To the Graduate Council:

I am submitting herewith a thesis written by John W. Kaufman entitled "Advanced and Innovative Wind-Wheel Turbine." I recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Engineering Science.

Walter Frost

Walter Frost, Major Professor

We have read this thesis
and recommend its acceptance:

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Vice Chancellor
Graduate Studies and Research

ADVANCED AND INNOVATIVE WIND-WHEEL TURBINE

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

John W. Kaufman

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ABSTRACT

An investigation was made to determine the response characteristics of an advanced and innovative wind-wheel turbine (WWT). This WWT was conceived by Mr. John W. Kaufman of the Marshall Space Flight Center, Alabama 35812 in April 1976. The basic apparatus is the subject of patent application NASA No. 880726 filed on February 24, 1978, pending before the U.S. Patent Board.

Two scale models of the WWT were fabricated and tested in the Space Sciences Laboratory at NASA-MSFC where this investigator is employed. A jet of air, in which the flow rate was varied and closely monitored, was directed toward each model in separate tests to acquire wheel speed and acceleration data. Hand held anemometers, a precision stop watch, an electronic counter, and other devices were used to determine jet air flow speeds and wheel revolution with time values.

Preliminary tests of these two models demonstrate the feasibility, practicality, favorable response, and potential benefits of using this WWT as a wind energy conversion system (WECS). Performance data show that the system is a high torque but low blade tip speed to wind speed device. In that the theoretical and empirical finds were quite favorable, plans are to build a precision and properly scaled model for wind tunnel and environmental testing.

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FIGURE PAGE

LIST OF SYMBOLS

A	Area perpendicular to wind direction
C	Cost
C_p	Power coefficient
C_T	Torque coefficient
D	Diameter
$\vec{F}_S$	Surface force acting on control volume surface
g_c	32.2 lbm/slug
h_T	Height of tower
I	Moment of inertia
$\vec{M}$	Torque
$\vec{M}_S$	Moment of surface forces acting on control volume
$\hat{n}$	Outwardly drawn normal to control volume surface
p	Pressure
P_R	Rated power
Q_R	Rated torque
r	Radius coordinate
R	Radius of rotor
S	Surface of control volume
t	Time
T	Torque
T_e	Equivalent thrust
$\vec{T}$	Surface stress vector
$\vec{u}$	Velocity
u_r, u_θ	Velocity components in cylindrical coordinates

V	Wind velocity
$\ddot{V}$	Volume of control volume
w	Width of rotor
W_S	Shaft work passing through control volume
$X,Y,Z,$	Coordinates
x,y,z	Coordinates
θ	Angular coordinate
λ	Blade tip speed ratio
ρ_a	Density of air
ω	Angular velocity
$\dot{\omega}$	Angular accelerate

CHAPTER I

INTRODUCTION

This advanced and innovative wind-wheel turbine (WWT) concept was conceived by John W. Kaufman, National Aeronautics and Space Administration (NASA), Marshall Space Flight Center (MSFC), Alabama, in April 1976. The preliminary development and WWT model testing was carried out to obtain turbine performance criteria as discussed in this study. The basic apparatus is the subject of patent application NASA number 880726, filed February 24, 1978, pending before the United States Patent Board.

The technical discussion of the innovative wind-wheel turbine concept is presented in two parts. In Chapter II a description of the WWT concept is provided along with identification of the key features and characteristics, the major unknowns, and the research which was carried out. In Chapter III the potential of the concept is discussed. The result of a preliminary analytical performance evaluation is presented, and a discussion of the inherent advantages of the system is provided [1].[1] A preliminary comparison with conventional wind energy conversion systems is also included.

All testing of the two WWT models was done by placing the models, one at a time, downwind from a controlled jet of air. Such tests were conducted on a workbench in the Fluid Dynamics Branch, Atmospheric Sciences Division of the Space Sciences Laboratory, NASA-MSFC.

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[1]Numbers in brackets refer to similarly numbered references in the Bibliography.

One special factor must be realized. WWT model number 1 is made of construction paper and pasteboard and is not built to scale. This model was put together prior to the "disclosure of invention" as documented under NASA case number 23515 which presents the initial design of the innovative system. Model number 2, also not built to scale, is a very heavy stainless steel model which was purposely built to withstand much higher wind speeds than the flimsy paper model.

Plans are to build a very precision scale model of the WWT and conduct wind tunnel and atmospheric environmental tests to generate some more accurate WWT response criteria.

As stated in the disclosure of invention documentation on the WWT, the system should be designed with the following special features:

1. An air escape vent should be built into the base of the main housing floor to permit the air to escape from the bottom of the main housing.

2. An electric generator could be a main part of the bladed wheel itself. Here, the blades would only have to be about one-half the outer radius of the wheel, allowing the center one-half radius of the wheel to contain a separate generator or be composed of generator components (i.e., stator, magnets, wire windings, commutator (if required), voltage regulator, etc.). A generator, or generators, can be made to operate in a number of ways, depending on the selected design.

3. The WWT is a "tuneable to wind speed system." Through testing it will be found that the internal air duct deflectors should be elevated or lowered, depending upon the mean wind speed. For example, the ducts (deflectors) must be opened to allow maximum possible air flow through them during periods of high wind speeds to keep the wheel

from spinning at an excessive turn speed. The deflector positions for duct sizing can be controlled by electromechanical devices or by wind pressure devices which would be determined through testing.

4. When operating in extremely high wind speed conditions, the WWT should be turned 180 degrees to its normal operating position. As a result, the wheel will turn only about one-fourth its normal operating speed but will still function and produce power satisfactorily.

5. The sides of the front ducts should project outward approximately 10 degrees. This is required so that approximately 65 to 70 percent of the wind (about $\pm 1\ \sigma$ of the wind direction variability) enters the forward ducts. The two side ducts should project approximately 30 degrees out from the parallel side walls of the main housing. Subsequently, for a wind direction variability of ±30 degrees from the mean direction is a significant amount of the flow will enter the side ducts. That is, the WWT system discussed herein is designed to operate optimally under a $\pm 1\ \sigma$ wind direction value of 10 degrees or a total variation of 60 degrees. It will be determined from wind tunnel and field tests of the WWT that the deflection angles of the ducts should be changeable according to wind direction and wind speed variability.

6. The WWT has eight blades. The optimum number of blades and actual pitch of blades (i.e., distance from tip of blade to innermost radial surface) must also be determined from tests of the WWT.

7. The optimum width of the wheel as compared to the radius must be determined through analytical and/or actual experimental tests to provide adequate surface area for wind loading.

8. The cowling, or the top surface of the front portion of the main housing, must be turned upward to deflect the flow of air

accordingly. In doing so, the wind is not permitted to blow directly onto the leading edge of the blades when they are in the horizontal forward position. The shape and curvature of cowling also must be determined from actual testing.

Other design features will be discussed in the following chapters; however, two things should be mentioned at this point. First, a full-scale WWT must have lightning arrestors to protect the overall system and personnel from lightning discharges. Second, the top half of the wheel may be covered with a specially designed and securely fastened hood. This is a possible requirement to protect the wheel and other internal subcomponents from snow and ice. If the hood cover for the wheel is not practical, then means to heat the wheel may be possible to alleviate wheel icing. A small amount of electrical power generated by the WWT system can be used to provide such heat. It is not known at this point whether wheel icing will be a problem, but it is anticipated.

In summary, the preliminary WWT performance analyses included in this study must not be considered optimum. This is due to the lack of properly scaled models and the models were only given a bench test where a jet of air from a pressurized tank was used as the wind source. These and other factors must be overcome in future WWT fabrication and testing.

CHAPTER II

CONCEPTUAL DESCRIPTION

I. SYSTEM DESCRIPTION

The conceptual basis of the wind-wheel turbine (WWT) was closely coordinated with Dr. Walter Frost of The University of Tennessee Space Institute (UTSI), Tullahoma, Tennessee [1]. Dr. Frost has been this investigator's advisor from the onset of his graduate studies at UTSI in 1975. The basic apparatus is the subject of a patent application, NASA number 880726, filed on February 24, 1978, as is pending before the United States Patent Board. The disclosure of invention is documented under NASA Case number 23515 and presents the initial design of the innovative system. Preliminary tests described subsequently indicate, however, that the final optimized system design may be different, depending on additional analytical and experimental studies of a properly scaled model.

The WWT is illustrated in Figures 1 through 4. The main parts of the WWT are: (1) bladed wheel, (2) main housing, (3) two forward ducts (front concentrators), (4) one duct on each side of the main housing (side concentrators), (5) an elevated base to support the functional parts of the WWT (a columnar support is indicated although appreciable study must be devoted to determine the best design), and (6) electrical/mechanical systems to be determined. The apparatus plurally directs air (wind) onto the blades of the rotatable wheel through the multiple duct units, and by direct impingement of wind onto

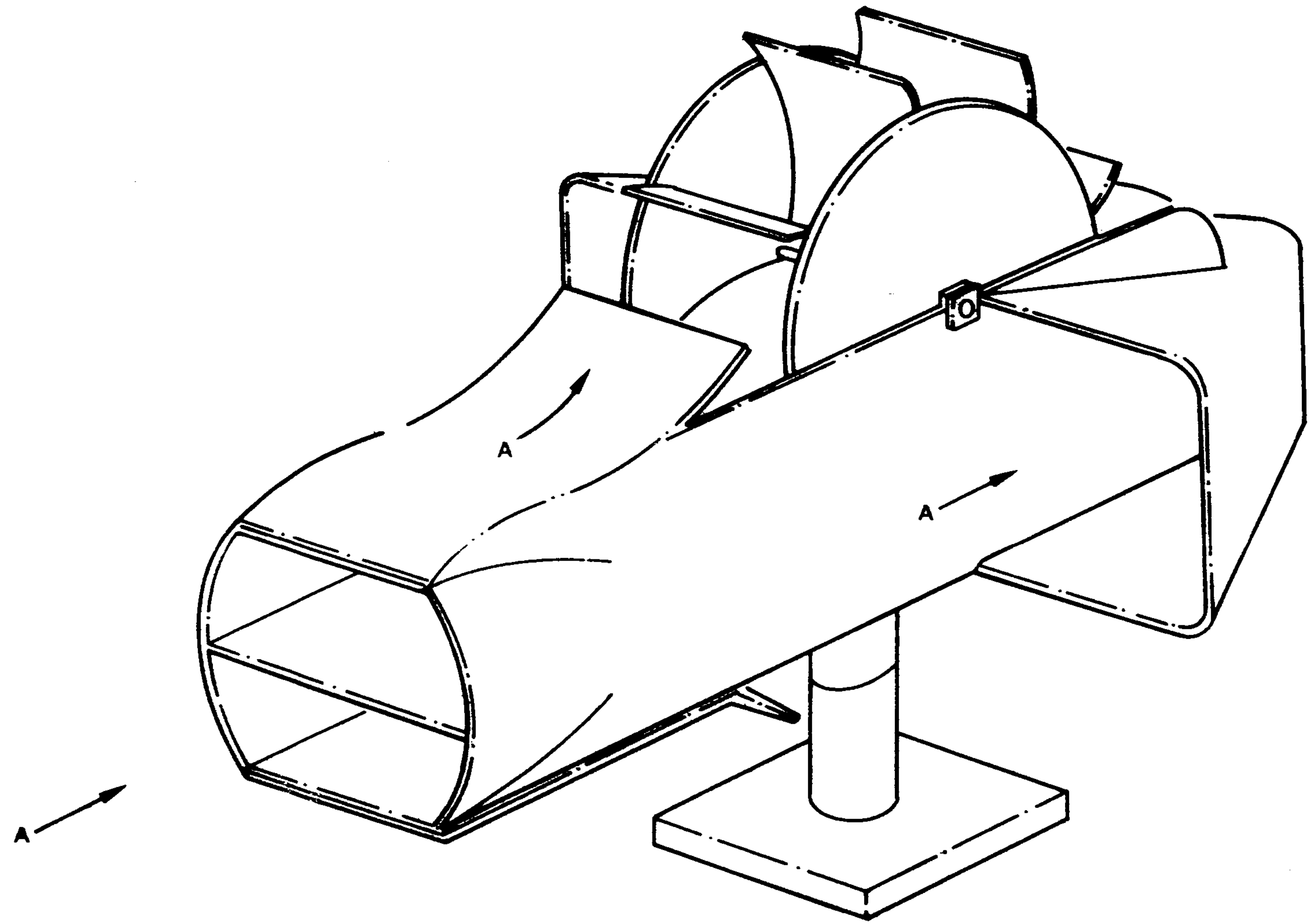

Figure 1. Wind-wheel turbine preliminary design.

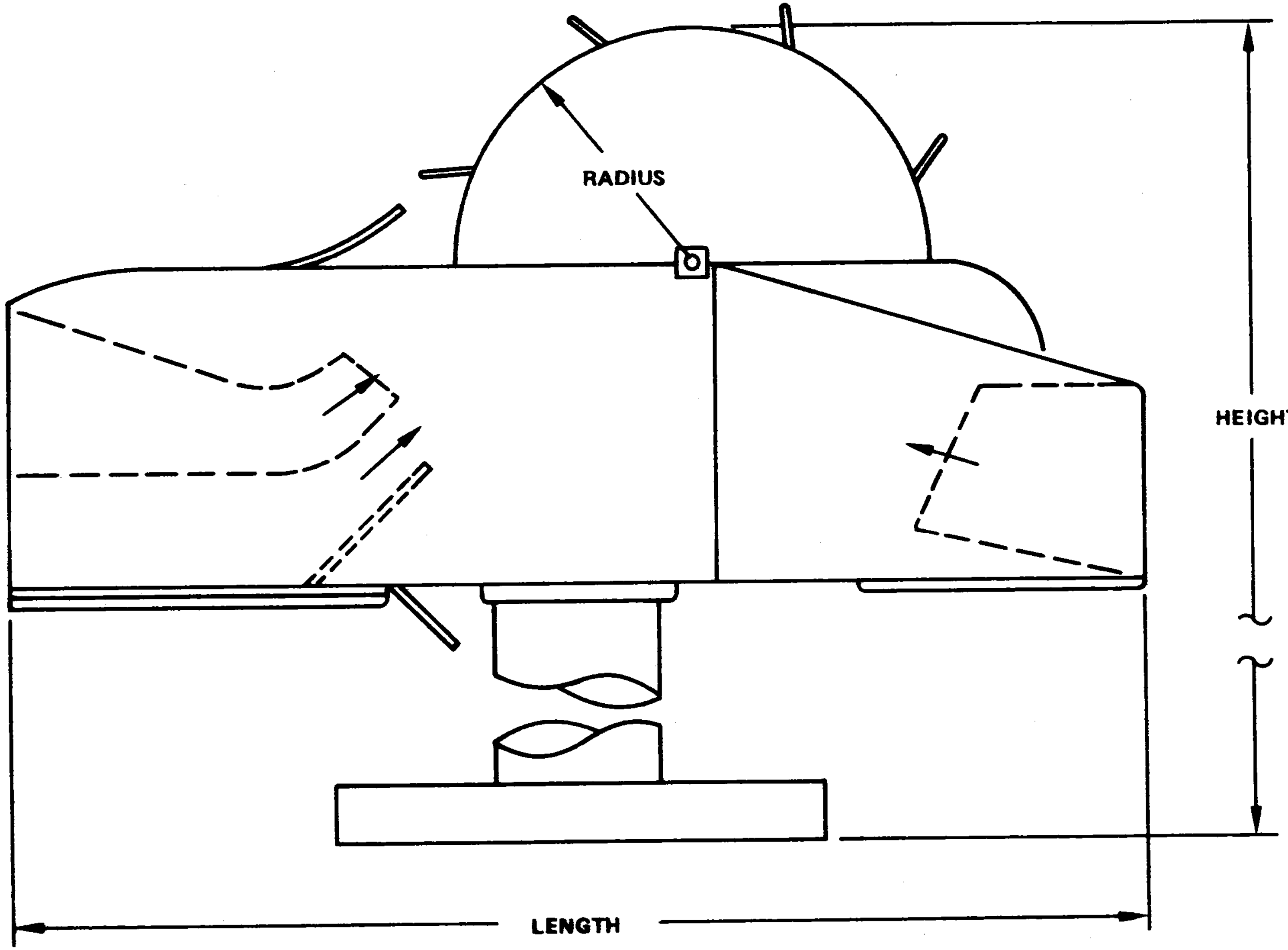

Figure 2. Side view of wind-wheel turbine.

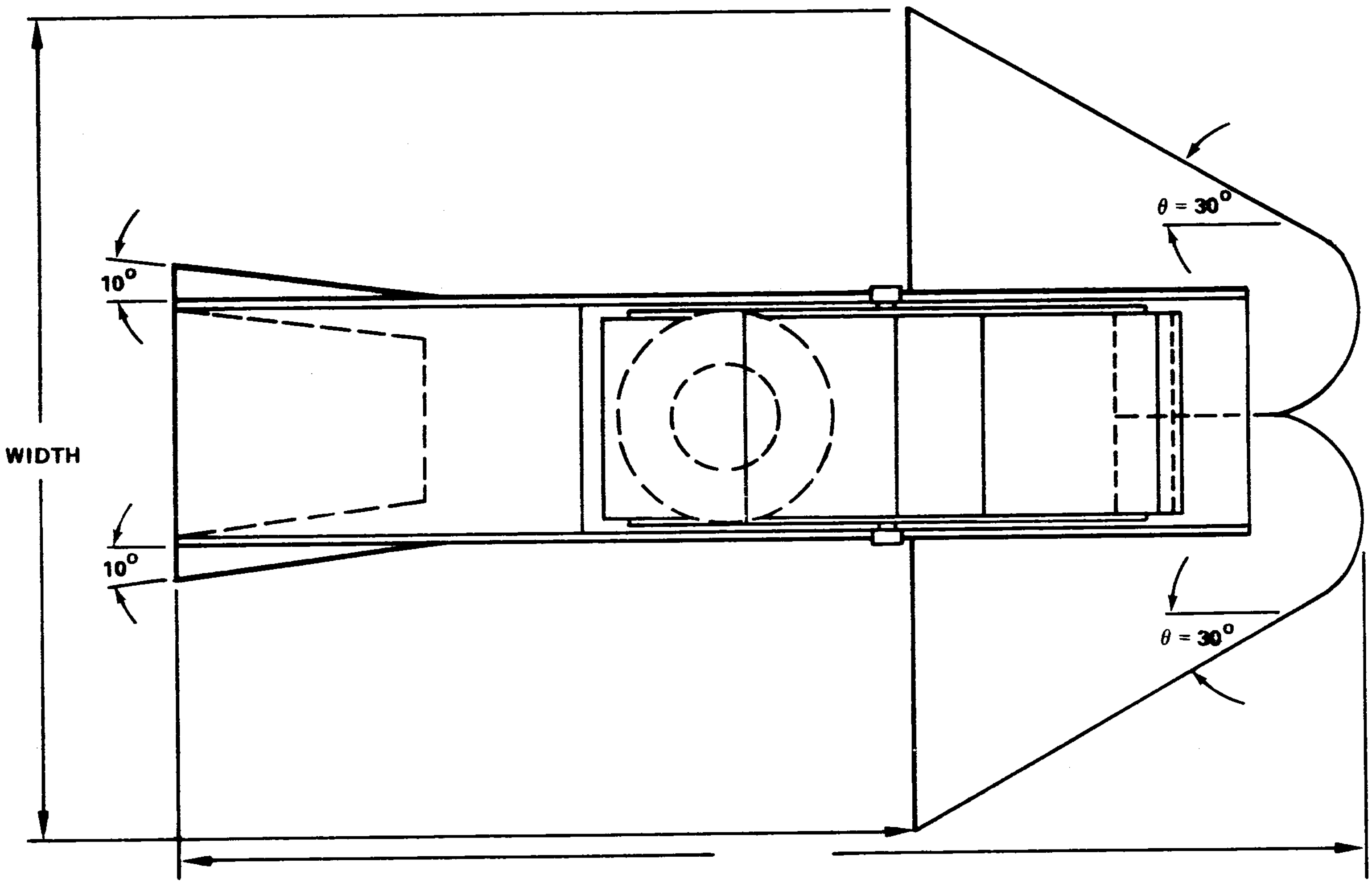

Figure 3. Top view of wind-wheel turbine.

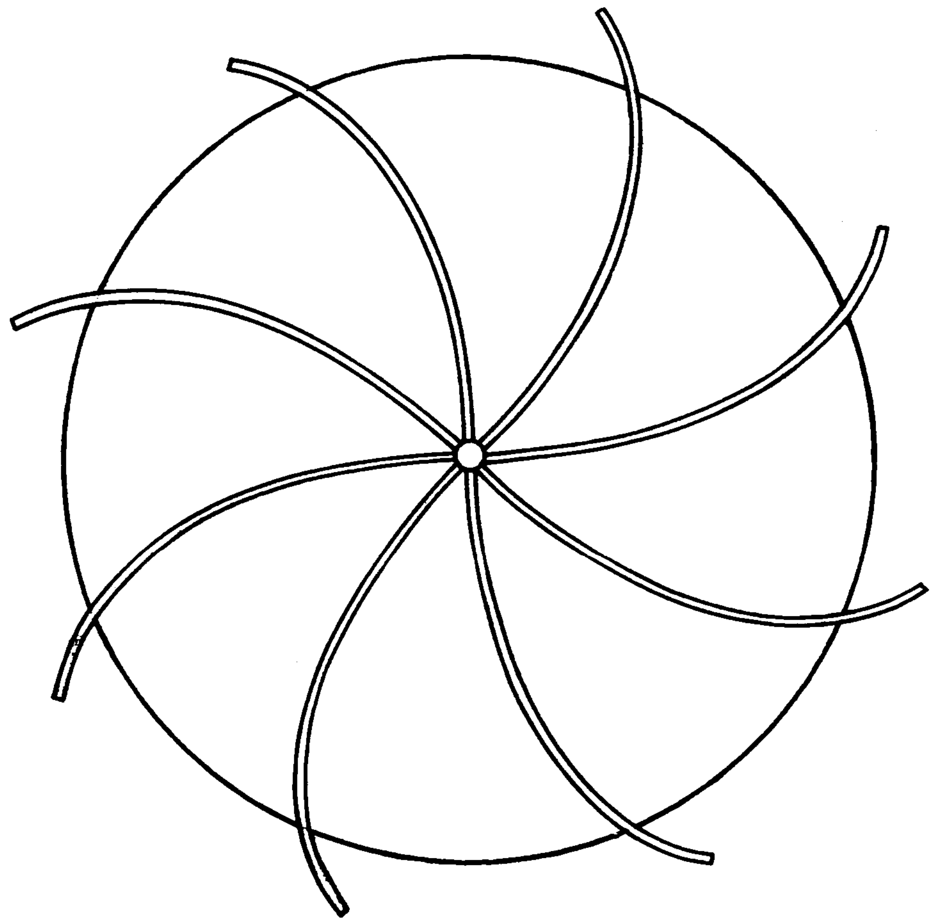

Figure 4. Illustration of wheel of the wind-wheel turbine.

the top half of the exposed bladed wheel as illustrated in Figure 5. Air entering the forward scoop is divided and accelerated by two venturi tubes which funnel it upward to the blades. Air flow across the top impinges on the exposed blades at the top of the wheel. Air flowing past the sides is scooped into the venturi tube which turns it 180 degrees and expels it against the blades at the bottom of the wheel. As the wheel turns, "used" air leaves the large opening (exhaust vent) at the bottom of the main housing. The WWT is mounted on a pivot so that it can weather vane to face the wind.

Figure 4 is an illustration of the wheel of the WWT. The major effort of future research will be the final design of the wheel, i.e., blade shape, number of blades, sizing, and other features. The use of segmented blades which will provide enhanced forces through lift or favorable pressure distribution is anticipated for the final design.

Analytical and experimental studies may demonstrate that the blades should have a variable pitch control so as to have the wheel rotate at a constant speed. Also, the length of the blades (i.e., from outer tip to inner connect point on the axis) may only have to be a portion of the radius of the wheel as will be determined from analytical and wind tunnel testing. Exhausting the air out of the bottom of the duct or alternately through the axle of the wheel may have certain optimizing advantages which will be investigated [1].

Two preliminary studies of the wind-wheel concept have been carried out. Photographs of a paper model (Model number 1) and of a stainless steel constructed model (Model number 2) are shown in Figures 6 and 7. These tests are described in detail in Section III.

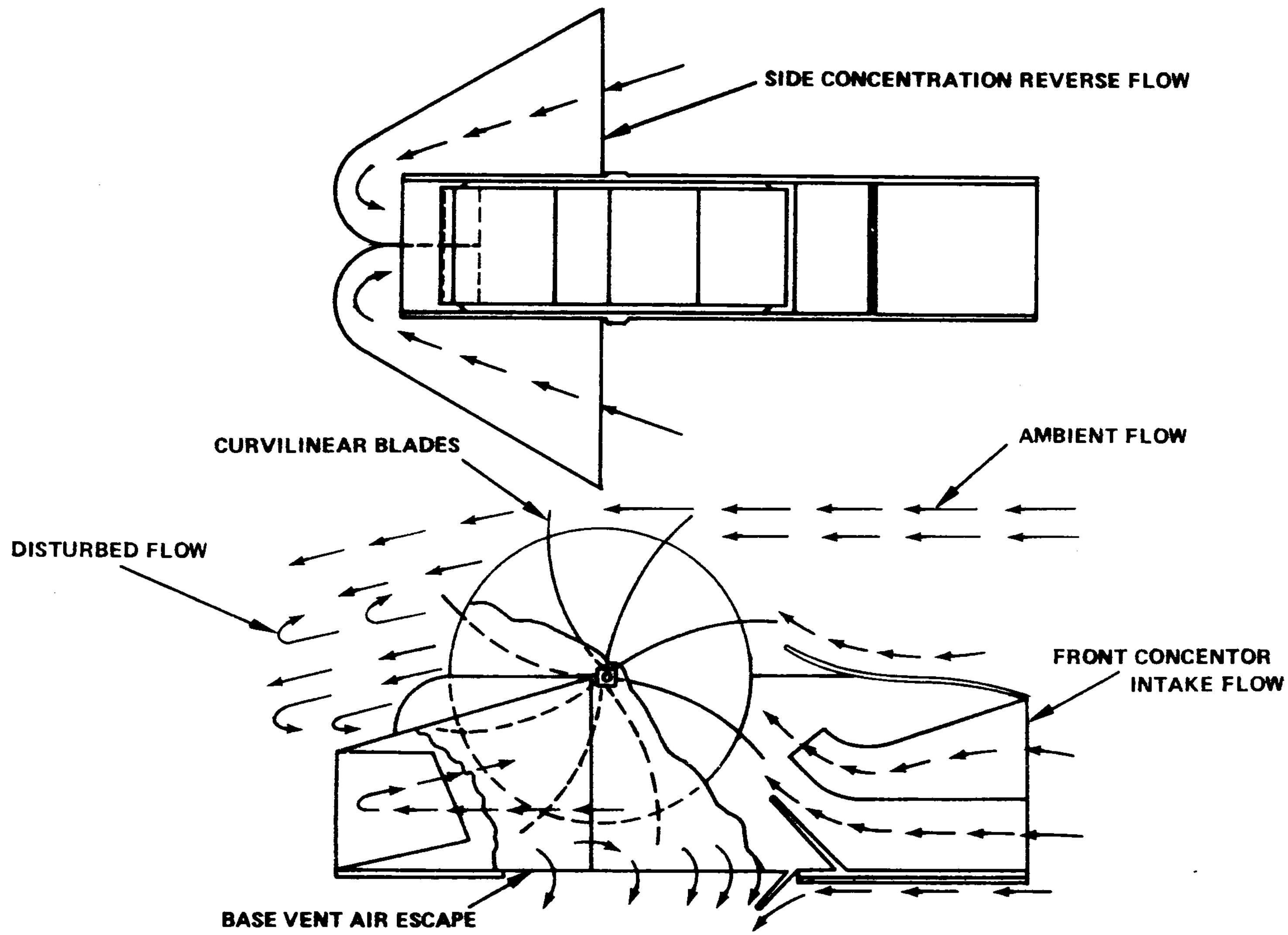

Figure 5. WWT internal-external airflow.

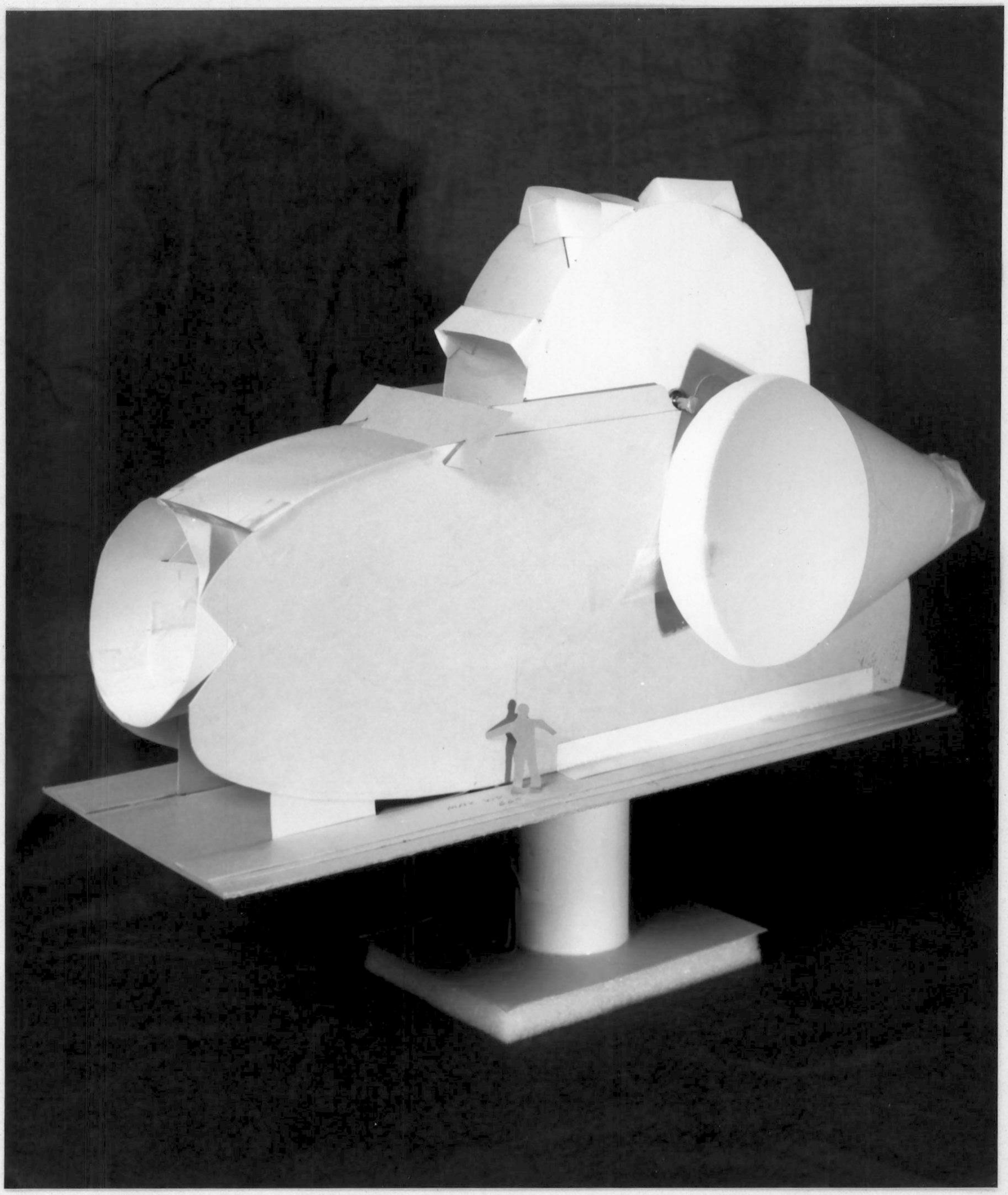

Figure 6. Wind-wheel turbine (Model No. 1).

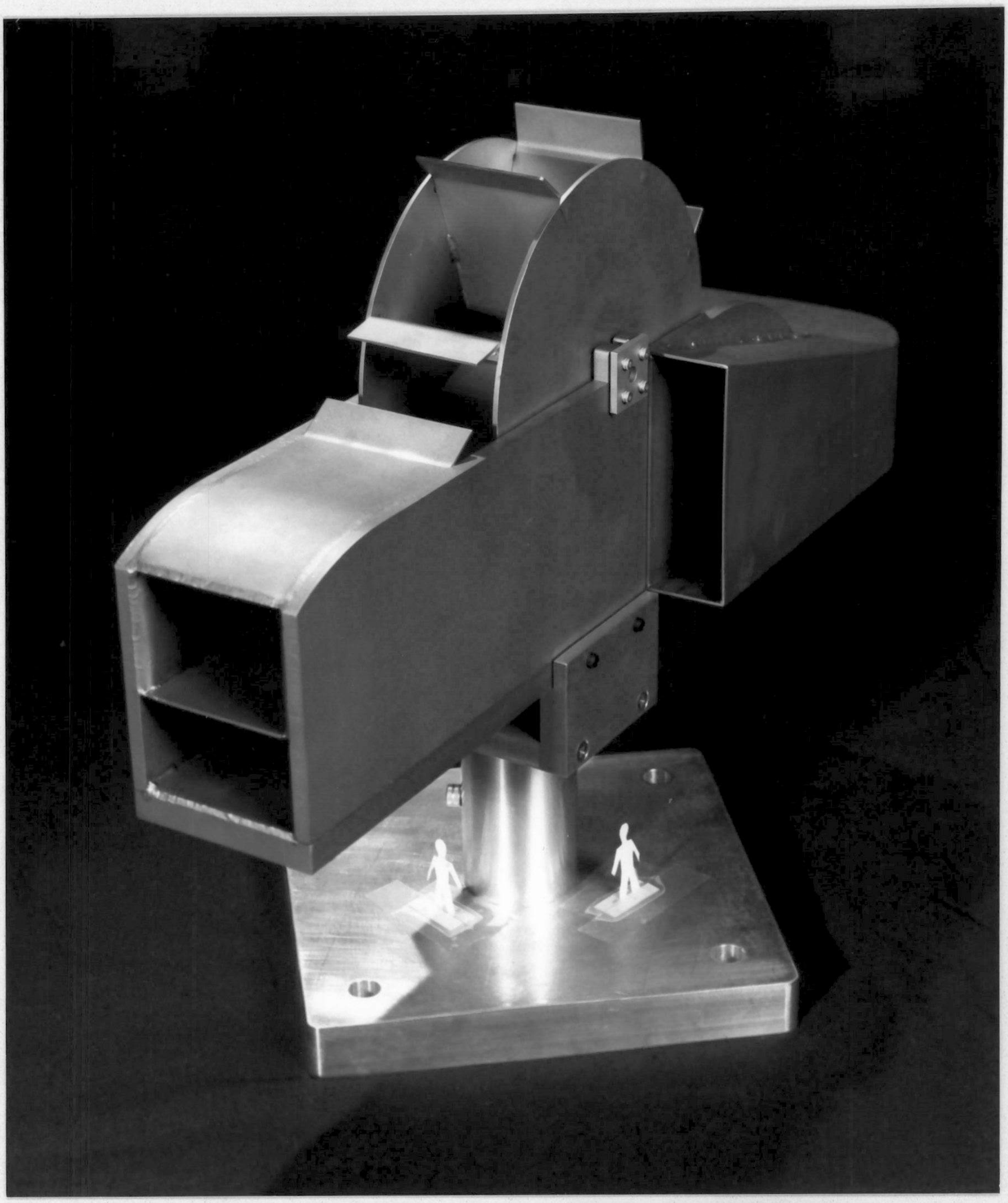

Figure 7. Wind-wheel turbine (Model No. 2).

II. KEY FEATURES AND CHARACTERISTICS

Although wind driven machines bearing some superficial resemblance to a water turbine have been developed in the past, the proposed concept has certain distinguishing features, which are pointed out here. The efficiency of the machine will be improved by achieving a higher output from a relatively small rotor utilizing a number of fixed ducts to increase the wind speed striking it from different wind directions. Preliminary tests will study the fixed bladed wheel; however, subsequent design will take the form of multilifting surfaces or of turbine-type rotors utilizing favorable pressure distributions to enhance rotor speed. Ducts will be constructed with diminishing cross sections to accelerate the wind. Thus the size of the rotor required for a given power output can be reduced in size. This is important to a device of the proposed type because for a fixed blade wheel the maximum tip speed to wind speed ratio is unity. Hence, if high rpm is desired, the smaller the wheel radius, r, the higher the angular rotation, ω, for a given wind speed, V (i.e., $\omega = V/r$).

It is generally concluded that without the lifting blades (to be developed) a wind-wheel turbine of this type has the disadvantage that the moving surface must always travel at a lower speed than that of the wind. The speed of revolution of the rotor is thus low, resulting in the need for expensive gearing when a high-speed electrical generator is to be driven. Preliminary analysis shows, however, that rotational speeds on the order of 25 to 30 rpm can be achieved, which is not appreciably less than that of existing wind energy systems. Also, the WWT is expected to be extremely useful in mechanical drive applications such as pumping water where there is a requirement for very high starting torques. In these cases, high rpm operations are not necessary because it is generally better to pump a large quantity of water slowly than it is to pump a small quantity rapidly. This reduces resistance to water flow in the pipes. Large diameter, slower moving pumps require a slow turning, high torque WWT.

In summary, the key features of the proposed WWT which has inherent construction advantages relative to conventional wind energy systems are: (1) ducted airflow, producing a venturi effect which accelerates the air prior to impinging on the blades and side, and (2) front air scoops which effectively and significantly increase the volume of air captured and converted to rotational energy.

III. MAJOR UNKNOWNS

The final design of the wheel will require answers to a number of questions. The shape of blades, number of blades, sizing, and other features will be dependent on the results of the analytical and experimental studies. It may be determined that the blades should have a variable pitch control so that the wheel will be forced to rotate at a constant speed to turn a constant speed subsystem. Also, the length of the blades (i.e., from outer tip to inner connect point on the axle) may only have to be a portion of the radius of the wheel and can be designed to produce lift or favorable pressure distributions. These factors will be determined from analytical and wind tunnel research.

Components and subcomponents of the WWT must be sized so that they do not vibrate at matched frequencies. Very rugged and efficient bearings must be used in the wheel axle assembly as well as in the internal column, rotatable main housing assembly. In turn, these bearings must be very low friction, and a careful study is required to assess the optimum bearing design. Research will also be carried out on the possible use of air bearings to determine their practicability for use with the WWT. Advanced bearing technology will play a decisive role in future research.

Careful investigation of the ducting is required. The optimum duct contours required to achieve maximum enhancement of flow acceleration through the concentrators must be studied. The appropriate flare of the side ducts is important not only to the capture of the most air possible but also to the turning of the WWT into the oncoming wind. Also, the desirability of having variable flare angles to account for wind direction variability is unknown. It is believed these side ducts can be designed to provide the necessary weather vaning of the apparatus. For instance, the standard deviation of wind direction is much different over rough terrain than it is for coastal, onshore winds near the ground.

The WWT is a "tunable to wind speed system." Through testing it may be found that the internal air duct deflectors (especially the segments connected to the floor of the main housing) should be elevated or lowered depending upon the mean wind speed. For example, the ducts (deflectors) will be fully opened to allow maximum possible airflow during low wind conditions and partially closed during periods of high wind speeds to keep the wheel from spinning above load limits. The deflector positions for the ducts can be controlled by electromechanical devices or by wind pressure devices which would be determined through testing.

As depicted in Figure 3, page 8, the sides of the front ducts should project outward approximately 10 degrees. This is required so that approximately 65 to 70 percent of the wind (about ±1 σ wind direction variability) enters the forward ducts. Also, as shown in the figure, the two side ducts should project approximately 30 degrees out from the parallel side walls of the main housing. For a wind direction variability of ±30 degrees from the mean direction, a significant amount of the flow will thus enter the side ducts. The preliminary WWT system design based on these angles is to operate optimally under a ±1 σ wind direction value of ±10 degrees with a ±3 σ variation of ±30 degrees. It will be determined from analysis and wind tunnel tests what degree the WWT deflection angles of the ducts should be changeable according to wind direction and wind speed variability.

The design of the base mount is extremely important. It must provide means for the exhausted air to escape from underneath the windwheel without creating high back-pressure. Coupled with this, however, the base must be sufficiently strong to withstand high winds.

Finally, a careful performance analysis is required relative to coupling the device with the electrical power generator or the mechanical subsystems. This analysis will establish the size, weight, necessary speed controls, mounting height, and other design parameters from which a detailed cost and performance comparison with a competitive size conventional wind energy system can be carried out.

An electrical generator subsystem may be a main part of the bladed wheel itself. Here, the blades would only need to be about one-half the outer radius of the wheel, allowing the center one-half radius of the wheel to contain a separate generator or to be composed of generator components (i.e., stator, magnets, wire windings, commutator (if required), voltage regulator, etc.). A generator, or generators, can be made to operate in a number of ways, depending on the selected design. These and other possible coupling mechanisms will be included in future studies.

IV. PREVIOUS RESEARCH AND OTHER TECHNICAL INFORMATION

A series of preliminary tests on two models, which were not exactly to the scale proposed in Section I, were carried out. These tests were made without funding and were in a sense a makeshift experiment. However, they have provided useful information that does demonstrate a potential for the WWT. The results of these tests are described in the following paragraphs. First, a description of Model number 1 and test results are given. Model number 2 is then described and its preliminary performance verification discussed.

WWT Model Number 1 Design and Test Results

Model number 1 of the WWT (Figure 6, page 12) is made of paper and pasteboard. The only metallic parts of the model are two small bearings and a metal axle. The model stands approximately 36.0 cm (14 in) high. The discs on the side of the wheel have a diameter of 15.24 cm (6.0 in). The blades extend 0.95 cm (0.375 in) beyond the outer circumference of the outer wheel discs. The blades are 6.0 cm (2.36 in) in width and join at the small center axle point. The model is not to scale as to shape and size of the ducts, wheel, main housing, and other features.

Limited testing of the paper model was carried out at low-speed airflow conditions. The model was exposed to airflow ranging from

1.7 to 2.9 $m\ s^{-1}$ where the air source was an overhead air conditioning ventilation system. The flow was measured repeatedly during the tests by the anemometer manufactured by Hastings-Raydist, Inc., Hampton, Virginia. Figure 8 shows the number of WWT wheel revolutions per minute versus various exposure-geometric conditions. These WWT test conditions were: (1) all ducts open with airflow directed onto the front of the WWT system; (2) the front ducts were covered with a fitted piece of construction paper; (3) all ducts (front and side ducts) were covered to allow no air to flow into the ducts; (4) all ducts were open (uncovered) and a hood, made of construction paper, was placed over the top half of the exposed wheel, allowing no air to flow directly onto the upper part of the wheel; and (5) the WWT Model number 1 was turned 180 degrees with respect to the mean wind direction. For each test condition 30 1-minute samples of data were recorded. The number of revolutions were determined by counting the number of times an ink spot, marked on the outer edge of the wheel disc, appeared for a 1-minute time period recorded with an electronic stop watch (Cronus 3-5, manufactured by the Cronus Precision Products, Inc., Santa Clara, California). The arithmetic mean values were computed and are shown in Table 1 and Figure 8. Table 1 also includes the ratio of the mean number of revolutions per minute of the wheel operating in an "all ducts open" normal mode (condition number 1) to the remaining four modes. It is interesting that the wheel turns in a normal rotational manner when the WWT model is turned backwards into the mean flow but only at one-fourth of the rotation speed.

It is especially significant to note the improvement in performance due to the front ducts, although the advantage of the side scope

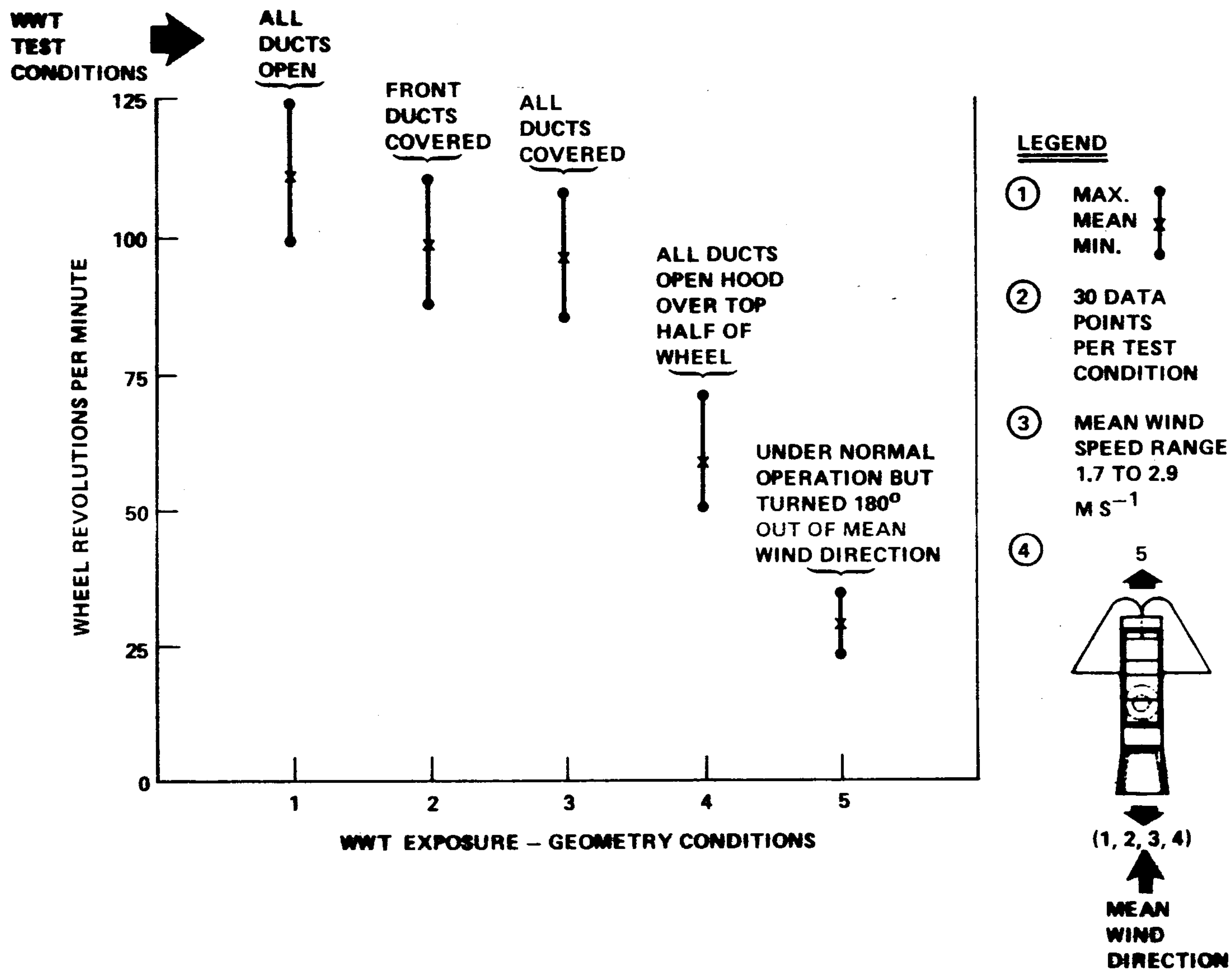

Figure 8. Revolutions of WWT (Model No. 1) wheel per minute versus various exposure-geometry conditions.

TABLE 1

MINIMUM, MEAN,* MAXIMUM NUMBER OF WWT (MODEL NUMBER 1) WHEEL REVOLUTIONS PER MINUTE WITH RESPECT TO EACH TEST CONDITION

WWT Test Condition	Wheel (rpm) Min.	Mean*	Max.	Ratio: Mean of Test No. 1 / Mean of Each Other
1. All ducts open	101	112.6	125	1.00
2. Front ducts covered	87	98.2	110	1.15
3. All ducts covered	85	96.2	108	1.17
4. All ducts open, hood over top half of wheel	51	57.7	68	1.95
5. Under normal operation but WWT turned 180 degrees out of mean wind direction	24	29.2	34	3.86

*Arithmetic mean of 30 1-minute data values.

appears marginal. Careful design of these is expected to enhance their effects.

A special set of wheel speed data was acquired using the WWT Model number 1 which provides additional information as to how the wheel turns as it is positioned differently in regard to the mean wind direction.

Figure 9 shows the WWT (Model number 1) wheel revolutions per minute versus orientation of the model to the mean wind direction. The mean wind speed was approximately 2.0 m s^{-1} with a range of 1.5 to 2.4 m s^{-1}. Again, these wind speed measurements were made with the Hastings anemometer in the same air ventilation situation as used to obtain the data for Table 1.

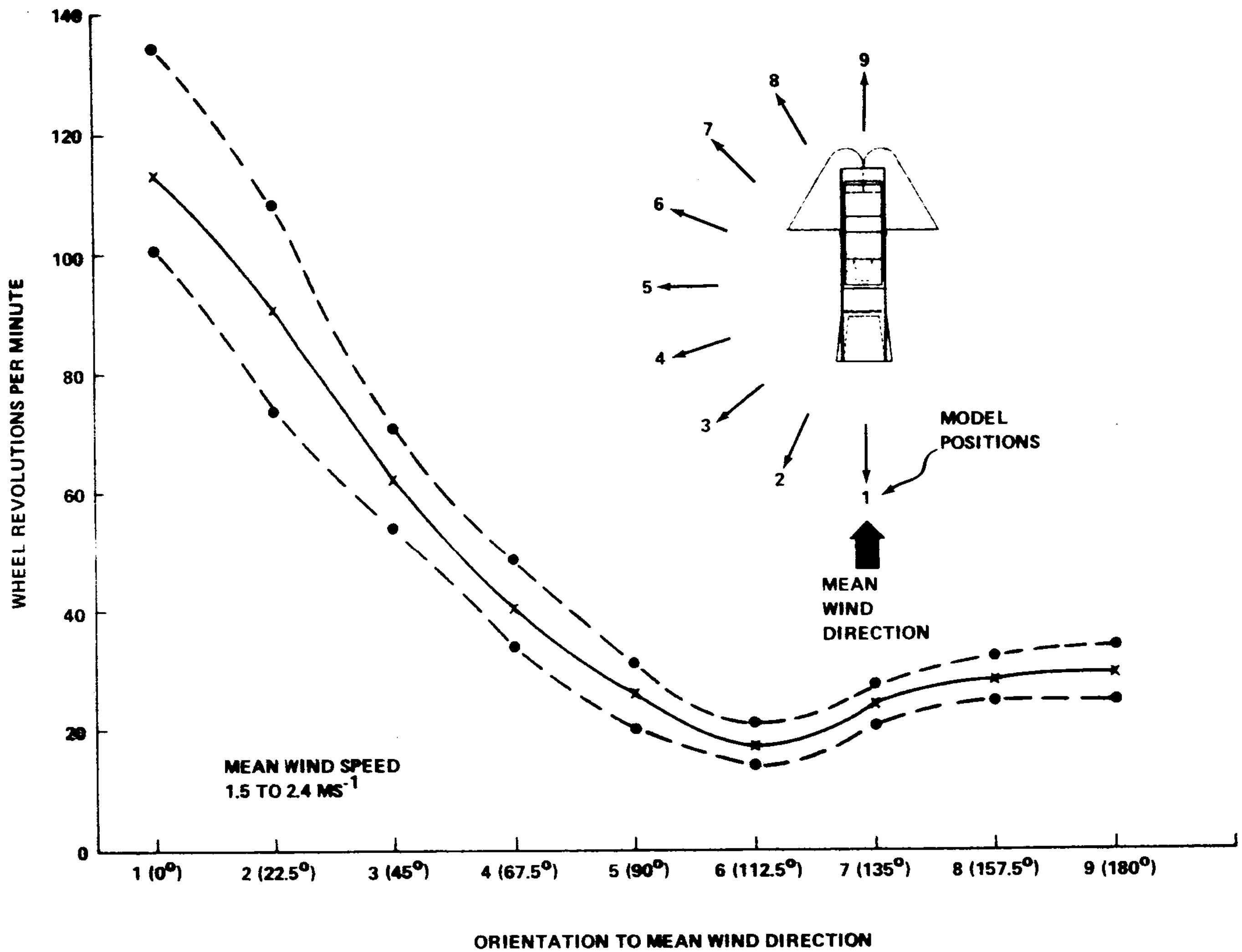

Figure 9. WWT (Model No. 1) wheel revolutions per minute versus orientation to mean wind direction.

The model (with duct open) was turned 22.5 degrees at a time to the mean wind direction to position number 9 where the model faced 180 degrees out of the mean flow. Again, 30 1-minute samples were taken at each position or orientation point.

The data illustrated in Figure 9 show that the lowest number of wheel revolutions per minute was at the orientation of 112.5 degrees to the mean wind direction. The interesting fact is that the wheel rotates regardless of its orientation to the mean flow. Also, the wheel rotates about four times faster when the WWT points into the flow than when it points 180 degrees "out-of-phase" with the flow. It will be interesting to study the wheel rotation speed versus mean wind direction of a true-to-scale model since Model number 1 deviates appreciably from the presently proposed design.

WWT Model Number 2 Design and Test Results

The stainless steel WWT Model number 2 is shown in Figure 7, page 13. Again the model is not built to the presently conceived proportions. Several laboratory and field experimental tests have been conducted using Model number 2. It was soon learned that it responded quite erratically to wind direction variability. After a couple of field exposures in moderate wind speed conditions (6 to 12 m s^{-1}), it was decided to mount a temporary vaning tail onto the system. The tail was about the same size, in area, as the side-of-wheel discs and was placed directly to the back center of the main housing assembly. With the temporary, lightweight, fiberboard tail attached, the model functioned quite well in regard to vaning into the wind. The main reason this heavy model responded erratically without the tail is that the side ducts (concentrators) do not project outward far enough (i.e., the outer edge of the side ducts should project out about 30 degrees with respect to a plane parallel to the main housing walls) to generate adequate drag on the downwind end of the model.

The first field tests of Model number 2 were made in a field which contained an array of wind towers. The tests provided a few data points during outflow of wind from rather immature thunderstorms (data not shown herein), but very limited field testing was done.

A laboratory experiment was then conducted. Figure 10 shows the basic equipment used to gather wheel spin-up, spin-down, and steady state revolution data. The stopwatch (Cronus Electronic Model 3-S) and hand-held anemometer (Belfort Instrument Company, SM-C-367329) are not shown in Figure 10. However, the electronic counter was a Hewlett-Packard 5245L unit, the airflow meter was Parker-Hannifin, and the other apparatus consisted of standard laboratory items. The photocell pulse detector was made from on-shelf electronic components. After several trials to determine airflow speed, placement of air source with respect to the model, stability in acquiring accurate revolutions of the wheel with time, etc., the experiments were started. It should be mentioned that the air pressure remained very stable throughout all testing, which made the results more accurate. To determine airflow speed (wind speed in the figures to be discussed), the hand-held anemometer was positioned approximately 40.0 cm away and directly in the mainstream of the jet (plume) of air. Measurements were made just prior to and after recording all data sets, and especially for similar groups of wheel turn data. To obtain the anemometer measurements, the air source was turned into the "open room" and measurements were not taken in front of or in line with

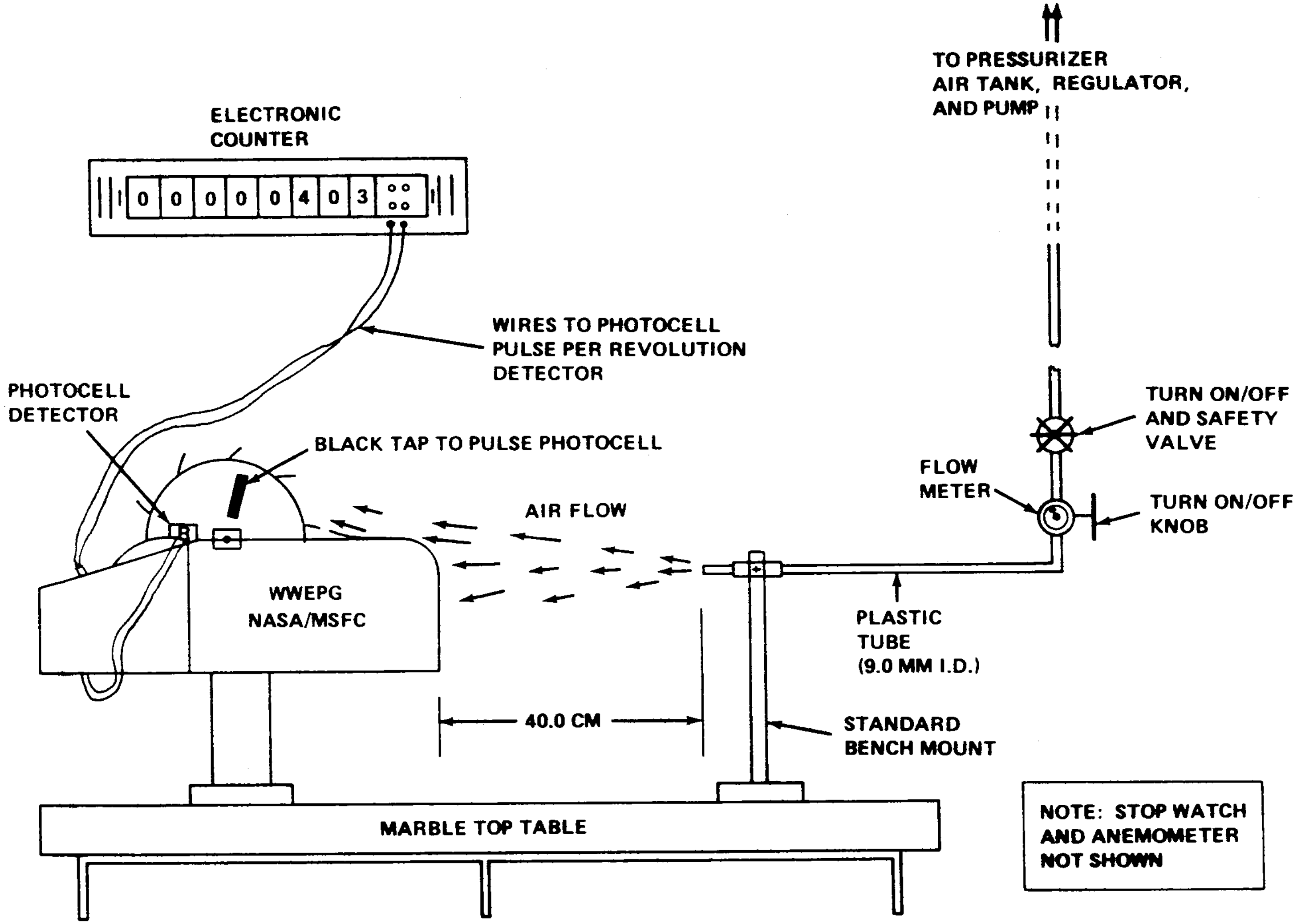

Figure 10. Laboratory apparatus used to obtain spin-up, spin-down, and steady state WWT (Model No. 2) wheel pulse per revolution data.

any nearby obstacles. The laboratory air temperature was always approximately 25 °C (77 °F) throughout the testing.

Model Number 2 Performance Data Acquisition

Model number 2 is made of stainless steel (Figure 7, page 13). The overall weight of the model is approximately 18.0 kg (40.0 lb). It stands 35.0 cm (14.0 in) high, has a wheel that weighs approximately 2.25 kg (5.0 lb), a blade length (i.e., radius of wheel) of 10.0 cm (3.9 in), blade width of 6.0 cm (2.4 in), and an axle of 0.7 mm in diameter. The wheel bearings were made by the Fafnir Bearing Company, New England, Connecticut.

Three sets of data were acquired to determine the basic performance of the Model number 2 wheel. These were spin-up data, steady state, and spin-down data. First, how much wind is required for the wheel to complete 20, 40, and 60 revolutions starting from rest and as a function of mean wind speed (actually a jet or plume of air from a pressurized tank and nozzle) was determined. Figure 11 shows the raw data and fitted curves for the three fixed revolution conditions. This figure also shows that it takes 3.1 m s^{-1} of wind speed for the wheel to begin rotating (i.e., threshold start speed). As can be seen on the ordinate of Figure 11, it requires approximately 30 sec in a 4.0 m s^{-1} mean wind speed to cause the wheel to rotate 20 times; whereas, it only takes 10 sec for the wheel to rotate 20 revolutions in a 16.0 m s^{-1} mean wind speed.

The second set of data, with a curve fit, is shown in Figure 12. These data show the wheel revolutions per minute as a function of steady state wind speed. The wheel threshold speed of 3.1 m s^{-1} is shown;

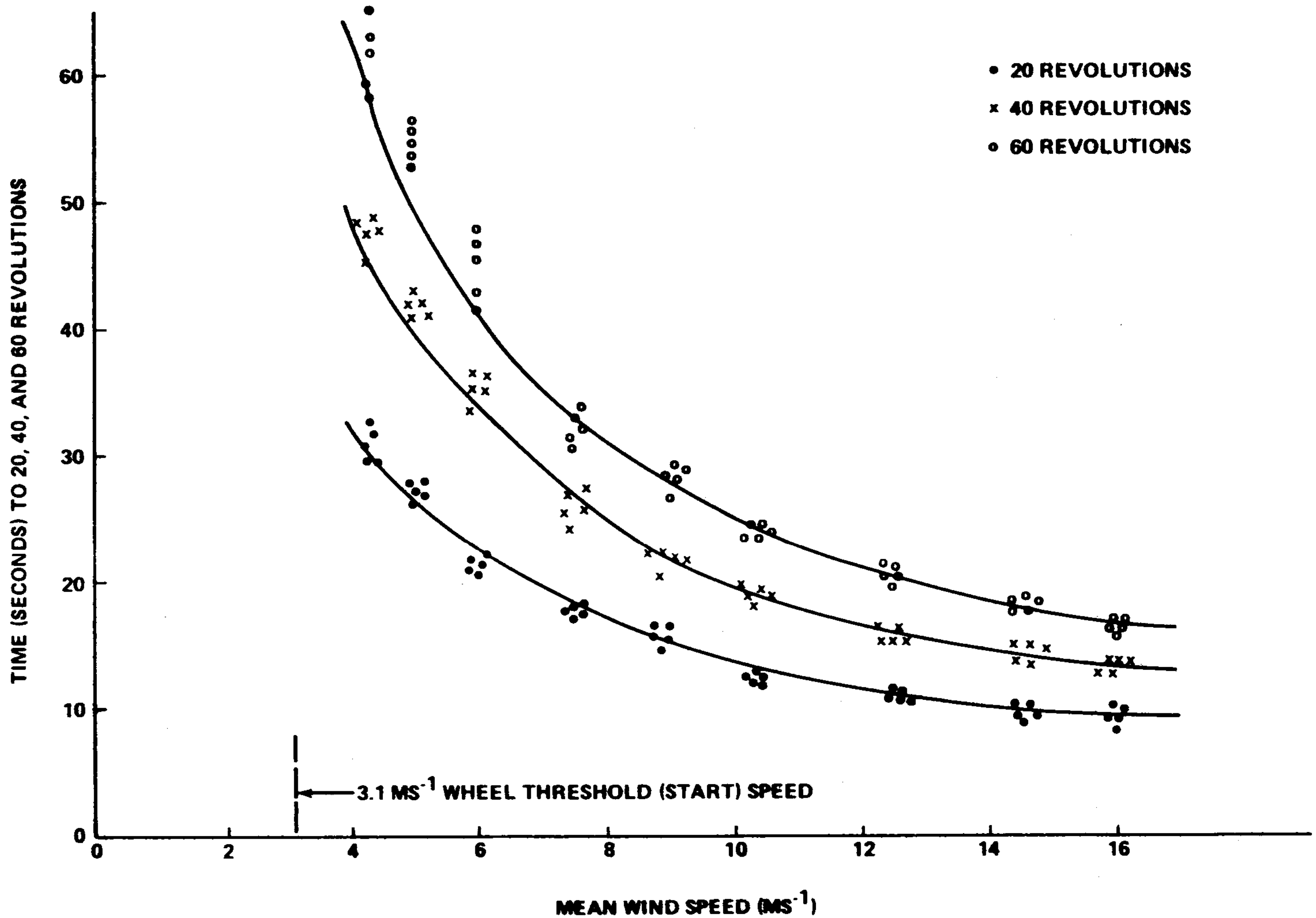

Figure 11. WWT wheel time to reach 20, 40, and 60 revolutions versus mean wind speed (Model No. 2).

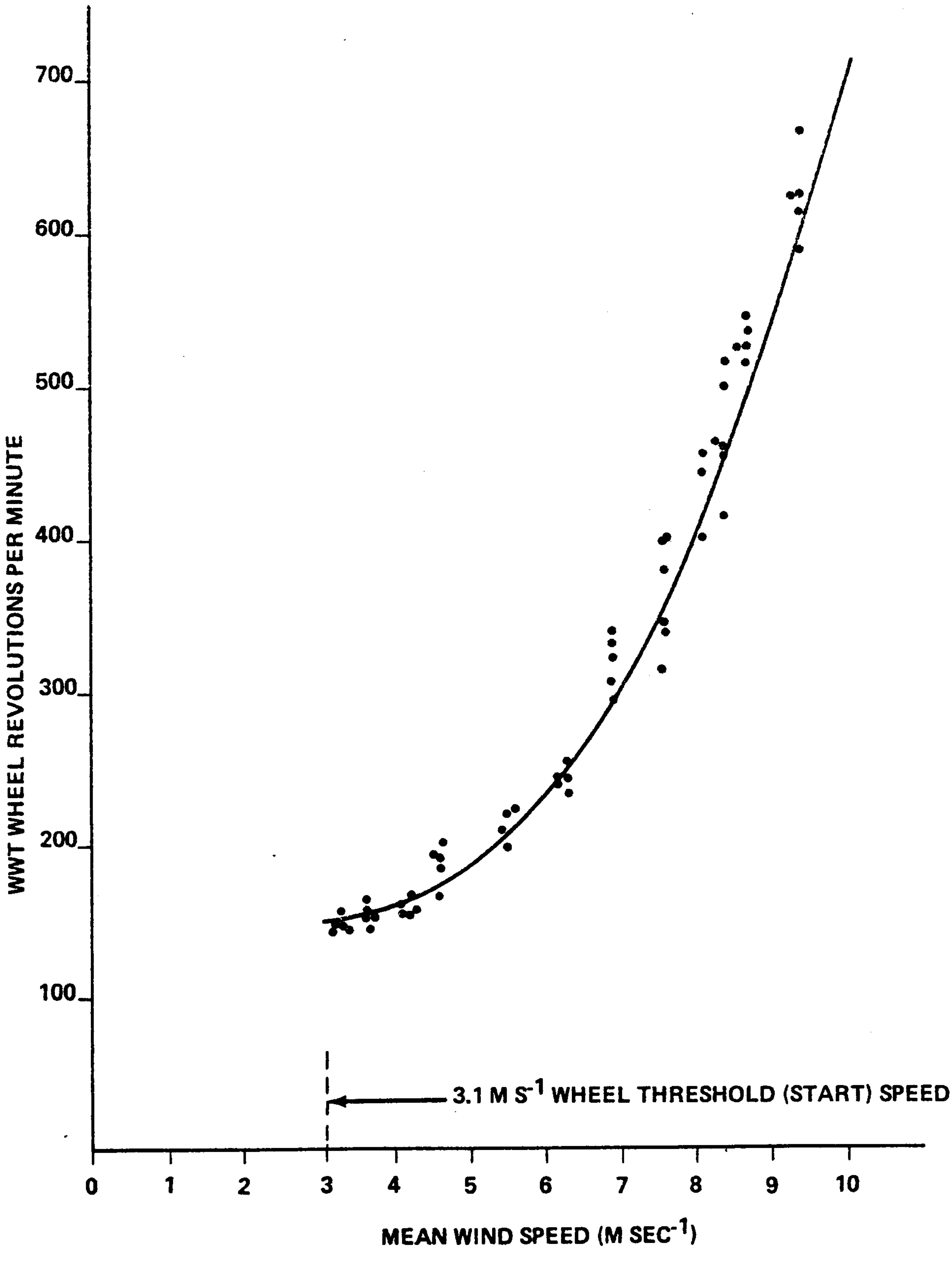

Figure 12. WWT wheel revolutions per minute versus mean wind speed (Model No. 2).

however, the wheel rpm values are only given to about 650. The reason higher rpm values were not obtained for winds above 10.0 m s^{-1} was to avoid ball bearing heating.

The third set of data consists of the wheel stop-time or spin-down data. The data and curve fit are shown in Figure 13. It is interesting to note the appreciable difference in wheel slowdown time versus rpm. As shown, it takes approximately 50 sec for the wheel to stop if it is given a constant rpm of 500, compared to approximately 400 sec to stop when at a speed of 150 rpm. This indicates that the wheel has a very significant drag force to still air when rotating at high speeds.

Analyses of WWT Model Number 2 Performance Data

A simple analysis of the blade allows an estimate of the performance to be made. Using impulse moment theory, the data of Figure 11 should obey the relationship

$$Mt = I\dot{\theta}/g_c \tag{1}$$

where M is moment due to wind forces and I is the moment of inertia and the dot indicates the time, t, derivative. Carrying out integration gives

$$\frac{g_c M}{I} = \frac{2\theta}{t^2} \tag{2}$$

The relationship for the moment divided by the moment of inertia can then be expressed from a curve fit of the data in Figure 11 as:

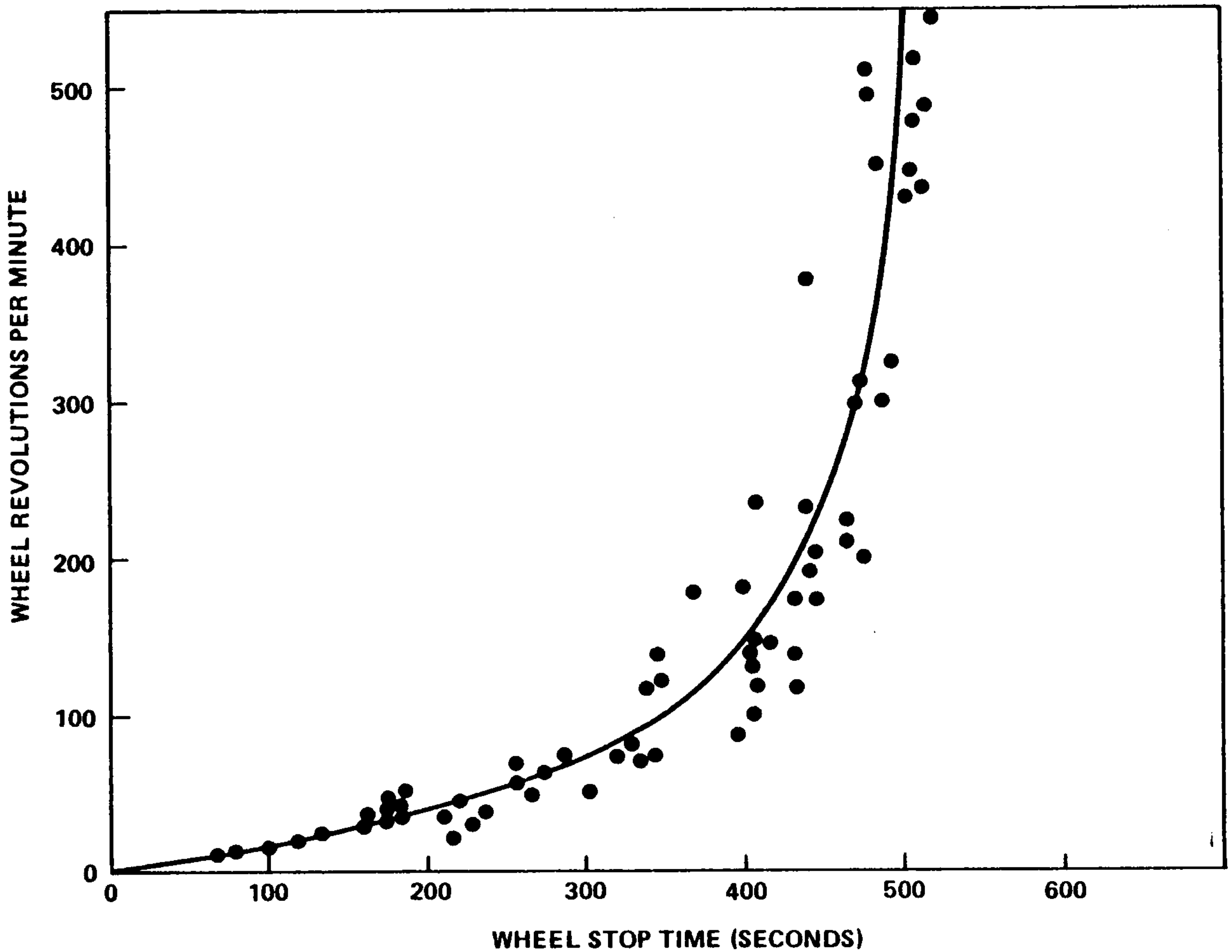

Figure 13. WWT wheel initial revolutions (turn speed) versus stop time (Model No. 2).

$$\frac{g_c M}{I} = 0.016V^{1.91}; \quad (V \text{ in } ms^{-1}) \tag{3}$$

The power of the system is given by ωM where the maximum value of ω is given by a tip speed ratio of unit, i.e., $\omega = V/R$. Substituting these values into the equation for power and carrying out the algebra results in the power coefficient expression given by

$$\text{Power} = 0.016V^{1.91}\ IV/g_c R \tag{4}$$

The C_p for this simple device thus reduces to

$$C_p = \frac{0.032I}{\rho V^{0.09} AR} \tag{5}$$

Table 2 shows some tentative power coefficients for the device.

TABLE 2

VALUES OF POWER COEFFICIENTS INTERPOLATED FROM SIMPLE EXPERIMENTS

V	4 m/s	10 m/s	16 m/s
C_p	0.058	0.053	0.051

These should not be construed as the maximum that can be obtained from this device because there were inherent problems with the bearings becoming hot during the study, and the model design is not at all optimized. The anticipated power coefficient for this system is given in Chapter III, Section I, Potential of the Concept. Moreover, it should be noted that power coefficients in themselves do not demonstrate cost-effectiveness. The important parameter is the ¢/kW-h.

The steady state data shown in Figure 12, page 28, was also analyzed to compute the tip speed ratio for the given device. By calculating the value of V^2/ω from the steady state curve, one determines that for values of wind speed 6.0 m s^{-1} to 10.0 m s^{-1}, this ratio is constant at 0.15. The steady state tip speed ratio for the device is thus given by

$$\frac{\omega R}{V} = 0.011V \tag{6}$$

which once again represents a relatively low value but is also a strong function of the bearing resistance.

Finally, one can see the very strong influence of the bearing drag in Figure 13, page 30, which illustrates a very rapid decay from steady state speed once the forcing wind function has been removed.

Results of this study, although very preliminary, do illustrate that the device has potential and that the system is a viable technique for converting the kinetic energy of the atmospheric wind into rotational energy. Further evaluation of the potential of the system based on a more optimum design analysis is given in Chapter III, Section I, Preliminary Performance Analysis.

In summary, data acquired on a bench setup using a jet source of air is not an optimum procedure; however, the preliminary data do provide a basis for confidence in the preliminary WWT concept.

Citations to the Literature

A thorough review of the literature reveals that no studies are reported for a wind-wheel turbine concept as cited herein.

Golding [2] makes reference to a 500 kW wind driven device developed in Italy. This device appears to have some small resemblance to the ducted concept proposed in the present study, the difference being that a three-blade propeller turning on a vertical axis was utilized. Although the information is limited, it would appear that a subterranean ducting system was used.

No other information on a similar device is available in the current literature. For this reason, it is believed important to carry out additional performance analysis. The analysis of necessity must begin from basic principles and be extended to the degree of sophistication necessary to provide a meaningful demonstration of the system capabilities. A major effort of a future study will be to establish analytical models of the WWT performance.

CHAPTER III

POTENTIAL OF THE CONCEPT

The potential of the concept is described in this chapter. In Section I a preliminary performance analysis is given which demonstrates that the WWT can achieve C_p's comparable to those obtained with conventional propeller- and Darrius-type wind energy systems. A qualitative argument is then given in Section II to demonstrate that the cost of fabrication, materials, and tower construction will be less for the WWT due to the sturdier and simpler design. Finally, in Section III a preliminary comparison of the WWT with the MOD-0 wind turbine is given to provide a "feel" for the size of the WWT for comparable power output.

I. PRELIMINARY PERFORMANCE ANALYSIS

A preliminary analysis of the proposed WWT performance, based on a simple engineering model described in Appendix A, was carried out. Figure 14 schematically illustrates the model. Only four blades are considered for simplicity; however, this is not expected to be the optimum number. The engineering model is represented effectively by three jets of wind acting on the turbine rotor. Jet 1 is the freestream wind speed, V_1; jet 2, resulting from the front duct concentrator, has a speed of V_2; and jet 3, resulting from the side scoops redirecting the flow onto the backside of the turbine rotor, has a speed of V_3. The front duct will be constructed with a contoured configuration which will accelerate the flow. The acceleration is expressed by the ratio $\zeta_2 = V_2/V_1$. Although detailed calculations, proposed as part of the study,

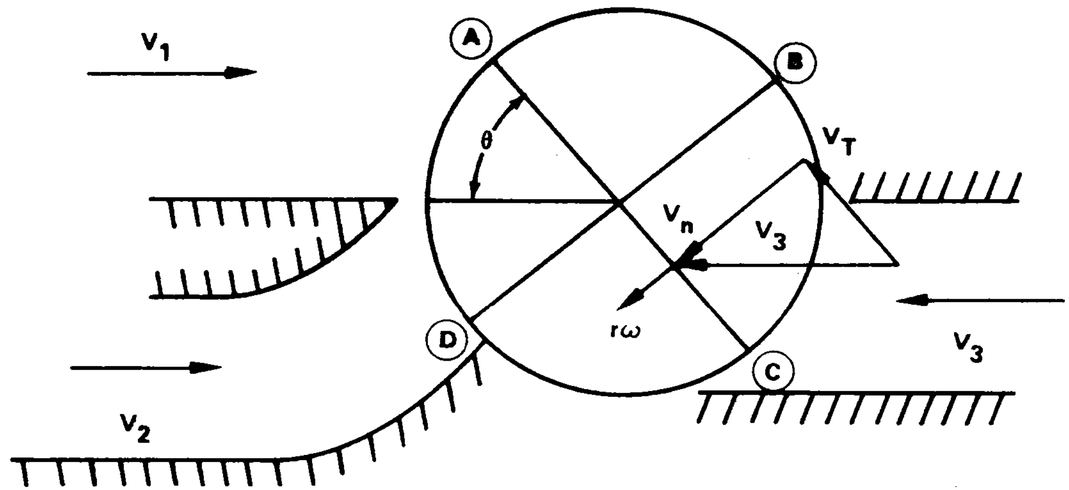

Figure 14. Definition of jets and blade interaction.

are needed in order to determine the magnitude of ζ_2, it is estimated that values between 1 and 2 are realistic. When $\zeta_2 = 0$, the duct is closed.

Similarly, the velocity of flow impinging on the rear side of the blade from duct 3 is less than the freestream velocity due to the deceleration caused by turning through 180 degrees. This deceleration is represented by $\zeta_3 = V_3/V_1$. To assign a specific value to ζ_3 also requires detailed analysis; however, ζ_3 will be less than unity and is estimated to lie between 0.1 and 0.5. Again, $\zeta_3 = 0$ represents a closed duct.

Forces due to all three jets acting on the rotor create a torque which is expressed in coefficient form as

$$C_T = 2\text{Torque}/\rho wRV_1^2 \tag{7}$$

where ρ is the density of the air; R is the radius of the rotor; w is the width of the wind turbine wheel; and V_1 is the freestream wind speed.

The power coefficient of the WWT is then related to the torque coefficient by the relationship

$$C_p' = \lambda C_T \tag{8}$$

where λ is the ratio of tip speed to freestream wind speed (i.e., $\lambda = \omega R/V_1$) and ω is the rate of the angular rotation of the wind turbine wheel. The C_p' given by Equation 8 is based on a flow area of ωR. However, the actual area of the wind flow entering the various ducts is estimated (without having specific dimensions at this time) as approximately 3.3 ωR. Hence,

$$C_p = C_p'/3.3 \tag{9}$$

Values of C_p reported throughout this section are therefore based on an effective area of $A_e = 3.3\ \omega R$.

Computed values of C_p measured with the rotor at the angular position of $\theta = 45$ degrees (see Figure 14, page 35) for given values of the tip speed ratio are shown in Figure 15. This figure illustrates the influence of the front duct venturi which produces a significant increase in the C_p values. These values of C_p are thus competitive with conventional propeller-type wind turbine generators. That the influence of the air drawn in through the side scoops is relatively small is also indicated in the figure. Therefore, through the remainder of this section a value of $\zeta_3 = 0.1$ has been used although methods of increasing ζ_3 will be part of the proposed study. It is anticipated that through careful design of the front venturi duct, values of $\zeta_2 = 2$ can realistically be obtained.

Figure 16 shows how the value of C_p varies with rotor angular position. The data are plotted only for θ ranging from 0 to 90 degrees since for a four-bladed rotor C_p varies cyclically with a period of 90 degrees. The nonuniformity of C_p with angular rotation can be reduced by utilizing additional blades on the wind turbine wheel. The optimum number of blades will be determined from the proposed study, which will consider the trade-off between uniformity of power output and cost.

As noted earlier, low tip speed ratio devices such as the WWT generally have associated with them high coefficients of torque. Figure 17 illustrates the starting coefficient of torque for various values of acceleration through the front concentrator. The starting torque is very

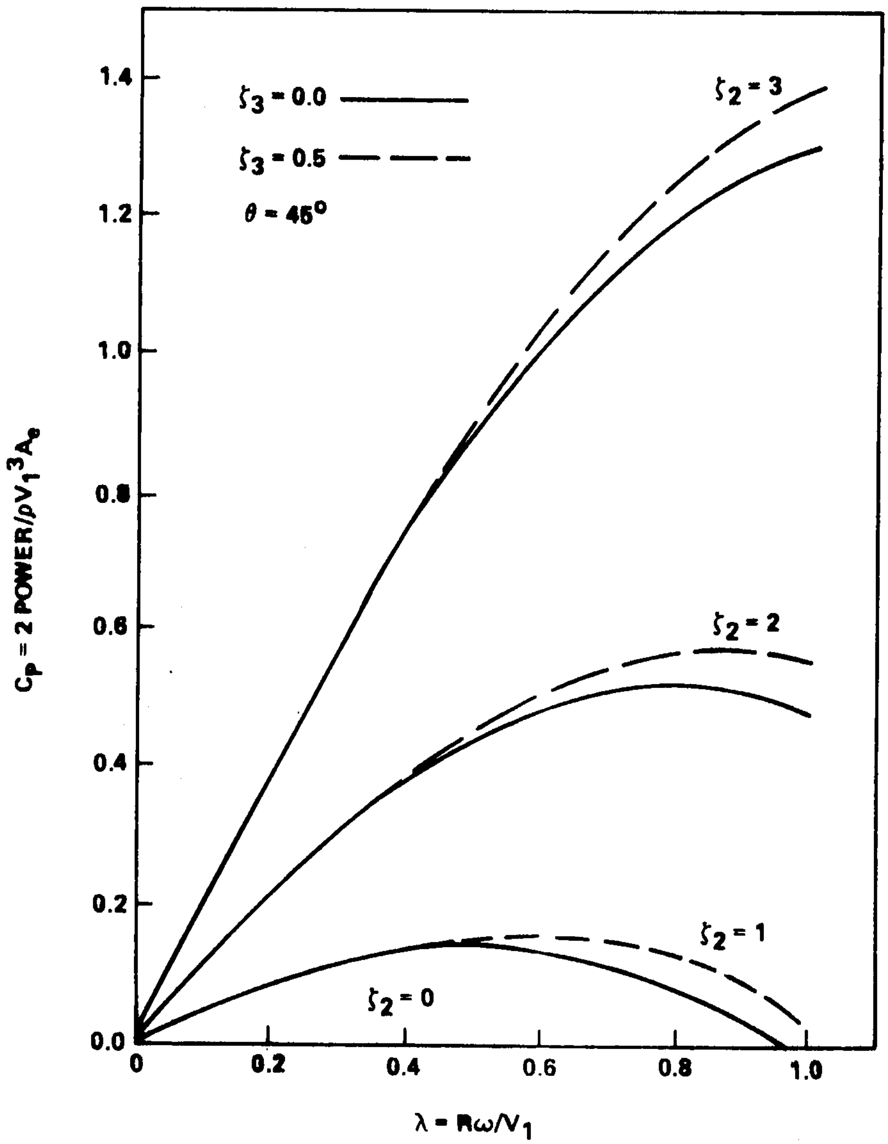

Figure 15. Influence of accelerated flow through front duct.

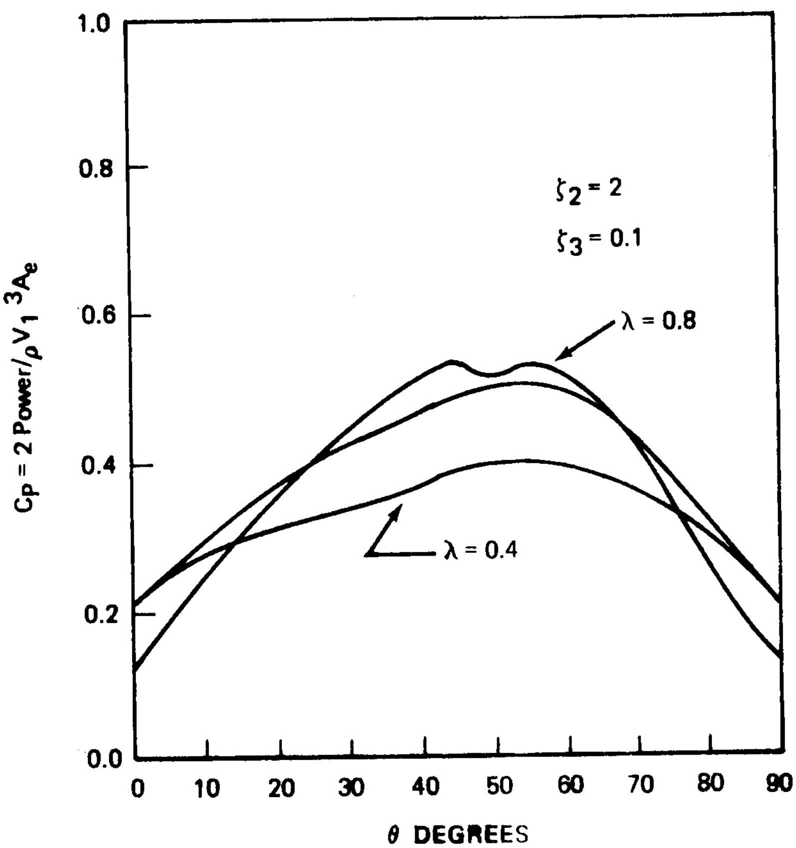

Figure 16. Variation of the power coefficient with rotor position.

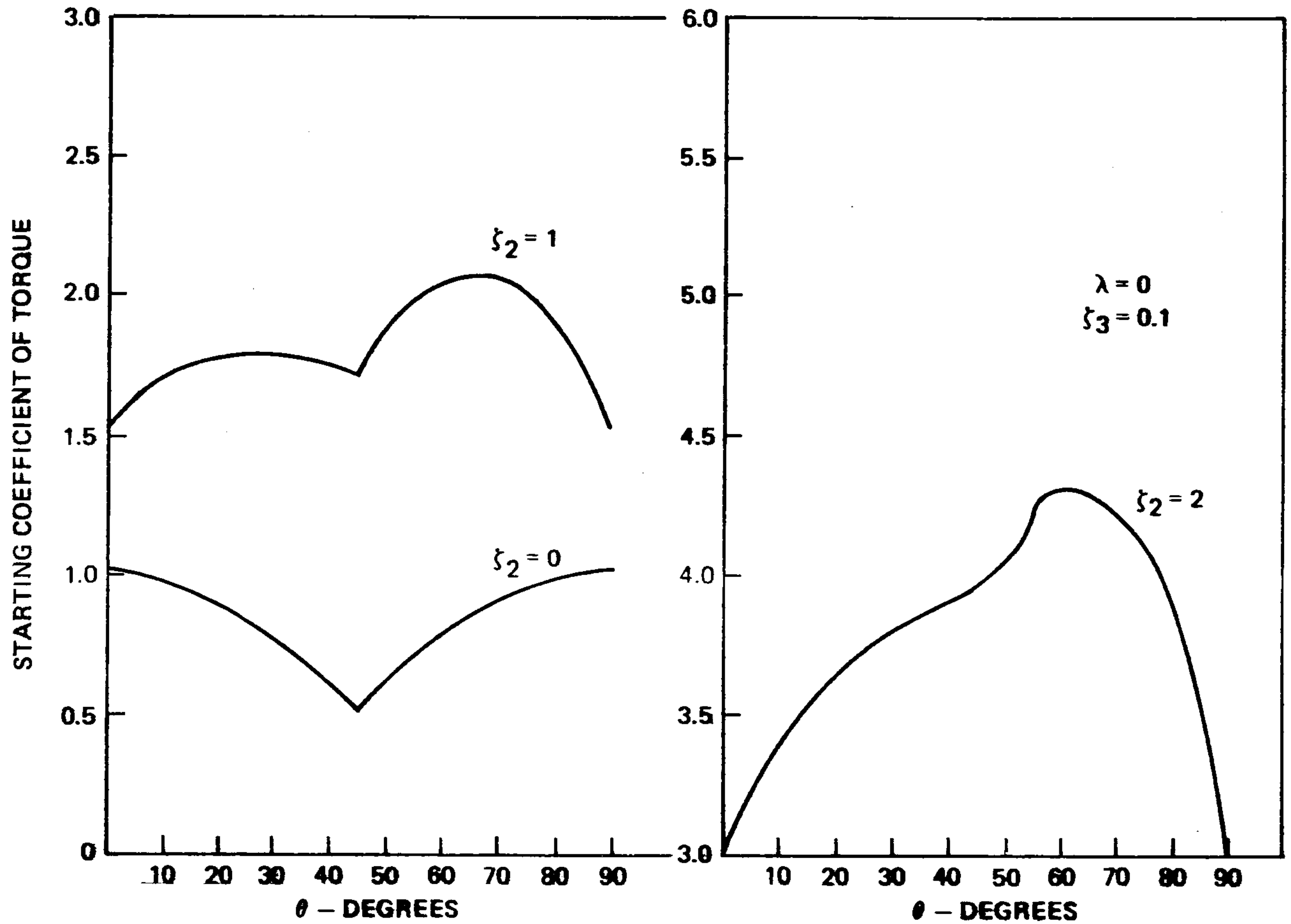

Figure 17. Influence of flow through front duct on starting torque.

high and demonstrates the good characteristic of the system for applications to pumping and other mechanical subsystems requiring high torque.

Figure 18 illustrates how the torque varies on an individual blade throughout a complete 360 degrees of rotation. Relatively high cyclic loading occurs which will be a fatigue factor that must be carefully considered in the blade design. This cyclic loading, however, is no greater than that of a conventional propeller rotor. Moreover, the wind turbine wheel will be less susceptible to high fatigue moments because of the sturdier construction of each blade as contrasted to the cantilever construction of propeller-type rotors.

II. QUALITATIVE COST COMPARISON

The preliminary performance analysis given in the preceding paragraphs demonstrates that the wind wheel turbine concept has the potential of being competitive with conventional systems in terms of performance. Moreover, although cost analyses have not been carried out in any detail because of the numerous unknowns which must be resolved prior to a specific conceptual design, it is inherently obvious that the construction costs of the wind wheel turbine will be less than that of a conventional propeller- or Darrius-type system. Standard manufacturing processes such as welding and bolting will make the cost of assembling the WWT rotor considerably less expensive than that of the intricate fabrication process called for in construction of large wind turbine propellers. Propeller-type rotor construction costs are higher because of the requirement for careful moulding and precisely contoured shapes, whereas the wind wheel does not require nearly as carefully contouring

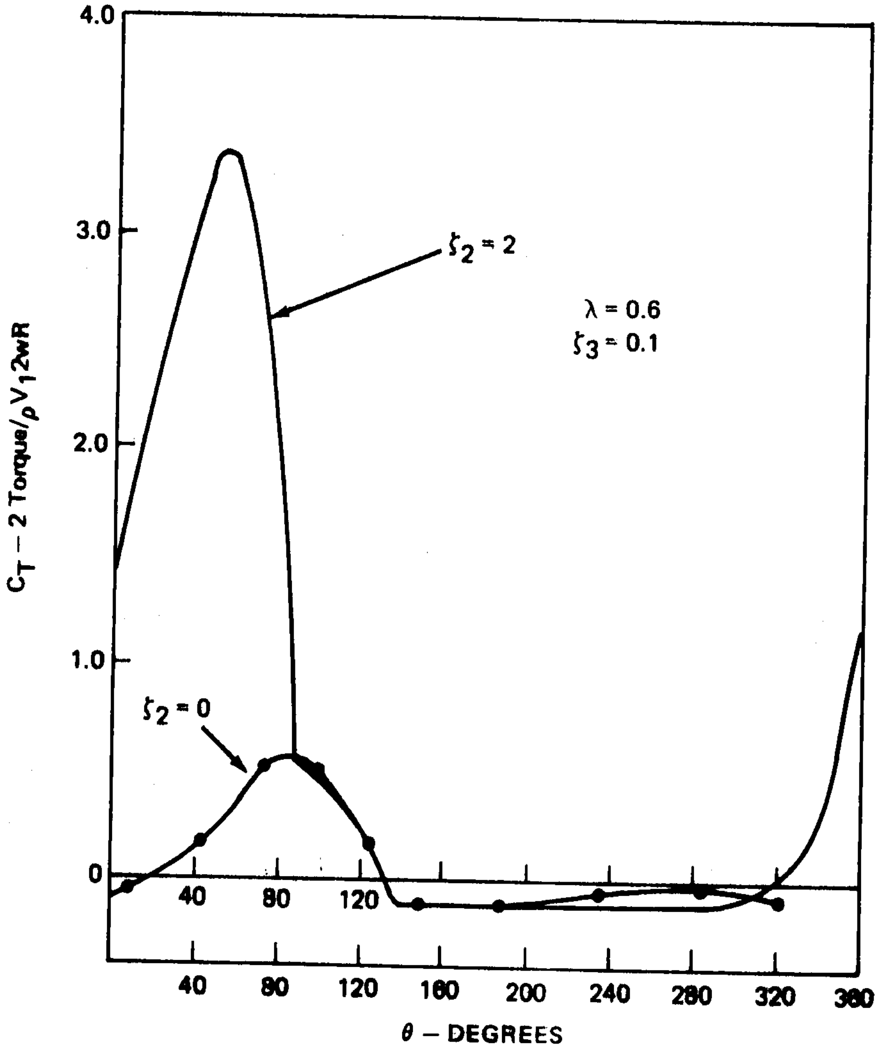

Figure 18. Influence of front concentrator on torque of individual blade.

and can be made and balanced in a standard jig. Trade-off studies as to size, weight, and subsystem components are proposed as part of a future study. These studies will indicate whether the WWT has savings of cost in terms of materials. It is expected that this will be true.

In summary, although this will be carefully studied in the proposed work, it is believed that equivalent power output from the WWT can be achieved at lower costs than those of conventional propeller- and Darrius-type wind turbine generators. Savings will be achieved in cost of construction, in particular, but also in materials which will be less exotic and in terms of tower construction which, due to the configuration of the WWT, can be built more sturdily, perhaps as a three-legged structure or as a cylindrical concrete structure. Note the problem of tower shadow or of the blades flexing into the structure does not exist for the WWT.

Relative to subsystem components, similar commercially available transmissions and generators can be utilized with the WWT as currently utilized with the conventional wind turbine generators (WTG). Moreover, a control system for the WWT will be inherently simpler than blade pitch control used for propeller-type wind anemometers. Simple mechanisms of opening and closing the frontal ducts can be used to carefully control power output of the WWT. Also, the self-weather vaning characteristics of the WWT provide cost-effective control systems which do not extract power from the generator output to position the rotor into the wind as happens with horizontal axis propeller wind turbines. Note also that the rotor can act as a flywheel storing energy which can be used during short periods of low wind speed and thus maintain a more constant power output without additional control inputs necessary.

The above arguments, although subjective in nature, provide confidence that the WWT can produce energy at lower ¢/kw-h than conventional systems.

III. PRELIMINARY COMPARISON WITH CONVENTIONAL WIND ENERGY SYSTEMS

A comparison of the performance of the WWT relative to other systems is shown in Figure 19. The shaded area illustrates the range of C_p's that are potentially achievable from the innovation system. Although careful study is required to verify the exact curve, the WWT is competitive in terms of the C_p coefficients for other systems. Additionally, part of this study will be directed towards the development of blades that have additional life and will extend the performance to higher tip speed ratios.

To provide some concept of the size of the WWT relative to other systems, a comparison is made with the MOD-0, 100 kW experimental WTG characteristics. This model is chosen because of the abundance of data relative to its performance. Table 3 lists the data utilized in making the comparison.

The power output of both systems was assumed equal to 100 kilowatts and both systems were treated as operating in an 8 m s^{-1} mean wind speed. To establish the relative dimensions of the two systems, their characteristics can be equated as shown:

$$[C_p \rho \pi R^2 V_i^3/2]_{MOD-0} = [C_p \rho A_e V_i^3/2]_{WWT} \tag{10}$$

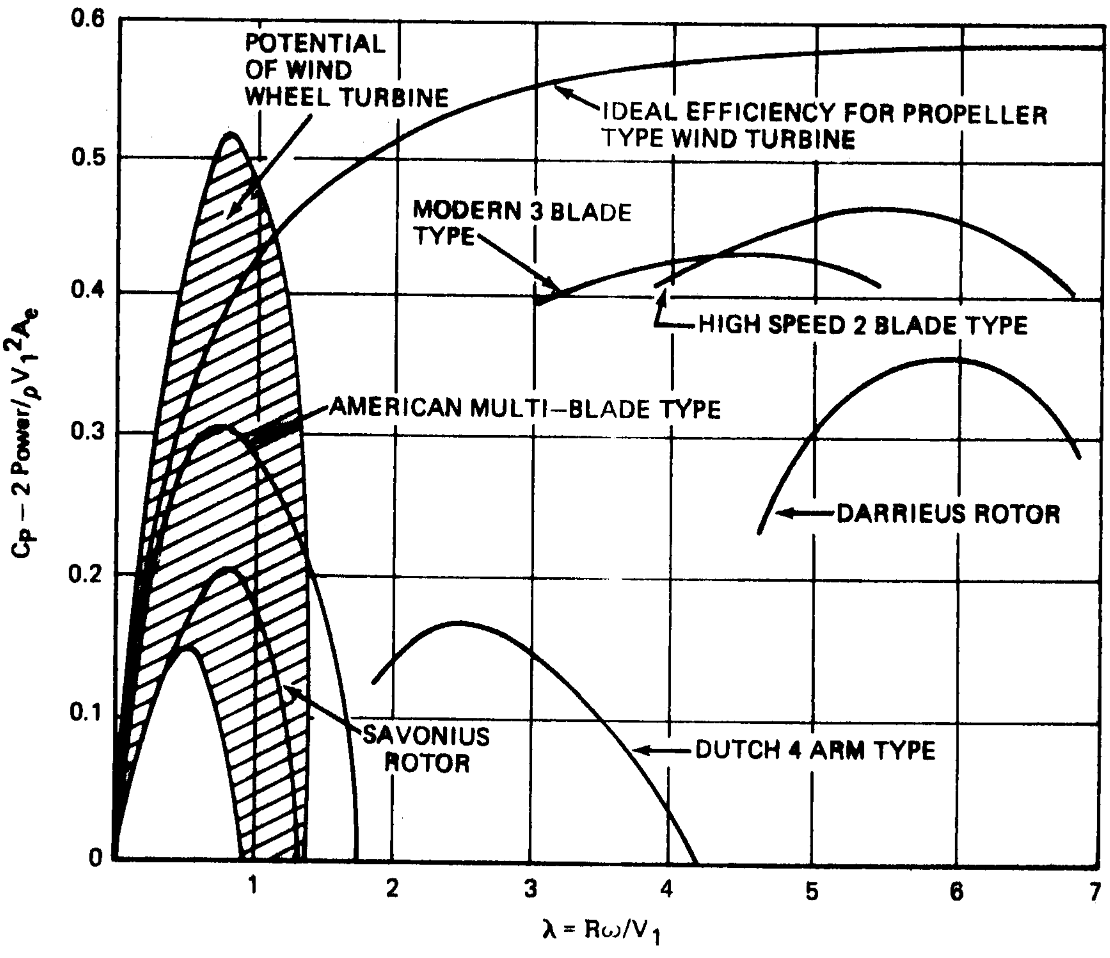

Figure 19. Comparison with performance of several wind machines.

TABLE 3

DATA USED IN COMPARISON OF MOD-0 WITH WWT

	MOD-0	WWT
Power Coefficient	0.375	0.30
Tip Speed Ratio	1.600	0.50

Carrying out the algebra and introducing a $C_p = 0.375$ and $C_p = 0.3$ for the MOD-0 and WWT, respectively, and radius of 19 m for the MOD-0 rotor dimension into Equation 10 gives

$$A_e = (3.3\omega R)_{WWT} = (1.25\pi R^2)_{MOD-0}$$

Assume a radius for the wind-wheel turbine of 15 m. This is selected somewhat arbitrarily and will be determined more rigorously during the proposed study. Introducing R = 15 m into the above expressions gives an effective width of the wind turbine wheel of approximately w = 30 m.

Based on these dimensions, an approximate to-scale comparison of the size of the WWT relative to the conventional MOD-0 wind turbine generator is illustrated in Figure 20. This clearly shows that the material quantity required for constructing the WWT is greater than that of the conventional wind turbine system. However, the rotor construction costs will be considerably less. Moreover, the fatigue load of the propeller-type rotor has shown a tendency to cause stress cracks which occur on the order of 1000 hrs or 50 days. This calls for frequent replacement of the most costly component. The sturdier wind-wheel rotor will be relatively maintenance-free which will more than compensate for the additional

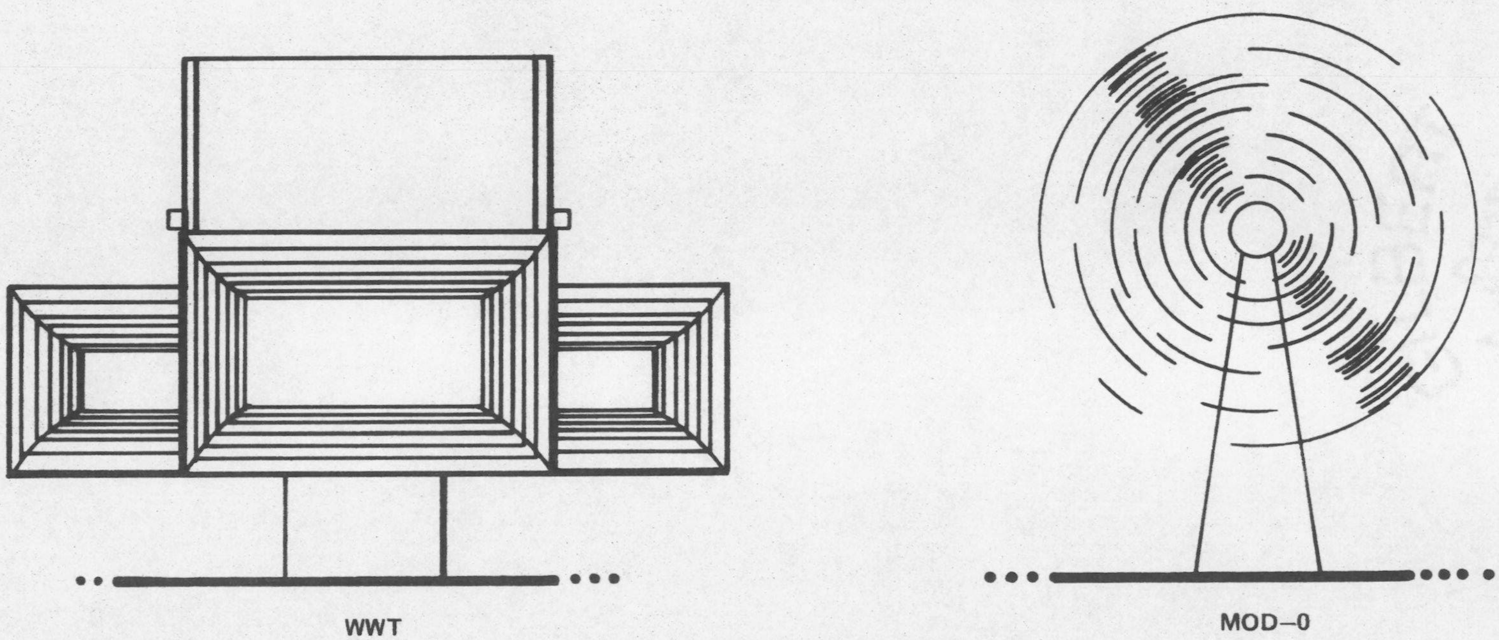

Figure 20. Approximate size comparison.

material costs if indeed they are significant in the cost comparison.

The true comparison of the MOD-0 to the WWT will not be known precisely until some actual wind tunnel and field tests are conducted with a full-scale WWT. Also, there is no reason to initiate work with such a large WWT unit until further research is conducted from both an analytical and experimental effort on optimum configuration, sizing, wheel blade number, etc. It is anticipated that a WWT to provide the power needs for a single dwelling home would actually be quite small.

CHAPTER IV

PROPOSED RESEARCH PLAN

There is a need to perform four basic tasks in order to define the optimum WWT configuration, to determine the feasibility and economic viability of the innovative wind-wheel turbine concept; and to compare it with conventional wind energy systems.

I. GENERAL PLAN AND WORK TASKS

Based on the technical approach the four technical tasks are described below.

Task 1: Performance and Design Analysis

A performance analysis of the innovative concept must be carried out and the necessary design criteria established. The analysis must include determination of the governing relationships for predicting torque and power output, wheel configuration, duct design, and other features. The possibility of using segmented blades designed to provide additional lifting force would also be studied, along with their fundamental relationships. Flow through the multiple duct units will be analyzed, and the optimum duct configuration for the front and side duct concentrators selected. The analysis must be supported by an experimental program as described in Task 2.

With the governing relationships in hand, definitions of applications and requirements of the system must be specified and a conceptual design carried out. The conceptual design must match subsystem components required for a complete system for converting wind energy into electrical/mechanical energy. These components consist of, essentially, a turbine, transmission, generator, and tower. Also, a control system is required. Static and dynamic load analysis of the final configuration must be made, and mathematical relationships between system performance and cost must be incorporated into a computer program. The computer program would optimize the system to provide minimum costs of power output. This computer program must be used in Task 3 to carry out parametric analysis relative to comparing the innovative new system with conventional wind energy systems.

Task 2: Experimental Verification of the Model

An experimental program must be carried on simultaneously with Task 1 to verify the analytical modeling and to develop empirical relationships. The experimental program would first extend the preliminary data by testing the existing conceptual model. This testing would verify the necessity and efficiency of the various aspects of the design including blade configuration, duct configuration, support loading, and starting wind speed. The experimental study must include verification of the optimum configuration of the forward venturi concentrator, the side scoop concentrators, the top cowling, and the exhaust duct under the rotor, and other factors which are determined analytically in Task 1. The usefulness of other modifications suggested by the analytical program will also be investigated, thus integrating the efforts of Task 1 with the experimental program. Tests must then be performed on the final conceptual configuration determined from the analysis and the experimental tests on the initial design concept. The results of the tests will provide empirical input and physical understanding to the computer program to be developed under Task 1.

Task 3: Comparison with Conventional Systems

A set of guidelines based on system requirements must be established. Utilizing the computer program developed on the basis of the performance analysis carried out in Task 1 and experimental data resulting from Task 2, a parametric analysis to optimize the conceptual design must be carried out. The optimum design would then be compared with conventional propeller-type or Darrius-type wind energy conversion systems. The comparison should be made on the basis of low power generation costs and high system reliability.

Task 4: Future Plans

Based on the results of Tasks 1, 2, and 3 above, a preliminary design for future development must be developed, if the preceding described study establishes that it is warranted. The preliminary design would provide a prototype and a detailed plan for field testing. Alternately, should additional work relative to analysis and experiment in the laboratory be required, the program must be carefully defined, justified, and the expected result delineated.

The tasks described above have many interrelated lines of action, and several phases of each task must be carried on simultaneously. To illustrate the interrelationships in the study approach, Figure 21 is provided.

The technical approach on what it will take to accomplish the four major tasks is given in the following sections. Section II describes the analyses to be carried out relative to optimizing the

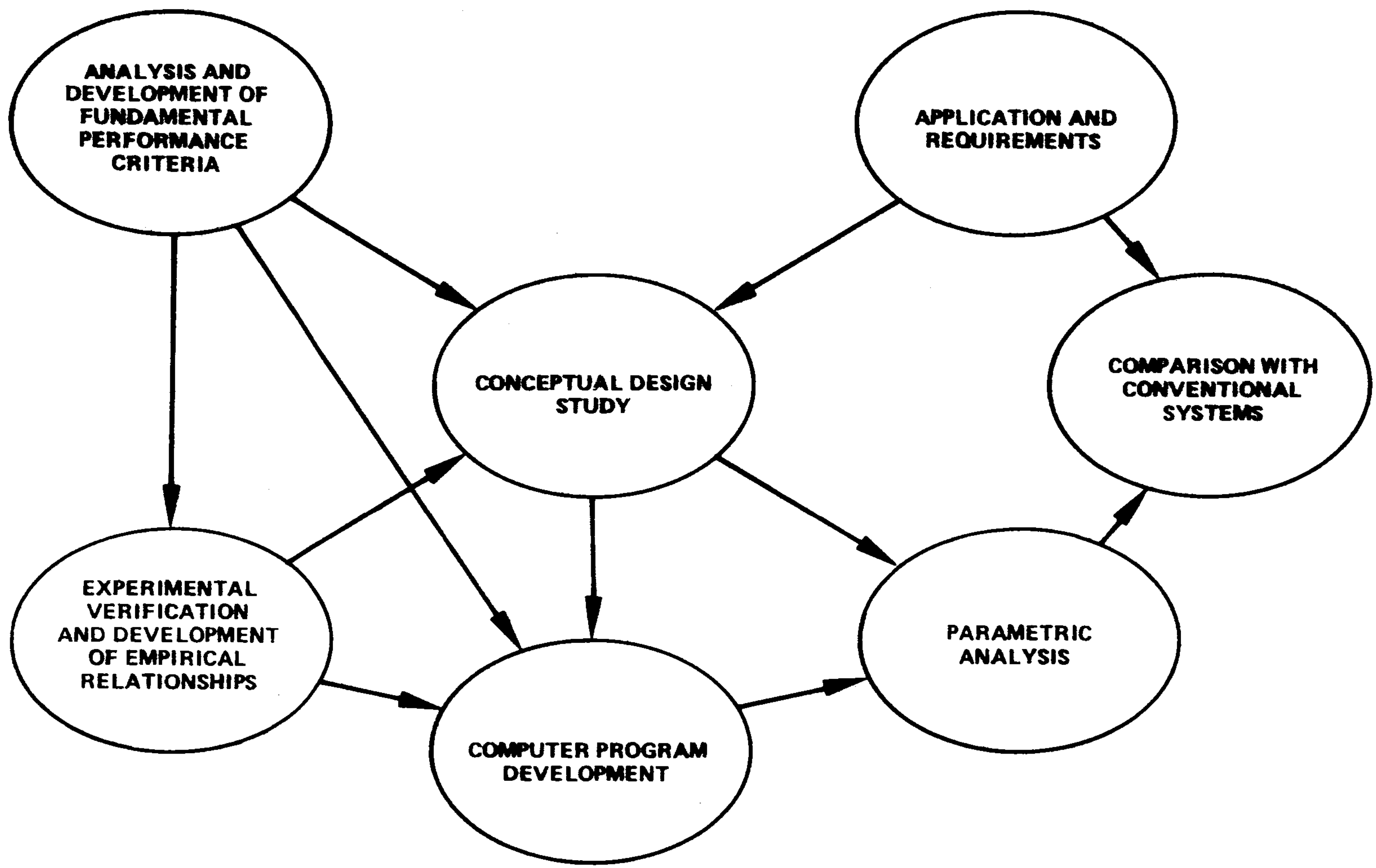

Figure 21. Study approach.

design of the WWT rotor and ducting system; to determine the size, weight, and estimated construction cost from analytical and dynamic load analyses; and to determining the coupling between a generator/mechanical subsystem. Sections III and IV address the experimental program to verify the preliminary design concept and to provide experimental data relative to the final optimized design configuration. Section V describes how comparisons between conventional competitive systems, both from a cost and performance point of view, will be carried out. Finally, Section VI is a brief discussion on the future plans to perform additional research on the WWT system.

II. PERFORMANCE AND DESIGN ANALYSIS

The analytical analyses should be carried out in essentially three phases. The initial effort could develop analytical models which will allow predictions of overall performance capabilities of the system, i.e., power and torque coefficients as a function of wind speed, tip speed, etc. These efforts will require mainly fluid dynamic analysis of the flow through the ducts and the forces exerted on the rotor by the wind. Once these fundamental relationships are developed, a conceptual design can be determined based on certain established applications and requirements of the system. Matching of the system with the electric generator or mechanical subsystem and determining the optimum transmission and tower arrangements can then be carried out. This conceptual design phase must establish one or two feasible WWT configurations which will be selected for more detailed analysis. Static and dynamic load analysis of the final concept will be carried out to determine the size and weight and method of construction.

With this information in hand, cost estimates can be developed in terms of material volume, fabrication cost per pound, etc., and a computer program must be written to optimize cost versus power output. Sufficient understanding of the proposed WWT system would then be available for comparison with conventional systems which would be carried out as Task 3.

Development of Performance Criteria

The detailed analysis must be carried out relative to the fluid mechanics of the system. These anlyses would basically extend the mathematical model described in Appendices A and B. The simple, "back-of-the-envelope" engineering calculations carried out in Appendix A include several simplifying assumptions that must be relaxed for a more rigorous analysis as briefly outlined in Appendix B. Thus considerable additional detail will be included in the analytical model development.

For example, the flow over the blades requires a momentum balance rather than simply relating force to the square of the relative velocity squared. The moemntum balance would indicate the momentum loss due to escaping air and thus define the required exhaust ports. Figure 22 illustrates some simple concepts for reducing the back pressure on the airflow while retaining the major portion of the momentum for transferring force to the blade.

Boundary layer effects must also be considered. These will result in additional loss of momentum on the blade due to friction forces. The boundary layer buildup and the momentum transfer would then identify the optimum blade configuration required to provide the

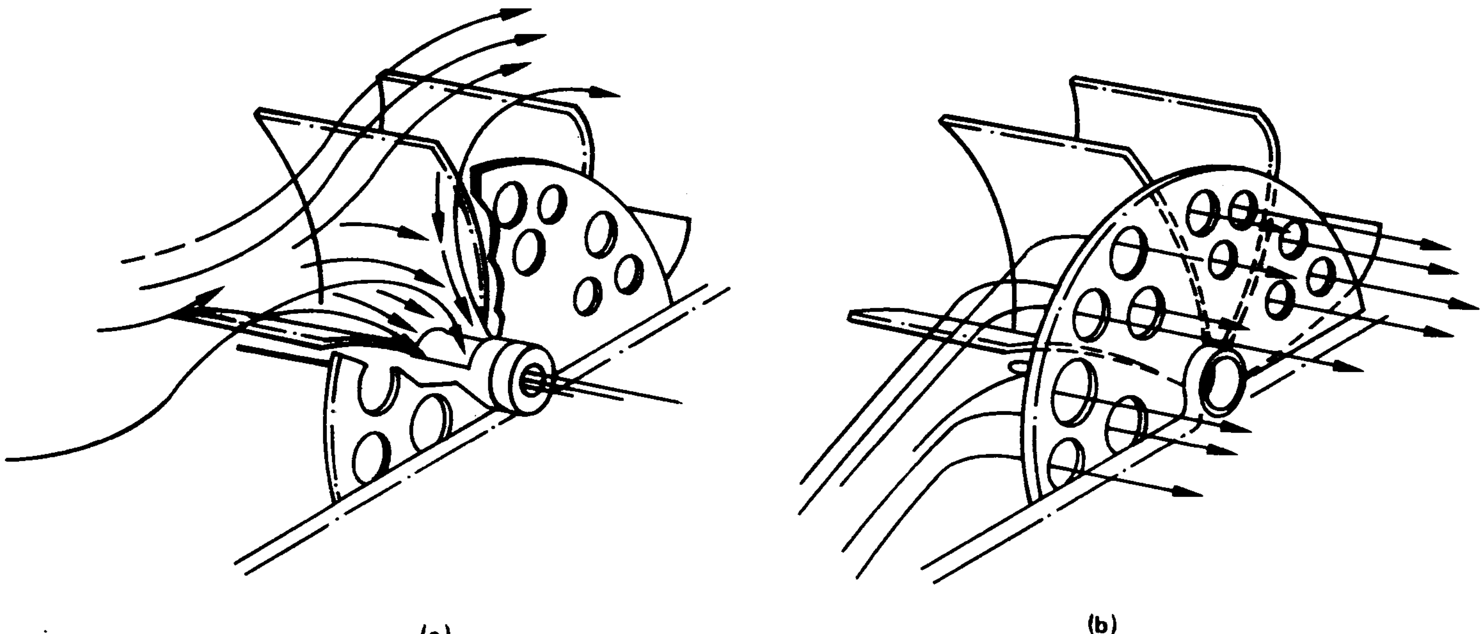

Figure 22. Concepts for exhausting air flow.

most efficient flow and thus transmit the highest force possible. Boundary layer analysis should also be applied to the friction drag created by the fluid flow between the plate and the supporting walls. This analysis must indicate the optimum separation distance between the rotating wheel and the supporting side walls.

Flow through the front and side concentrators must also be analyzed in detail. The effect of pressure losses due to friction and turning in the duct must be carefully analyzed in order to achieve the maximum flow acceleration while redirecting the flow to impact the blades as desired. These analyses will define the duct configurations and sizes necessary to capture the maximum kinetic energy of wind. Also, an important consideration in the duct analysis will be the angle of the side scoops. These ducts must provide strong weather vaning forces and must be appropriately designed to remain stable forces under varying wind conditions.

The mathematical models of the flow process would then be applied to determine the optimum wind wheel configuration, including the number of blades, the blade contour, and other factors necessary to provide a high and uniform torque. In this regard, it may be feasible to couple the motion with a flywheel in order to maintain uniform rotational speed.

A similar analysis of the fluid mechanics could be carried on possible lifting surface type blade configurations. If a lifting blade concept such as is illustrated schematically in Figure 23 can be determined, it would permit tip speed ratios greater than unity and thus extend the capabilities of the wind turbine wheel closer to the range of tip speed ratio for conventional wind turbine generators.

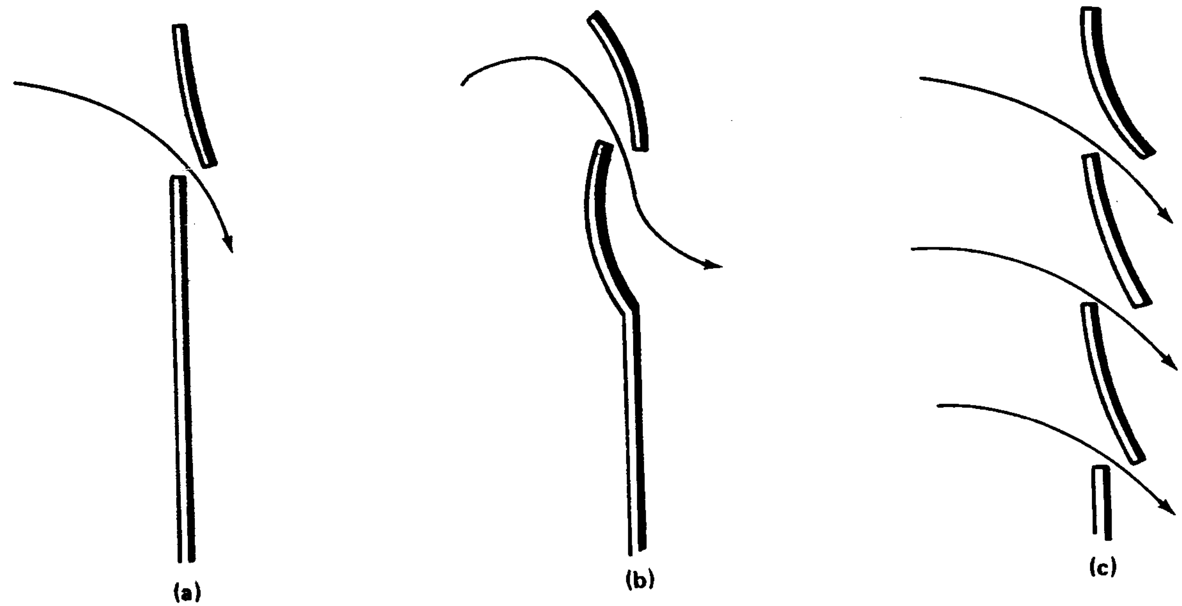

Figure 23. Possible configurations of blade tips to provide pressure distribution which augments the force acting on the blade.

Other fluid mechanics problems in the system must also be analyzed at this time; for example, handling of high wind speeds (i.e., 100 year winds), self-starting characteristics, and other fluid mechanics related problems.

Conceptual Design

This phase of Task 1 would be focused upon evaluating the various alternative concepts which could be used in the WWT system. The objective must be to select those concepts which offer low power generation costs and high system reliability for eventual use in preliminary design.

The conversion of wind energy into electrical/mechanical energy requires the following types of components: a turbine, transmission, generator, tower, and control system. The relationship between wind velocity, wind-wheel diameter, blade arrangement, and shaft power output will be determined in Task 1. Once the wind energy is converted to rotating shaft power, the shaft speed, n_r(rpm), must be made compatible with the electrical generating equipment via the power transmission. The electrical generator then converts the transmission output shaft power to useful electric output. The definition of useful output will depend upon certain design guidelines and requirements of the system which must be designed before the conceptual analysis can be carried out and before a comparison can be made. The applications and requirements of the system will be defined through Task 3 efforts which would be carried on simultaneously with Task 1.

A control system is also required to regulate and protect the system. It is not certain at this time to what degree the control system will be developed; however, very sophisticated, fully automatic models are outside the scope of the proposed study.

Rotor. In the conceptual design, the operational mode of the WWT must be determined which would set certain requirements for the rotor. Two operational modes are common, i.e., either constant rpm output of the system which requires some controls or variable rpm. Methods of controlling the WWT rpm include changing the azimuthal orientation of the WWT (yaw), changing the configuration of the blades, or opening and closing ducts, etc. (see Figure 7, page 13, and Figure 8, page 20). Each method must be investigated. Other parameters involved in the conceptual design study would be rotor dimension and characteristics including structural design and fabrication.

Transmission. The power transmission system must be designed to be compatible with the operating characteristics of the turbine and the generator. Fixed ratio and variable ratio transmission will be considered.

Generator. A number of electrical generator options are available to convert mechanical power into electrical power. The electrical power generated must be compatible with the applications and requirements set down as guidelines for the conceptual development and for the comparison with conventional systems. The generator selection would then depend primarily on cost to meet the requirements and on the machine efficiency in the conversion process. The wind-wheel turbine can be designed and operated to provide mechanical power either at a variable or at a constant shaft speed (rpm). With the variable speed rotor, it is possible to extract more power from the wind, but a variable speed transmission or a variable speed generator, which are expensive, will be needed. A constant speed rotor requires a more elaborate control system, but permits the use of both a fixed gear ratio transmission and a constant speed generator. Therefore, both constant and variable speed generators will be considered in the conceptual design studies. Also, the trade-off between induction and synchronous generator options must be assessed.

Tower. In developing the tower concept, cost will be a major consideration, although the importance of aesthetics must not be overlooked. A number of candidate tower designs concepts have potential. For example, reinforced concrete, truss tower, tube shell, pole, multiple pole, etc. Concrete and truss concepts are illustrated in Figure 24. The importance of selecting rotor/tower natural frequencies in order to arrive at a low cost, technically acceptable solution is recognized and must be carefully considered.

Thus, from this part of Task 1, configuration, operating mode, and subsystem selections must be determined and included in the development of a computer program for parametric analyses.

III. COMPUTER PROGRAM DEVELOPMENT

The large quantity of independent and dependent cost variables associated with the design of a wind turbine generator necessitate a detailed parametric analysis with sophisticated optimization methods to resolve the most cost effective system. Mathematical relationships between system performance and cost must be incorporated into a computer program which would search for the minimum cost per kilowatt-hour by selecting the appropriate WWT operating conditions for the assumed site.

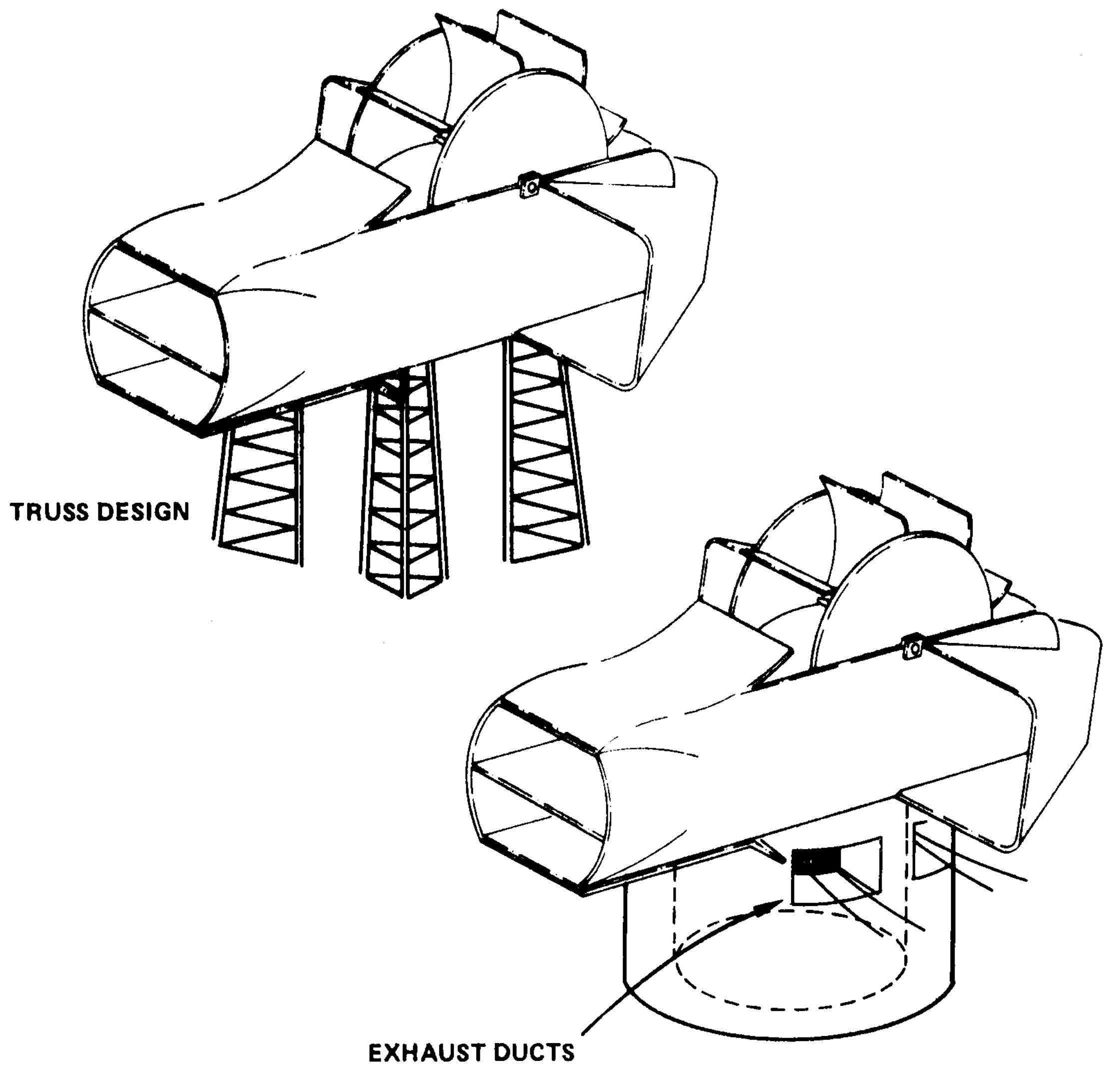

Figure 24. Illustration of tower concepts.

The computer program would characterize the system component requirements, operational conditions, and cost as a function of any specific site/wind environment. The functional objective for the computer program must be to determine the minimum cost per kilowatt-hour, to identify the system, and to evaluate the system cost sensitivity for a wide range of performance and design variables. It is envisioned that the analytical model programmed will operate on a given set of independent variables and calculate the cost per kilowatt-hour. The optimization technique would assess the cost sensitivity to each independent variable and simultaneously select new independent variables for the second perturbation of the optimization. Major inputs to the program must include the assumed site wind statistics, wind turbine wheel aerodynamic performance characteristics, and system component cost/weight equations as a function of appropriate system performance variables. Major program output would be minimum ¢/kw-h, subsystem cost and weight breakdown, optimum performance and design parameters, and a breakdown of the annual power and energy production.

Computer Program Assumptions

Performance assumptions would include the environmental aspects, wind turbine wheel aerodynamic characteristics determined in Tasks 1 and 2, mechanical efficiencies of power transmission, and the electrical efficiency of power regulation and energy conversion. The economic assumption must include the equipment capital costs, the methods and costs of capitalization, costs of system integration, installation, operation and maintenance, and the effects of system life and annual production rate on costs. A third category of input assumptions is that of subsystem component weight which would indirectly affect both the performance and economics of the system optimization.

Performance Assumptions

Environmental. The environmental assumptions must include a wind duration curve with characteristic annual median wind speed, a wind shear profile which is site dependent, and a characteristic site air density [3,4].

Aerodynamic characteristics. Aerodynamic characteristics for the wind turbine wheel in the form of power and thrust coefficients as a function of velocity ratios would be modeled in the computer program.

Mechanical. The computer would analytically model the main shaft axial thrust interaction with the tower through the bearings and the mechanical power transmission through a gearbox. Efficiencies must be determined from vendors' catalogs from which commercially available components would have been selected.

Electrical. The generator efficiency variations and the limiting generator operating conditions must be programmed from manufacturers' specifications.

Economics

The significant portion of the analytical model of the computer program must be devoted to the calculation, integration, and summation of cost information. Cost assumptions would be included in equation form; these expressions would relate cost data with the appropriate systems' environmental, performance, and economic variables. Table 4 shows typical expressions for capital costs which have been used in previous wind turbine generator design. These equations would require modifications for the current proposed program.

TABLE 4

ILLUSTRATION OF CAPITAL COST EXPRESSIONS WHICH CAN BE USED TO SHOW WHICH SYSTEM VARIABLES AFFECT A GIVEN COST AND TO WHAT DEGREE

Subsystem/Component	Capital Cost, Expression Used During Parametric Analysis, $
Rotor	$C = (33.77\ D^{2.22}/n^{0.11}) + 20{,}000$
Gearbox	$C = 2.987\ Q_R^{0.795}$
Main Shaft	$C = 0.01835\ Q_R^{1.097}$
Flexible Couplings	$C = 0.0324\ Q_R^{1.030}$
Main Bearings	$C = 0.150\ Q_R^{0.8645}$
Primary Brake	$C = 145.7\ P_R^{0.300}$
Generator (Synchronous)	$C = 84.12\ P_R^{0.835}$
Generator Accessories	$C = 169.1\ P_R^{0.446}$
Controls	$C = 3636 + 9.2\ P_R$
Truss Tower	
($T_eh_T < 2 \times 10^6$ nm)	$C = (8078 + 4.24 \times 10^{-3}\ T_eh_T)1.56/n^{0.084}$
($T_eh_T > 2 \times 10^6$ nm)	$C = 11{,}894 + 2.40 \times 10^{-3}\ T_eh_T)1.56/n^{0.084}$
Concrete Tower	
($T_eh_T < 2.4 \times 10^6$ nm)	$C = 0.65[9397 + 2.33 \times 10^{-3}\ T_eg_T]/1.13\frac{n-1}{n}$
($T_eh_T > 2.4 \times 10^6$ nm)	$C = 0.45[5053 + 4.41 \times 10^{-3}\ T_eh_T]/1.13\frac{n-1}{n}$

C = Cost

D = Diameter

Q_R = Rated torque

P_R = Rated power

T_e = Equivalent thrust

h_T = Tower height

n = Quantity

Studies to investigate cost sensitivity to parameters such as the following must be carried out [3,4].

1. Environmental: a) Annual median wind speed
   b) Wind shear characteristics
   c) Altitude
2. Design: a) Tower height
   b) Tower type
   c) Wind-wheel characteristic and geometry
3. Operational: a) Wind-wheel rpm
   b) Cut-in/cut-out velocities
   c) Rated wind speed
4. Economic: a) System design life
   b) System production quantity

IV. EXPERIMENTAL VERIFICATION

The experimental portion of the research study must serve several purposes. First, only preliminary and incomplete data are available for the conceptual model, and the first tests are needed to the complete testing of this model. These tests would be used to verify the necessity and efficiency of various aspects of the design including the forward venturi, the side scoops, the top cowling, the exhaust duct under the rotor, and the angle of the blades outside the rotor walls. Second, the experimental program would make it possible to examine the usefulness of modifications that are suggested by the analytical program. In this respect, the analytical and experimental programs must complement each other. Finally, tests must be performed on what appears to be the optimum configuration. The results of these final tests would be used to assist in making a performance and cost comparison with conventional wind systems.

The experimental, procedural plan follows:

1. An analysis must be made of the mechanics of the wind turbine and the applicable scaling laws determined. These scaling laws must be used for determining the model size, weight, and the test conditions.

2. A wind tunnel model would be built based on these scaling laws. The model will be made in the fashion of an "erector set" so that it can be disassembled easily in different configurations. The first configuration to be studied is that shown in Chapter II. Provision must be included for applying a load to the rotor shaft and measuring the torque and rotational speed. Also, the base would be connected to a three-component force balance that would be used to measure the fluid dynamic force and moment on the wind turbine structure.

Several different techniques are available for measuring the rotor power. First, a torque meter (or dynamometer) can be used to load the shaft and measure the force applied at the end of an arm. The force and moment arm give the torque an independent measurement of the shaft rotational speed needed to give power. Another method would be to connect the shaft to a generator and load the generator with a variable number of light bulbs. A wattmeter would then be needed to measure the power. A commercial unit would be chosen for measuring the rotor power.

3. The initial configuration must be examined in a low speed wind tunnel to investigate its performance under load. At present,

only very preliminary and incomplete data have been taken on this configuration and, as a starting point, it is important to completely test the configuration. The performance data needed include torque and power for various wind speeds, performance at various wind orientations, and performance under a rapid change of wind direction. The damping of the motion of the wind turbine in varying wind directions needs to be known.

4. An investigation must be made of the flow characteristics and necessity of various aspects of the initial configuration, including the entrance venturi, the side scoops, the top cowling, and the tip geometry of the blades. At this point, it may be useful to use a transparent plexiglas model in a water tunnel to make sure that separated or stalled flow regions do not occur in the ducts. If such flow regions occur, then the geometry would be modified to eliminate them.

The ducting is designed to reorientate the direction of the air so that it must impinge upon the turbine wheel in such a manner as to increase the wheel torque and, insofar as possible, to increase the flow velocity. It is important that the ducting perform these tasks properly for the wind-wheel turbine to realize its greatest efficiency. The increase in velocity possible form the venturi would depend upon the pumping action of the turbine rotor. Therefore, it will be necessary to measure the pressure at the exit to the venturi as well as at the exit to the side scoops. The geometry of the ducts will depend in large measure upon the pressures which are generated in the wake and underneath the wheel, and these pressures will be difficult to estimate theoretically. Therefore, the geometry would be modified as a result of the pressures measured in the wake and under the wheel.

5. An examination must be made of the effect of configurational changes that are suggested by the analytical studies. The purpose of this phase of the testing would be to simplify the geometry (to reduce construction costs) while increasing the performance. Unless complicated configuration changes can be shown to lead to substantial performance increases, they would be abandoned.

6. When the optimum WWT configuration has been decided upon from Task 1, it must be completely tested to prove input to the economic analysis, which would be the major consideration in comparison of the system with conventional wind systems. This comparison would depend upon the results of the above tests, namely, whether the wind turbine appears to be more useful as a power device, or as a torque device. The most likely use of the turbine would be taken into consideration in the conceptual design study.

V. COMPARISON WITH CONVENTIONAL SYSTEMS

Task 3 has, essentially, three phases. First, the applications and requirements of the wind energy systems to be compared with the innovative concept must be carefully spelled out and guidelines for comparison established. Second, a parametric analysis utilizing the computer program developed in Task 1 would be carried out to optimize the wind-wheel turbine system subject to the requirements. Data on conventional wind energy systems, in particular propeller and Darrius-types which also meet the applications and requirements defined, would be obtained either from the literature or specified through discussions with the SERI technical monitor [5,6]. A comparison of the systems would then be made on the bases of ¢/kw-h capabilities of each system and with consideration of performance reliability.

Applications and Requirements

A meaningful comparison with a conventional system would require establishing the requirements of the wind energy conversion system. Both the conventional system and the innovative system must meet these requirements with similar assurance of reliability. Typically, a low power system is connected to a local existing utility network or to separate loads dictated by its application, i.e., multi-unit farms located in favorable wind locales, etc., whereas a high power system is tied into a major public utility network. Once the capabilities of the wind-wheel turbine are fully understood from the conceptual design, these requirements and potential applications can be established.

Considerations which must be made in the requirements specifications include the following:

Operations. Operations including start-up, shut-down, and power regulation over the full range of wind operations must be equivalent or superior to those of conventional systems.

Design life. All static components including the tower should be designed for a service life of several years. All dynamic components should be designed for a service life of 30 years and may include periodic maintenance and replacement. The effect of design life on cost relative to the conventional and innovative system must be parametrically studied.

Parts and components. Comparisons must be made with material requirements and fabrication processes of the two systems. Also, off-the-shelf subsystem components with proven performance will be utilized in making comparisons of reliability.

Maintenance and serviceability. Ease of maintenance should be considered in comparing the two systems, such as platforms, stairs, removable covers, etc.

Assembly. A comparison of the shop assembly time versus field assembly time must be made.

Transportability and erection.

Appearance.

Environmental. The units must be designed to withstand a range of atmospheric environments. Considerations of operations in snow, rain, lightning, hail, icing conditions, saltwater vapors, wind blown sand and dust, and in temperature extremes are prime considerations when comparing systems.

Wind data. To determine the power and/or energy extracted from the comparable systems, the wind velocity duration curves must be utilized.

Design requirements for the other subsystem components of the conversion system would be defined relative to comparing the innovative concept with conventional systems.

Parametric Analysis

Utilizing the computer program developed under Tasks 1 and 2 and the design requirements "spelled out" further established in the first part of Task 3, a parametric analysis of the WWT system must be carried out to optimize its performance for the applications and requirements specified. Existing conventional systems would be selected that also meet the requirements spelled out, and it will be assumed that the design of these systems have been optimized relative to their documented performance specifications. Comparison of the systems would then be made.

Comparison with Conventional Systems

Having completed the work described above, comparisons of the systems would be relatively straightforward. The cost per kilowatt-hour provided by the two systems would indicate which system is the most cost-effective. The theme which would dominate the comparison procedures would be the minimization of capital and maintenance costs while maximizing the energy capture. A comparison of the use of commercially available components with established records of reliability would be compared for the two systems. How each design allows for ease of maintenance and the simplicity and versatility of the control system must also be assessed. It is projected at this time that the comparison should look at two possible systems; one for high power performance and another for lower power performance. The innovative system may prove to have advantages in one application while having less advantage in another application. The power level selected for making the comparisons of the two systems would be based on the specified yearly mean wind speeds at a site with $z_0 = 0.05$ given in the RFP. The wind velocity at which the machines would be rated is a consequence of the assumed wind site. The predicted annual operating time for each of the systems must also be a controlling factor in the comparison. These would be dependent upon the wind duration curves given. In this respect, capacity factors which are the energy generated within a specific time frame divided by the energy which could be generated if rated power was attained continuously would be figures of merit for comparison of the two systems.

Of course, simplicity of turbine rotor design, mechanical power transmission, suitable generator types, and tower construction must enter the comparison primarily through cost benefits, but the aesthetics of the tower and system would not be completely neglected.

VI. FUTURE PLANS

The results of Task 3 would indicate whether the system has advantages over conventional systems and whether further development should be carried out. If the comparison shows that the two systems are marginally competitive, then it may be necessary to perform additional research and experiment relative to the innovative system to either perfect it further or fully substantiate certain features and characteristics. The effort of Task 4 would thus be directed towards determining whether a prototype should be built and field tested or whether additional analysis and testing of the concept are required, and finally whether the concept should simply be ruled out as a candidate for wind energy conversion.

If the comparison indicates a tremendous advantage of the innovative system over conventional systems, a prototype preliminary design would be outlined and a plan for field testing documented. The degree to which this would be completed under the first year's effort would depend appreciably upon adherence to the time schedule for the relatively ambitious efforts described in Tasks 1, 2, and 3.

CHAPTER V

SUMMARY AND RESULTS

With the excessive use of oil, natural gas, coal, etc., it is imperative that countries of the world place more emphasis on the use of our natural energy sources such as the wind. The necessity for this is constantly being stated by energy resource engineers everywhere. As is known, one of the cleaner sources of energy is the wind. Therefore, when this wind-wheel turbine is properly designed, tested, and the results are favorable it could be utilized to supply a fair amount of energy for private as well as commercial needs.

The WWT Model No. 1 test results have shown, (1) that the response of the device to wind is very pronounced, (2) that various combinations of duct closures reduce the wheel turn speed as anticipated, (3) that the wheel turns in the proper direction, but at less speed, regardless to how it is oriented into the mean wind direction, (4) that the wheel continues to rotate, but at a lesser speed, with a hood placed over the top half of the wheel and, (5) if extreme winds were expected the overall functional system could be turned 180 degrees into the wind and the wind energy conversion would continue but at a lesser amount.

The WWT Model No. 2 test results showed that, (1) the wheel response is a function of the square of the wind velocity, (2) the threshold (start) wind speed is about 3.1 m s^{-1}, (3) the relationship for the moment of wind force (torque) divided by the moment of inertia can be expressed by a term which includes the square of the mean wind speed, (4) the power coefficients, although quite low for the device, are about what would be expected for the poorly designed WWT Model No. 2, and (5) once given a significant number of wheel turns per time the wheel stops quite suddenly due to bearing friction and air resistance. Of course, this testing was done in still air environment.

The findings from these WWT model tests are considered to be quite satisfactory where, (1) the models were not built to scale and (2) only room ventilator air and a jet of air was used to make the models function.

As for the results from the preliminary analytical performance analysis of a four bladed wheel (see Appendix A) it was determined, (1) that the forces acting on each blade will be different in each quadrant but will repeat itself in subsequent quadrants, therefore, only the force acting on each blade during a 90 degree arc needed to be calculated, and (2) when the torque coefficients for each blade is summed, the torque varies cyclically over each 90 degree period of rotation.

In Appendix B, Discussion of Duct and Rotor Analyses, the interaction of the air with the wind-wheel turbine and the flow of air through the ducts can be described by the Navier-Stokes equations. That, with moment equations, shows that the flow about the rotor is always unsteady and difficult to analyze. However, as the number of blades is increased the rotor flow can be seen to be approximately steady. The actual velocity distribution of the air leaving the rotor will have to be determined either from a solution of differential equations or from actual measurements. In future studies the standard boundary layer approximation to the differential equations will be used to find the viscous torque. The analysis can then be used to modify the disk geometry to lower the viscous torque.

It is anticipated that this WWT will operate as well if not better than the conventional bladed wind turbines to convert the kinetic energy of the wind into useable power. This fact will be proven through modification of the design, if determined necessary, and test of properly scaled models and fully scaled field units.

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APPENDICES

APPENDIX A

PRELIMINARY PERFORMANCE ANALYSIS

The torque and power coefficients for a straight four-bladed wind-wheel are estimated. Consider the wind-wheel with three wind jets V_1, V_2, and V_3 acting upon it as shown in Figure A-1. The forces acting on the baldes A, B, C, and D will be different in each quadrant but will repeat in subsequent quadrants. Therefore, the forces acting on each blade during a 90° arc need only be calculated. Table A-1 shows typical values of the torque coefficient for the blades individually and as a total. The torque coefficient is defined as

$$C_T = \text{Torque}/\rho V_1{}^2 R^2/2 \tag{A-1}$$

Calculation of these forces is straightforward but care must be taken to assure that all interaction of the blades and jets are taken into account.

Starting from $\theta = 0$, blade B is acted upon only by jet 1. After a 45° rotation it is shielded by blade A. During the remaining 45° of travel of blade B, it has a negative force resulting from displacing the air in front of it at a speed of $r\omega$. The expression for C_{T_B} is:

$$C_{T_B} = I_4 + I_5$$

where

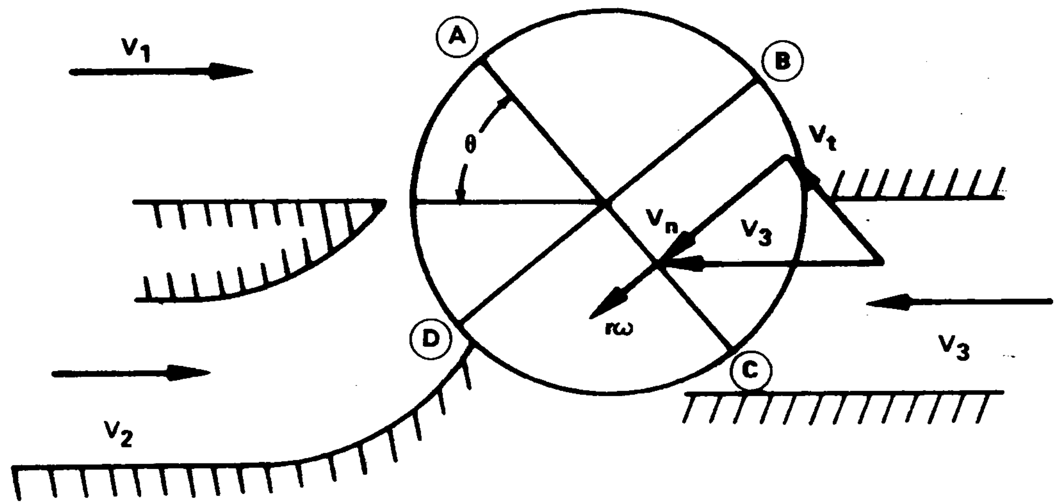

Figure A-1. Definition of jets and blade interaction.

TABLE A-1

REPRESENTATIVE RESULTS OF COMPUTED PERFORMANCE CHARACTERISTICS

```
LAMBDA=0.0  XI2=0.0  XI3=0.1
   THETA= 0.  CT=  -0.090  CP=  -0.054  CPP=  -0.016
CTA=-.160E+00  CTB=0.360E+00  CTC=-.180E+00  CTD=-.110E+00
   THETA= 5.  CT=  -0.033  CP=  -0.020  CPP=  -0.006
CTA=-.116E+00  CTB=0.366E+00  CTC=-.173E+00  CTD=-.110E+00
   THETA=10.  CT=  -0.012  CP=  -0.007  CPP=  -0.002
CTA=-.704E-01  CTB=0.336E+00  CTC=-.166E+00  CTD=-.111E+00
   THETA=15.  CT=  -0.021  CP=  -0.013  CPP=  -0.004
CTA=-.358E-01  CTB=0.286E+00  CTC=-.160E+00  CTD=-.112E+00
   THETA=20.  CT=  -0.053  CP=  -0.032  CPP=  -0.010
CTA=-.107E-01  CTB=0.224E+00  CTC=-.154E+00  CTD=-.113E+00
   THETA=25.  CT=  -0.097  CP=  -0.058  CPP=  -0.018
CTA=0.903E-02  CTB=0.154E+00  CTC=-.148E+00  CTD=-.113E+00
   THETA=30.  CT=  -0.143  CP=  -0.086  CPP=  -0.026
CTA=0.279E-01  CTB=0.805E-01  CTC=-.142E+00  CTD=-.109E+00
   THETA=35.  CT=  -0.179  CP=  -0.108  CPP=  -0.033
CTA=0.501E-01  CTB=0.514E-02  CTC=-.137E+00  CTD=-.971E-01
   THETA=40.  CT=  -0.197  CP=  -0.118  CPP=  -0.036
CTA=0.789E-01  CTB=-.756E-01  CTC=-.133E+00  CTD=-.674E-01
   THETA=45.  CT=  -0.194  CP=  -0.116  CPP=  -0.035
CTA=0.114E+00  CTB=-.180E+00  CTC=-.128E+00  CTD=-.334E-07
   THETA=50.  CT=  -0.331  CP=  -0.198  CPP=  -0.060
CTA=0.154E+00  CTB=-.180E+00  CTC=-.125E+00  CTD=-.180E+00
   THETA=55.  CT=  -0.285  CP=  -0.171  CPP=  -0.052
CTA=0.196E+00  CTB=-.180E+00  CTC=-.121E+00  CTD=-.180E+00
   THETA=60.  CT=  -0.241  CP=  -0.145  CPP=  -0.044
CTA=0.237E+00  CTB=-.180E+00  CTC=-.118E+00  CTD=-.180E+00
   THETA=65.  CT=  -0.199  CP=  -0.120  CPP=  -0.036
CTA=0.276E+00  CTB=-.180E+00  CTC=-.116E+00  CTD=-.180E+00
   THETA=70.  CT=  -0.162  CP=  -0.097  CPP=  -0.030
CTA=0.311E+00  CTB=-.180E+00  CTC=-.114E+00  CTD=-.180E+00
   THETA=75.  CT=  -0.132  CP=  -0.079  CPP=  -0.024
CTA=0.340E+00  CTB=-.180E+00  CTC=-.112E+00  CTD=-.180E+00
   THETA=80.  CT=  -0.109  CP=  -0.065  CPP=  -0.020
CTA=0.362E+00  CTB=-.180E+00  CTC=-.111E+00  CTD=-.180E+00
   THETA=85.  CT=  -0.095  CP=  -0.057  CPP=  -0.017
CTA=0.375E+00  CTB=-.180E+00  CTC=-.110E+00  CTD=-.180E+00
   THETA=90.  CT=  -0.090  CP=  -0.054  CPP=  -0.016
CTA=0.380E+00  CTB=-.180E+00  CTC=-.110E+00  CTD=-.180E+00

LAMBDA=0.0  XI2=2.0  XI3=0.3
   THETA= 0.  CT=   3.099  CP=   0.000  CPP=   0.000
CTA=0.200E+01  CTB=0.100E+01  CTC=0.000E+00  CTD=0.900E-01
   THETA= 5.  CT=   3.309  CP=   0.000  CPP=   0.000
CTA=0.224E+01  CTB=0.965E+00  CTC=0.664E-03  CTD=0.866E-01
   THETA=10.  CT=   3.481  CP=   0.000  CPP=   0.000
CTA=0.245E+01  CTB=0.940E+00  CTC=0.271E-02  CTD=0.846E-01
   THETA=15.  CT=   3.615  CP=   0.000  CPP=   0.000
CTA=0.266E+01  CTB=0.866E+00  CTC=0.603E-02  CTD=0.779E-01
   THETA=20.  CT=   3.717  CP=   0.000  CPP=   0.000
CTA=0.287E+01  CTB=0.766E+00  CTC=0.105E-01  CTD=0.689E-01
   THETA=25.  CT=   3.795  CP=   0.000  CPP=   0.000
CTA=0.308E+01  CTB=0.643E+00  CTC=0.161E-01  CTD=0.579E-01
   THETA=30.  CT=   3.853  CP=   0.000  CPP=   0.000
CTA=0.329E+01  CTB=0.500E+00  CTC=0.225E-01  CTD=0.450E-01
   THETA=35.  CT=   3.896  CP=   0.000  CPP=   0.000
CTA=0.349E+01  CTB=0.342E+00  CTC=0.298E-01  CTD=0.308E-01
   THETA=40.  CT=   3.930  CP=   0.000  CPP=   0.000
CTA=0.370E+01  CTB=0.174E+00  CTC=0.372E-01  CTD=0.156E-01
   THETA=45.  CT=   3.959  CP=   0.000  CPP=   0.000
CTA=0.391E+01  CTB=0.596E-07  CTC=0.450E-01  CTD=0.536E-08
   THETA=50.  CT=   4.154  CP=   0.000  CPP=   0.000
CTA=0.407E+01  CTB=0.000E+00  CTC=0.528E-01  CTD=0.304E-01
   THETA=55.  CT=   4.295  CP=   0.000  CPP=   0.000
CTA=0.411E+01  CTB=0.000E+00  CTC=0.604E-01  CTD=0.121E+00
   THETA=60.  CT=   4.370  CP=   0.000  CPP=   0.000
CTA=0.403E+01  CTB=0.000E+00  CTC=0.675E-01  CTD=0.268E+00
   THETA=65.  CT=   4.373  CP=   0.000  CPP=   0.000
CTA=0.383E+01  CTB=0.000E+00  CTC=0.739E-01  CTD=0.468E+00
   THETA=70.  CT=   4.295  CP=   0.000  CPP=   0.000
CTA=0.350E+01  CTB=0.000E+00  CTC=0.795E-01  CTD=0.714E+00
   THETA=75.  CT=   4.132  CP=   0.000  CPP=   0.000
CTA=0.305E+01  CTB=0.000E+00  CTC=0.840E-01  CTD=0.100E+01
   THETA=80.  CT=   3.880  CP=   0.000  CPP=   0.000
CTA=0.248E+01  CTB=0.000E+00  CTC=0.872E-01  CTD=0.132E+01
   THETA=85.  CT=   3.533  CP=   0.000  CPP=   0.000
CTA=0.179E+01  CTB=0.000E+00  CTC=0.893E-01  CTD=0.165E+01
   THETA=90.  CT=   3.090  CP=   0.000  CPP=   0.000
CTA=0.100E+01  CTB=0.000E+00  CTC=0.900E-01  CTD=0.200E+01
```

$$\left.\begin{aligned} I_4 &= 2\int_{\tan\theta}^{1.0} \cos\theta - \lambda\tilde{r}|(\cos\theta - \lambda\tilde{r})\tilde{r}d\tilde{r} \\ I_5 &= -2\int_{0}^{\tan\theta} \lambda^2\tilde{r}^3 d\tilde{r} = \frac{-\lambda^2\tan^4\theta}{2} \end{aligned}\right\} \; ; \; 0 \leq \theta \leq 45 \qquad \text{(A-2)}$$

$$\left.\begin{aligned} I_4 &= 0 \\ I_5 &= -2\int_{0}^{1.0} \lambda^2\tilde{r}^3 d\tilde{r} = -\frac{\lambda^2}{2} \end{aligned}\right\} \; ; \; 45 \leq \theta \leq 90$$

Blade C experiences a force due to jet 3 only. This jet results from turning the air through a 180° by the side ducts and a large portion of the wind velocity will be lost. The velocity of jet 3 is therefore $V_3 = \zeta_3 V_1$, where ζ_3 is a small factor. The equation for blade C is:

$$C_{T_C} = I_6$$

where

$$I_6 = 2\int_{0}^{1.0} |\zeta_3\sin\theta - \lambda\tilde{r}(\zeta_3\sin\theta - \lambda\tilde{r})\tilde{r}d\tilde{r} \qquad ; \; 0 \leq \theta \leq 90° \qquad \text{(A-3)}$$

Blade D is initially acted upon by jet 3 exactly as blade B. After 45 degrees of rotation, jet 3 no longer acts on blade B; however, at that point jet 2, the air being ducted in from the front and turned through a 45° angle, begins to interact with blade D. This action continues through the remaining 45 degrees of the blade travel. The expression for C_{T_D} is

$C_{T_D} = I_7 + I_8$

where

$$I_7 = 2\int_{\tan\theta}^{1.0} |\zeta_3\cos\theta - \lambda\tilde{r}|(\zeta_3\cos\theta - \lambda\tilde{r})\tilde{r}d\tilde{r} \quad ; 0 \le \theta \le 45°$$

$$= 0 \quad ; 45 \le \theta \le 90° \tag{A-4}$$

$$I_8 = 0 \quad ; \quad 0 \le \theta \le 45°$$

$$= 2\int|\zeta_2\sin(45 - \theta) - \lambda\tilde{r}|(\zeta_2\sin(45 - \theta) - \lambda\tilde{r})\tilde{r}d\tilde{r} \quad ; 45 \le \theta \le 90°$$

Blade A has the most complicated interaction being acted upon by jets 1 and 2 simultaneously. It is assumed in this simple analysis that the wind speed from jet 1 and that from jet 2 can be superimposed over the portion of the blade upon which they impinge. The velocity of jet 2 is anticipated to be higher than that of jet 1 due to the fact that it has passed through a venturi and accelerated. Thus, $V_2 = \zeta_2 V_1$ where $\zeta_2 \ge 1.0$. The expression for C_{T_A} is given by

$C_{T_A} = I_1 + I_2 + I_3$

where

$$I_1 = 0 \quad ; 0 \le \theta \le 45°$$

$$= 2\int_0^{\tan|45° - \theta|} |\sin\theta - \lambda\tilde{r}|(\sin\theta - \lambda\tilde{r})\tilde{r}d\tilde{r} \quad ; 45° \le \theta \le 90°$$

$$I_2 = 2 \int_{LB}^{\cos 45/\cos|45° - \theta|} |\sin\theta + \zeta_2 \cos|45° - \theta| - \lambda\tilde{r}| \times (\sin\theta + \zeta_2\cos|45° - \theta| - \lambda\tilde{r})\tilde{r}d\tilde{r} \qquad \text{(A-5)}$$

where

$$LB = \begin{cases} 0 \quad ; \quad 0 \leq \theta \leq 45° \\ \tan|45° - \theta| \quad ; \quad 45° \leq \theta \leq 90° \end{cases}$$

$$I_3 = 2 \int_{\cos 45°/\cos|45° - \theta|}^{1.0} |\sin\theta - \lambda\tilde{r}|(\sin\theta - \lambda\tilde{r})\tilde{r}d\tilde{r} \quad ; \quad 0 \leq \theta \leq 90°$$

To determine the overall torque coefficient for the four-bladed wheel, the torque coefficients for each blade are summed. The torque varies cyclically over a 90° period of rotation.

Power coefficients for the WWT may be computed directly from the torque coefficient as follows:

$$\text{Power} = T\omega$$
$$C_p = \text{Power}/(\rho V_1{}^3 A/2) \qquad \text{(A-6)}$$

substituting Equation 1 gives

$$C_p = \lambda C_T$$

The integral form of Equations A-2 through A-5 were integrated numerically with a Simpson's rule integration scheme for a range of the parameters λ, ζ_2, ζ_3, and β. Some of these results are plotted in Figures 15-17, pages 38-40, and Figure 18, page 42.

APPENDIX B

DISCUSSION OF DUCT AND ROTOR ANALYSES

The interaction of the air with the wind-wheel turbine and the flow of the air through the ducts is described by the Navier-Stokes equations. These equations are a set of nonlinear partial differential equations which are very difficult to solve. Whenever possible the inegral form of the equations will be used. The integral equations must be applied to an appropriate volume called the "control volume." Some control volumes that could prove useful in the present case will be rotating with the rotor and are thus in a non-inertial frame of reference. This must be taken into account. Approximations will have to be made in order to solve either the differential or integral equations.

First consider the differential equations that could be used to examine the details of the flow about the rotor. Use coordinates fixed to the rotor and assume that the flow is incompressible (ρ = constant), inviscid (only pressure surface forces) and has no velocity component in the direction of the rotor axis. This last assumption is equivalent to assuming a rotor with infinitely long blades. Using the coordinates shown in Figure B-1, the continuity and momentum equations are as follows.

$$\frac{\partial(ru_r)}{\partial r} + \frac{\partial u_\theta}{\partial \theta} = 0 \qquad \text{(B-1)}$$

$$\frac{\partial u_r}{\partial t} + u_r \frac{\partial u_r}{\partial r} + \frac{u_\theta}{r}\frac{\partial u_r}{\partial \theta} - \frac{{u_\theta}^2}{r} - r\omega^2 - 2u_\theta\omega = -\frac{1}{\rho}\frac{\partial p}{\partial r} \qquad \text{(B-2)}$$

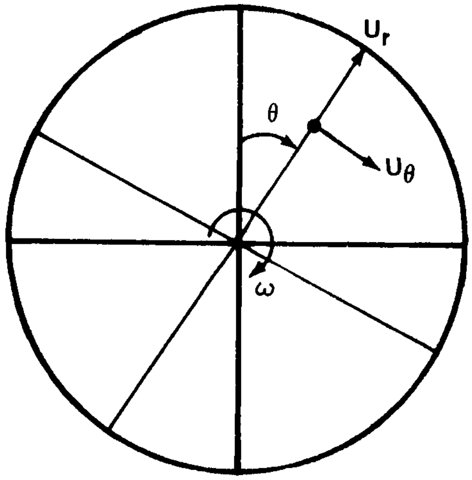

Figure B-1. Coordinate system.

$$\frac{\partial u_\theta}{\partial t} + u_r \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_\theta}{\partial \theta} + 2u_r \omega + \frac{u_r u_\theta}{r} = - \frac{1}{\rho} \frac{\partial p}{\partial \theta} \tag{B-3}$$

The need for the coupling of the results of these equations to the integral equations will be explained later.

The integral equations result from the integration of the above equations over appropriate volumes which are fixed relative to the earth or to a portion of the rotor. The latter control volume is not in an inertial coordinate frame and the equations must be modified accordingly. Some control volumes appropriate to the wind-wheel turbine will be shown after the equations are given. In the following equations, $\vec{u}$ is the velocity relative to the control volume, $\hat{n}$ is the out outward normal to the surface, S, of the control volume, $\tilde{V}$.

$$\frac{\partial}{\partial t} \int_{\tilde{V}} \rho d\tilde{V} + \int_S \rho u \cdot \hat{n} \, dS = 0 \tag{B-4}$$

Momentum equation in an inertial frame:

$$\vec{F}_S = \int_S \rho\vec{u}(\vec{u} \cdot \hat{n})dS + \frac{\partial}{\partial t}\int_{\hat{V}} \rho\vec{u}d\tilde{V} \tag{B-5}$$

where $\vec{F}_S$ is the surface force acting on the control volume due to pressure and viscous stresses.

Momentum equation in a non-inertial frame (see Figure B-2):

$$\vec{F}_S - \int_V [\ddot{\vec{R}} + 2\vec{\omega} \times \vec{u} + \dot{\vec{\omega}} \times \vec{r} + \vec{\omega} \times (\vec{\omega} \times \vec{r})]\rho d\tilde{V} = \int_S \rho\vec{u}(\vec{u} \cdot \hat{n})dS + \frac{\partial}{\partial t}\int \rho\vec{u}d\tilde{V} \tag{B-6}$$

In this equation u is the velocity in the moving coordinate system relative to the control volume. For the wind=wheel turbine the most useful moving coordinate system is one fixed on the rotor, rotating with it. This equation is valid for a control volume fixed in the moving control volume.

Moment-of-momentum equation in an inertial frame:

$$\vec{M}_S = \int_S (\vec{r} \times \vec{u})\rho\vec{u} \cdot \hat{n}dS + \frac{\partial}{\partial t}\int_V \rho(\vec{r} \times \vec{u})d\tilde{V} \tag{B-7}$$

where $\vec{M}_S$ is the total moment about the origin of the radius vector r of the distribution of surface forces acting upon the surface of the control volume.

This is a vector equation and only one component need be examined to determine the moment about the rotor axis.

An equation similar to Equation B-7 can be written for a non-inertial frame but it will not be given here. However, it could prove useful for some parts of the analysis.

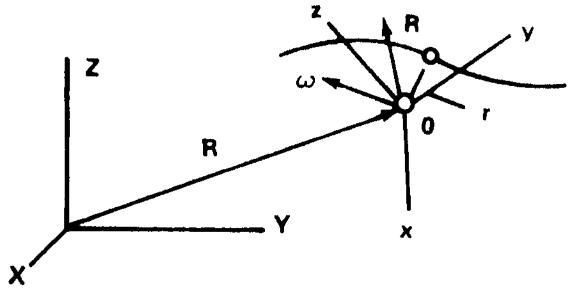

Figure B-2. Rotating coordinate system.

Energy equation, neglecting internal and potential energy:

$$- \frac{dW_S}{dt} + \int_S \vec{T} \cdot \vec{u} dS = \int_S \rho \frac{u^2}{2} (\vec{u} \cdot \vec{n}) dS \qquad \text{(B-8)}$$

where W_S is the rate of shaft work passing from the control volume to the surroundings and $\vec{T}dS$ is the surface force which equals $-\rho\vec{n}dS$ if viscous stresses can be ignored.

The use of these equations to aid in designing several parts of the wind=wheel turbine will now be illustrated. First, consider the forward ducts, Figure B-3. Place a control volume just inside the duct. Then flow crosses the control volume only at areas 1 and 2. The flow through the duct can be analyzed using the integral continuity and momentum equations, Equations B-4 and B-5. Some approximations need to be made relative to (a) the distribution of velocity across areas 1 and 2, (b) the pressure distribution across face 1, and (c) the pressure across face 2. Also, the flow can be taken as steady. For a small duct the assumption of uniform properties across any cross-section of a duct is frequently made and will be used as a first approximation. This can only be assumed as a rough first estimate, however. Also, the

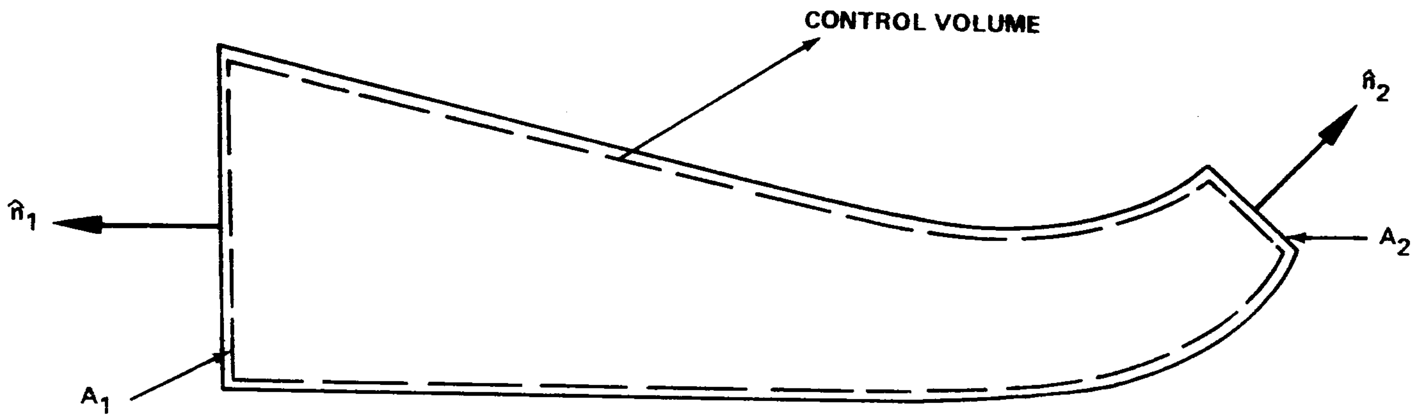

Figure B-3. Control volume.

static pressure approaching area 1 is atmospheric pressure, but in order for the duct to work properly the pressure across area 2 must be less than the atmospheric pressure due to the pumping of the rotor. This pressure is difficult to determine but can be obtained from the differential equations in a simplified form (approximations, Equations B-1 and B-3 applied to the rotor) or experimentally determined pressures can be used. The pressure across area 2 is most certainly a function of the geometry, number of rotor blades, and rotational speed. Therefore, the ducting geometry will have to be optimized for the expected number of blades and operating rotational speed. The side scoops can be analyzed in the same manner.

The result of the duct analysis will be the velocity crossing the duct exit surfaces. These velocities can then be used to examine the flow about the rotor, Figure B-4. The flow about the rotor is always unsteady and therefore difficult to analyze. However, as the number of blades increase the rotor flow can be seen to be approximately steady. If that assumption can be made then the torque can be obtained from the moment-of-momentum equation (Equation B-7) applied to the fixed control volume 1. Before that can be solved the distribution of momentum leaving the rotor must be known. That will depend upon the final rotor geometry decided upon, i.e., upon whether air is allowed to leave through the sides of the rotor or through the shaft, etc. This leaving velocity distribution will have to be determined either from a solution of differential equations or from measurements.

It might also be useful to apply the integral moment-of-momentum equation to rotating control volume 2 which encloses the volume between two rotor blades. The flow in this control volume is unsteady.

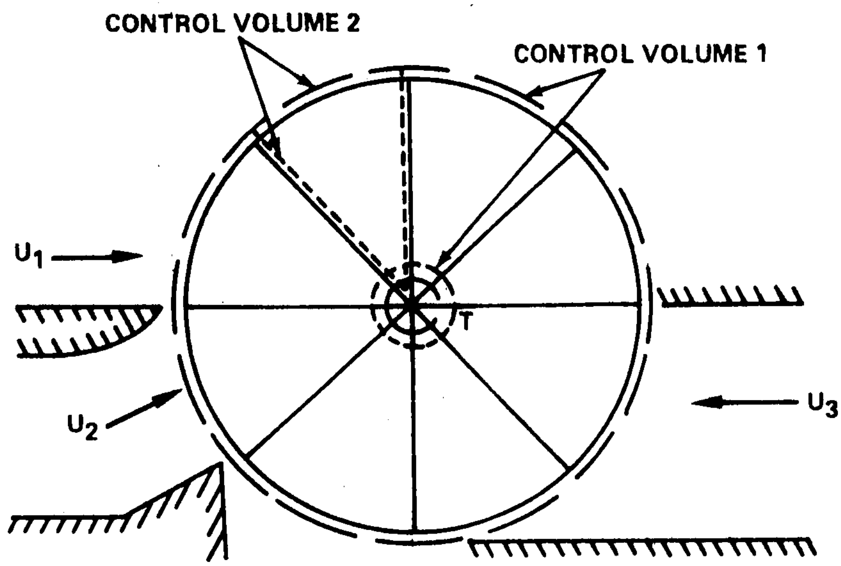

Figure B-4. Control volumes on rotor.

All of the above analysis techniques ignore many of the finer details of the flow which have important influences upon the torque. The first is the energy required to form the vortex system that is shed from each blade. The estimation of this energy loss is difficult and probably will not be attempted for this preliminary work. The second is the viscous drag experienced by the rotating disks that form the sides of the rotor. The viscous torque on a rotating disk in free space is well known. However, in the wind-wheel turbine the disk will be separated from the support walls by only a small distance. Then the standard boundary layer approximation to the differential equations will be used to find the viscous torque. The analysis can be used to modify the disk geometry to lower the viscous torque.

VITA

John Wesley Kaufman was born in Kittanning, Pennsylvania, on June 30, 1927. He attended public school in the Armstrong County (Sherret and Kittanning) and McKean County (Bradford) school systems. In June 1946, he entered the U.S. Signal Corps for a three-year enlistment. In September 1949, he entered Pennsylvania State University in University Park, Pennsylvania, and received a Bachelor of Science Degree in August 1953. He did additional undergraduate and graduate work at The University of Utah, The University of Arizona, New Mexico State University, The University of New Mexico, and The University of Alabama in Huntsville. Since that time, he was an instructor in meteorology for a six-month period at the U.S. Signal Corps School at Fort Monmouth, New Jersey, in 1957-1958. He has nearly 28 years of Government service of which he has been employed by the National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Alabama, since October 1961.

Mr. Kaufman entered the graduate school at The University of Tennessee Space Institute in the fall of 1975 and received the Master of Science Degree with a major in Engineering Science in December 1979. He has been a member of many scientific work groups during his past years in Government service. He has devised and invented a few special devices which are useful in the field of meteorology and in altitude balloon research and technology.

He is married to the former Marita B. Adams of Tularosa, New Mexico. They have four children--Deborah, Sherry, Lori, and Becky. Kamilla, the daughter of Deborah, is the only grandchild.