speed and thus are expressed as 1 per rev., 2 per rev., etc. For the Giromill system, these excitations can be characterized as follows:

1/rev. excitation - caused by mass and aerodynamic unbalance, this will be a strong input

2/rev. excitation - caused by two blade reversals per revolution, also this is the first harmonic of the 1/rev

3/rev. excitation - caused by aerodynamic unbalance from the three support arm-blade configuration, this will be a strong input

4/rev. excitation - this is a multiple of the two/rev. excitation

6/rev. excitation - caused by two blade reversals per revolution in conjunction with three blades, also first harmonic of the 3/rev.

While higher harmonic excitations may be present in the system, it is assumed that their amplitudes will be small enough such that they need not be considered in this study.

In order to simplify the computation of the resonant frequencies of the Giromill system, it was divided into three components or sub-systems; blades, support arms, and tower. The resonant frequencies of these components were calculated and, where appropriate, were then inertially or elastically coupled to give the resonant frequencies of the Giromill system. A short description of each of these resonances and its coupling follows.

Blades - Blade uncoupled vibration modes are flap bending, lag-lead bending, and torsion. Blade torsion must be considered under two conditions, clutched and declutched. With the blade in the clutched condition the pitch change mechanism provides a relatively soft torsional restraint at one end of the blade, and the mode of vibration is essentially rigid blade flexing the pitch change mechanism. In the declutched condition the first torsion mode is blade free-free motion.

Support Arms - Support arm vibration modes can be divided into two types, arm vertical bending and arm horizontal bending. Arm vertical bending can further be divided into symmetrical (all arms responding in-phase) and anti-symmetrical (one arm moving up and the other two moving down and vice-versa). Arm horizontal bending modes can be subdivided into in-phase modes (top, center, and bottom arms moving in the same direction) and out-of-phase modes (top and bottom arms moving in opposite directions with the center arm standing still). These modes can further be described as collective (all three sets of three arms moving in-phase with a resulting torque) or cyclic (a phase of 120 degrees between each set of arms with zero resulting torque).

Tower - Tower vibration modes that were investigated are tower first bending and tower first torsion. Tower first bending must include the effects of foundation flexibilities as well as the tower flexibilities. Tower torsion will be influenced by the polar moment of inertia of the generator gear box combination as well as by the polar moment of inertia of the support arms and blades.

These component modes must be coupled to give total system modes. For example, blade lag-lead bending may couple with support arm horizontal bending. Support arm vertical bending out-of-phase can couple with tower bending. Indeed, many other combinations are possible and were considered in the analysis. This type of an analysis was carried out because at this time the scope of the investigation is simply to determine the feasibility of the Giromill concept. Feasibility of the concept has been proven from a dynamics standpoint by investigating two typical Giromill sizes, a 120 kW and a 500 kW machine, selected as having desirable aerodynamics and performance characteristics. The dynamics of these two configurations were then determined with regard to blade flutter, mechanical instabilities, and forced resonances.

5.6.2 Structural Dynamics Results - Figure 56 is a summary of the preliminary flutter investigation for the blades of the 120 kW configuration 2 Giromill. The flutter mechanism investigated here using simplified methods involves blade first bending and rigid blade torsion supported by the pitch change mechanism. As can be seen from the figure, the flutter velocity is very sensitive to the torsional stiffness of the blade pitch change mechanism. At the present time the nominal torsional stiffness of the pitch change mechanism is 2.5×10^5 in-lb/rad. With this stiffness a blade flutter speed of 180 fps is indicated which is substantially above the operating speed of the Giromill. It should be noted, however, that the nominal torsional stiffness is in a very

Figure 56
Blade Flutter Velocity Restraint Stiffness

sensitive region with regard to flutter. More detailed flutter analysis of the blade is required and will be conducted as appropriate parameters become better defined. While results for the 500 kW configuration 12 are not reported here, similar trends are anticipated. Another possible flutter conditions involving blade elastic modes not shown on Figure 56 is the case where the blade is in a declutched condition. In this case the blade may be subject to wind velocities up to 120 MPH. However, the flutter involves blade second bending and blade first free-free torsion. The flutter speed for this case has been determined to be very high and should therefore not be a problem. Single degree of freedom flutter of the declutched blade about its pitch axis will be eliminated for pitch axis locations at or aft of the quarter chord.

Figure 57 is a summary plot showing the rotational speed ranges of the mechanical instabilities investigated. This plot shows the resonant frequencies for tower bending and arm lag-lead bending as a function of rotational speed. Several points are labeled on the figure as regions of mechanical instability or as forced vibration. The region labeled as A-B represents the potential instability region involving tower bending and support arm lag-lead bending which was described earlier. The point labeled as H corresponds to the shaft whirl instability. Since the nominal rotational speed for this configuration is 0.4 rev/sec., it can be seen from Figure 57, that the above described mechanical instabilities occur well above the operating speed. Figure 58 presents similar results for the 500 kW configuration 12 Giromill which has a nominal rotational speed of 0.2 rev/sec. While the resonant frequencies are lower and mechanical instabilities occur at lower rotational speeds relative to those given in Figure 57, the rotational speed at the operating point is also lower. Therefore, this configuration also has no mechanical instabilities in the operating range.

Figure 59 is a plot of the resonant frequencies of the 120 kW configuration 2 Giromill as a function of the rotational speed range of interest. Because of the programmed pitch angle of the blades with respect to rotor azimuth angle, the blade modes are plotted as being in the stationary coordinate system. Also shown are the excitation lines up to the 8/rev. As can be seen from this plot, while the system has many modes of vibration, only the four lowest modes are of interest from the standpoint of forced vibration. Modes above 4 Hz would require excitations well above the 8/rev. before they could be excited and for this investigation, the strengths of the higher

Figure 57
Tower and Blade Support Arm Dynamics

Figure 58
Tower and Blade Support Arm Dynamics

harmonics are considered small enough to be neglected. Figure 60 is similar to Figure 59 except that the results for the 500 kW configuration 12 are presented. In this case, however, because of the low rotational speed of the configuration only the two lowest modes need be considered for forced vibration computations.

Figure 61 is similar to Figure 59 except that the vertical scale has been expanded to show the frequency range of interest. Where lines corresponding to the resonant frequencies cross lines representing the excitation frequencies, regions of forced vibration exist in the rotating components and may exist in the stationary components. These points should be as far removed as possible from the operating point shown here as a vertical line at 0.4 RPS. As the machine is started and stopped, several of the forced vibration points will have to be crossed. The load amplification associated with these points is important in determining the fatigue life of the system. As can be seen from Figure 61, the present configuration has tower bending resonance points near the operating speed. Further studies will be conducted to investigate structural changes required to shift this mode from the present frequencies. Figure 62 shows similar results for the 500 kW configuration 12 machine. However, it shows a situation of more mild load amplification for tower bending and for tower torsion at the operating speed (higher harmonic excitations).

5.6.3 Conclusions - Giromill configurations 2 (120 kW) and 12 (500 kW) have been investigated to determine their dynamic characteristics. In this regard it has been determined that neither machine has any mechanical instabilities in its operating range. Further, configuration 2 (120 kW) has been investigated to determine the bending torsion flutter characteristics of its blades. The preliminary results show an adequate flutter speed margin within the operating range, but a high sensitivity to the blade pitch restraint stiffness. Each of the Giromill configurations has resonance points near the operating speeds. The highest load amplification is likely to be at the tower bending frequencies in the 120 kW systems.

Figure 59
Giromill Component Resonant Frequencies

Figure 60.
Giromill Component Resonant Frequencies

Figure 61
Giromill Component Resonant Frequencies

Figure 62
Giromill Component Resonant Frequencies

6. CONTROL SYSTEM

The control system measures both Giromill and environmental parameters and, based upon the interrelationships between these parameters, controls blade rocking angles in a manner consistent with deriving maximum power output from the generator without exceeding system power rating. The control system must provide signals for starting and synchronizing the Giromill and shutting the system down when events exceeding the capabilities of the Giromill or its control system occur, as well as providing for normal operating mode functions. This section provides a discussion of the functional requirements for the control system, proceeds to a discussion of the hardware elements used to implement the functional requirements, and concludes with an integrated system discussion.

6.1 Functional Requirements

The main functional requirement of the control system is to enable the Giromill to generate the maximum rated power output without exceeding the available structural and electrical load limitations. The control system performs this function primarily by controlling the blade rock angles so that the desired Giromill rotor rpm is maintained in the presence of variations in wind velocity and power load.

Other control system functions are to provide for the startup and power interrupt or shutdown situations as they occur and to insure that safe operating conditions are met. The control system functional requirements can be most suitably described in terms of the following three main control modes: (1) startup, (2) normal operation, and (3) power interrupt mode. All three of these modes can be best visualized with the aid of the overall control system functional diagram of Figure 63. This diagram shows where the sensor outputs are utilized, includes a tabulation of the system switching logic, and indicates what other functions are performed to generate the rocking angles, θ_{R_1} , θ_{R_2} , and θ_{R_3} for the three Giromill blades. Figure 64 shows an expansion of the control system detail for the normal Giromill operating mode. Discussion of these control system diagrams for the three control modes follows.

6.1.1 Start Up Operation - Giromill rotor start up can be initiated any time the wind velocity is between V_{CO} and V_{CI} . The start up condition is signified by condition A in the block diagram of Figure 63. At start up the blades are clutched to their actuator and driven in accordance with the rock angle profile previously described in Section 3.6 through the start up rocking

Figure 63

Giromill Control System Functional Diagram

Figure 64

Blade Rocking Angle Control

angle processor. During this time the generator is declutched from the rotor. As the rotor rpm is brought up such that λ becomes greater than 1.33, the normal rocking angle processor will be switched in (Selector switch II). When the rotor rpm and the rpm of the generator, driven as a motor, are within a specified tolerance, the generator to rotor clutch will be engaged. This is signified by condition B in Figure 63.

6.1.2 Normal Operation - In normal operation the rotor control system positions the blade rocking angles so that the net aerodynamic torque on the rotor varies as necessary to balance the electrical load torque and any superimposed torque disturbances. The sum of total torques from the blades controls rotor rpm. The specific computations performed for rocking angle control in normal operation are depicted in block diagram form in Figure 64. In normal operation the individual commanded blade rocking angles, shown in Figure 64, are labeled θ_{RN_1} , θ_{RN_2} , and θ_{RN_3} . The commanded rocking angles are computed as a sum of the following three angle components: (1) the feathered rocking angles (computed by function block N-1), (2) the effective nominal blade angles of attack (computed primarily as a product of outputs from the functional blocks N-2 and N-3), and (3) the induced blade angles of attack (computed as a product of function blocks N-4 and N-5). The purpose of the induced angle of attack increments, α_{i1} , α_{i2} , and α_{i3} , is to compensate the nominal angles of attack for the effect of airstream vortices generated by the Giromill blades. The effective angle of attack is incrementally corrected by the difference between the actual rotor rpm and the desired rpm. This correction serves to provide the necessary control torque increment and thus maintain the rotor rpm and generator output.

The control concept described is considered feasible but further stability and response studies of this approach will be needed to establish the system stability and response characteristics. It is anticipated that such studies will be a part of follow-on control work.

6.1.3 Power Interrupt Condition - The events leading to interruption of generator power output are best described based on Figure 63. The two events which are most likely to cause temporary power interruption are either insufficient wind velocity, denoted as condition D of Figure 63, or excessive wind, labeled as condition C. In these two power interruptions the blades are declutched and are allowed to weather vane. The starting sequence and clutch control will enable automatic restart when wind velocity again becomes usable.

The remaining three power interrupt conditions shown in Figure 63 are included as safety features to be utilized in case there is a partial system malfunction. The three conditions considered are excessive vibration (condition E), excessive error in one or more of the blade rocking angles (condition F), and out-of-tolerance generator power output (condition G) occurring while other available sensors indicate essentially normal Giromill operation. The last three power interrupt conditions represent a potential system malfunction, therefore the reclutching of the blades will be prevented in these cases until required system checks or repairs are performed.

6.2 Hardware Requirements

Implementation of the functional requirements of the preceeding section requires four types of hardware elements: sensors, blade actuators, power supplies, and computing and control elements. This section describes the requirements imposed on the hardware elements and discusses the technique used to satisfy those requirements.

6.2.1 Sensors - Sensors are required to measure environmental conditions of wind speed and wind direction and Giromill parameters for rotor position, blade rocking angle, tower vibrations, and generator control signals.

The wind speed transducer is a direct current permanent magnet generator, driven by a 3-cup wheel arrangement attached to the armature shaft. The voltage output is directly proportional to cup wheel rotation speed. Any wind speed exceeding 1.5 knots is sufficient to start the transducer. The output from the generator is buffered, smoothed (to minimize gusting effects), and converted to a digital signal for use in the processor. The accuracy of this measurement system is $\pm 1.5\%$ over the entire environment to which it will be exposed.

Wind direction is measured by a dual wiper potentiometer with 3 pickoff points on the resistance element. The wipers are mechanically connected directly to a weathervane which rotates to the prevailing wind. A direct current voltage is applied through the wipers to the resistance element. The 3 pickoff point outputs are electrically loaded, buffered, and converted to digital signals by an analog to digital (A/D) converter for subsequent use in the processor. The minimum wind speed which will provide accurate direction outputs is 1.5 knots. Direction measurement accuracy is ± 1 degree for a constant direction input.

Rotor position is measured in a manner similar to that for wind direction

except that the wipers are mechanically coupled to the rotor shaft through a 1:1 gear train. Once again, a 3-point output pickoff passes the direct current voltage to conditioning circuits which prepare it for input to the processor.

Blade rocking angle is determined by a rotary variable differential transformer (RVDT) which is incorporated in each actuator. The RVDT is connected via gear train to the actuator motor shaft. The RVDT output has two regions of 140 degrees each where the output is very linear: the remaining 80 degrees are non-linear. Measurement accuracy using this device is ± 1 degree after the ambiguity resulting from two angles having the same output voltage is resolved. Ambiguity resolution is provided by the processor.

Tower and rotor vibrations are measured to detect mechanical conditions (such as blade icing) which could cause tower or rotor destruction if the Giro-mill is not placed in a powered-down, weathervane mode. Frequencies of importance for vibration measurement are quite low - nominally 0.25 to 100 Hz. Therefore, the vibration transducer selected is a force balance type unit wherein a pendulous mass offset induced by vibration generates an error signal to a servo amplifier to drive the mass back to rest. The current required to drive the mass back to rest is directly proportional to acceleration. Current is converted to a voltage analog which is converted by the A/D converter for input to the processor.

Generator control sensing is performed differently during startup than it is during the operating mode. The generator on-line/off-line decision is made as a function of rotor rpm and generator-versus-grid phase coherence during the startup mode. Rotor rpm is computed from successive measurements of rotor blade position. Generator and grid phase coherence is measured in the generator system. Once in the operate mode, the parameters of interest change and now the measured parameters for generator control consist of power-delivered analogs. These signals are buffered and converted to digital signals for use by the processor.

6.2.2 Blade Actuators - The blade actuator consists of a brushless DC motor, electrical disconnect clutch, a rotary variable differential transformer position indicator, interconnecting control and power gearing, plus cast aluminum housings. The motor drives the dual output shaft through three stages of planetary reduction gearing and one state of spur gearing to the electrical disconnect clutch and output shaft.

The use of planetary reduction gearing for the high speed gear reduction

uses the lubrication recirculatory characteristics to achieve long life. The motor mounts directly to the planetary ring gear which in turn mounts to the main gear housing cover. The output of the planetary gear section drives a spur gear pinion in the main gear housing and the pinion mates with a gear attached to the input section of the electrical disconnect clutch.

The electrical disconnect clutch is mounted directly on the dual output shaft to achieve the lowest possible backdriving drag and allow the declutched blade to "weather vane." A tooth-type disconnect clutch is used to keep the clutch size within reasonable limits and a stationary coil type of clutch selected for long life characteristics.

A rotary variable differential transformer was selected for position indication to meet the long life requirement and is geared directly to the output shaft to maintain position monitor during "weather vane" operation.

The block diagram is contained in Figure 65. Basically, the system functions as an analog blade position control. The Giromill processor determines the blade angle (rock angle) with respect to the rotor arm. This signal is sent to the blade system as an analog voltage. The system compares this voltage to the feedback voltage from the position indicator (RVDT). The RVDT is coupled directly to the output shaft by one gear mesh. The error signal is acted upon by the compensation network and then is used to generate two control signals. One is a two-position signal which commands actuator direction (CW or CCW). The other is an analog signal whose amplitude commands motor excitation voltage.

The sequencing and driver circuit accepts the direction command signal as well as the commutator interrupter signals. By means of logic and gating, two coil drivers are energized to cause motor rotation in the proper direction. As the motor rotates, the commutator disc energizes the appropriate two coils to cause continued rotation. The direction of rotation is changed only by the signal generated in the analog circuitry. The drivers and associated circuitry are interlocked in such a manner that only the appropriate two coils can be energized at one time.

The error amplitude signal is accepted by the converter circuit and commands a DC voltage level. Basically, the converter is composed of a full wave bridge operating from 115 VAC. Two legs of the bridge are silicon controlled rectifiers which can control the resultant voltage at the motor windings by phase

Figure 65
Blade Actuator Block Diagram

angle control. The applied voltage is compared to the command voltage and the converter adjusts the phase angle accordingly. In this manner, the voltage applied to the motor coils is proportional to the error signal.

The actuator is capable of being uncoupled from the blade by means of an electrical disconnect clutch. In this manner, the wind will rotate the blade to a feathered position when the clutch is de-energized.

6.2.3 Power Supplies and Switching - Power supply and switching functions consist of switching 115 VAC single phase power to the actuators; providing regulated direct current power at voltage levels of +5, +15, +25 and -15 VDC; and storing for maintenance purposes to use in case of abnormal shutdowns.

Whenever the Giromill is in the startup or operate modes of operation and a declutching sequence is not in progress, 115 VAC power is switched to each of the blade actuators. Approximately 450 watts of AC power is required for this function.

The actuators require a source of regulated +25 VDC for their commutator circuitry - all other control system circuit elements operate from +5 and ± 15 VDC power. Figure 66 delineates the regulated DC power estimates.

The DC power requirements will be provided by series regulator-type devices which will achieve conversion efficiencies on the order of 50%. Approximately 50 watts of 115 VAC are required for the DC supplies. Short circuit and overvoltage protection will be provided to ensure that a failure in either the power supply or the using circuits will not adversely affect the other.

The power supply and switching networks also contain electromechanical fault indicators which are latched during an abnormal shutdown sequence to preserve system status information for maintenance purposes. These devices are manually reset by maintenance personnel after the fault-causing shutdown (such as excessive vibration) has been serviced. Three of these devices are provided.

6.2.4 Computing and Control Elements - Computing and control functions are effected by a processor together with the digital to analog (D/A) converters, A/D converters, and latches necessary to preserve processor input and output data.

The processor directs all system functions for the control system, except when the standby mode is in effect. It is a micro-computer consisting of a microprocessor, programmable read-only memory (PROM) program and permanent data storage, a random access memory (RAM) scratch pad area, and input/output

ESTIMATED CURRENT~MILLIAMPERES			
Control System Function	+5VDC	+15VDC	-15VDC
Wind Speed Sensing		15	15
Wind Direction Sensing		10	
Rotor Position Sensing		10	
Blade Position Sensing		180	180
Vibration Sensing		40	
A to D Conversion	30	20	26
D to A Conversion	180	48	48
Power Switching		100	
Processor	<u>1200</u>		<u>200</u>
TOTAL	1410	423	469

Figure 66
Control System DC Power Requirements

multiplexers required to communicate with circuit elements outside the processor itself. The multiplexers are transistor-transistor-logic (TTL) devices from the 5400 series of devices, while the CPU, PROM and RAM are MOS-type integrated circuits. The three D/A converters receive the commanded rocking angle (θ_{RC}) from the processor, retain it in latches and convert it to the analog voltage for driving the three actuator assemblies. The A/D converter is a successive approximation-type device which receives its analog input from an analog multiplexer having a tri-state output. The digital output from the A/D is directed to the input multiplexer of the processor.

6.3 Control System Integration

Figure 67 presents a hardware block diagram of the Giromill control system showing signal generation and flow among the various hardware elements. All control functions originate in the processor except when the system is in the standby mode. During the standby mode of operation, the control is exercised by the wind speed circuits. The only decision made here is to command the system to start up whenever $V_{CO} > V_W > V_{CI}$.

Several elements of the control system have special mounting requirements - among them the wind speed and direction transducers, the vibration transducers, and the blade position sensor.

Figure 67
Giromill Control System Block Diagram

Both the wind speed and direction transducers must be mounted so that they are free from induced effects of the Giromill itself and so that the measured data can be modified to the proper values for selecting rocking angle commands. Both units mount on top of 1.25 inch pipe.

Vibration transducer mounting must be such that the vibrations induced by the undesirable effects can be accurately and reliably discriminated from the normal operating mode vibrations. Current mounting location envisioned is one unit near the top of the tower and one unit located near the blade actuator.

The rotor position sensor must be mechanically attached to the Giromill rotor axis ahead of the speed increaser.

The actuators are physically mounted between blade sections with the dual output shaft providing torque to each blade section.

All remaining portions of the control system are housed together in a single, sealed container whose volume is approximately 48,000 CC (2929 cubic inches). If possible, this unit will be mounted on the stationary structure since it will greatly simplify the signal interfacing requirements. Signals which must be passed across the stationary/rotating interface are the blade actuator and position signals, emergency pyro blade declutch, and vibration transducers: all remaining units are attached to stationary structural members.

The control system supports electrical interfaces between itself and the generator system and between itself and the emergency pyro-actuated declutching device.

The generator system furnishes 115 VAC power for the control system, provides instrumentation outputs analogous to generated power and phase coherence, and accepts control system inputs for going ON-line/OFF-line.

The emergency pyro actuator declutching interface is accommodated in the control system by a high current pulsed output generator with suitable protection to guard against energy build-up which might result in an undesired ignition of the device.

7. ENERGY OUTPUT EVALUATION

This section discusses the various input parameters which are required to determine the total energy output of the Giromill systems. Location of high windspeeds (favorable to the use of wind energy power) within the United States are identified in Section 7.1. The study groundrules specify use of two typical wind speed distributions for the Giromill parametric analysis. One such distribution having a mean wind speed of $\bar{V} = 5.4$ m/sec (12 mph) is utilized for the 120 kW and 500 kW systems and a second with $\bar{V} = 8.1$ m/sec (18 mph) is utilized for the larger 1500 kW and 3000 kW systems. Wind duration curves for these $\bar{V}$ values are discussed in Section 7.2 and are utilized in Section 7.3 with minimum wind speed for Giromill operation (cut-in speed) criteria and the system power output versus wind speed input to determine total energy output.

7.1 High Wind Speed Areas

The annual mean wind speed $\bar{V}$ at the Giromill site is of paramount importance in determining the amount of energy converted annually. Large values of $\bar{V}$ are desirable for two reasons. First, the maximum available system power output varies as the cube of the local windspeed. The second is related to the capricious nature of the wind. Wind speed at any site is a continuously changing commodity, which, during the course of a year, varies from zero to extreme values. In general, the higher the annual mean wind speed, the greater the proportion of the time that the wind speed, and hence power output, exceeds an acceptably low value.

To achieve high system power output for specific Giromill size or weight constraints, locations must be found that have a high annual mean wind speed. Figure 68 shows the long term annual mean wind speed ($\bar{V}$) for the 48 contiguous states. Note the location of areas of high local wind speed $\bar{V}$ in the Great Plains and the belt of high $\bar{V}$ along the southeastern shores of Lake Erie and Lake Ontario. In these areas $\bar{V}$ often exceeds 5.85 m/sec (13 mph) over a large area.

Empirical data indicate that in many cases the wind blowing over hills experience a Bernoulli type acceleration resulting in $\bar{V}$ at the top of a hill being significantly greater than the $\bar{V}$ for the surrounding area. These local areas of very high wind speed within the bands of high $\bar{V}$ should be sought out as potential locations to maximize the energy recovery from the wind. It is reasonable to expect that a number of sites having local wind speeds up to and exceeding 8.1 m/sec (18 mph) will be found; i.e., equal to or exceeding the values

Figure 68
Mean Annual Wind Speeds (MPH)

of mean wind speed assumed for the large 1500 kW and 3000 kW Giromills.

7.2 Wind Duration Curves

Wind duration curves for two values of mean wind speed, 5.4 m/sec (12 mph) and 8.1 m/sec (18 mph), are to be utilized in the Giromill study. These curves are representative of actual site wind speed distributions and, as presented in Section 2.4.1, show the number of hours per year that the wind exceeds a specified value of wind speed for two values of assumed mean wind speeds.

These wind duration curves were defined for an altitude of 9.14 meters (30 ft). The center distance of the Giromill blades will be at a considerably higher elevation and, consequently, the analysis will assume an average height of 30.48 m (100 ft). The wind duration curves were, consequently, scaled to this altitude by using the power law of turbulent boundary layers, i.e., the one seventh power of velocity relationship:

$$V_{30.48 \text{ m}} = V_{9.14 \text{ m}} \times \left(\frac{30.48 \text{ m}}{9.14 \text{ m}} \right)^{1/7} = 1.188 V_{9.14 \text{ m}}$$

These results are shown in Figure 69 for the increased mean wind speeds of 6.4 m/sec (14.25 mph) and 9.6 m/sec (21.4 mph) at this higher altitude, and will be used throughout this study.

7.3 System Operating Envelope

The Giromill system operating envelope and boundary constraints are discussed in this section. Systems with power outputs of 120, 500, and 1500 kW are considered in this study. These values denote the system power output when the wind speed is at the design speed, i.e., the rated speed V_R . The power output for wind speeds exceeding V_R is constant up to a maximum operating wind speed, or system cut-off speed of V_{CO} . At wind speeds above V_{CO} , the Giromill is stopped to prevent physical damage, and the blades will be feathered. Section 3.5 defined V_{CO} as 13.4 and 20.1 m/sec (30 and 45 mph) for the 5.4 and 8.1 m/sec mean wind speed, respectively. When scaled to the 30.48 m height, V_{CO} becomes 15.9 and 29.9 m/sec. As observed in Figure 69, the number of hours per year that the wind exceeds V_{CO} is very small for both wind speed distributions used in this study. Consequently, V_{CO} will not appreciably reduce the total annual power output for the Giromill system in this study.

Another value of wind speed must be considered in defining the operating envelope, that of the wind speed at which the generator begins producing power, cut-in speed, V_{CI} . For the present analyses the V_{CI} will be assumed as $0.6 V_R$.

Figure 69

Wind Speed Duration Curves for Two Values of Mean Wind Speed

- $V_{\text{RATED}} = 9.6 \text{ MPS} = 1.5 \times \bar{V}$
- $V_{\text{CUT IN}} = 0.6 \times V_{\text{RATED}}$

Figure 70

Estimation of Annual Energy Output

The approach to determining the total annual energy output is illustrated in Figure 70. This was derived for a $\bar{V} = 6.4$ m/sec (14.25 mph) and considers power output which is varying as the cube of the wind speed. A rated speed of $V_R = 1.5 \times \bar{V} = 9.6$ m/sec (21.4 mph), which results in the power output being constant at the rated value (P_R) for all wind speeds exceeding this value, will occur 1500 hours per year. Assuming a cut-in speed of $0.6 V_R$ ($V_{CI} = 5.8$ m/sec (12.8 mph)), which occurs for approximately 4950 hours per year, the total annual energy output is proportional to the shaded area of Figure 70. The ratio of the shaded area under this curve to the total annual output available if the Giromill were operating at V_R during the entire year (8760 hours) multiplied by 8760 is the system's Specific Output (θ) which is in kWhr/Yr/kW.

$$\theta = 8760 \times \frac{\text{Area of oacfg}}{\text{Area of oade}}$$

where oacfg is the shaded area.

The total annual energy output (E) for a Giromill system is the product of the power output (P_R) at rated wind speed (V_R) and the specific output (θ).

$$E = P_R \times \theta$$

Specific output can be related directly to the ratio of $(V_R/\bar{V})$. This assumes that for any value of time during the year, the ratio of wind speeds for two different mean wind speed curves is the same as the ratio of their $\bar{V}$ values. Comparison of the $\bar{V} = 6.4$ m/sec and $\bar{V} = 9.6$ m/sec curves indicates that this is a good approximation.

The approach discussed above estimates specific output and, subsequently, total annual energy output, assuming the power output for wind speeds below the Giromill rated speed V_R is proportional to the wind speed cubed. This simple V-cubed relationship provides an upper limit of energy output obtainable. Section 3.5 developed the relationship between wind speed ratio (V_W/V_R) and power output for the Giromill for wind speeds below V_R . This relationship is shown in Figure 71 which, as previously discussed, illustrates that power is not generated below the cut-in speed of $0.6 V_R$. Using this relationship for windspeeds below V_R and considering that power output above V_R is constant at the system rated power, the specific output was determined for ratios of

$(V_R/\bar{V})$ ranging from 0 to 2.5. These results are shown in Figure 72. This has been used to determine Giromill annual power output as the example in Figure 73 illustrates.

Figure 71
Comparison of Giromill Power Output/Wind Speed Relationship

Figure 72
Calculation of Annual Giromill Energy Output

Giromill Power Output = 120 kW

Mean Wind Speed ($\bar{V}$) = 6.4 mps @ 100 ft

Rated Wind Speed (V_R) = 9.6 mps

Therefore, ($V_R/\bar{V}$) = 1.50

From Figure 72, θ = 2980 kWhr/kW/Yr

Annual Energy Output $E = P_R \times \theta$

$$E = 120 \text{ kW} \times 2980 \text{ kWhr/kW/Yr}$$

$$E = 357,600 \text{ kWhr/Yr}$$

Figure 73

Calculation of Annual Giromill Energy Output

8. COST ANALYSIS

This section discusses the total production costs and operating and support (O&S) cost components associated with the Giromill subsystem. These have been developed for production unit 100 and considering a design life of 50 years for static components and a design life of 30 years for dynamic components, e.g., rotor blades.

Costs considered in the life cycle of a new system are, generally comprised of the following elements.

- (1) Research, Development, Test and Evaluation (RDT&E)
- (2) Production (includes purchase and preparation of land)
- (3) Recurring Costs (cost of capital and operation and support (O&S))

RDT&E and production costs comprise total program or acquisition costs. The present study will not address RDT&E elements but will concentrate on identifying the production and O&S costs.

Production costs include the cost of the various components and raw materials comprising the Giromill system, fabrication of raw materials, assembly and installation of the entire system. In addition, these costs include cost of special support equipment (SSE), if necessary, and cost of acquisition of the land, site preparation, and provision for security. The present analysis is intended as a feasibility assessment, and does not assume a specific application for the Giromill system such as a power generating source to be input to an existing electrical power plant. Consequently, the cost to hook-up to a power generating source using transformers, cables, and other equipment is not considered in this analysis.

Two major areas of recurring costs to be considered, are cost of capital and other Operation and Support (O&S) costs. The cost of capital includes the categories typically called investment charges, i.e., interest on bonds, dividends, depreciation, income taxes, and insurance. This cost of capital represents a large share of the cost of generating electricity and it is common practice to estimate these costs as a percentage of the total cost of the system installation.

Operation and Support (O&S) costs consider those components of direct personnel or labor, spare parts, supplies, and supervision. For a conventional electrical generating facility, fuel costs are a major component of the operating costs. The basic structure of O&S costs will be addressed in this section, but a more complete analysis will be included in the final report.

For cost analysis purposes, the Giromill is assumed comprised of several interacting subsystems. These subsystems are:

- o Foundation and Tower Structure
- o Electrical Components
- o Mechanical Power Transmission
- o Blades and Blade Supports
- o Blade Control

Individual components of these various subsystems are identified below. In cases where off-the-shelf components are available, the vendor costs have been obtained and used to derive cost estimating relationships (CER), adjusted to 1975 dollars, relating cost with size, weight, power requirements or other physical and/or performance characteristics. These identified relationships are subsequently used in production costs. Engineering judgement is utilized to assess production costs of non-shelf items such as the foundation and tower structure and the blade supports. McDonnell Aircraft Company producibility personnel were consulted to determine the material cost for the tower and both material and fabrication costs associated with developing the blade system. Considerable emphasis was placed on the design of the blade control system, and cost estimates for producing this system were determined by McDonnell Douglas Electronics Company.

Subsystem and component costs are discussed in Section 8.2. The total cost of producing energy for the 120 kW and 500 kW systems are presented in Section 8.3. Total production costs (dollars), system cost per rated output (dollars/kW) and cost per annual energy output (cents/kWhr) are presented in that section. Point design configurations were investigated, and their system costs determined. This analysis identified the range of system variables to be included in future studies to optimize the system design.

8.1 Basic Cost Elements

Basic cost elements discussed in this section are those components of cost that are related to the total system and are not identifiable with a specific component or subsystem. These cost elements are (1) cost of capital, (2) incidental costs, and (3) operation and support costs.

Cost of Capital - The cost of capital for the Giromill installation will be based on a percentage of the total system installed cost. A rate of 15% of the total installed cost per year is assumed to cover the following annual charges:

- o Depreciation
- o Cost of indebtedness, interest on bonds and notes
- o Federal income taxes
- o State and local taxes
- o Cost of equity, common and preferred stock

Figure 74 illustrates a possible financing structure for a Giromill system resulting in the aforementioned 15% annual cost of capital. The total financing as shown is comprised of bonds (60%), preferred stock (10%) and common stock (30%). The 15% Cost of Capital agrees with values used for other current wind power generation programs.

Incidental Costs - The incidental cost of final assembly and total shipping costs are assumed at 10% of the costs for all subsystem components. At the subsystem level some installation and fabrication costs have already been included for such elements as the tower and blade supports and rotor blades. For example, both tower and blade support structure are assumed as \$1.00/lb, whereas the cost of the channels and I beams comprising most of the tower structure is on the order of \$.13 to \$.16/lb and pipe for the blade supports on the order of \$.22/lb.

Operations and Support Cost - Annual operations and support cost are assumed to be 4% of the total Giromill installed cost. A more detailed breakdown of the components of support cost will be defined during the remainder of the study. Specifically, the following elements will be considered:

- o Salary for maintenance and supervisory personnel
- o Maintenance and overhaul costs
- o Administrative and general expenses
- o Insurance
- o Miscellaneous expendables

This value of 4% is conservative when compared with O&S costs for electrical utilities but consistent with those for other wind power generation programs.

8.2 Production Costs for Giromill Subsystems

Production costs discussed in this section will include the following cost elements:

- o Raw materials
- o Off-the-shelf components
- o Manufacturing and assembly costs
- o Purchase and preparation of land, security provisions

ELEMENT OF COST OF CAPITAL ANALYSIS (ANNUAL)	PORTION OF INSTALLED SYSTEM COST
INCOME GENERATED	.15000
LESS: DEPRECIATION (30 YEARS)	.03333
LESS: BOND INTEREST	.04800
INCOME BEFORE TAXES	.06867
LESS: FEDERAL INCOME TAX @ 48%	.03296
INCOME AFTER TAXES	.03571
LESS: STATE AND LOCAL TAXES @ 4%	.00143
LESS: PREFERRED STOCK DIVIDENDS	.00800
COMMON STOCK EQUITY	.02628
RETURN ON COMMON EQUITY	8.76%

ASSUMED CAPITAL STRUCTURE:

60% BONDS @ 8% INTEREST
 10% PREFERRED STOCK @ 8% DIVIDEND
 30% COMMON STOCK

Figure 74
 Cost of Capital Analysis for Typical
 Capital Structure

The productions costs are based on costs associated with the Giromill production unit 100 and based on current 1975 prices. For cost analysis purposes it was assumed that all manufactured components, such as rotor blades, are manufactured within a time span of three years. Costs for major off-the-shelf components such as generators, speed increasers, couplings and bearings were obtained either from vendor catalogs or from the vendors directly. Cost estimating relationships (CER) were developed for some of these components to permit efficient calculations for point design analyses of the Giromill systems. The following Giromill subsystems are discussed below.

- o Foundation and tower structure
- o Electrical Components
- o Mechanical power transmission
- o Blades and blade supports
- o Blade control

Foundation and Tower Structure - Cost analysis for the foundation and tower structure began with selection of the appropriate Giromill tower structure, i.e., concept, weight, material. An inexpensive carbon steel, such as A36, was selected for this structure with I beams and channels for the major structural members. Final assembly at the site will require welding and/or bolting to join the members. Raw material costs were obtained for typical channels and I beams, and from these cost per pound of material was determined for large quantity purchases. An approximate cost per pound of raw material was determined for this analysis. Final tower assembly will be based on a total of \$1/pound for material and installation consistent with previous MCAIR experience for windmill towers and with other current wind power system programs. Typically, cost of tower foundations have been estimated as a fraction of tower costs. In the present analysis, a value of 25% of the tower cost, e.g., both the lower tower and the upper rotating tower, was assumed for the foundation.

Electrical Component Costs - The electrical components as well as some components of the mechanical transmission will be protected from direct contact with the environment by a simple shroud. Consequently, they will not be totally protected from the environment and may experience temperature extremes, for example, from -40°F to 120°F, depending upon location. No environmental control system (ECS) or heaters for the generator are considered. The major cost components for the electrical system is the generator. In addition, a voltage regulator and simple control panel and indicators will be required.

The CER developed for 100 kW and 500 kW generators is shown in Figures 75 and 76. A 15% increase over vendor data costs was applied to these data to account for the higher cost of mounting the generator vertically as suggested by a generator vendor. Cost data for the 100 kW generator will be utilized in this study for the 120 kW system because only a minor cost increase is anticipated to increase output from 100 kW to 120 kW. The total cost of all electrical components other than the generator is assumed to be the same as the generator for the 120 kW system, i.e. \$4025.

Mechanical Power Transmission - The mechanical power transmission system is described and shown for the 120 kW Giromill in Section 4.4. Primary components of this system include:

- o RPM speed increaser
- o Couplings - generator to speed increaser and speed increaser to drive shaft
- o Bearing - combination on drive shaft
- o Adaptor - shaft to rotor
- o Drive shaft

CER developed for these major components of the mechanical power transmission are shown in Figures 77 through 80. The speed increaser cost shown in Figure 77 has been increased by 15% over vendor data to account for the additional expenses of vertical mounting. As shown, the speed increaser cost increases as the gear ratio increases, i.e., as the Giromill rpm decreases, and this will be considered in costing alternate point designs. Again, the 100 kW data will be utilized for analysis of the 120 kW system. Costs for couplings versus diameter are shown in Figure 78; costs for bearings versus diameter are shown in Figure 79 (large size) and 80 (small size). Cost versus diameter for the couplings is also utilized in determining the shaft to rotor adaptor cost. Figure 81 summarizes characteristics, weight, cost and cost per unit weight for the components of the 120 kW and 500 kW system. With the exception of the speed increaser, whose cost varies with Giromill rpm, the costs for all components described in Figure 81 are the same for all point designs with the same Giromill output power.

Blade and Blade Support - The fabrication of rotor blade supports emphasizes the use of simple member types (tubular cross-section) and an inexpensive material, for example, A36 carbon steel. This approach should enable the support members costs to be closely approximated by the tower structure cost of \$1.00/lb. Raw material costs are approximately \$.22/lb.

Westinghouse Generator Unit Costs
Costs Increased by 15% for Vertical Mounting

Figure 75
Cost Relationship for 100 kW Synchronous Generator

Westinghouse Generators

Costs Increased by 15% for Vertical Mounting

Figure 76

Cost Relationship for 500 kW Synchronous Generator

Falk Speed Increasers

Costs Increased by 15% for Vertical Mounting

Figure 77

Cost Relationships for 100 kW and 500 kW Speed Increasers

Morse A-218 Industrial Radial Couplings
 1969 Prices Increased by 30%

Figure 78
 Cost Relationship for Flexible Coupling

Rotex Bearings Unit Cost

- ⊙ Series 3000
- △ Series 10000

Figure 79
Cost Relationship for Large Diameter Combination Bearing

Messinger Combination Bearing Cost

Figure 80
Cost Relationship for Small Diameter Combination Bearings

- o 120 kW Configuration 2
- o 500 kW Configuration 14

COMPONENT	CHARACTERISTICS 120KW FIRST LINE 500KW SECOND LINE	120 KW			500 KW		
		WEIGHT (LB)	COST (\$)	($\frac{\$}{LB}$)	WEIGHT (LB)	COST (\$)	($\frac{\$}{LB}$)
SPEED INCREASER *	GEAR RATIO 50:1 (1200 RPM) GEAR RATIO 22.8:1 (450 RPM)	7100	19,000	2.68	8955	24,000	2.68
COUPLING - GENERATOR TO SPEED INCREASER	3.5" DIA 6.42" DIA	128	640	5.00	366	1860	5.08
FLEXIBLE COUPLING - GEN. TO DRIVE SHAFT	7.25" DIA 13.29" DIA	600	2400	4.00	2016	8064	4.00
ADAPTOR - SHAFT TO GENERATOR	7.25" DIA 13.29" DIA	188	877	4.67	632	2947	4.67
DRIVE SHAFT	6" O.D., 4" I.D. x 12" 11" O.D., 9" I.D. x 12"	53	106	2.00	107	214	2.00
THRUST / RADIAL BEARING	6" DIA 11" DIA	42	420	10.00	78	780	10.00
TOTAL (EXCLUDING SPEED INCREASER)		1011	4443	4.39	3199	13,865	4.33

* Includes 15% Cost Increase for Vertical Mounting

Figure 81
Mechanical Power Transmission Components
and Characteristics

Rotor blades were expected to be major contributors to the total Giromill system cost. As discussed in Section 4.2, a simple blade design is considered which does not require complexities of twist, camber, and taper associated with most conventional windmill designs. The McDonnell Aircraft Company Producibility department was consulted to obtain raw material and labor costs for rotor blade fabrication. Rotor blades for the 120 kW system identified as configurations 1, 2, and 6, were considered because they covered a wide range of blade span, chord, and weight per unit length. Results from these analyses indicated that total blade cost per unit weight varied only slightly with design. Assuming a manufacturing cost of \$12 per hour, the blade cost of approximately \$4.00/lb was determined.

Blade Control System - The conceptual blade control system utilizes individually modulated blades driven by a control servo motor. The McDonnell Douglas Electronics Company of Grand Rapids, Michigan, was consulted to prepare cost estimates for a rock angle actuator to direct the rotor blades to follow a prescribed rock angle history and provide torque to flip the blades where necessary. Cost estimates for actuators and associated electronics are summarized in Figure 82 for the 120 kW, 500 kW and 1500 kW systems based on recurring cost for the 300th unit, i.e. assuming 3 actuators for each of the 100 Giromills.

Giromill Size (kW)	Estimated Cost per Actuator
120	\$2400
500	\$3400
1500	\$4600

Figure 82
Estimated Costs for Rotor Blade Actuators

Actuators have been sized with adequate torque for turning the total blade span at each blade station. For the 120 kW system all the blades are in two sections having a support at the one-half span location, and use one control actuator. The 500 kW system, in general, requires three or four blade sections separated by supports. In determining the 500 kW control system costs,

two blade control actuators are assumed for Giromill configurations having three or more blade sections, even though only one actuator may be adequate.

Other components of the blade control system are currently being defined and cost estimates being prepared. These other components include the following sensors and devices:

- o Giromill rpm sensor
- o Blade angle position sensors
- o Rotor angle position sensor
- o Wind velocity sensor and direction sensor
- o Vibration sensors
- o Emergency pyrotechnic devices

These components are not expected to be significantly affected by Giromill power output, and a preliminary cost of \$10,000 for all system sizes is assumed for the cost analysis.

8.3 Total Giromill System Cost

Elements of the Giromill total system costs are presented. The major cost elements for the 120 and 500 kW systems are summarized in Figure 83. The cost of electrical system components other than generator are assumed to be the same as the generator for the 120 kW system i.e., \$4025. The cost of the site, site development and security is assumed to be \$15,000 for the 120 kW system and \$30,000 for the 500 kW system. In addition, final system installation cost is assumed to be 10% of the total of the subsystem costs. The cost of capital, as shown, is 15% of the total Giromill installed cost, and O&S costs are assumed to be 4% of the total.

Using the cost inputs from Figure 83, the CER of Figures 76 through 80, and structural weights (rotor blades, support arms and tower) from Section 5.5, Giromill total system costs and cost per unit of energy were derived for the 120 kW and 500 kW systems. Production costs for the seven 120 kW systems are summarized in Figure 84 with the corresponding cost of energy produced summarized in Figure 85. Using Configuration 2 as a baseline, 35% of the system cost is attributed to the foundation and tower structure, 18% to the power transmission equipment, and 13% to the rotor blades. The cost of a pair of main bearings for the Giromill have been separated from the other transmission components because of their large contribution to the total system cost and because the bearing costs vary significantly from configuration to configuration. Figure 85 summarizes Giromill cost, costs for site pur-

ELEMENT	MATERIAL OR DESCRIPTION	COST FACTORS	
		120 KW	500 KW
TOWER LOWER & UPPER (ROTATING)	CARBON STEEL WELDED OR BOLTED	\$1.00/LB	
FOUNDATION	CONCRETE	25% OF TOWER COST	
ROTOR BLADE SYSTEM BLADE SUPPORTS ROTOR BLADES MAIN BEARINGS (2)	TUBULAR CARBON STEEL WELDED OR BOLTED ALUMINUM DIAMETER OF UPPER TOWER	\$1.00/LB \$4.00/LB \$9 TO \$10/LB	
DRIVE SYSTEM COUPLINGS BEARINGS SPEED INCREASER*	SPEED INCREASER TO GEN., TO DRIVE SHAFT THRUST/RADIAL BEARING ON DRIVE SHAFT PRICE INCREASES WITH GEAR RATIO	\$4 TO \$5/LB \$9 TO \$10/LB \$15,000 TO \$22,000	\$4 TO \$5/LB \$9 TO \$10/LB \$26,500 TO \$33,000
ELECTRICAL GENERATOR *	120KW (1200 RPM), 500KW (RPM OPTIMIZED FOR MINIMUM GENERATOR PLUS SPEED INCR COST)	4025	\$23,000 TO \$27,000
OTHER COMPONENTS	CONTROL PANEL, GAUGES	4025	4025
CONTROLS SERVO ACTUATORS OTHER COMPONENTS	MDEC GRAND RAPIDS ESTIMATE SENSORS, ETC.	\$7200 ⁽¹⁾ \$10,000	\$20,400 ⁽²⁾ \$10,000
ADDITIONAL CAPITAL	INSTALLATION, ETC. LAND PURCHASE, SITE DEVELOPMENT, SECURITY	10% OF CAPITAL \$15,000	10% OF CAPITAL \$30,000
ANNUAL CHARGES COST OF CAPITAL	LOAN REPAYMENT, DEPRECIATION, BOND INTEREST STOCK DIVIDEND, INSURANCE, TAXES	15% OF TOTAL CAPITAL	
OPERATING & SUPPORT		4% OF TOTAL CAPITAL	
TOTAL		19% OF TOTAL CAPITAL	

* 15% INCREASE IN VENDOR PRICES FOR VERTICAL MOUNTING

(1) 3 ACTUATORS AT \$2400

(2) 6 ACTUATORS AT \$3400

Figure 83

Preliminary Cost Estimating Relationships for 120 kW and 500 kW Giromill
o 1975 Dollars

CONFIG. NO.	UPPER & LOWER TOWER.	FOUND- ATION	ELECTRICAL		POWER TRANSMISSION		CONTROLS	MAIN BEARINGS	BLADE SUPPORT	ROTOR BLADES	TOTAL
			GENER.	OTHER	SPEED INCREASER	OTHER					
1	37,330	9332	4025	4025	17,500	4443	17,200	11,400	8,220	21,720	135,195
2	37,330	9332	4025	4025	19,000	4443	17,200	11,400	8,330	16,920	120,005
3	37,330	9332	4025	4025	20,000	4443	17,200	11,400	8,390	19,000	135,145
4	31,020	7755	4025	4025	20,300	4443	17,200	8,200	12,430	15,800	125,198
5	27,280	6820	4025	4025	22,200	4443	17,200	5,900	21,050	15,800	128,243
6	50,350	12,588	4025	4025	21,100	4443	17,200	16,600	13,190	23,320	166,841
7	29,340	7335	4025	4025	15,000	4443	17,200	8,800	5500	12,800	108,468
CONFIG. NO. 2 PORTION	.283	.071	.030	.030	.144	.034	.130	.086	.063	.128	1.000

Figure 84
Production Cost Elements for 120 kW Giromill Configurations
o Major Components & Subsystems

CONFIG. NO.	GIROMILL COST	SITE DEVELOP. & INSTALL.	TOTAL INSTALLED COST	INSTALLED COST (\$/KW)	POWER OUTPUT		ANNUAL CHARGES			ENERGY COST ($\frac{\text{¢}}{\text{KWHR}}$)
					SPECIFIC HR	TOTAL $\frac{\text{KWHR}}{\text{YR}}$	COST OF CAPITAL 15%	OPS 4%	TOTAL 19%	
1	135,195	28,520	163,715	1364	2980	357,600	24,551	6549	31,106	8.70
2	132,005	28,201	160,206	1335	2980	357,600	24,031	6408	30,439	8.51
3	135,145	28,515	163,660	1364	2980	357,600	24,549	6546	31,095	8.70
4	125,198	27,520	152,718	1273	2980	357,600	22,908	6109	29,016	8.11
5	128,243	27,824	156,067	1301	2980	357,600	23,410	6243	29,653	8.29
6	166,841	31,684	198,525	1654	3750	450,000	29,779	7941	37,720	8.38
7	108,468	25,847	134,375	1119	2320	278,400	29,156	5375	25,531	9.17

Figure 85
Cost of Energy Produced for 120 kW Giromill Configurations

chase and development and system installation, cost per kilowatt, annual power output and annual charges and cost per kilowatt-hour. It is significant to note that the configuration having the lowest cost per system rating (\$/kW), i.e., Configuration 7, has the highest cost per unit output of energy (¢/kWhr). This is a result of Configuration 7 having a higher rated speed (V_R) that results in a significantly lower specific output (θ) and total annual energy output.

A similar presentation of costs for eight 500 kW Giromill configurations is shown in Figures 86 and 87. Using Configuration 14 as a baseline, a major portion of the total system cost, 28% (largest single cost item) is attributed to a pair of large main bearings. Design modifications to reduce this portion of the system cost are discussed in Section 5.1 and will be considered during the remainder of the study.

Results of energy cost (¢/kWhr) from the two system sizes (120 kW and 500 kW) are plotted in Figures 88 and 89 versus solidity ratio (σ), speed ratio ($V_R/\bar{V}$), and rotor aspect ratio (AR_R). These figures suggest that a reduced energy cost may be achieved within the range of variables shown in Figure 90 below. The results of such a cost sensitivity analysis for the Giromill systems will be considered during the remainder of the study.

VARIABLE	120kW	500kW
ASPECT RATIO	0.6 to 0.8	Less than 0.6
($V_R/\bar{V}$)	Less than 1.4	1.4 to 1.6
SOLIDITY (σ)	0.05 to 0.07	0.07 to 0.07 & less than 0.03

Figure 90
Range of System Variables for Minimizing Energy Costs

CONFIG. NO.	UPPER & LOWER TOWER	FOUND- ATION	ELECTRICAL		POWER TRANSMISSION		CONTROLS	MAIN BEARINGS (DIA)	BLADE SUPPORT	ROTOR BLADES	TOTAL
			GENER.	OTHER	SPEED INCREASER	OTHER					
8	127,490	31,873	25,000 (514)	4025	26,500	13,865	30,400	162,637 (222)	38,780	55,840	516,410
9	134,800	33,700	25,000 (514)	4025	28,500	13,865	30,400	162,637 (222)	37,060	67,840	537,827
10	134,800	33,700	25,000 (514)	4025	30,800	13,865	30,400	162,637 (222)	36,970	59,200	531,397
11	69,930	17,483	27,000 (450)	4025	30,200	13,865	30,400	72,000 (147.6)	79,100	29,400	373,403
12	83,550	20,888	27,000 (450)	4025	30,600	13,865	30,400	93,000 (168)	60,760	43,720	407,808
13	200,680	50,170	27,000 (450)	4025	30,800	13,865	30,400	229,997 (264)	58,750	77,320	723,007
14	95,840	23,960	23,300 (600)	4025	27,000	13,865	30,400	118,629 (189.6)	24,060	62,920	423,999
15	155,320	38,830	27,000 (450)	4025	32,700	13,865	20,200	162,637 (222)	45,260	85,120	584,959
CONFIG. NO. 14 PORTION	.226	.057	.055	.009	.064	.033	.072	.279	.057	.148	1.000

Figure 86
Production Cost Elements for 500 kW Giromill Configurations
o Major Components & Subsystems

CONFIG. NO.	GIROMILL COST	SITE DEVELOP. & INSTALL.	TOTAL INSTALLED COST	INSTALLED COST (\$/KW)	POWER OUTPUT		ANNUAL CHARGES			ENERGY COST (\$ /KWH)
					SPECIFIC HR	TOTAL KWHR YR	COST OF CAPITAL 15%	D&S 4%	TOTAL 19%	
8	516,410	81,641	598,051	1196	2980	1,490,000	89,708	23,922	113,630	7.63
9	537,827	83,783	621,610	1243	2980	1,490,000	93,242	24,864	118,106	7.93
10	531,397	83,140	614,537	1229	2980	1,490,000	92,181	24,581	116,762	7.84
11	373,403	67,340	440,743	881	2980	1,490,000	66,111	17,630	83,741	5.62
12	407,808	70,781	478,589	955	2980	1,490,000	71,788	19,144	90,932	6.10
13	723,007	102,301	825,308	1651	3750	1,875,000	123,796	33,012	156,809	8.36
14	423,999	72,400	496,399	993	2320	1,160,000	74,460	19,856	94,316	8.13
15	584,959	88,496	673,455	1397	2980	1,490,000	101,018	26,938	127,956	8.59

Figure 87
Cost of Energy Produced for 500 kW Giromill Configurations

Figure 88

Giromill Energy Cost Sensitivity for 120 kW and 500 kW Systems

- VARIATION WITH ASPECT RATIO
- $V_R/\bar{V} = 1.5$

Figure 89

Giromill Energy Cost Sensitivity for 120 kW and 500 kW Systems

9. CONCLUSIONS

A parametric design and cost effectiveness analysis of the 120 and 500 kW Giromill systems has been completed. The analysis of the 1500 kW systems is continuing.

Evaluation of the data generated to date indicates that the Giromill system is indeed cost effective competitive with a conventional windmill. This is indicated even though very little Giromill system optimization has been accomplished. System optimization of the identified high cost items should reduce the cost of the Giromill systems well below that of comparable conventional windmills.

Conservative design criteria have been applied to the Giromill designs. Present state of art construction techniques and materials have been assumed. Where possible off the shelf hardware has been used. The Giromill systems as defined can be built at this time.

10. REFERENCES

1. "Wind Generator System - Final Design Review," presented to NASA Lewis Research Center, 16 July 1975, Kaman Aerospace Corp.
2. "Wind Turbine Generator Study - Final Presentation," presented to NASA Lewis Research Center, 17 July 1975, General Electric Space Division.
3. "Windmill Sizing Study," McDonnell Aircraft unpublished data, 1975.
4. Abbott, I. H. and vonDoenhoff, A. E., "Theory of Wing Sections," Dover Publications, Inc.
5. Jacobs, E. N. and Sherman, A., "Airfoil Section Characteristics as Affected by Variations of the Reynolds Number," NACA TR 586, 1937.
6. Jacobs, E. N. and Abbott, I. H., "Airfoil Section Data Obtained in the NACA Variable Density Tunnel as Affected by Support Interference and Other Corrections," NACA TR 669, 1939.
7. "Structural Standards for Steel Antenna Towers and Antenna Supporting Structure," EIA Standard RS-222-A, November 1966.
8. Baker, Ralph D., "Wind Tunnel Test on Amplified High Pitch Cyclogiro," University of Washington, Aeronautical Laboratory, Seattle, Washington, Sept 7, 1946.
9. Kirsten, Frederick K., "Cycloidal Propulsion in Air," No. 79, Engineering Experiment Station Series, University of Washington, Seattle, Washington, 1935.
10. Wheatley, John B., Windler, Ray, "Wind-Tunnel Tests of a Cyclogiro Rotor," NACA TN 528, May 1935.