

1979

# Performance Matching of Hydraulic Energy Converters and Wind Turbines for Heating Purposes

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PF67025F  
UC-60  
UM-WF-TR-79-8

PERFORMANCE MATCHING OF HYDRAULIC  
ENERGY CONVERTERS AND WIND  
TURBINES FOR HEATING PURPOSES

Technical Report

by

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December 1979

Prepared for the United States Department of Energy  
and Rockwell International, Rocky Flats Plant, Golder,  
CO under Contract Number PF67025F.

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## ABSTRACT

An investigation of a fluid brake device designed to convert wind energy to heat is reported for small to moderate size wind turbine applications. Fluid devices of three different geometries with vaned rotors were examined for power capacity and operating range. Converter size and geometry was related to rotational speed through a parametric study of the wind turbine and fluid energy converters. Wind tunnel tests, laboratory experiments, and analytic techniques lead to the development of three candidate hydraulic converters for matching the 10 m diameter Wind Furnace Model Four.

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## LIST OF SYMBOLS

$A$	Rotor swept area
$B$	Damping term
$c$	Weibull scale parameter
$C_p$	Specific heat of working fluid
$C_p$	Power coefficient
$d$	Vane depth
$dT$	Time derivative of fluid temperature
$D$	Energy converter rotor diameter
$D_t$	Wind rotor diameter
$I$	Total system mass moment of inertia
$k$	Weibull shape parameter
$K$	Empirical proportionality constant
$K_s$	Torsional stiffness
$m$	Mass of working fluid
$N$	Speed ratio
$P$	Power
$P_{max}$	Theoretical power in the wind
$r$	Rotor radius
$T$	Torque
$V$	Wind speed
$\bar{V}$	Mean wind speed
$W$	Work
$T$	Fluid temperature change
$\theta$	Angular position

# LIST OF SYMBOLS (Continued)

$\pi$	Power number
$\rho$	Density of fluid
$\rho_t$	Air density at turbine
$\sigma$	Standard deviation of wind speed
$\omega$	Rotational speed of converter
$\omega_t$	Rotational speed of wind rotor
$\nu$	Viscosity
$\lambda$	Tip speed ratio
$\frac{\rho \lambda^3 N^3}{\rho_t C_p}$	System parameter

## INTRODUCTION

Man has always used energy resources with more regard for result than efficiency. The attitude stems partly from ignorance, but is primarily linked to tradition and lack of foresight. Imprudent planning saw the depletion of wood, charcoal, coal and more recently the short supply of natural gas and uranium. Insufficient supply of these natural resources and the subsequent social and economic hardships can only be alleviated by conservation, equitable distribution of conventional fuels and the use of renewable energy resources.

Careful energy use will slow the growing energy demand, but the increasing population of the industrialized world requires new energy sources. Concerned planners will develop renewable energy resources that will satisfy world needs for all future generations.

Renewable energy resources must be used judiciously. Traditional energy use shows little regard for environmental quality. Fossil fuels pollute air with waste products of combustion. All energy processes which convert matter into energy including combustion, fission, and fusion reject heat to the environment causing thermal pollution. Also, serious doubts concerning the long time safety of the nuclear energy process and radioactive waste products make carefully energy planning imperative for our generation and generations to come. Proper use of renewable energy means that environmental dangers like those associated with traditional energy resources can be avoided.

Solar energy is the most prevalent renewable energy resource. Solar energy is available everywhere on the globe in periodic quantities depending upon location, weather, day, and hour. An estimated  $1.05 \times 10^{18}$  Kwh of solar radiation<sup>1</sup> reaches the Earth's surface each year. Much of the energy is

reflected, but the remainder causes ocean currents, atmospheric circulation<sup>2</sup>, and heating.

The solar influx is a nearly inexhaustible resource. Any portion of the resource directed towards World energy demand can relieve environmental and economic burdens of conventional energy sources.

Exploitation of the solar resource is in its infancy. Today hydroelectric power is the only solar derived energy source used on a large scale providing approximately 4% of the United States energy supply<sup>3</sup>. Direct solar insolation provides energy and both passive solar architecture and greenhouses employ solar energy. Ocean currents and ocean thermal differences remain essentially untapped energy resources only to be exploited through large scale economic and material commitments. Wind energy presents an excellent opportunity for immediate energy development and great strides towards realizing the wind energy potential are now underway. In fact wind energy is easily accessible at any scale and it can provide renewable energy with little continuing environmental impact<sup>4</sup>.

The wind energy resource is dispersed, but in many regions the availability is good. The World Meteorological Organization estimates  $2 \times 10^7$  Mw wind power potential is available at suitable sites<sup>5</sup>. With appropriate technology exploitation of this energy resource can alleviate dependence on traditional fuels. Careful use of wind energy may diminish the waste and inefficiency associated with traditional fuels and build a sound energy structure fueled by solar insolation and the rotation of the earth.

The penetration of wind machines into the energy market will depend largely on the economics of energy conversion. Because renewable energy sources have

small continuing costs the initial costs, productivity, and long term fuel savings<sup>6</sup> will determine the economics of wind power. Practical energy conversion systems will convert significant quantities of energy to a useful task and perform simply and efficiently at low cost.

Wind energy promises to provide substantial fuel savings and environmental benefits. The choice of a simple wind machine of low life cycle cost and high productivity is very attractive. One such machine is the Wind Furnace Model Four.

The Wind Furnace Model Four converts wind energy to heat. The device employs a hydraulic energy converter driven by a wind rotor to heat a fluid by viscous friction. Heat from the machine can be used for space heating, hot water, agricultural, industrial, and commercial heating. The Wind Furnace Model 4 is a very simple, low cost, high productivity machine that converts energy to satisfy a common need for heat.

This paper will discuss the Wind Furnace Model 4 concept. Performance matching of wind turbines and hydraulic devices to convert wind energy to heat will be outlined together with important system design parameters. Development of the Wind Furnace Model 4 is directed towards identifying an alternative means of providing heat with no adverse environmental impact.

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# CHAPTER 1

## ENERGY DEMAND AND WIND ENERGY RESOURCE

The wind energy resource and the task for which the energy is harnessed will define the wind turbine design. Productivity of the wind energy conversion system depends upon the wind characteristics of speed, frequency, and duration. Size of the wind rotor and other components depends upon the magnitude and regularity of the energy demand. Often times short term energy storage can bridge the variable availability and the demand.

The available power in the wind is given by the following equation developed from momentum theory.

$$P_{\max} = \frac{1}{2} \rho_t A V^3$$

where  $P_{\max}$  is instantaneous power in the wind,

$\rho_t$  is the density of air, and

$V$  is the wind velocity.

Knowledge of the wind regime will allow an estimate of the yearly wind energy resource. Wind speed data from the National Climatic Center, meteorological sources, and other agencies may provide adequate information. Often wind speed data reduced to statistical distributions simplifies the wind resource estimate. Here the estimate becomes a calculator operation.

Perhaps the most straight forward approach to estimating the energy resource involves published wind data and an assumed Weibull probability distribution<sup>1</sup>. In regions where wind velocity is reported by mean and standard deviation, the conversion to Weibull probability distribution can be made by the means described by Sexton<sup>2</sup> and Justus<sup>3</sup>.



Data reduced on a monthly basis by meteorological organizations is preferred for the wind energy estimate. Published Weibull shape parameters are employed to determine the duration of windspeed during the month. The power in the wind is calculated for a unit area by means of Equation 1.1. The product of the power available at one wind speed interval and its respective duration in hours yields the monthly productivity for that interval. Summation of energy productivity for the whole spectrum of windspeeds results in the month's wind energy resource. Repeating the calculations for each month produces an estimate of the annual wind energy resource as shown in Fig. 1.1

Month to month variation of the wind energy resource is of great importance. Practical wind energy conversion systems direct all energy to perform a task. The Wind Furnace Model Four, for instance, can take advantage of the seasonal winds to provide space and hot water heating during cold and windy months in temperate climates. Fig. 1.2 shows the similarity of extracted wind energy and heating demand. Note that the system provides only a portion of winter heating and a surplus of heat for the remaining months. Here, the system is optimized to provide a large portion of the heating demand for a minimal capital investment.

Wind turbine performance and operating characteristics determine the quantity of energy extracted from the wind. Both system efficiency and operating range affect the fraction of wind energy converted to a useful task.

## CHAPTER 1

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Fig. 1.1 WIND ENERGY IN IOWA

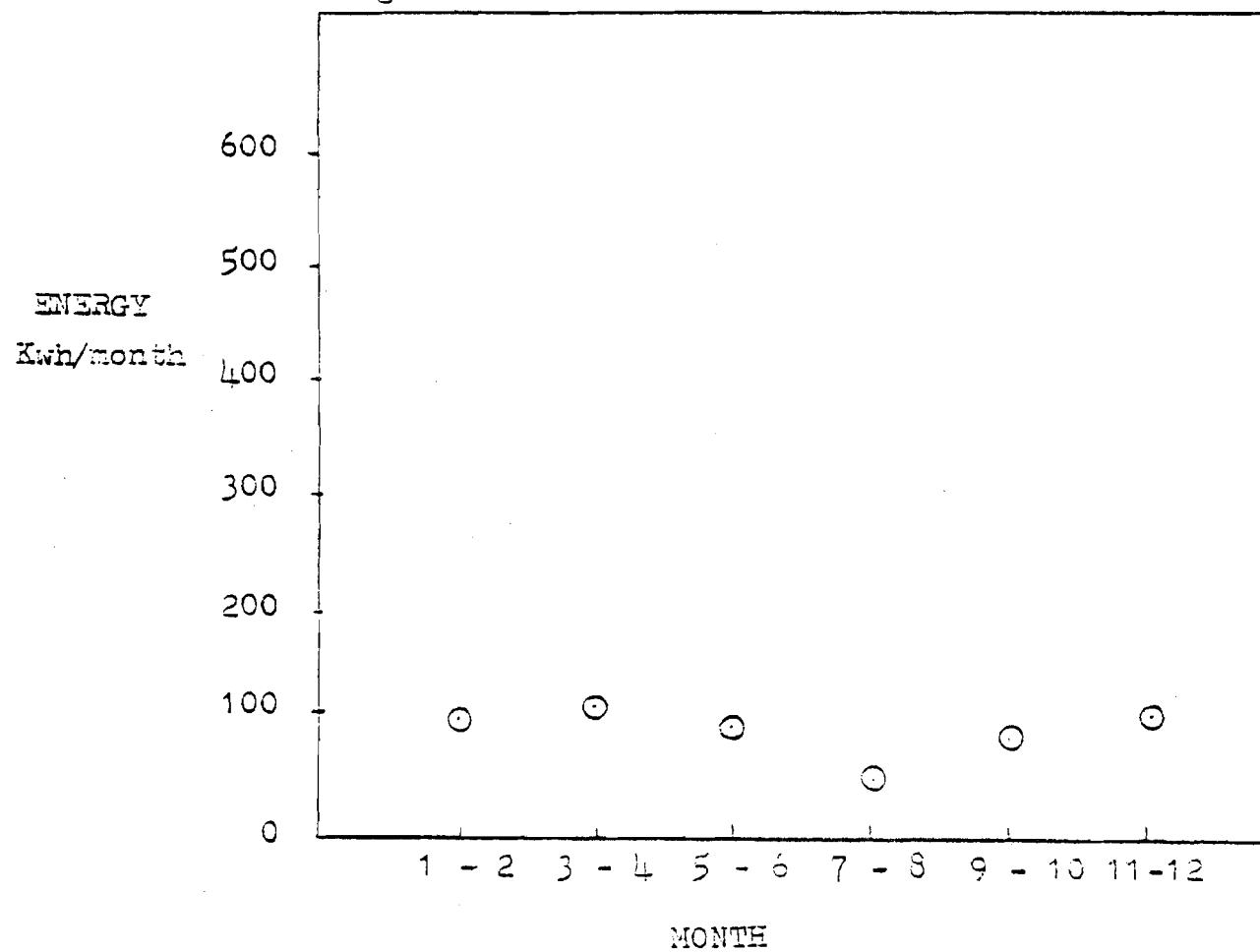
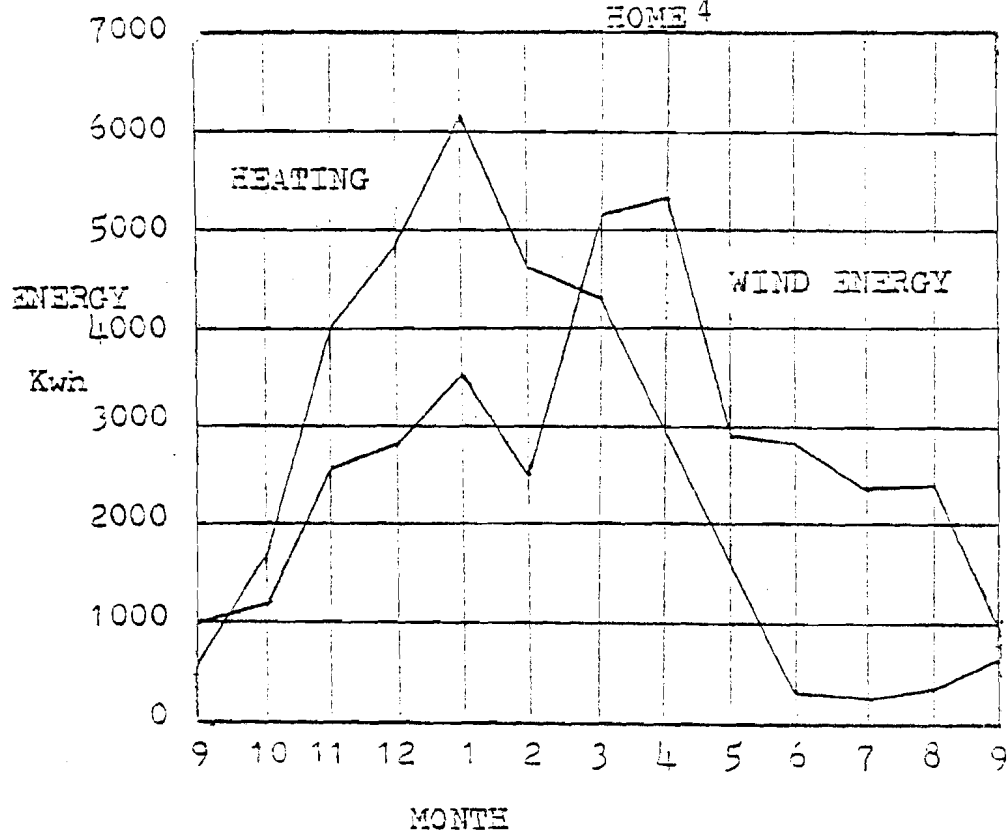


Fig. 1.2 ANALYTICAL COMPARISON OF WIND ENERGY  
AND HEAT DEMAND FOR AN AVERAGE NEW ENGLAND  
HOME <sup>4</sup>





## CHAPTER 2

### WIND TURBINE PERFORMANCE

The wind energy resource introduced in Chapter 1 must be modified to describe wind turbine performance and productivity. A general understanding of wind machine performance and operation can lead to a good estimate of wind turbine productivity. Also, careful design can optimize wind turbine costs without sacrifice of annual productivity.

Several mechanical characteristics affect the productivity of the wind energy conversion system. Rotor geometry, rotational speed, cut - in speed, cut - out speed, and time response of the wind turbine can all be optimized to produce an efficient energy conversion system.

Equation 1.1 shows that the power in the wind varies directly with the swept area of the wind turbine. Larger wind rotors in general extract more energy than smaller wind rotors, but efficiency of the device must also be considered in an estimate of machine productivity.

Performance of several different wind rotors is shown in Fig. 2.1. The power coefficient represents the fraction of power extracted from the wind to the power available in the wind as shown by the following equation.

$$C_p = \frac{P}{.5 \rho A V^3}$$

where P is the power of the wind turbine rotor, and  
 $C_p$  is the power coefficient.

Rotational speed of the wind turbine is represented by the tip - speed ratio: a ratio of the speed of the rotor blade tip to the free stream wind speed. The tip - speed ratio is given by the following

equation.

$$\lambda = \frac{R \omega}{V}$$

where,  $\lambda$  is the tip - speed ratio,  
 $R$  is the rotor radius, and  
 $\omega$  is the rotational speed.

High efficiency wind turbines maintain a constant tip - speed ratio for all wind speeds within the operating range. Horizontal axis propeller type wind turbines are capable of higher efficiency than most other wind turbines.

Tip speed tolerance or off - design performance is very important because wind turbines extract energy from the variable wind and seldom operate at the designed tip - speed ratio. The American Fan Mill wind turbine, for instance, performs with .28 power coefficient at .6 tip - speed ratio, but change the tip - speed ratio to 1.0 and the device produces no usable power. Broad and gentle curves in Fig. 2.1 are indicative of tip speed tolerant rotors: rotors that perform with high efficiency over a range of tip speed ratios.

Wind turbines generally begin rotating at low wind speed, operate near constant tip - speed ratio in moderate winds, and shut down in high winds. The lowest wind speed where power is extracted is termed the cut - in velocity. Cut -out velocity is the wind speed at which the machine is shut down.

Properly designed wind turbines do not sacrifice a significant portion of the available wind energy. The annual energy available below cut - in and above cut - out wind speed should be insignificant. However, in reference to estimating the productivity of wind energy conversion systems, the procedure must be altered to include the

affect of cut - in and cut - out velocity.

The revised procedure is tabulated in Table 2.1 for a hypothetical situation involving one month's data. The power is calculated assuming a constant power coefficient. Power duration is calculated by means of the Weibull cumulative probability distribution. Only wind speeds within the operating region are included in this estimate. Also, calculations are performed on the basis of a unit area. The swept area of the device can be sized to provide a specified portion of the energy demand according to monthly trends.

Performance of wind energy conversion systems depend largely upon the speed regulation provided by the energy converter. To maintain a high power coefficient the machine tip - speed ratio should be invariable in the operating region. The perfect energy converter should control the rotor speed by providing proper resisting torque for all wind speeds. The power of the wind turbine varies with cube of the wind speed according to Eq. 1.1.

Constant tip - speed ratio implies that the wind turbine power and speed should vary according to the cubic relationship shown below.

$$P = K \omega^3$$

where  $K$  is a proportionality constant, and  
 $\omega$  is the rotational speed.

An empirical performance curve of a high efficiency,  $C_p = .47$ , wind turbine is shown in Fig. 2.2 with the cubic power relationship.

Mechanical and electrical control devices are often used to regulate wind turbine rotational speed. Mass produced wind machines typically



sacrifice efficiency for design simplicity causing the power curves to deviate sharply<sup>1</sup> from the cubic speed - power relationship. These machines fail to extract from the wind all the energy that is available to the wind rotor.

Careful rotor design, including aerodynamic performance evaluation of the rotor blades, can provide a rotor with high tip - speed tolerance, but at a sacrifice in efficiency. Perhaps the most attractive wind energy conversion design incorporates a high efficiency wind rotor and energy converter with a cubic speed - power curve.

Fluid energy converters like centrifugal pumps and turbines follow the cubic speed - power relationship. As early as 1973 Heronemus suggested the use of a fluid device as a self regulating energy converter for wind turbines. Gunkel and Fury<sup>2</sup> and Esbensen and Strabo<sup>3</sup> demonstrated that a fluid churn device much like an imersed paddle wheel could be directly coupled to wind turbines so that wind energy could be converted to heat. However, the two demonstrations employed small 2 Kw wind rotors and crude energy conversion devices. The energy converter designs are large and bulky and appear unsuitable for matching with moderate and large scale wind turbines.

## CHAPTER 2

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TABLE 2. 1 CALCULATION OF WIND ENERGY CONVERSION  
BIRMINGHAM, ALABAMA

Wind Characteristics at 19.8 m Elevation

Mean Velocity	$\sigma$	$\sigma/\bar{V}$	k	c
3.71 ms <sup>-1</sup>	2.25	.605	9.15	1.71

Wind Turbine Characteristics

V Cut-in	V Cut-out	Elevation	Swept Area	Cp
2.0 ms <sup>-1</sup>	13.0 ms <sup>-1</sup>	19.8 m	1.0 m <sup>2</sup>	.40

VELOCITY ms <sup>-1</sup>	POWER kW	Hr. EXCEEDED	HOURS	Cp	ENERGY kWh/m <sup>2</sup>
1.0 - 2.0-	4.8	516.00	2.04	.4	0.00
2.0 - 3.0	16.2	400.00	116	.4	.75
3.0-- 4.0	38.4	275.00	125	.4	1.92
4.0 - 5.0	75.0	167.00	108	.4	3.24
5.0 - 6.0	129.0	83.30	83.7	.4	4.32
6.0 - 7.0	206.0	33.30	50.0	.4	4.12
7.0 - 8.0	307.0	25.00	8.30	.4	1.02
8.0 - 9.0	437.0	16.70	8.30	.4	1.45
9.0- 10.0	600.0	8.33	8.37	.4	2.01
10.0- 11.0	799.0	4.17	4.16	.4	1.33
11.0- 12.0	1040.0	2.08	2.09	.4	.87
12.0- 13.0	1320.0	1.00	1.08	.4	.57

TOTAL 21.6 kWh/m<sup>2</sup>

Per Month

Fig. 2.1 SUMMARY OF POWER COEFFICIENT  
AND TIP SPEED RATIO

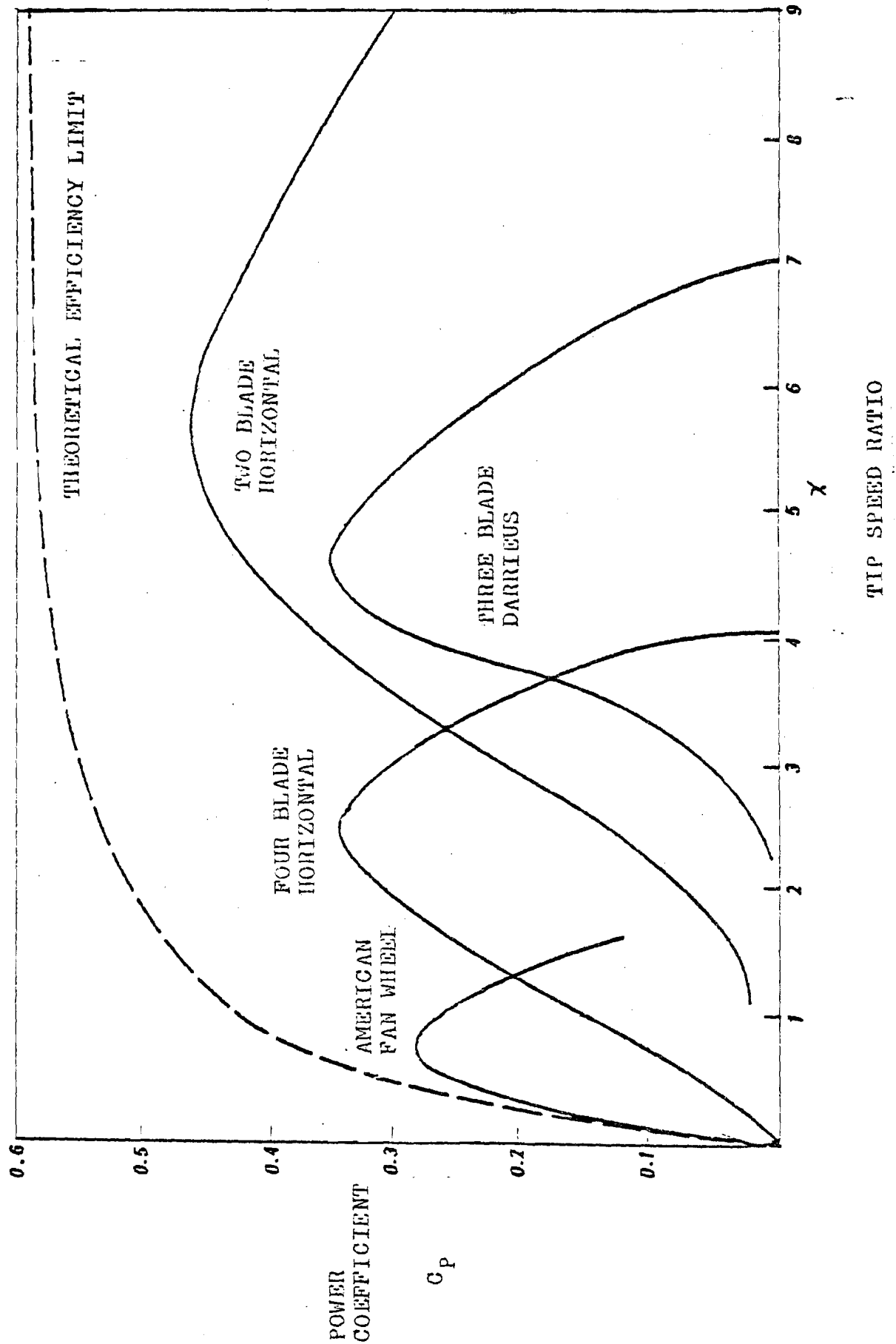
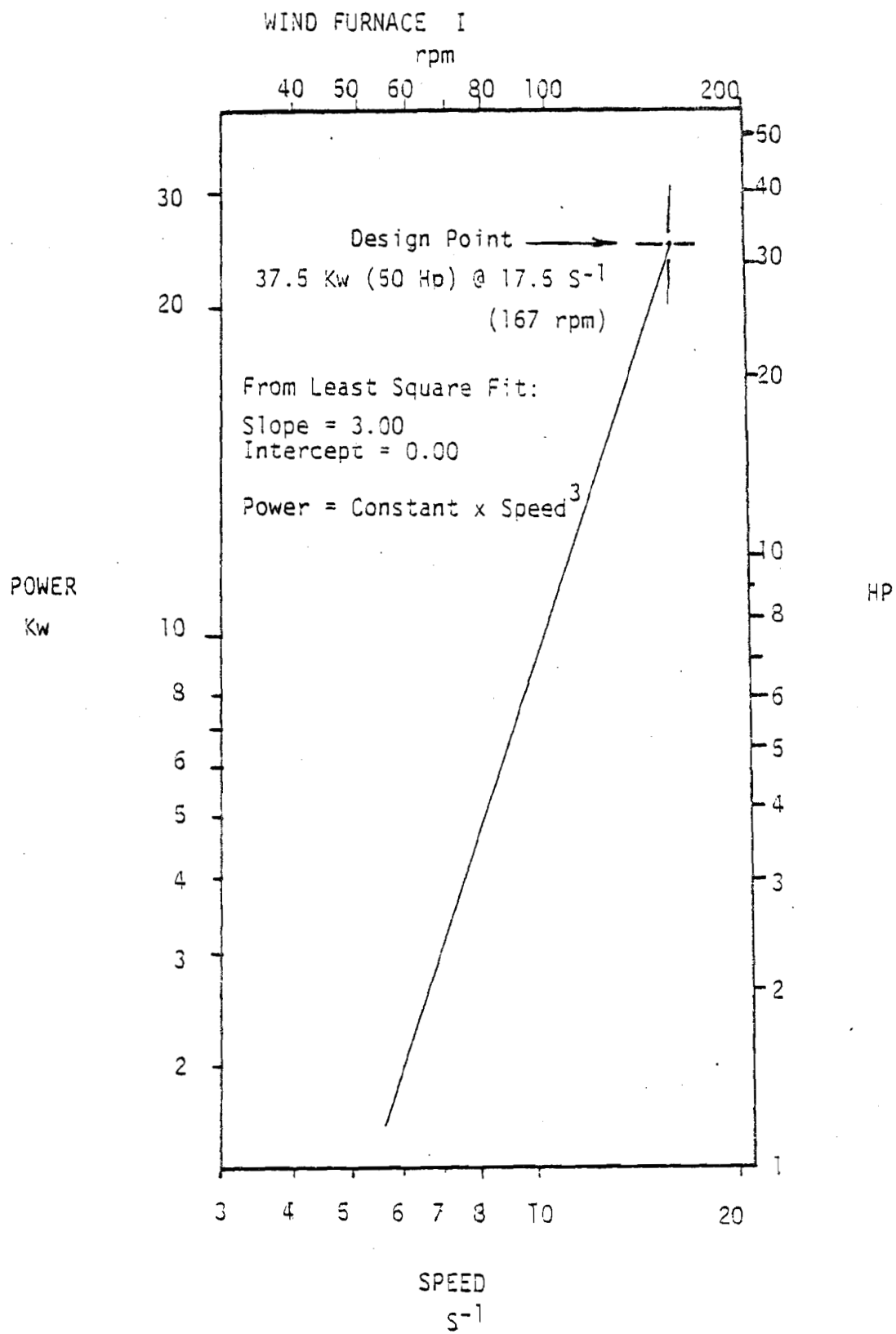


Fig. 2.2 RELATIONSHIP OF SPEED AND POWER FOR



## CHAPTER 3

### HYDRAULIC ENERGY CONVERTERS

Hydraulic energy converters traditionally turn mechanical energy to waste heat. Rotating machines known as water brakes, water churns, and fluid dynamometers are most often used to load or arrest prime movers and reject heat to the environment by means of a fluid circuit. Because the devices are largely employed in test equipment they tend to be very costly. They are also built to perform at high speeds common to most prime movers.

Other applications of hydraulic energy absorbers employ the device as a speed dependent brake. Cranes, trucks, oil rigs, glider tethers, and emergency airplane arresters utilize the device as a brake for intermittent service. The braking torque provided by the hydraulic energy absorber is self-governing and does not fade or change with extended use.

Rudimentary theory of hydraulic energy absorbers was documented by Froude<sup>1</sup> in 1877. Froude had pioneered ship theory and designed ship hulls for reduced drag. In an effort to measure the power delivered to the screws of large ships Froude designed a hydraulic energy absorber to be placed on the ship's propeller shaft while the ship was docked. The ship's engines could then be fired and the propeller shaft turned against the resisting torque of the energy absorber. Measurement of speed and torque for varying engine speeds could then be made while the ship remained at dock. From the gathered information future vessels would have a better match between ship drag and propulsion creating faster ships with greater cruising range.

Froude's experiments identified a novel braking device which others have improved. The overall power absorption for a given energy absorber increases with the complexity of the device's internal geometry. Figure 3.1 shows a simple hydraulic dynamometer and Figure 3.2 shows the cross-section of several hydraulic energy absorbers. The internal geometry is identified by the shape of the vanes. Note the energy absorber consists of a central rotating vaned impeller or rotor and fixed vane stator housing.

The cavity between rotor and stator is filled with a fluid that provides resistance to shaft rotation. The fluid motion is restrained by the vanes in such a way as to provide retarding torque by means of viscous shear, shock, and momentum transfer. The magnitude of retarding torque is directly related to the fluid motion within the energy absorber. Rounded internal geometry in the circular, elliptical, and Froude design enhances the fluid velocity and the mechanism for energy absorption.

Vanes in the elliptical and Froude dynamometers are inclined approximately  $45^\circ$  to the plane of rotation to take advantage of secondary flow. Figure 3.3 illustrates the inclined vanes of an elliptic geometry dynamometer. The complex geometry make these devices provide high torque at low rotational speeds. Machines of this type can be more compact than a rectangular or circular geometry device of the same speed and capacity.

### 3.1 Historical Examination

Froude identified the internal energy dissipating mechanism of his device in 1877. Froude's work was largely qualitative but still remains the rudiment of vaned dynamometer design. In fact, to date perhaps only one machine performs better than the device described by Froude.

The cavity between rotor and stator is in the shape of a torus. The torus is divided in plane at the center as shown in section in Figure 3.4 and half oval vanes are fixed inclined to the plane of rotation within the half torus troughs. The inclined vanes form pockets in both the rotor and stator as shown in Figure 3.4 and circulation of fluid within the pockets provides the resisting torque.

Froude chose to design two torus back to back counter balance the axial thrust created by the exchange of fluid from rotor to stator during operation. For this reason most dynamometers have vanes on both sides of the rotor and stator housing to provide two toroidal cavities.

When the cavity is filled and the rotor set in motion toroidal flow develops with rotational and radial directional components. Fluid on the rotor is accelerated radially outward within the rotor pocket and then directed to the stator pocket where the momentum of the fluid is exchanged and the velocity of the fluid reversed as the fluid flows inward and back to the rotor. Of course the fluid flow is continuous and the momentum exchange takes place continually across the dividing plane between the rotor and stator.

Froude found that he could interrupt the flow from stator to rotor by means of shutters and reduce the power absorption of the device. Reduction to seven percent of the maximum was reported. This phenomena shows the extent to which the toroidal flow contributes to the power absorption. Froude suggested the flow from rotor to stator was similar to fluid jets: one jet of axial and circumferencial velocity impinging upon the outer portion of the fixed stator, and one jet of axial velocity impinging upon the moving rotor. Each jet augments the flow which is slowed only by



viscous friction and eddy losses. Figure 3.5 shows a diagram of Froude's concept.

The pathline of a fluid particle in the Froude device is a toroidal helix like a helical spring wrapped in a circle with both ends connected. The fluid passes from one pocket to the next exchanging momentum from rotor to stator as it circulates and dissipates energy by friction and shock.

Froude further identified the fluid movement between rotor and stator as a vortex. The fluid velocity in Figure 3.5 could be imagined as many concentric stream tubes where the velocity varies from zero at the vortex center to a finite velocity near the outer edge of the vortex. The vortex speed bears a direct relation to the rotational speed of the turbine and the resistive torque varies as the time change of momentum with the square of the rotational speed.

From detailed analysis of the fluid kinematics Froude deduced the cubic speed-power relationship of the dynamometer device. He also proved a hypothesis that the power absorbed by the dynamometer is related to the fifth power of the rotor diameter. He attempted with some success to determine analytically the speed-power characteristics of his device. Unfortunately the mathematics and fluid dynamics of the time did not permit a solution to the problem of defining shock fronts in fluid flow. With some simplifications, dubious assumptions and graphical techniques the energy absorption of the device could be estimated; however, Froude cautioned that "careful experiment is still needed."

Professor Osborne Reynolds modified Froude's design and placed vent holes at the center of each vane<sup>2</sup> as shown in Figure 3.3. The vents assured that the pressure at the center of the fluid vortices would remain constant

insuring no change in the working fluid over a wide range of operating speeds. Reynolds' modification increased the operating range and stability of the device. Also cavitation may be reduced by this means because venting increases the pressure at the vortex center where without venting the pressure could decrease below vapor pressure. Reynolds' device has slightly higher power absorption than the Froude device.

E.P. Culver reported on several hydraulic dynamometers of the rectangular geometry in 1937. He experimented with 23 modifications of the device. Vane dimensions, number, angle, and side clearance effects were investigated by Culver who concluded the rectangular device, in Figure 3.2, was best for engine testing. Culver's device is simple and easy to fabricate, but it does not have the low speed-torque characteristics found in the more complex devices.

Culver's data can provide insight into the rudiments of the rectangular design, but the investigation is not thorough. Some contradictions exist and some tests are not well documented. No statistical evaluation is provided and effects of vane side clearance, termed negligible, can actually alter the performance.

Culver shows the cubic speed power relationship of the device and the fifth power relationship of power to diameter. He attempts to identify a linear relationship of several parameters that determine the power absorption. His final equation shown below can be used only for a first approximation.

$$H = K \rho^{.92} \gamma^{.08} N^{2.92} D^{4.4}$$

where H is horsepower

K is a proportionality constant

$\gamma$  is the fluid viscosity

$N$  is rotational speed, rpm, and

$D$  is diameter of rotor.

### 3.2 Limitations of Hydraulic Energy Converters

Time response and cavitation limit the applications of vaned hydraulic energy converters. The devices provide torque by momentum exchange and fluid friction: both velocity dependent mechanisms. Should the rotational speed change suddenly the internal flow changes gradually until the moment of momentum within the device obtains equilibrium. High speeds can lead to cavitation and departure from the cubic speed-power relationship because the overall fluid density decreases.

CHAPTER 3  
LIST OF REFERENCES

1. William Froude; "On A New Dynamometer For Measuring The Power Delivered To The Screws Of Large Ships." Proceedings. Institution Of Mechanical Engineers. Vol 72, July 1877 pp. 237 - 252.
2. A. H. Gibson; Hydraulics And Its Applications. Van Nostrand Company Inc. New York 1934, 4th Ed. pp.781 - 786.
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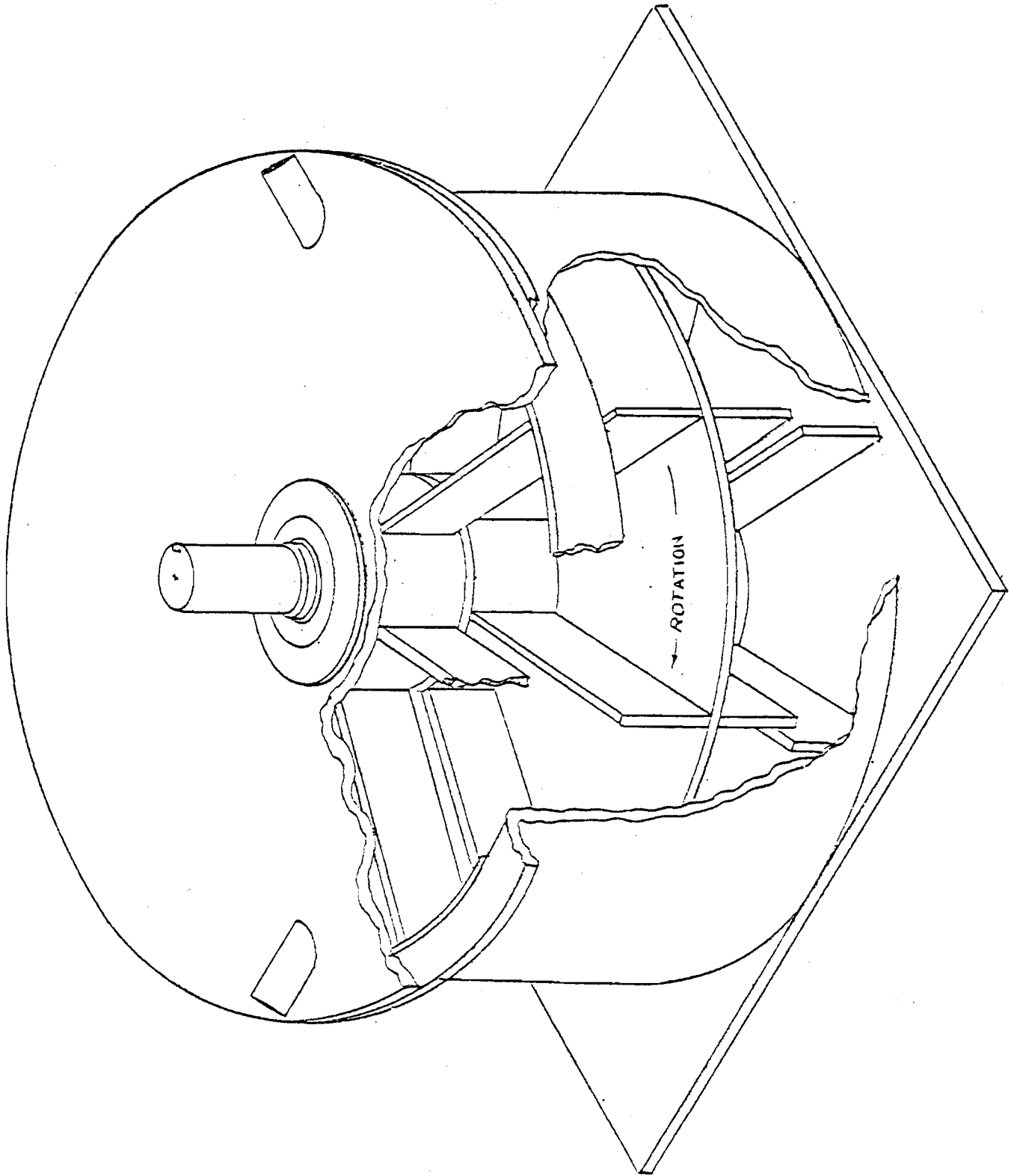
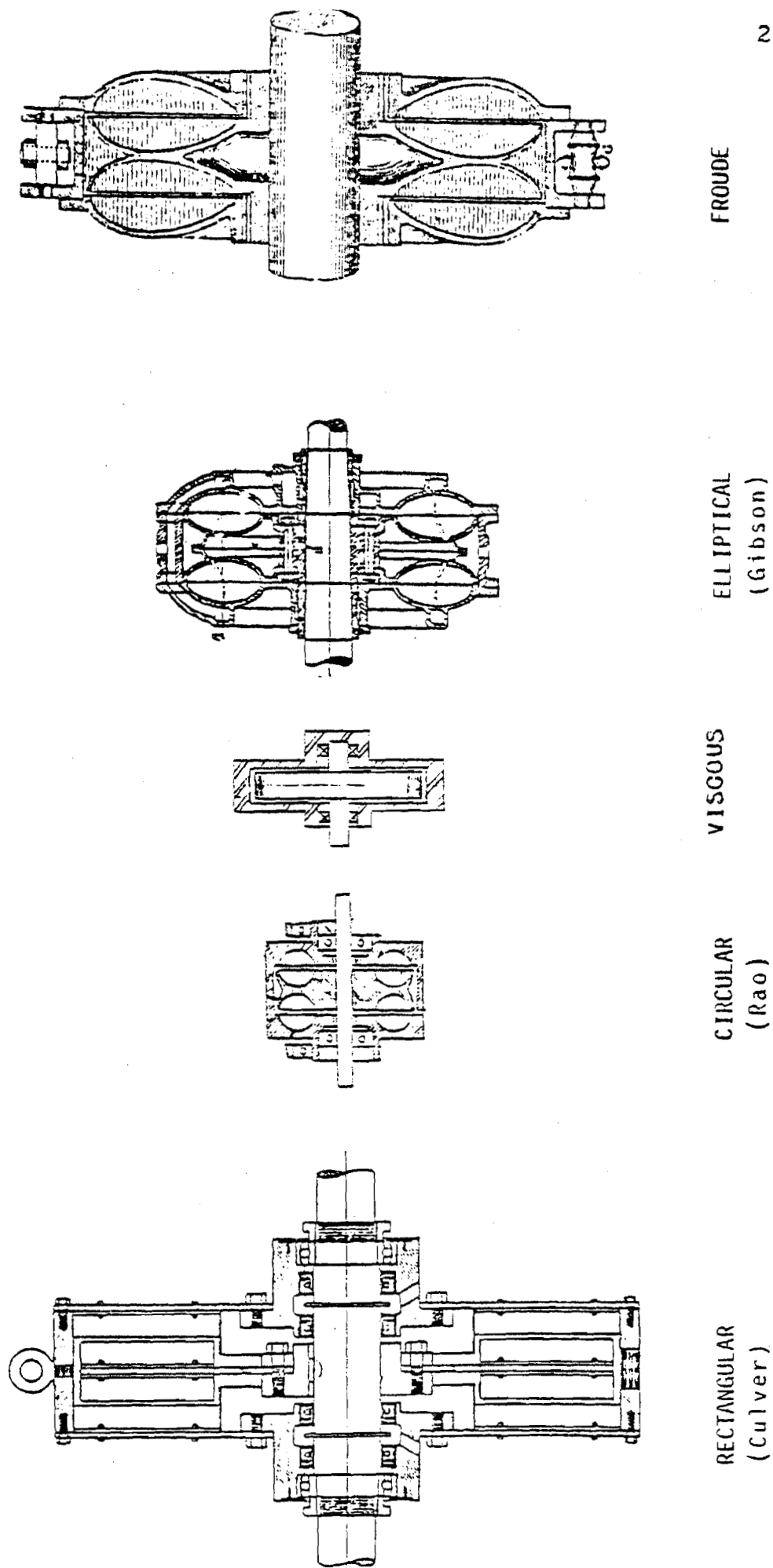


Fig. 3.1 WATER TWISTER® 12

Fig. 3.2 GEOMETRY OF SEVERAL HYDRAULIC DYNAMOMETERS



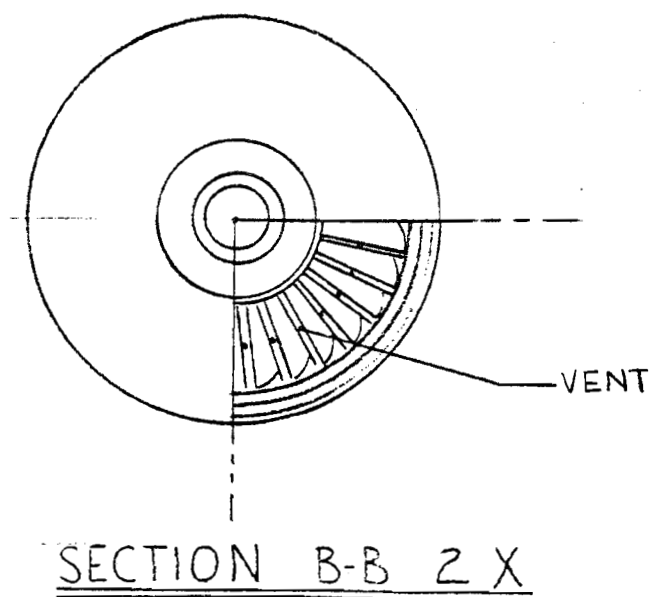
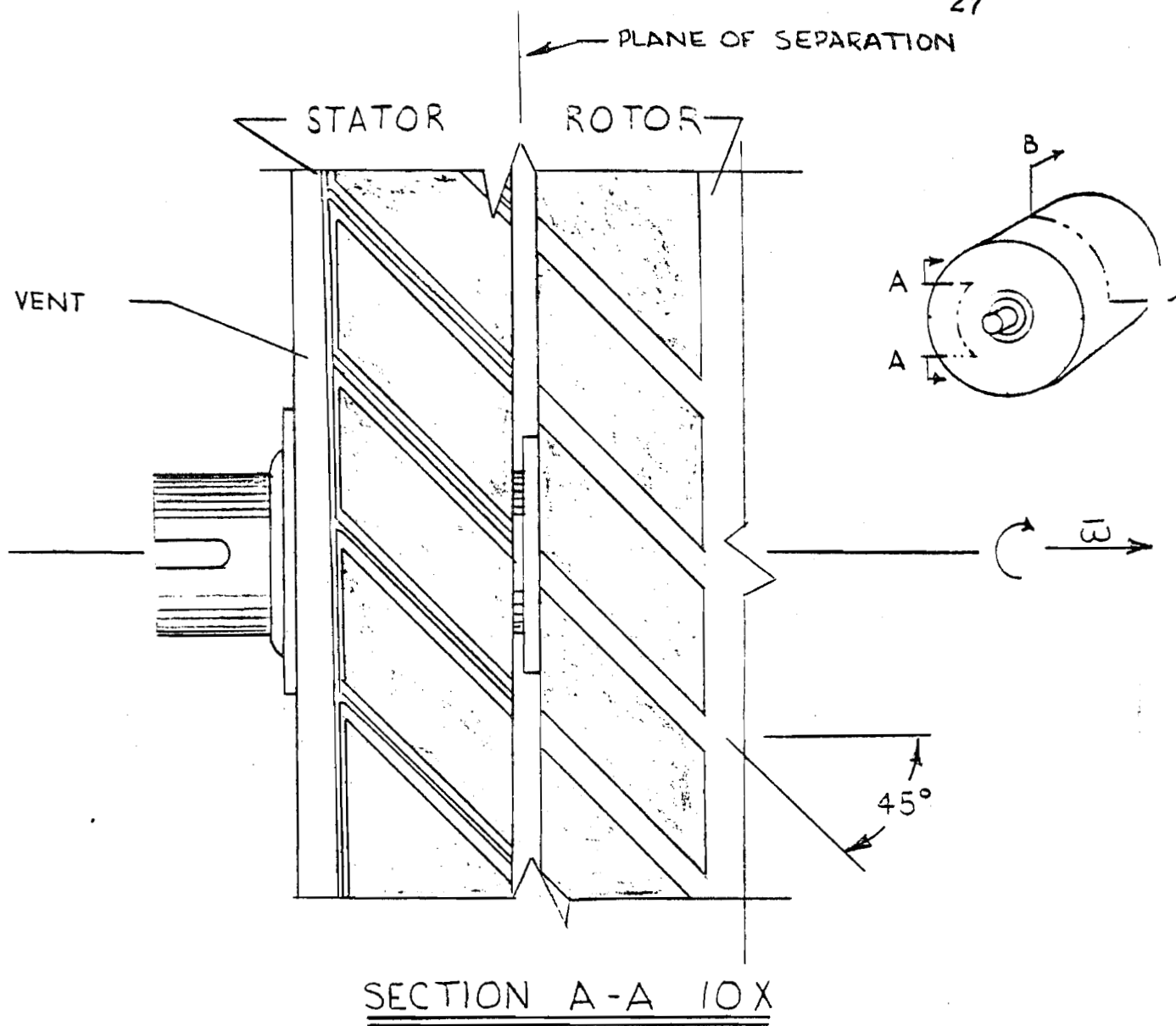


FIG. 3.3 VANE DETAIL OF REYNOLDS DYNAMOMETER

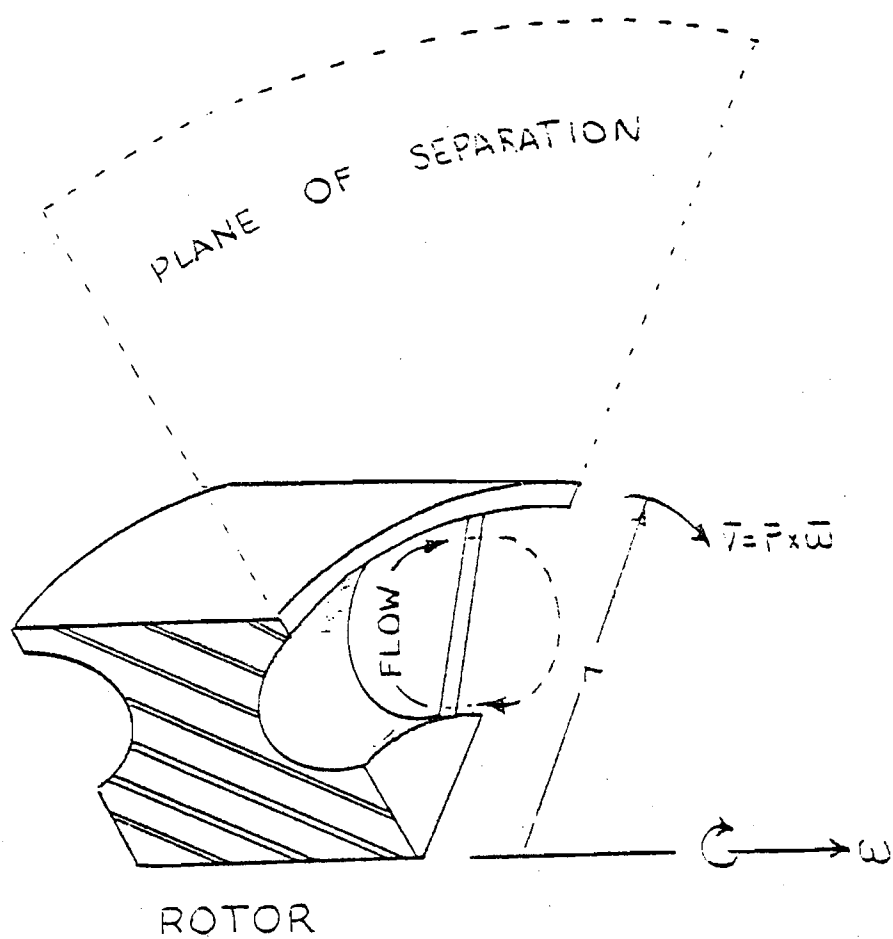


Fig. 3.4 TORUS DETAIL



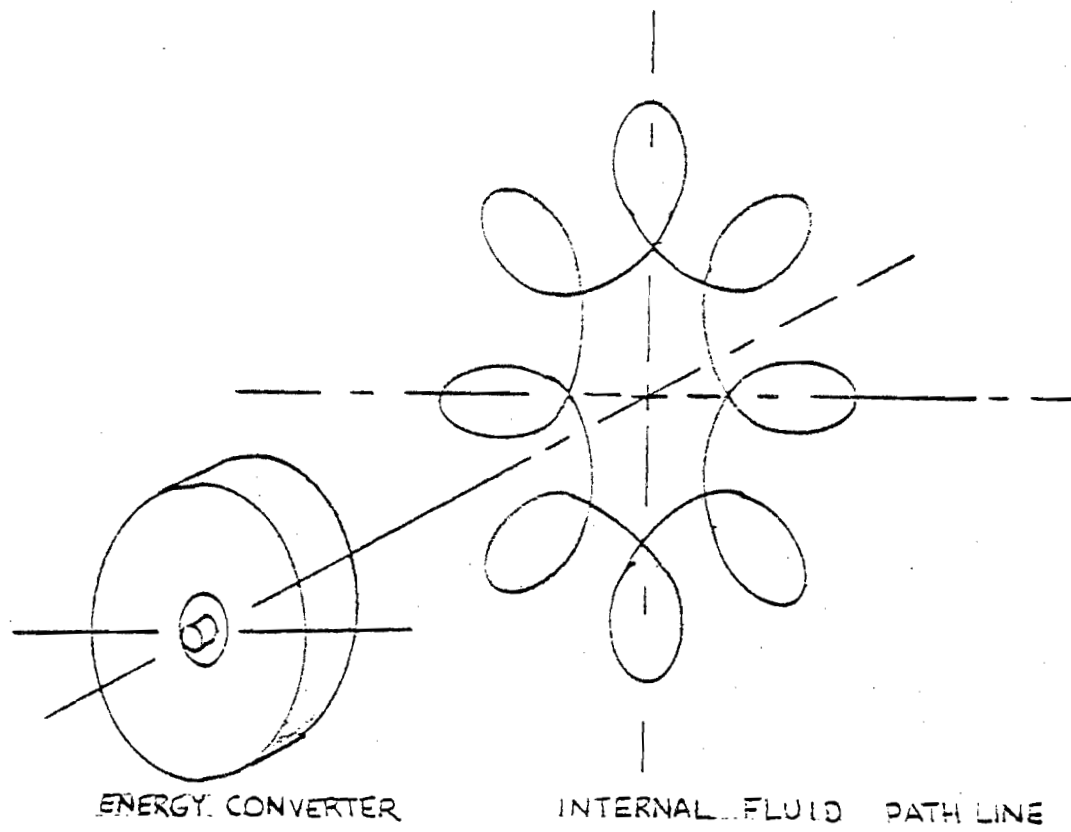


FIG. 3.5 STREAM LINE FLOW

## C H A P T E R 4

### ANALYTICAL PERFORMANCE EVALUATION OF HYDRAULIC ENERGY CONVERTERS

Hydraulic energy converters can be described in terms of size, geometry, and capacity. These three characteristics are interrelated and serve as the basis for comparison. Dimensional analysis simplifies the comparison by relating different size devices. Fluid kinematics and empirical data help to test the relationship of the three characteristics and explain the analytic results. Results of the performance estimate are quite good; however, limitations due to numerical precision, effects of scale, and changing fluid properties must be considered.

Froude<sup>1</sup> described the cubic speed - power relationship and the fifth power relationship of diameter and power for hydraulic dynamometers. Clearly this relationship is valuable in describing the interdependence of size, speed, and capacity.

In the comparison of energy converter performance, the size of the device is the rotor size. Speed is the rotor rotational speed and capacity is the power absorption capability. The comparison requires a consistent set of units.

#### 4.1 DIMENSIONAL ANALYSIS

Dimensional analysis methods lead to a non-dimensional formulation that relates several independent variables. The method identifies the relationship of variables and serves as an engineering model. Dynamic similarity of fluid flow is the underlying principle in dimensional analysis and deviation of machine performance from the analytic performance estimate is largely due to a difference in fluid kinematics. Off design performance is detailed later in this

chapter, but first, the basic parametric relationship is presented.

For the hydraulic energy converter a power coefficient or power number<sup>2</sup>,  $\pi_1$ , can be derived that relates the machine parameters.

$$\pi_1 = \frac{P}{\rho \omega^3 D^5} \quad (4.1)$$

where  $\pi_1$  is the power number,

$P$  is the shaft power,

$\rho$  is the fluid density,

$\omega$  is the rotational speed, and

$D$  is the rotor diameter.

Test results from Froude<sup>3</sup>, Culver<sup>4</sup>, and this author affirm the validity of this model. The power number is near constant for all operating speeds and relates size, speed, geometry, and fluid density to the power capacity of the hydraulic energy absorber. All devices of similar internal geometry, working fluid, and relative internal surface roughness will have similar power numbers independent of size and speed.

Figure 4.1 shows power numbers for several devices of different geometry. High power numbers indicate high energy conversion capacity. The power number is also indicative of the speed and diameter of the device. High power number devices are compact and operate at slow rotational speed.

Froude and Reynolds type dynamometers have higher power numbers than the Culver type device and the viscous disk dynamometer. Figure 4.1 and Table 4.1 are helpful in the choice of a hydraulic energy converter. Also, manipulation of Eq. 4.1 and insertion of speed and fluid data can provide the designer with several choices of machine

size and geometry to fulfill a particular application. However, some of the devices may prove unacceptable due to speed limitations of the device.

#### 4.2 LIMITATIONS OF THE DEVICE

Figure 4.2 shows a typical power performance curve for the hydraulic energy converter. Note the cubic profile where the power number describes the machine performance. This is the normal operating range of the device. Above the operating range the performance deviates from the cubic curve due to a change in fluid properties. Hydraulic energy converters operated above the operating region experience high wear due to cavitation. Carefull design and the proper fluid choice will insure that the device remains in the operating range for all anticipated speeds. Determination of the fall off point shown on Fig. 4.2 can be obtained from empirical data or by calculating the speed where the vortex pressure approaches the fluid vapor pressure.

### 4.3 OPERATING RANGE AND WORKING FLUID

Consistent performance of the hydraulic energy converter depends upon the working fluid properties. Fluid density and viscosity normally vary with fluid temperature. Water density at atmospheric pressure changes 22% between room temperature and the boiling point.

The hydraulic energy converter is a momentum exchange device so fluid density should be high. Also the absolute fluid viscosity should be low so as not to impede fluid flow. Consequently the kinematic viscosity, a ratio of the absolute viscosity to the density, should be low. Change of the kinematic viscosity and the density should be low to insure uniformity in the machine performance. Figure 4.3 shows the water and ethyl alcohol possess these desirable characteristics.

The temperature and pressure must also be considered for the hydraulic energy converter. The fall off point is largely dependent on the fluid vapor pressure. Pressurized devices or fluids of high vapor pressure can achieve high temperatures. Often the operating range can be extended by change of fluid or increase in fluid pressure.

The power number remains as a good indication of performance. However, variations in fluid density, viscosity, and pressure can change the performance and limit the operating range. A truly objective comparison of hydraulic energy converters demands that the power number be calculated for devices operating at steady state under full load at identical temperature with identical fluid.

#### 4.4 OFF DESIGN PERFORMANCE

Often prototype hydraulic energy absorbers perform differently from similar models and published data. Discrepancies are due largely to dissimilarities in fluid flow. All too often available data does not include the characteristics necessary to build a machine and maintain similarity.

Effects of scale can be an important consideration. Relative roughness of internal cavities, the ratio of surface irregularity to rotor diameter, must be identical if two machines of identical geometry are to have similar power numbers. All internal angles must also be identical and internal dimensions should be proportional to the diameter of the machine.

If the above criteria for dynamic similarity are satisfied, the machine performance may still differ from available data. Entrained air in the fluid may effectively reduce the fluid density and reduce the power number. An air scoop or expansion tank can be placed on the inlet of the hydraulic energy converter to reduce the quantity of entrained air.

Proper design, fabrication and installation of the prototype cannot guarantee identical power numbers for prototype and the model, but the correlation is very good. Published data and manipulation of the power number equations can provide adequate data for performance matching of hydraulic energy converters and wind turbines; however the precision of the calculations should be inspected for proper matching.

#### 4.5 NUMERICAL PRECISION

The power number attempts to relate the physical dimensions and fluid characteristics to performance, but the power number equation can be misleading. Computation of the power number and the relative error illustrate the precision of the power number. Table 4.2 illustrates the error of each parameter and the cumulative error in power number for a hydraulic energy converter.

The 15% error is common for power numbers calculated from textbook data. If raw data is provided or the error of each parameter is reported the error can be calculated. Careful measurement of all parameters and tabulation of the relative error is necessary for prototype and model testing. The relative error is very important because it is a measure of the operating tolerance of the hydraulic energy absorber. Representation of the power number and the tolerance can be used to predict an interval where the power number of similar machines may lie.

Precise, well documented data are the key to accurate performance estimates of hydraulic energy converters. Still even with good precision and controlled conditions variation from the predicted performance may exist. Only fine tuning by means of baffles, gates, shutters or adjustable side clearance can provide uniformity of several machines.

#### 4.6 TIME RESPONSE AND INERTIA

This analysis is presented as a simplified description of dynamic interaction between a prime mover and the hydraulic energy converter. An equivalent moment of inertia is developed to facilitate calculation of the critical speed for the wind energy conversion system. This analysis is intended only as a first approximation to provide insight to the parameters that control system dynamic behavior and response.

Consider the system of Figure 4.6 where  $T_i$  is the driving torque,  $K$  is the torsional spring constant,  $\theta$  is angular position,  $I$  is the system moment of inertia, and  $B$  is the damping term. By analogy to the wind energy conversion system  $T_i$  is the rotor torque,  $K_s$  is the drive train stiffness, and  $I$  and  $B$  represent the moment of inertia and damping of the hydraulic energy converter.

The equation of motion for the system is shown below with boundary conditions.

$$I \ddot{\theta} + B \dot{\theta} + K_s \theta = T_i$$

$$\dot{\theta}_{(0)} = \dot{\theta}_i$$

$$\ddot{\theta}_{(0)} = 0$$

The equation can be solved by classical solution methods once the coefficients are determined. However, the linear ordinary differential equation cannot adequately describe the hydraulic energy converter response because, for the converter, the damping varies as the square of the rotational speed.

Two simple solution techniques exist. A Taylor series expansion technique<sup>5</sup> can simplify the problem and allow use of a piecewise



linearization for the non-linear term. As an alternative, the non-linear differential equation can be solved by numerical techniques. Both solution techniques lend themselves to computer formulation.

For the linear approximation, the damping ratio shown below can provide considerable insight into the time response of a wind turbine hydraulic energy conversion system.

$$\zeta = \frac{B}{2\sqrt{IK}}$$

where  $\zeta$  is the damping ratio.

The damping ratio is always positive for the energy converter. For  $\zeta$  between 0.0 and 1.0 response is oscillatory and damped, but for  $\zeta$  equal to or greater than 1.0, the system is over damped. Power extraction from gusty winds may be significantly reduced by too high a damping ratio. High oscillatory stresses may occur with very low damping ratios. Damping ratios close to .65 are suggested<sup>6</sup>.

Should the damping ratio exceed 10.0 the system will respond very much like a first order system because the damping term has a significantly greater effect than the inertial term. In this case the time response to a step input is given by the following:

$$\tau = \frac{B}{K}$$

where  $\tau$  is the time constant.

Approximations of the damping term and the moment of inertia for the hydraulic energy converter are necessary for the previous analysis. Both terms can be determined analytically. The damping term is derived from the power number and is given by the following equation.

$$B = \pi_1 \rho \omega D^5$$

Formulation for the moment of inertia is somewhat more complex due to the moving fluid entrained within the rotor. No universal technique exists to evaluate the contribution of internal flow to the angular momentum. As an approximation the moment of inertia of the dry rotor disk can be used. This approximation is satisfactory for analysis of small oscillations and critical speeds; however, the approximation may not be adequate to describe machine response due to abrupt changes of speed and torque.

Angular momentum of fluid within the device can augment the moment of inertia for the rotor. Run on or continued rotation in absence of external torque and evidence of a lead - lag phenomenon for speed and torque suggest that analysis of the fluid angular momentum be included in the study of time response. Unfortunately, empirical data is generally not available.

The fluid nature of the device dampens oscillations. However, devices driven by long elastic shafts may experience large torque excursions. Solutions of the equation of motion by linearization or computer technique can provide insight to these problems.

Thus, the basic elements for a first approximation of the dynamic behavior of the wind turbine and hydraulic energy absorber can be derived analytically. The analysis should be tabulated for a spectrum of operating speeds for easy evaluation of the damping ratio and the time response. Computer programming can greatly facilitate the design process by performing repetitive computations with speed and accuracy.

## CHAPTER 4

## LIST OF REFERENCES

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6. Ibid

Table 4.1 Power Characteristics Of Several Hydraulic  
Dynamometers

NAME	TYPE	$\Pi$	MANUFACTURE	REFERENCE
Model 14	Rectangular	$1.94 \times 10^{-2}$	All American Engineering Co Wilmington DE	Author
Model 12	Rectangular	$2.57 \times 10^{-2}$		Author
Reynolds	Elliptical	.973	Mather- Reynolds	Gibson
Froude	Elliptical	.949	Heenan-Froude	Rao
Unknown	Circular	.226	Worcester Unknown	Rao
Unknown	Rectangular	.025	Unknown	Rao
Prototype	Rectangular	.001	Author	Author
Prototype	Rectangular	.022	Culver	Culver
CHL1153	Viscous	$1. \times 10^{-3}$	Unknown	Knudsen

Table 4.2 Calculation Of Power Number And Relative Error

	PARAMETER	VALUE $\pm$ ERROR
CHARACTERISTICS	P	.010 $\pm$ .001
	$\omega$	10. $\pm$ .1
	$\rho$	986 $\pm$ 108
	D	.1 $\pm$ .001
RESULT	$\Pi$	.010 $\pm$ .0015

Fig. 4.1    EMPIRICAL POWER NUMBER OF SEVERAL  
DIFFERENT HYDRAULIC DYNAMOMETERS

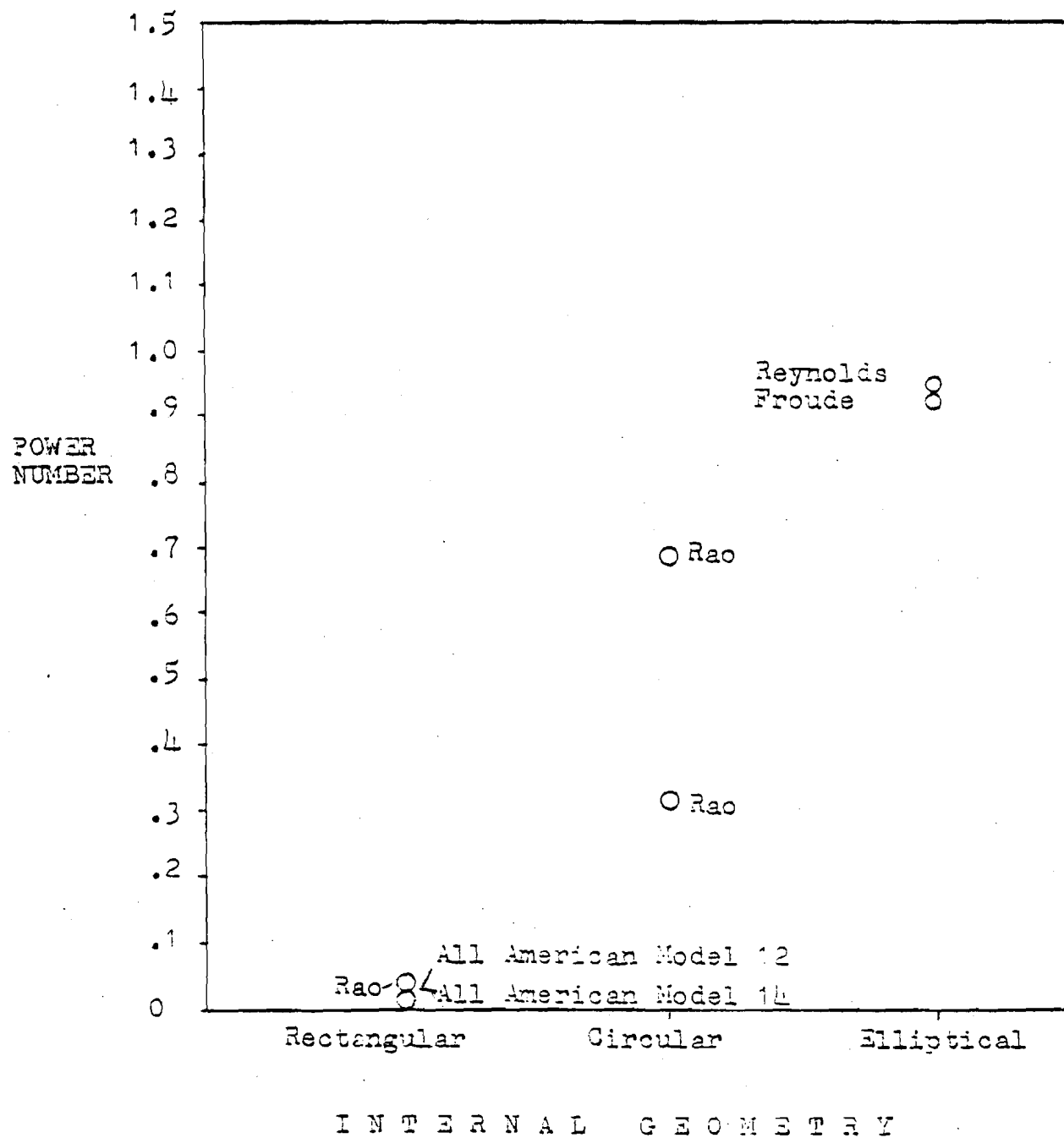
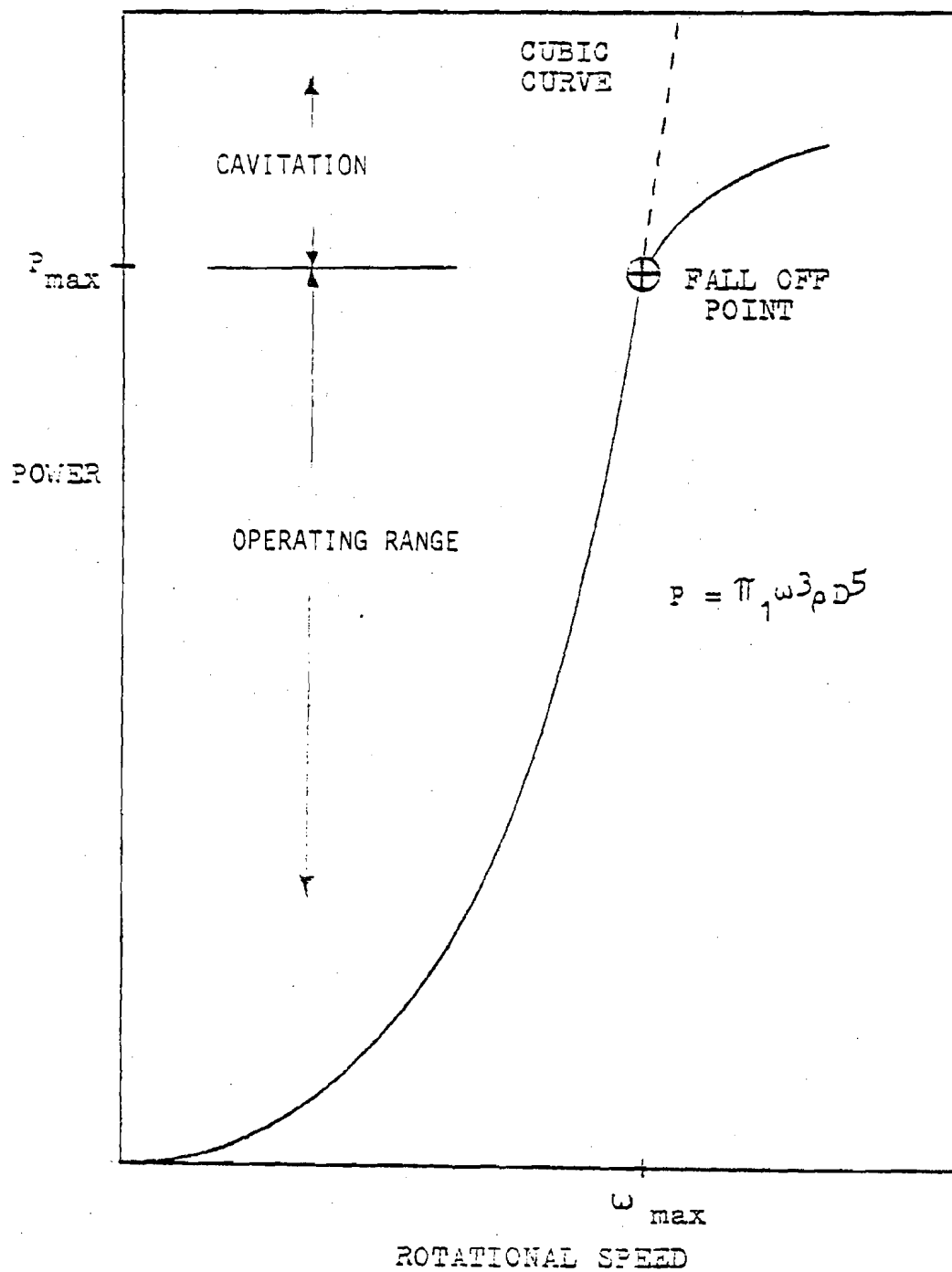


Fig. 1.2 TYPICAL PERFORMANCE OF A HYDRAULIC DYNAMOMETER



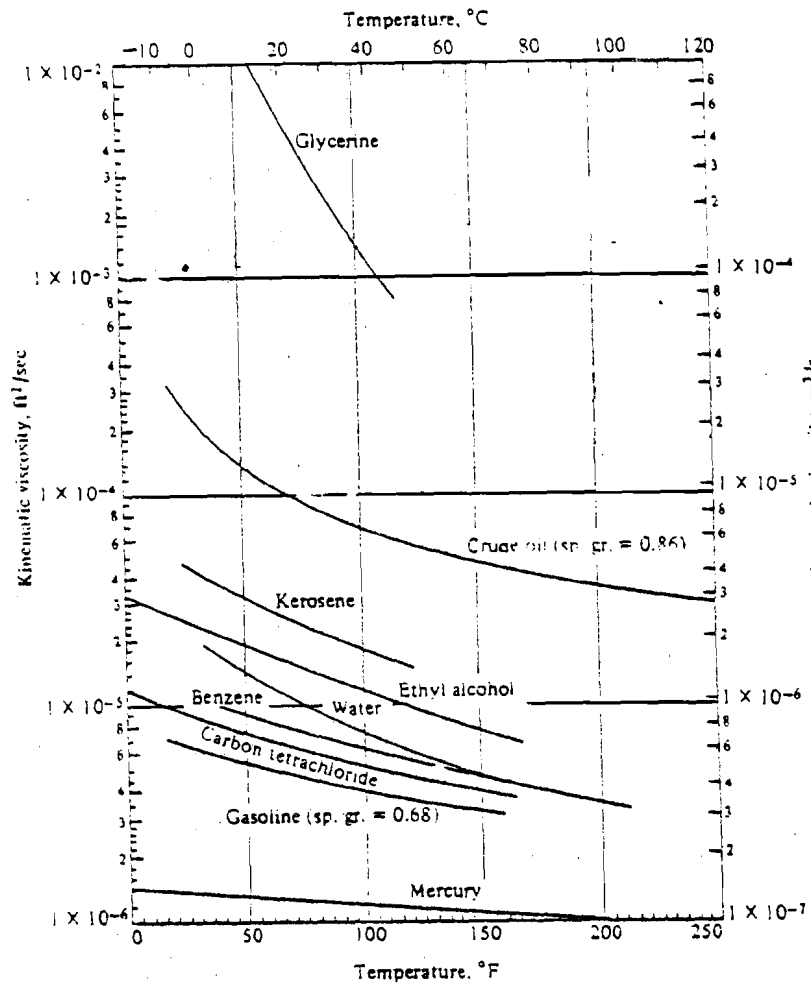


FIGURE 4.3 KINEMATIC VISCOSITY OF SEVERAL FLUIDS

[Adapted from V. L. Streeter, Fluid Dynamics  
Fifth Ed., McGraw-Hill, New York, 1971]



PLANE OF

SEPARATION

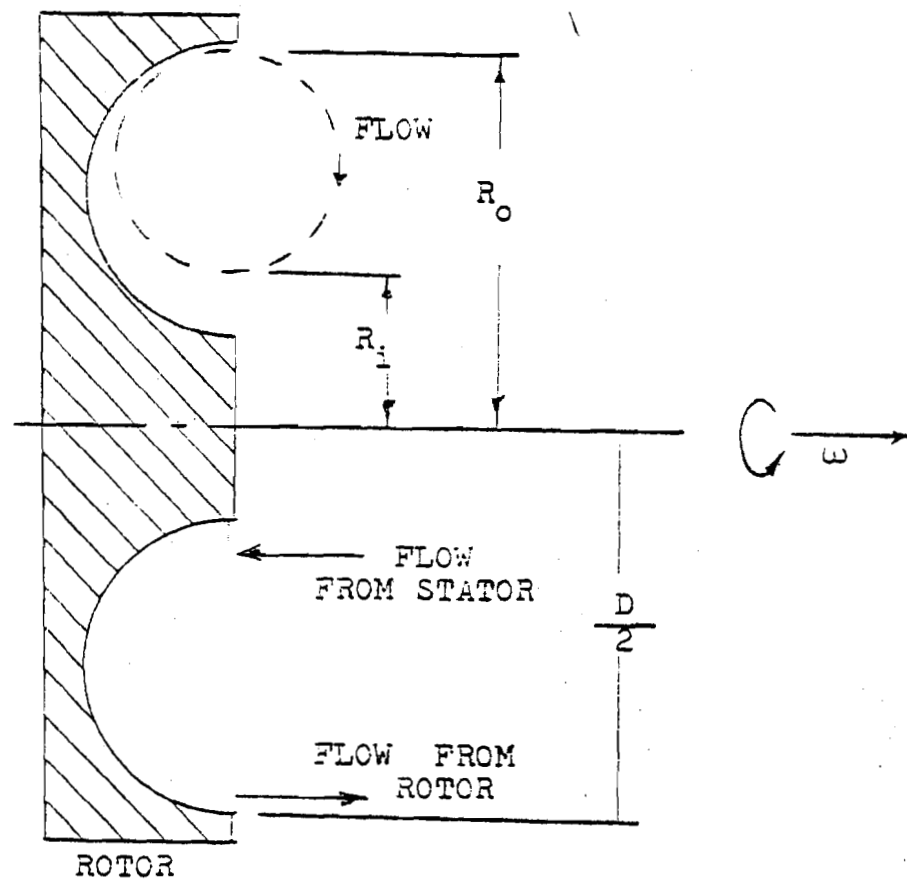


Fig. 4.4 TORUS GEOMETRY

## CHAPTER 5

### MATCHING OF WIND TURBINES AND HYDRAULIC ENERGY CONVERTERS

Wind turbines and hydraulic energy converters can be matched by identifying the power characteristics of each machine. For matching purposes the power number can be very useful, but it becomes cumbersome when evaluating many design candidates. The difficulty arises because the power number calculations must be performed for each wind turbine and hydraulic energy converter configuration. To simplify the design process, a unified dimensional analysis is presented that provides insight into the parametric relationship of wind turbines and hydraulic energy converters.

The result of the analysis is shown below:

$$\pi_1 = \frac{\pi}{64} C_p \frac{\rho_t}{\rho} \left(\frac{n_t}{D}\right)^5 \left(\frac{1}{xN}\right)^3$$

where  $\pi_1$  is the power number or systems power coefficient,

$C_p$  is the wind turbine power coefficient,

$\rho_t$  is the fluid (air) density at the wind turbine,

$\rho$  is the density of the energy converter fluid,

$D_t$  is the diameter of the wind rotor,

$D$  is the diameter of the energy converter,

$x$  is the wind turbine tip speed ratio, and

$N$  is the ratio of the converter speed to turbine speed.

Note that the dimensional terms are retained for clarity.

The system power coefficient is identical to the power number of the hydraulic energy converter. Also the system power coefficient is unique because it defines an explicit relationship of system parameters.

Model and prototype relationships of scale can be determined by the formulation. Also, this equation can be employed to match and compare system components. For example, the formulation can be used to estimate the necessary gear ratio to match a specific wind turbine and hydraulic energy converter. Similarly, the proper power number and diameter of a hydraulic energy converter for a direct drive wind turbine energy conversion system may be estimated.

The system power coefficient simplifies the parametric analysis. Physically the density ratio and the diameter ratio are limited and the speed ratio should be low to minimize transmission losses. In this light, high tip speed ratio wind turbines coupled to high power number energy converters can be compact and generally inexpensive.

Figure 5.1 shows the parametric relationship of wind turbines and hydraulic energy converters. The range of power numbers for each geometry is superimposed upon the graph as a guide to the designer.

## 5.1 Design Scheme

The wind turbine and hydraulic energy converter system will start easily and operate with constant tip speed ratio. The self regulating feature continues as the wind speed increases and the power remains a

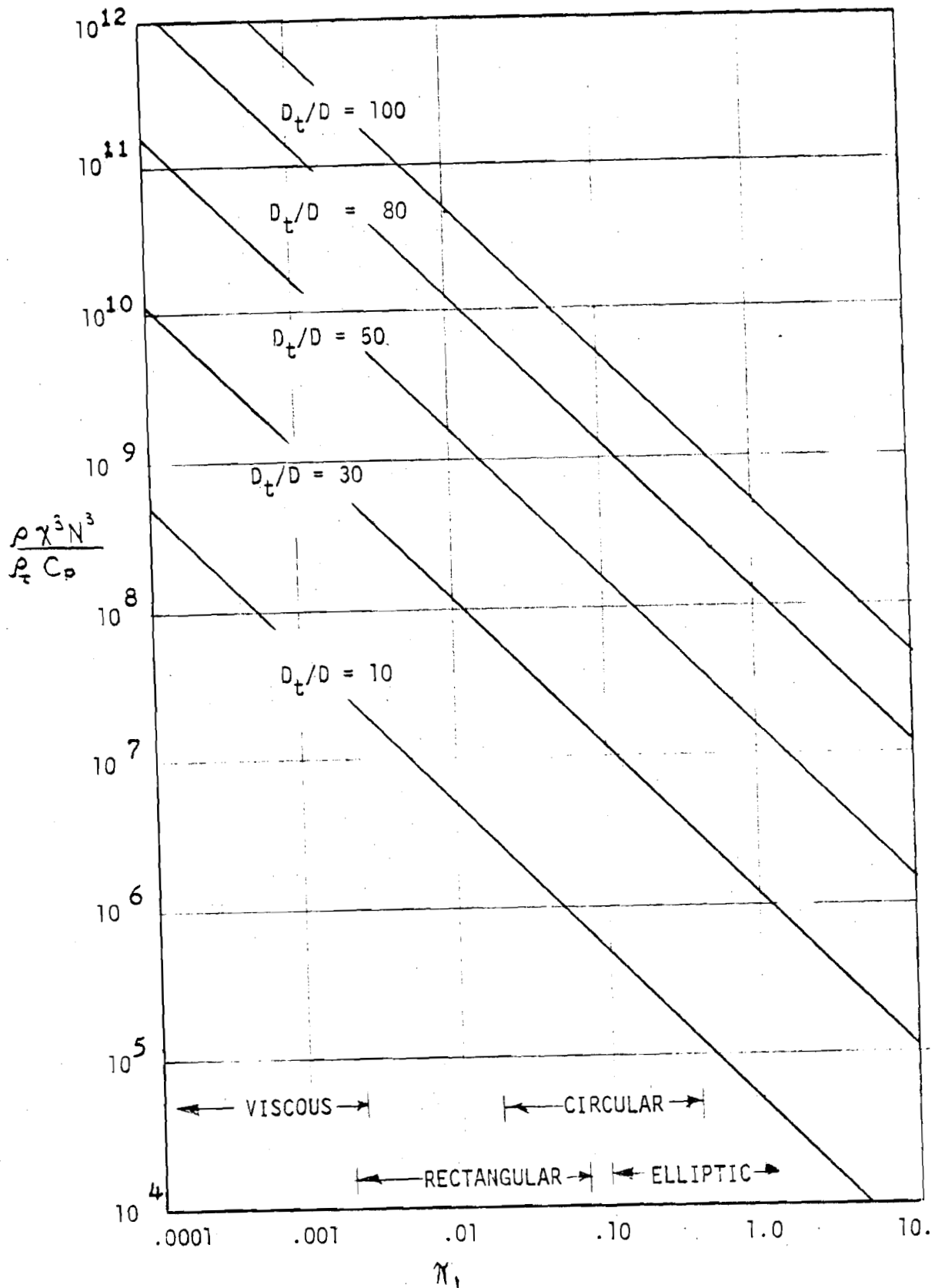
function of the wind speed cubed up to the shut down speed. Should the machine be subjected to such high wind speeds that the energy converter exceeds the operating range, the rotor will overspeed. The overspeed and accompanying increase in tip speed ratio is indicative of cavitation within the converter device.

Clearly some means of turbine shut down is necessary. Driving the energy converter above the operating region in an effort to lower rotor performance and reduce power has been suggested. Tests of this concept were unsuccessful. The idea bears merit, but it necessitates high converter wear and increased centrifugal blade loading.

Perhaps some means of dynamic braking is necessary. Conceivably, the energy converter could be oversized with some provision for adjustment of the device for run and brake modes. Essentially the brake mode would reduce the rotor tip speed ratio and the rotor performance to slow the device in high winds. Alternately, a means of blade or blade tip pitch control could be used to reduce rotor speed.

Care is necessary in evaluating the slow down scheme because the converter is not a brake. The converter is a speed dependent damper and must rotate to exert torque. Changing the converter or the wind rotor essentially changes the system power coefficient: that is, the system seeks equilibrium at a different rotational speed. Tests show that the rotor can be stalled in winds  $10 \text{ ms}^{-1}$  to  $30 \text{ ms}^{-1}$  and maintained at slow rotational speed by a hydraulic energy converter of twice the system power coefficient. Only when the wind rotor torque becomes zero will the system come to rest.

FIGURE 5.1 PARAMETRIC RELATIONSHIP OF WIND TURBINES AND  
HYDRAULIC ENERGY CONVERTERS



## CHAPTER 6

### PERFORMANCE EVALUATION OF A SCALE MODEL

Tests performed at the University of Massachusetts Wind Rotor Test Facility proved that a wind rotor could drive a hydraulic energy converter. This wind energy conversion system starts easily at low wind speeds, performs with speed regulation for high efficiency operation, and maintains stability in changing winds.

The ease of starting can be predicted from analytic study of the energy converter. Retarding torque is due to friction of seals and bearings, viscous shear, and momentum transfer within the fluid all directly related to rotational speed. The retarding torque at start up is very low and increases as toroidal flow and associated high fluid velocity enhance the mechanism for energy dissipation.

Automatic speed regulation controls the rotor speed to operate at a constant tip speed ratio within the operating region. In the operating region retarding torque varies as the square of the wind speed so that the energy converter loads the rotating wind rotor according to the available power. Tip speed ratio may vary due to both fluid density changes within the working fluid and non-steady state operation.

Mechanical stability of the system varies with the system parameters. Inertia, stiffness, and damping of the rotor, energy converter and intermediate drive train must be considered. The hydraulic energy converter in general provides a soft dynamic coupling with some damping that makes the drive system tend towards stability. Stability of a prototype system is investigated as part of the test procedure.

Different internal clearances between rotor and stator of the hydraulic energy absorber should allow fine tuning of the system so that the rotor tip speed ratio can be optimized. Tests of several side clearances prove the validity of this tuning scheme.

## 6.1 TEST PROCEDURE

A hydraulic energy absorber was designed by method of Chapter 5 to match a 1.12 m diameter wind rotor which tests the performance of the Wind Furnace Model Four concept. The energy absorber has a torrus diameter of .127 and a power number of .001 necessary to match the design point of 185 w at  $90 \text{ rs}^{-1}$  for the three blade rotor described in Figure 6.1.

The rectangular geometry is chosen for the absorber for this application because of material constraints and the necessary power characteristics. Figure 6.2 shows the details of the energy absorber. Note however that the hydraulic energy absorber shown in the drawing is designed for a 2.0 kW rated wind assisted domestic hot water system and required alteration to be used with the 185 w rotor at the Wind Rotor Test Facility. For the tests the blade depth,  $d$ , shown in Figure 6.3 was reduced to 4.5 mm and the calculated power number was estimated to be .001.

A test stand was modified to accommodate the wind rotor and permit direct drive through a flexible coupling to the hydraulic energy converter. The hydraulic energy converter was placed in trunion bearings and restrained in torsion by a force transducer. Rotational speed could be measured by means of an optical interrupt and digital count/divide circuit detailed in Appendix I. Details of the test device are shown in Figure 6.4.

The test stand was placed in the test area of the wind tunnel at the Wind Rotor Test Facility. Details of the wind tunnel are shown in Table 6.5. A pitot tube and slant tube manometer were used to determine the air velocity at the test region. Test equipment, origin, and precision are tabulated in Table 6.6.

To randomize experimental error the wind speed controlled by the wind tunnel inlet vane damper was positioned on a scale according to a table of random numbers. Data including air temperature, barometric pressure, manometer height, rotational speed, torque, and variation of each parameter for each damper position were recorded.

The energy converter dimensions were changed for each test. Vane depth, and the clearance between rotor and stator were altered in an effort to maximize the energy conversion of the wind rotor. Theoretical work suggests a maximum power coefficient,  $C_p$ , of .41 at tip speed,  $X$ , of 5.0 for the device.

Losses due to bearing friction and windage are shown in Figure 6.7 and working fluid density change were considered very small because of the small temperature variation of the working fluid. In the tests performed the energy absorption is given by equation

$$W = m C_p dT \quad (6.1)$$

where  $W$  is the work done on the fluid,

$m$  is fluid mass,

$C_p$  is fluid specific heat at constant pressure, and

$dT$  is the change of fluid temperature.

For small temperature changes or constant specific heat the equation can be written



$$W = m C_p \Delta T \quad (6.2)$$

where  $\Delta T$  is the change between initial and final temperature .

Upon substituting values for the energy converter, working fluid, and typical work input for an hour of testing the temperature rise within the working fluid is given by

$$\Delta T = \frac{75 \text{ J/hr}}{4. \text{Kg} \cdot 4.186 \frac{\text{KJ}}{\text{Kg}^\circ\text{C}}}$$

$$\Delta T = \frac{4.5^\circ\text{C}}{\text{hour}} .$$

Thus the small temperature change alleviated the need for a cooling circuit to keep the working fluid at constant density.

## 6.2 Results

Data for test 6-4-79 is shown in Table 6.8. The hydraulic energy absorber maintains the wind rotor tip speed ratio at  $4.2 \pm .1$  over the wind velocity range  $4. \text{ ms}^{-1}$  to  $17. \text{ ms}^{-1}$  the power coefficient is approximately  $.29 \pm .04$  over the wind velocity range as shown by Figure 6.9. Thus the device exhibits automatic speed regulation for constant tip speed ratio and reasonable power coefficient.

Analytic evaluation of the wind rotor predicts optimum performance at a tip speed ratio of 5.0 and a power coefficient of .41. Unfortunately the tests performed on the rotor were limited to a tip speed of 4.3 because the energy absorber is oversized. To achieve the tip speed ratio of 5.0 the vane depth must be reduced. A reduction of vane depth was deemed inappropriate because the vanes were already shallow and further cutting might obscure the view of internal flow within the prototype energy absorber.

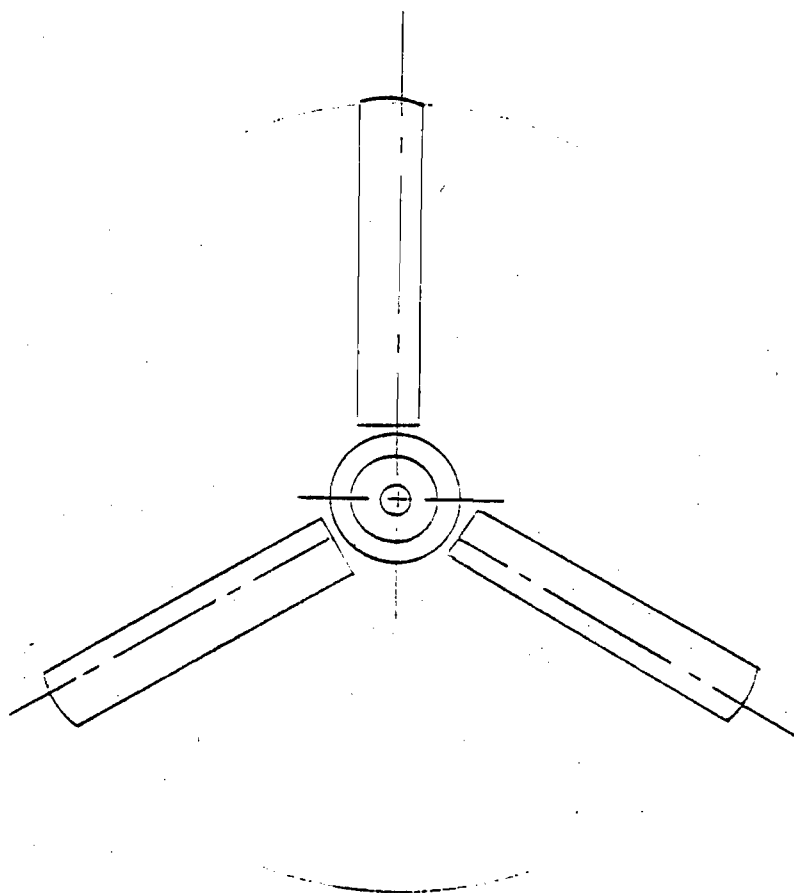
The inability to achieve the tip speed ratio of 5.0 can be attributed to the inaccuracy of the analytical rotor performance, the design procedure employed for the hydraulic energy absorber, and inefficiency of the wind rotor. The hydraulic energy absorber was designed by means of dynamic similarity with the analytic performance of the wind rotor. The actual performance of the system differed from the anticipated performance. However, the prototype performance is quite close to the predicted performance, as shown in Fig. 6.9, and illustrates the capabilities and limitations of the design procedure.

The energy converter permitted start up for all tested wind speeds and quick acceleration to the operating point and at no time did the device fail to reach the operating speed in less than 40 seconds. Also while changing velocity no overshoot of the tip speed ratio was evident indicating that the system response is critically damped.

It is possible to fine tune the energy converter to constrain the wind rotor to a particular tip speed ratio by adjusting the rotor side clearance.

### 6.3 Conclusions and Remarks

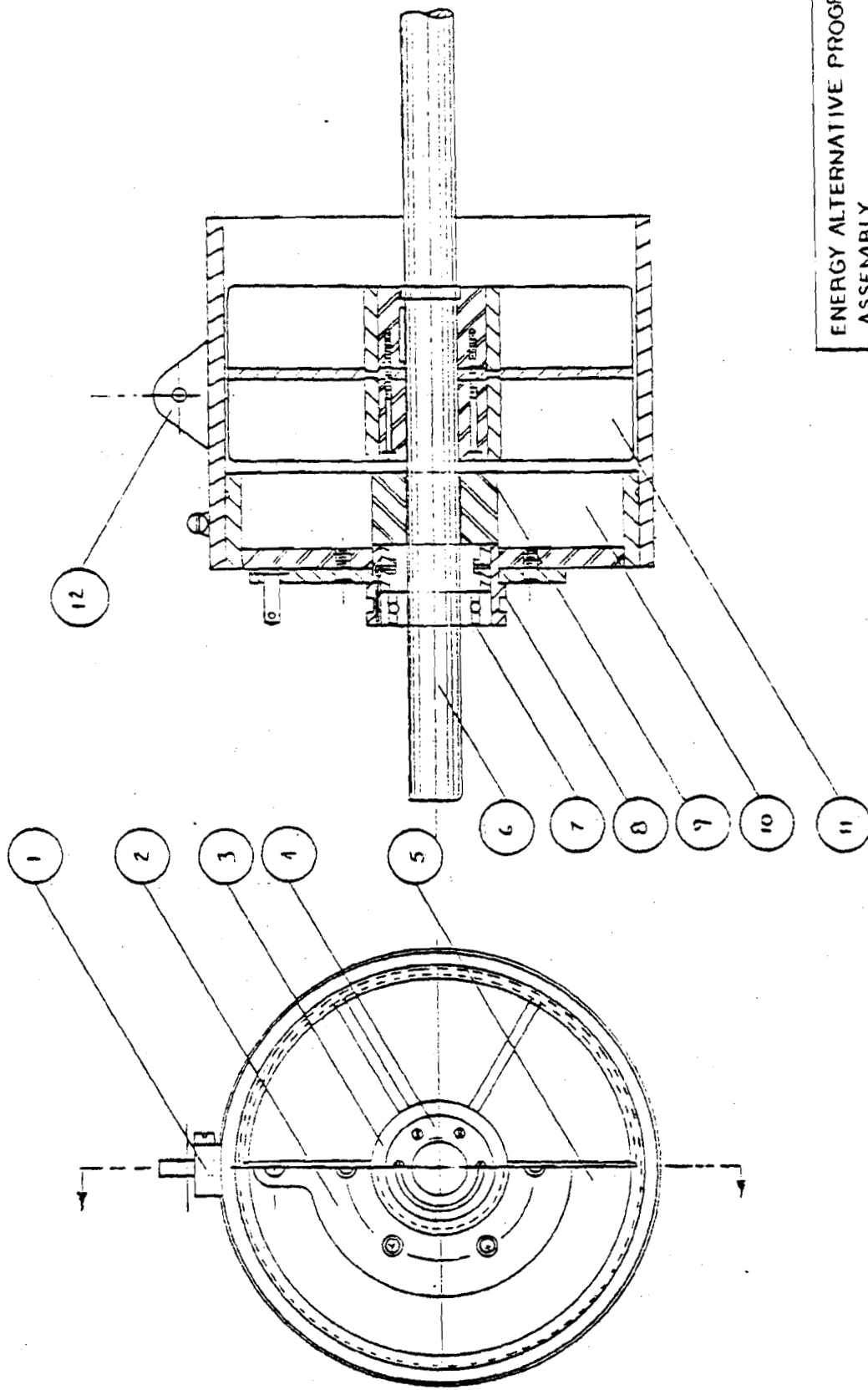
The design procedure that employs analytic and empirical performance of hydraulic energy absorbers and wind rotors may be used to design a wind furnace model four prototype. The wind energy conversion system with the hydraulic energy converter starts easily, maintains speed regulation, and is dynamically stable. Manufacturing tolerances and surface quality may alter the design characteristics of the system and a means of tuning the system such as altering wind rotor diameter, pitch, or changing the energy absorber internal side clearance is necessary.



## R O T O R   D E T A I L S

Diameter	1.12 m
Chord	.114 m
NACA	4415
Taper	None
Taper	None
Blades	3

Figure 6.1



ENERGY ALTERNATIVE PROGRAM  
ASSEMBLY

DWG NO. 1400  
DATE: 10-26-78 MGR SCALE: 1:2

Figure 6.2 PROTOTYPE

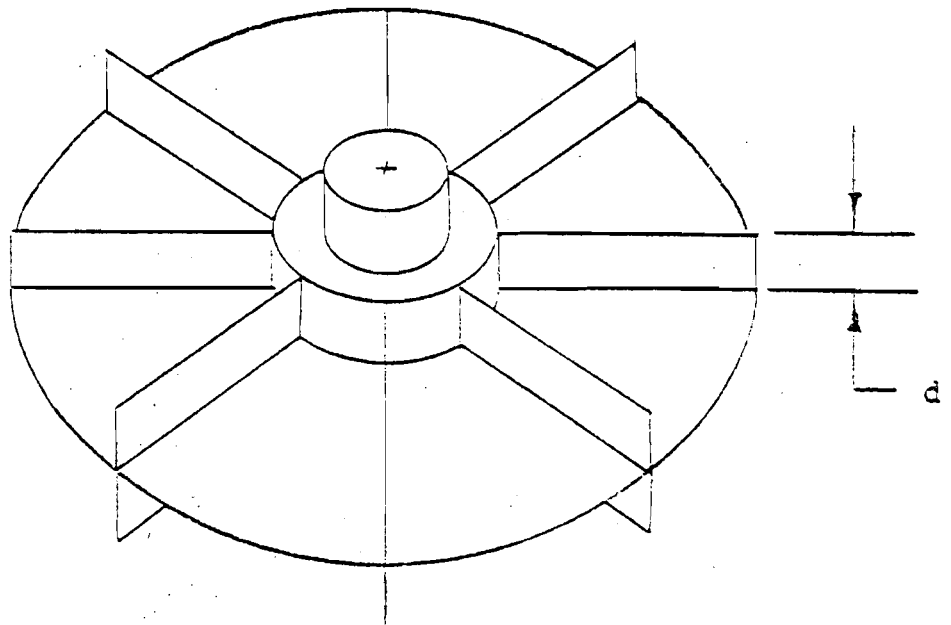


Figure 6.3 VANE DETAIL

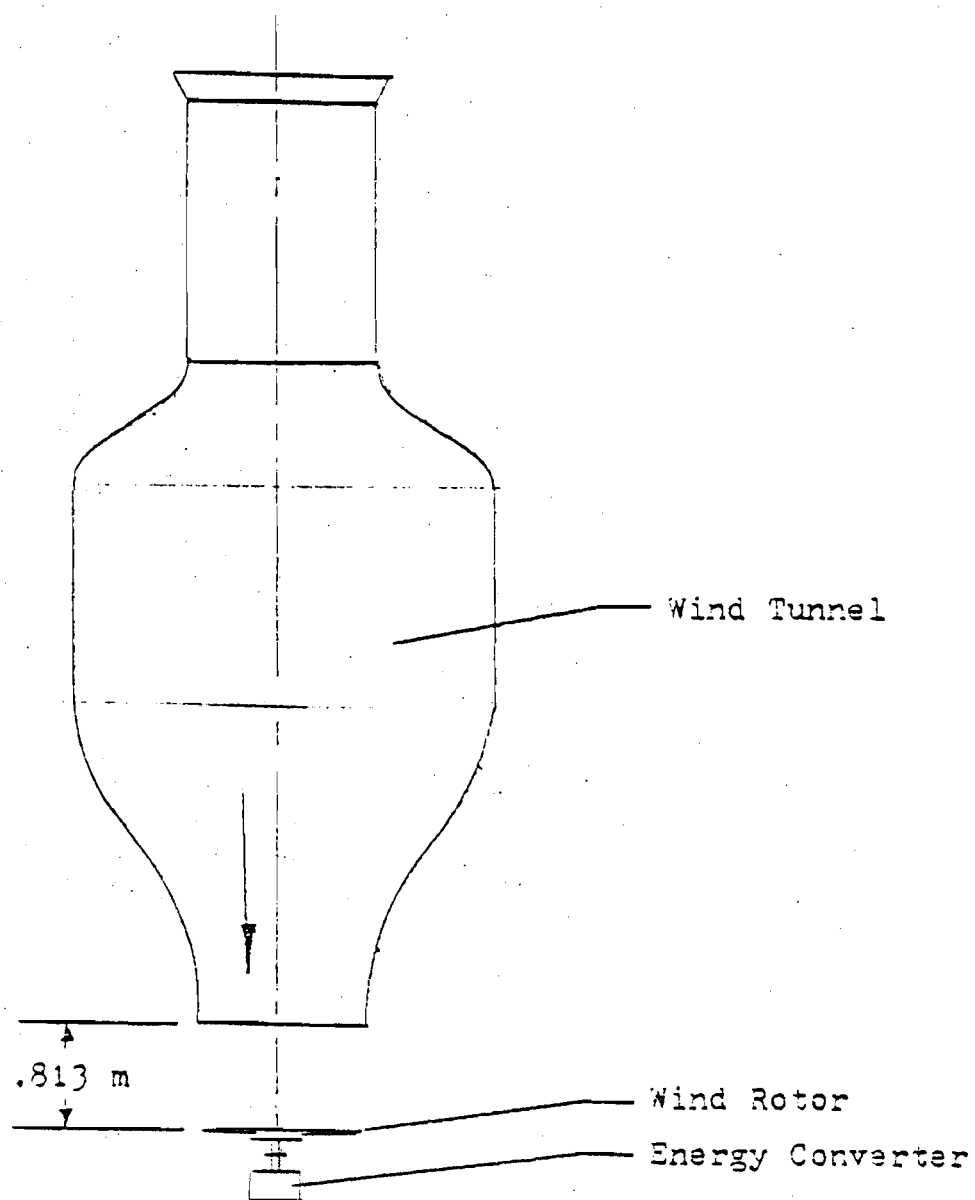


Figure 6.4 Test Detail

Table 6.5 DETAILS OF THE WIND ROTOR TEST FACILITY

Tunnel Construction	Open environment test section Non-recirculating horizontal flow
Fan	Axial flow Buffalo Forge variable vane pitch
Motors	2 56 Kw 3phase Synchronous
Test Section	Open throat 1.22m x 1.22m 1.49 m <sup>2</sup> area
Aerodynamic Data	Wind velocity of test section 1-30.5 ms <sup>-1</sup> Turbulence and expansion unknown
Measurement Equipment	Fluid manometer, pitot tubes, Strain gauge torque transducer, Opto-interrupt digital count divide Tachometer

Table 6.6 DETAILS OF TEST EQUIPMENT

INSTRUMENT	ORIGIN	PRECISION
Manometer	E. Vernon Hill	$\pm .005m$
Manometer	Gilmont Micro	$\pm 13 \text{ um}$
Pitot Tube .15m 9mmØ	Unknown	
Pitot Tube PAC8K1	United Sensor	
Bridge Meter	Ellison	$\pm$ as indicated
Tachometer	Sass, Wally U. Mass '79	$\pm 1\% \text{ Max.}$



Fig. 6.7 Windage for test apparatus

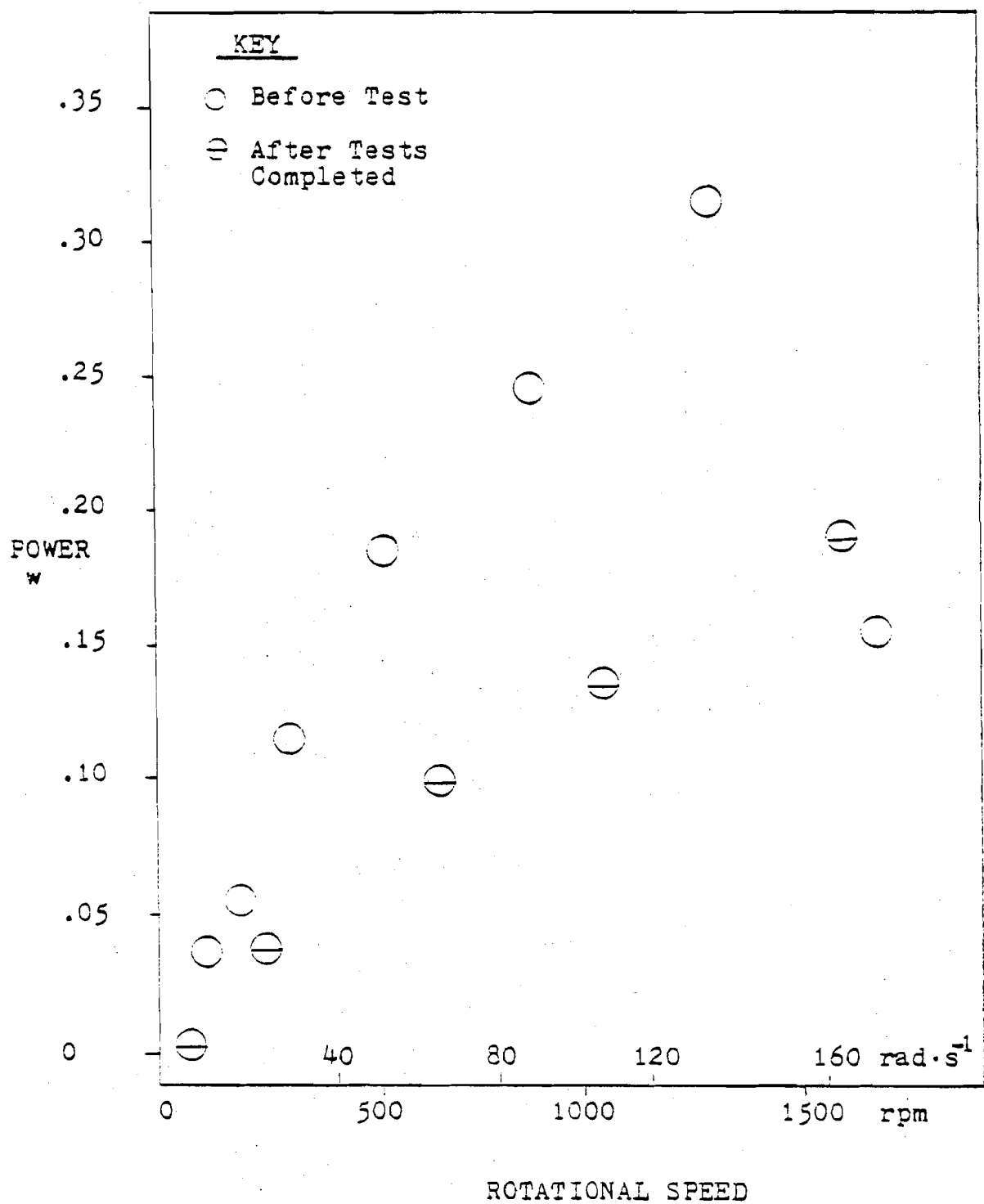
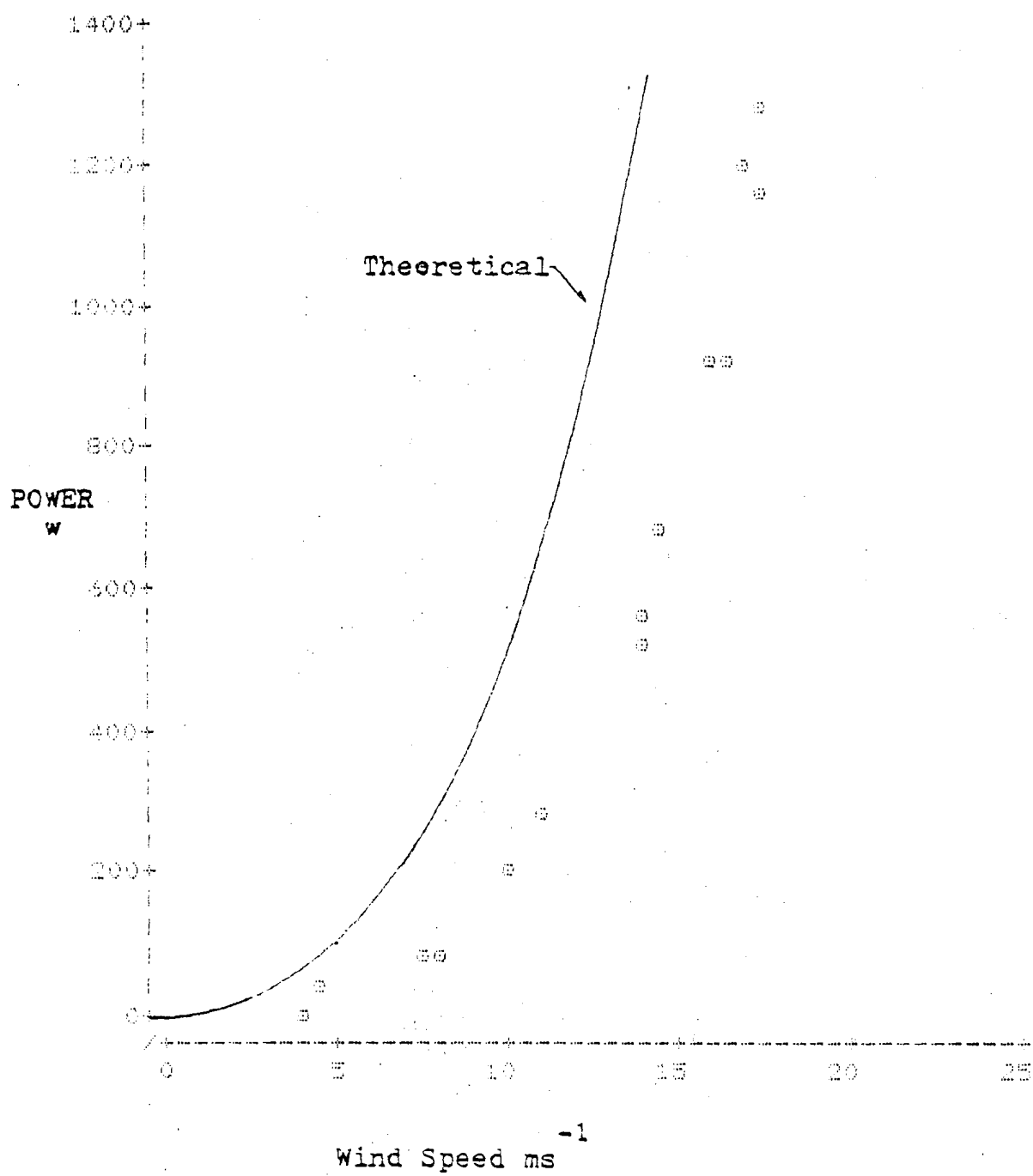


Table 6.8 SAMPLE DATA

JOB NO.	T	FEED	H	VELOCITY	W	TORQUE	POWER	R	CP
297,400	100940.000	0.001	4.200	38.432	0.422	16.215	4.184	0.307	
299,400	100940.000	0.015	14.598	130.899	5.289	692.299	4.100	0.312	
299,700	100940.000	0.002	4.695	41.155	0.491	20.191	4.008	0.274	
299,700	100940.000	0.021	17.240	162.629	7.290	1185.646	4.313	0.325	
298,500	100940.000	0.022	17.613	155.456	7.654	1266.329	4.295	0.325	
298,000	100940.000	0.004	7.820	71.104	1.403	99.770	4.157	0.293	
298,500	100940.000	0.022	17.380	162.420	7.173	1164.995	4.273	0.312	
298,500	100940.000	0.001	3.992	35.186	0.343	12.084	4.030	0.267	
299,700	100940.000	0.007	9.798	89.526	2.061	184.062	4.168	0.275	
300,200	100940.000	0.014	13.765	126.187	4.239	534.891	4.191	0.288	
300,800	100940.000	0.019	16.457	151.738	6.172	936.513	4.215	0.295	
300,800	100940.000	0.004	7.720	68.696	1.354	93.020	4.009	0.284	
301,400	100940.000	0.009	11.010	99.483	2.669	265.514	4.131	0.279	
301,400	100940.000	0.001	3.875	36.547	0.196	7.172	4.312	0.173	
301,800	100940.000	0.018	15.921	151.529	6.201	939.681	4.351	0.327	
301,800	100940.000	0.014	14.020	123.595	4.406	566.552	4.194	0.289	

Fig. 6.9 COMPARISON OF ANALYTIC AND EXPERIMENTAL RESULTS



## CHAPTER 7

### PERFORMANCE EVALUATION OF THE WIND FURNACE MODEL FOUR

The University of Massachusetts Wind Furnace wind turbine is designed to accept a variety of energy conversion schemes. One version, the Wind Furnace Model Four<sup>1</sup> concept, employs a hydraulic energy converter to convert wind energy to heat. The following chapter will outline a general design procedure and present several candidate designs for the Wind Furnace Model Four.

Figure 7.1 illustrates a logical design sequence for the Wind Furnace Model Four Project. Several crossroads exist in the sequence because parallel research changed the project direction.

At inception the project proposed coupling of a hydraulic energy converter or centrifugal pump and manifold to convert wind energy to heat for space heating and domestic hot water. The system promised automatic speed control and the direct conversion of wind energy to heat. Simplicity and low cost seemed the primary advantages of the system while power transmission losses brought about serious questions concerning overall efficiency.

Three design concepts were identified as possible design configurations for attachment to the Wind Furnace. Version A, shown in Fig. 7.2, employs the Wind Furnace right angle speed up drive to drive a rotating vertical shaft attached to an energy converter submerged in a subterranean heat storage tank. Version B, shown in Fig. 7.3, placed the energy converter aloft affixed to the tower and driven by the right angle speed up drive. Figure 7.4 shows version C with an aloft energy converter driven directly or by a speed up

transmission, from the wind rotor. Version A transmits mechanical kinetic energy to the storage medium and versions B and C transport thermal energy in heated fluid through coaxial insulated pipe to the energy storage tank.

Candidate designs are evaluated according to cost, simplicity, integrity, efficiency, special requirements, and safety features.

Version A costs are presented in Appendix II. This scheme requires a vertical line shaft to run the height of a tower that must provide access to the rotating shaft. A yaw drive or damper is also necessary. Low maintenance, high efficiency and use of standard components are important advantages of this device. Susceptibility to torsional vibrations are a disadvantage.

Machine B is essentially a hybrid of versions A and C. It incorporates the right angle drive and the yaw driver as well as the coaxial piping. Thermal losses and the necessity for yaw drive are primary disadvantages of this system, cost of main components is outlined in Appendix II.

Version C offers the greatest simplicity. A cost summary for this device is shown in Appendix II. Disadvantages of this scheme include thermal losses and necessity of a swivel fluid union between the fixed coaxial tube and the yawing wind turbine.

Version C is chosen because of low cost and simplicity. This machine could operate with only 2 moving parts if the wind rotor and energy converter are direct drive. Unfortunately thermal losses may make cold and calm days unproductive.

## 7.1 The Water Twister<sup>®</sup>

A Water Twister<sup>®</sup> dynamometer device of the Culver<sup>2</sup> design manufactured by All American Engineering Company, Wilmington, Delaware was tested to determine if the device could be matched to the Wind Furnace. Results of the test are shown in Figure 7.5. The data suggest a power number of .026 for the Water Twister<sup>®</sup> Model 12. A transmission speed ratio of 4.1 is required to match the device with the wind turbine rotor.

Figure 7.5 shows the speed performance of Water Twister<sup>®</sup> devices.

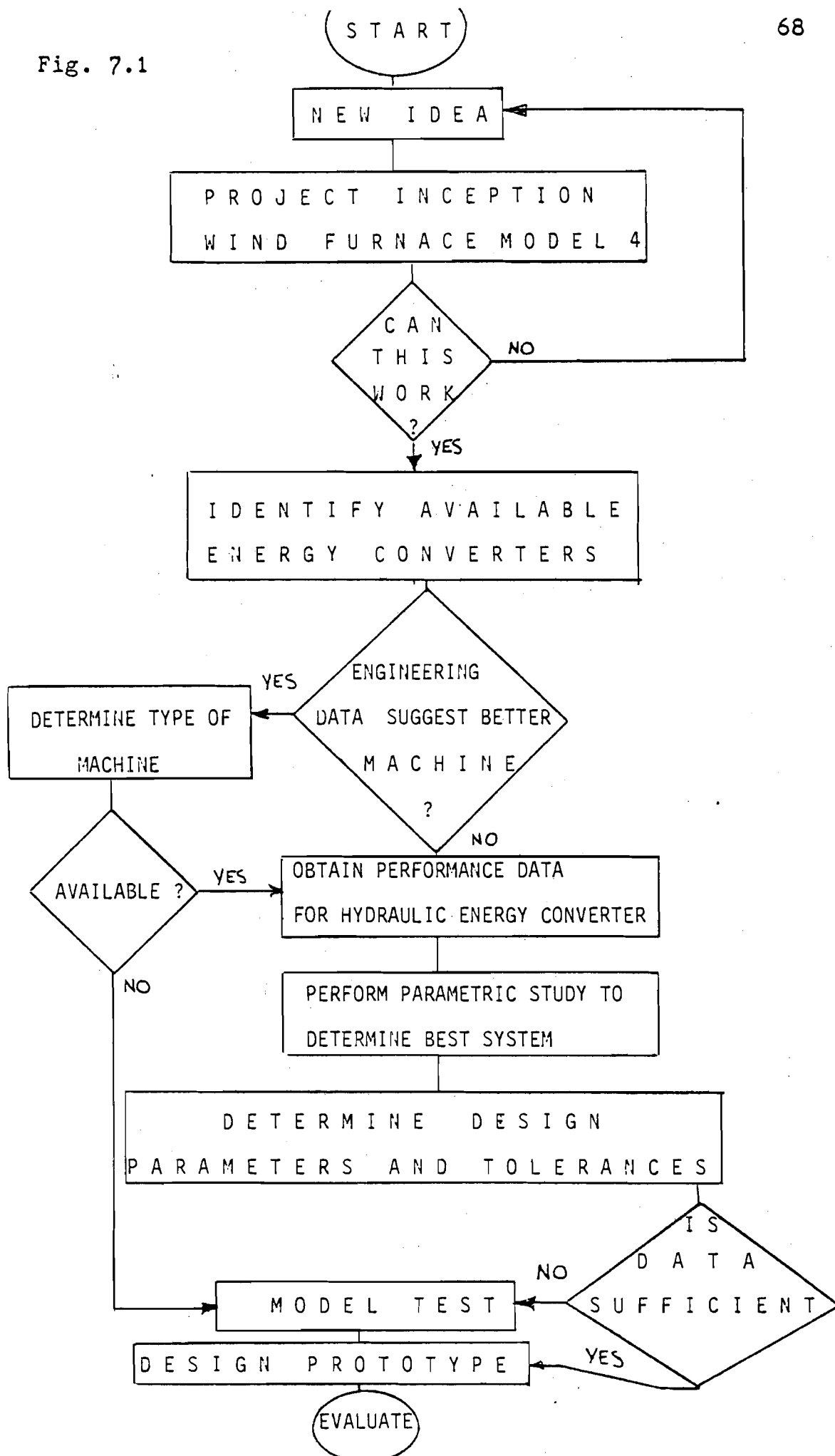
## 7.2 The Wind Furnace Model Four

Careful analysis suggests that a direct drive wind turbine and hydraulic energy converter device can be a very simple system. An energy converter of the Froude or Reynolds geometry of .457 m (18 inch) diameter can match the performance of the Wind Furnace wind rotor. Figure 7.6 depicts the primary system components.

C H A P T E R 7  
LIST OF REFERENCES

1. W.E. Heronemus, Personal communication January 1977.
2. E. P. Culver, "Investigation Of A Simple Form Of Hydraulic Dynamometer." Mechanical Engineering 1937. p. 749.

Fig. 7.1





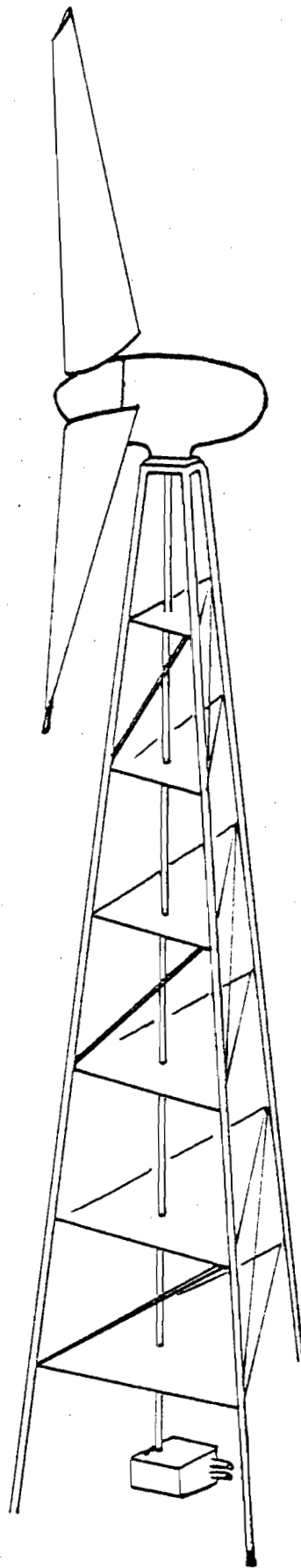


Fig. 7.2  
MODEL FOUR  
VERSION A

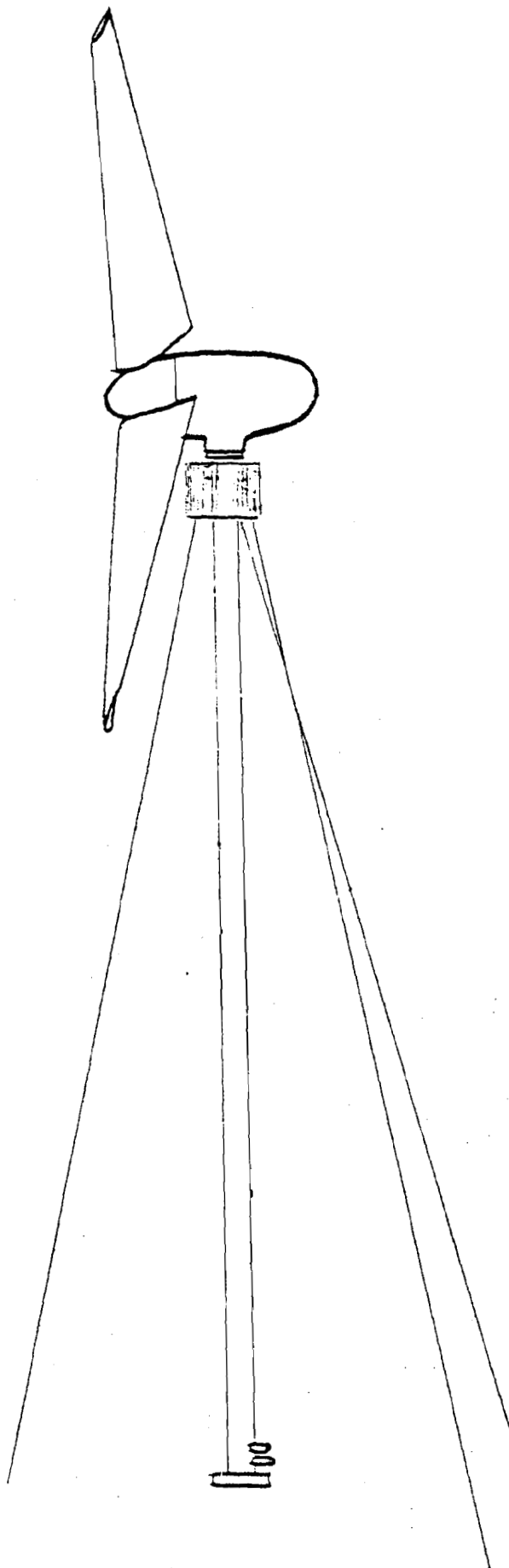


Fig. 7.3  
MODEL FOUR  
VERSION B

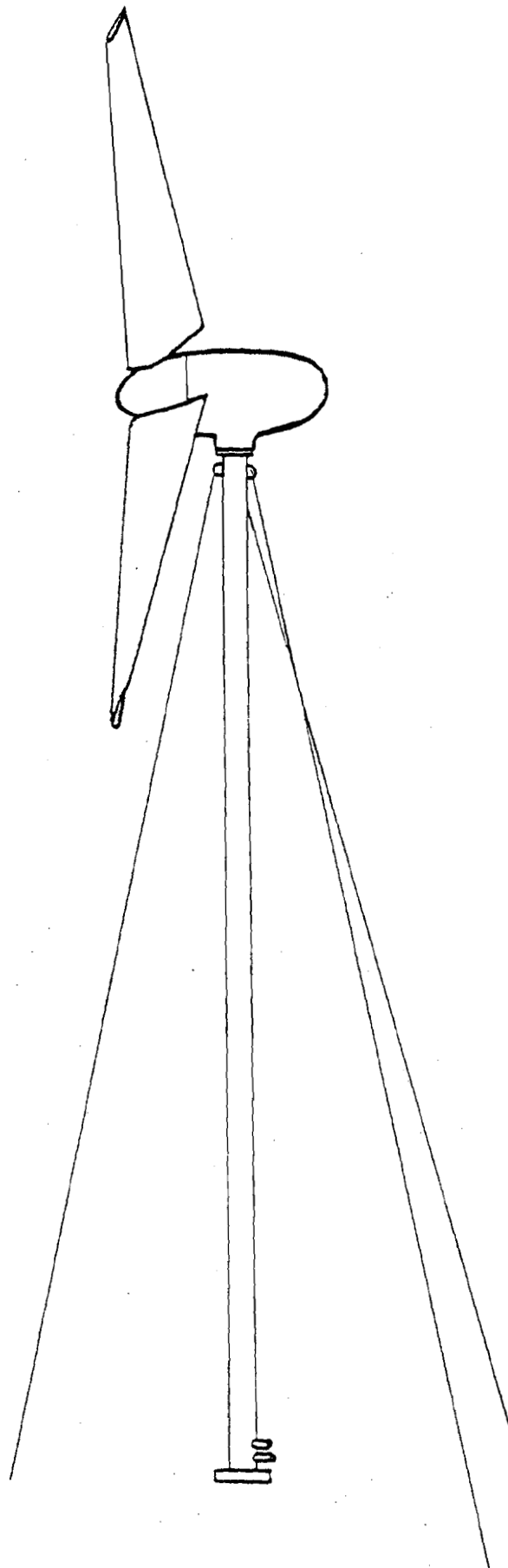
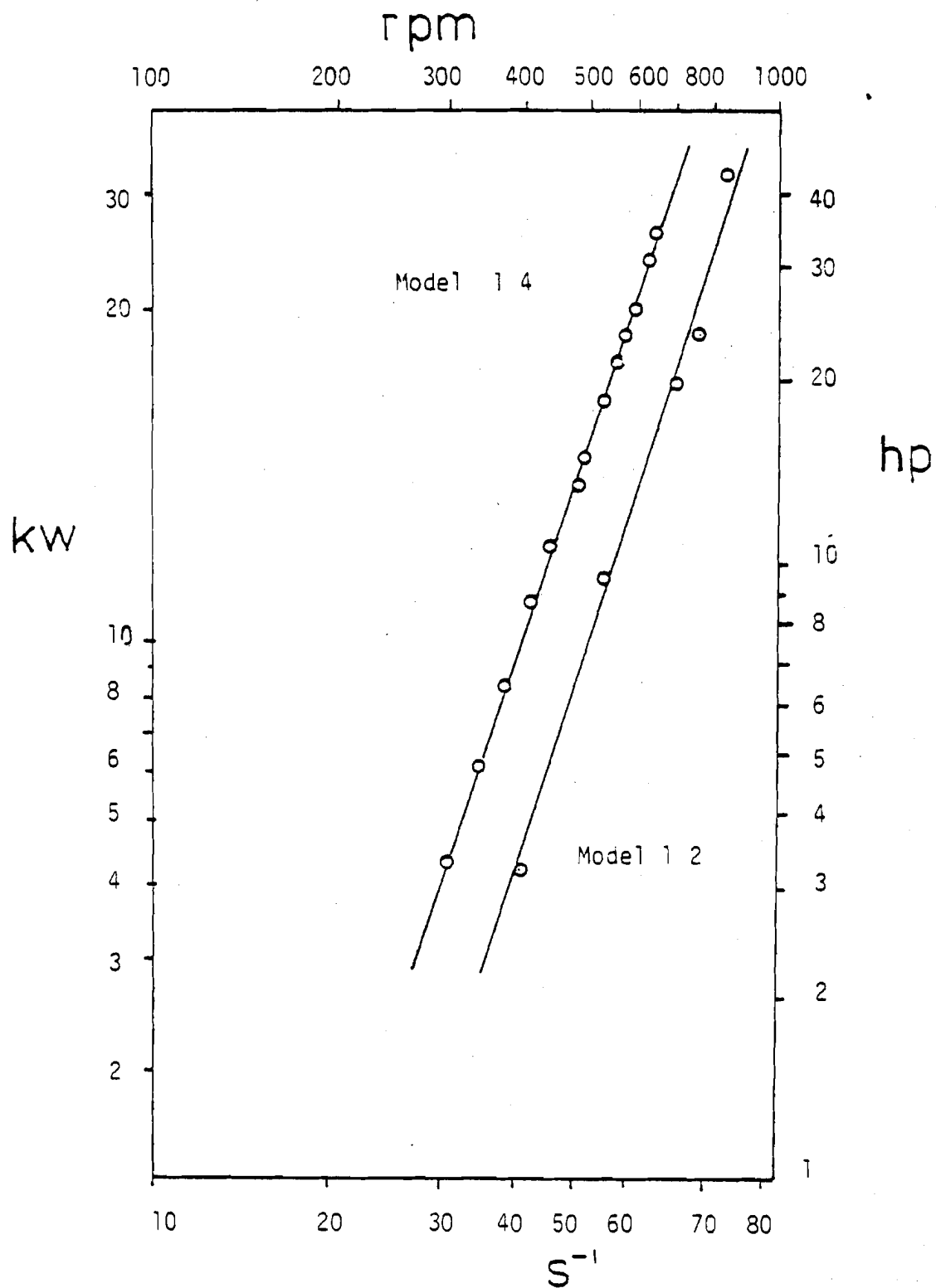


Fig. 7.4  
MODEL FOUR  
VERSION C

Fig. 7.5 RELATIONSHIP OF SPEED AND POWER FOR  
WATER TWISTER MODEL 1 2 AND MODEL 1 4



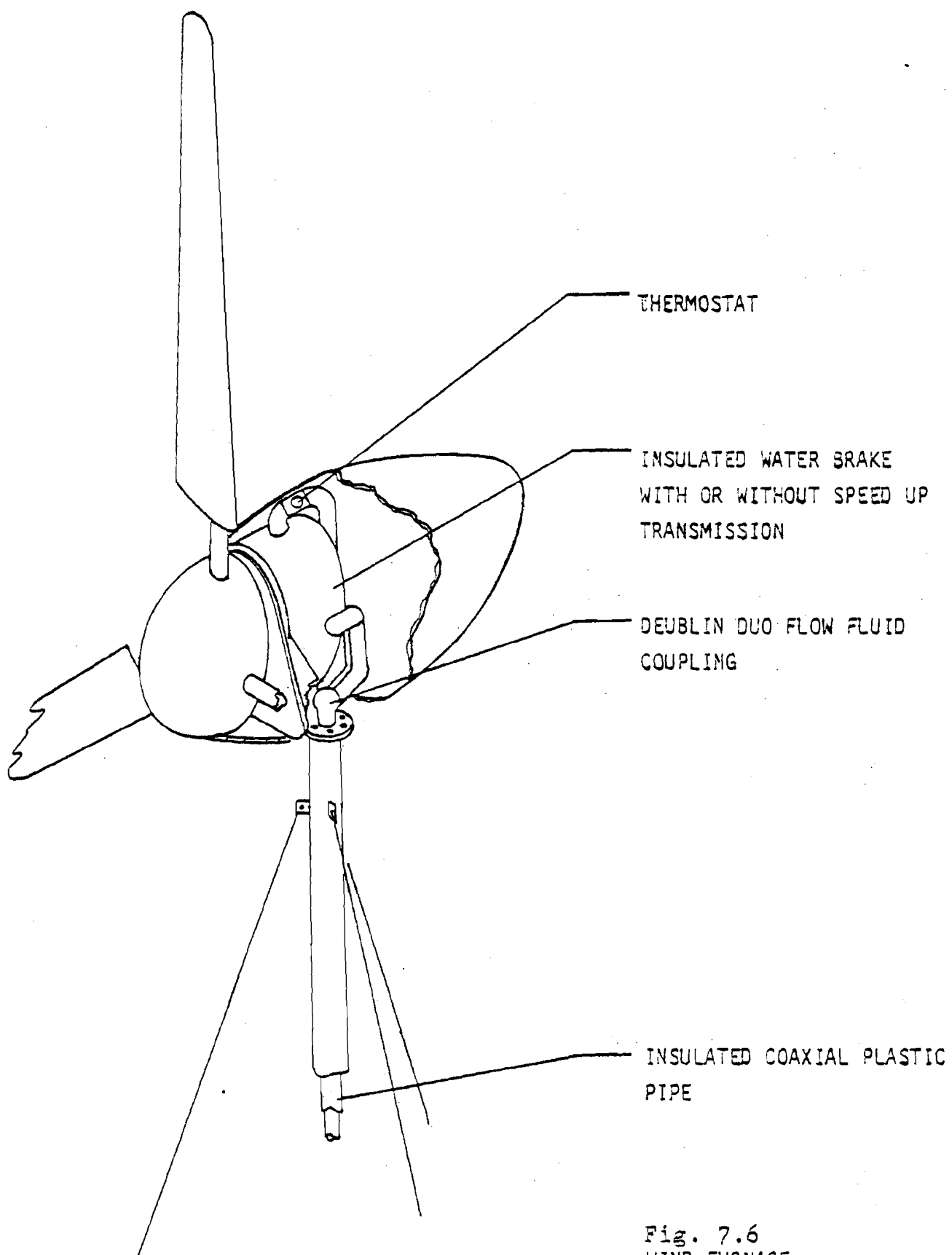


Fig. 7.6  
WIND FURNACE  
MODEL FOUR  
VERSION C

## RECOMMENDATIONS AND CONCLUSIONS

The Wind Furnace Model Four can readily convert the kinetic energy of the wind to heat. The device is self-contained and self-regulated and allows a significant materials savings above wind electric systems.

The design process outlined here required accurate data for both wind rotor and energy absorber. Still, some means of fine tuning the wind energy conversion device should be incorporated to achieve maximum performance.

In general hydraulic energy absorbers should be coupled with high tip speed ratio wind rotors with broad tip speed power curves. This tip speed tolerant combination will insure high system efficiency despite changes in the working fluid and power characteristics.

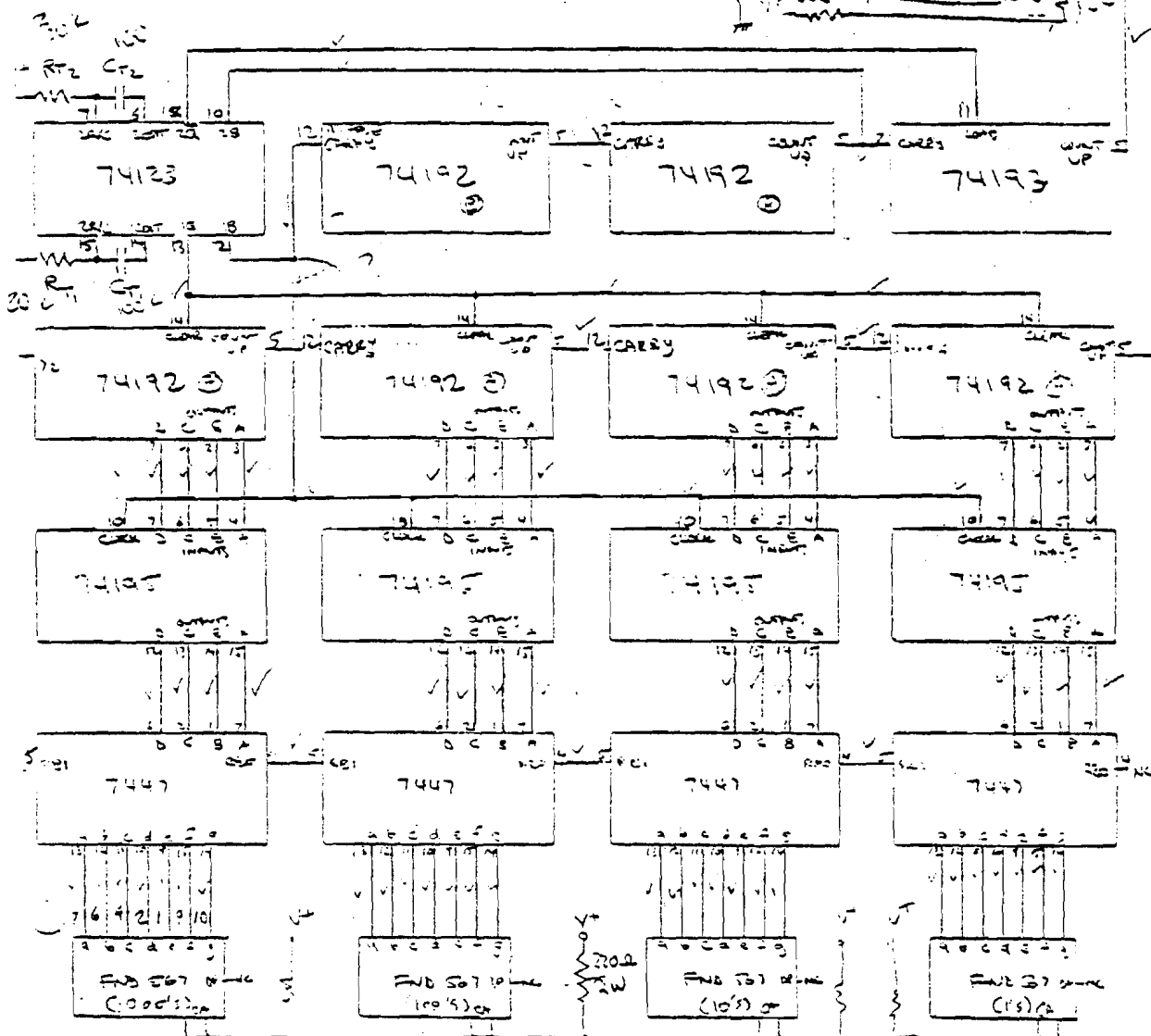
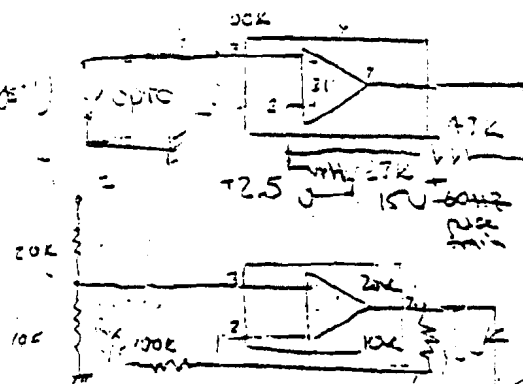
Future studies on dynamic behavior of the Wind Furnace Model Four might investigate the dynamics of mechanical elements and the detailed thermal efficiency of the system. More data from circular and elliptic geometry energy converters may be necessary to complete the study.

Although no detailed economic study is available, the overall simplicity of the Wind Furnace Model Four may provide significant cost savings above the wind electric systems. Through system modeling, careful cost estimates, and wind resource data, a thorough economics analysis can be performed. Of course, the true economics can only be ascertained through construction and operation of the Wind Furnace Model Four.

$$\frac{125}{2175}$$

0-9999 RPM MAX. RANGE  
Designed for 100 RPM - 1500 RPM (10 s<sup>-1</sup> - 150 s<sup>-1</sup>)  
X5% accuracy (±0.5%)

AC device frequency reference - no adverts.  
10 second refresh time



A P P E N D I X    II  
WIND FURNACE IV CANDIDATES

A. Version A

1. Costs
2. Diagram

B. Version B

1. Costs

C. Version C

1. Costs
2. Water Twister with Planetary Transmission
3. Diagram, Proposed Wind Assisted Electric Hot Water System
4. Tower Costs
5. Tower Assembly



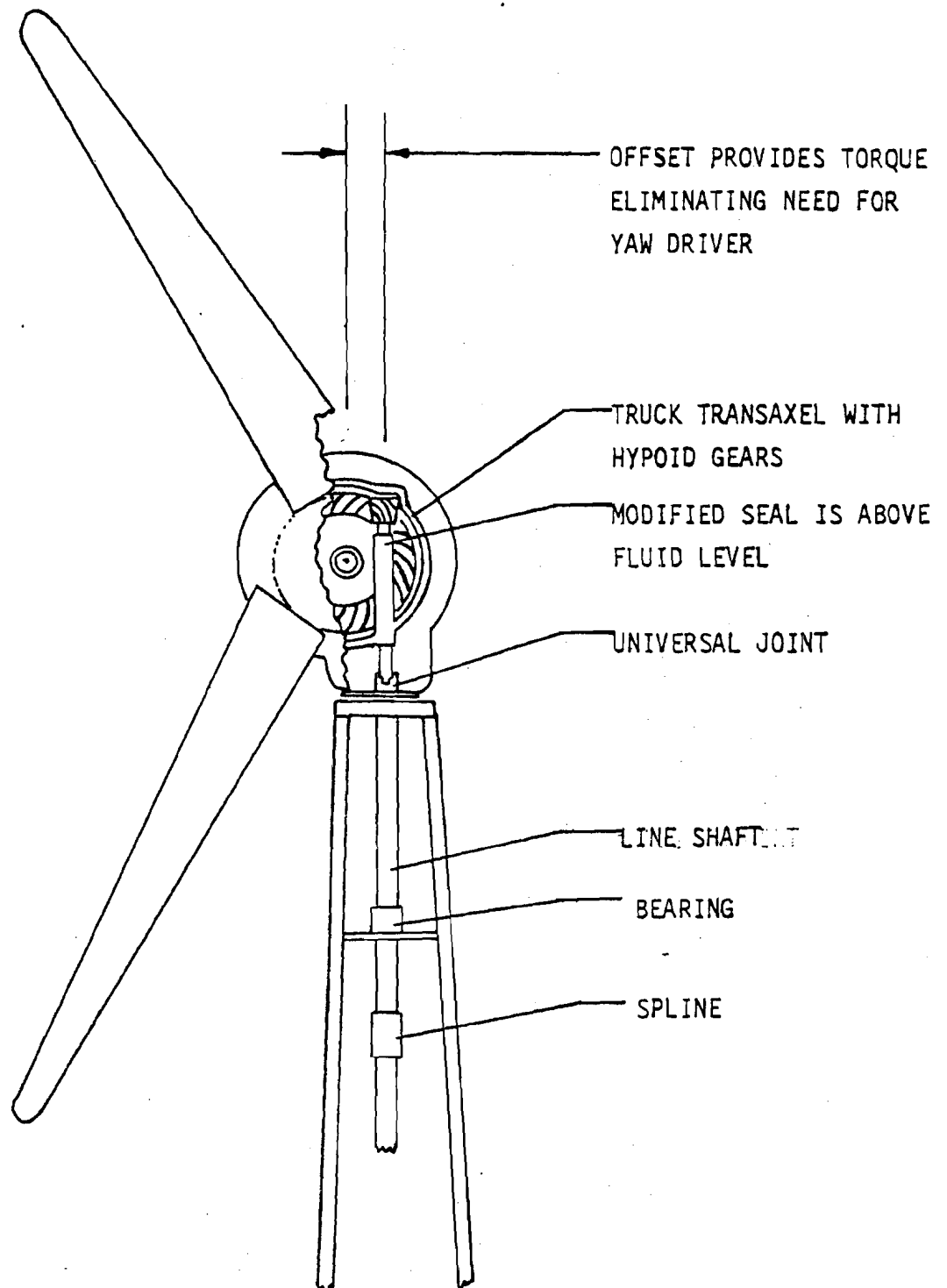
# BILL OF MATERIAL — WIND FURNACE IV

REV. 0

WORKING GROUP WF VERSION A

SHEET NO.

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST 1000
1	REAR AXEL	SPECTER	1	2500
2	UNIVERSAL JOINT	WATSON	10	100.
3	YOKE FOR JOINT	WATSON	20	200.
4	4" INTERMEDIATE CONDUIT	HAMPDEN, SPED	100 ft	325.
5	1/2 T.D. BALL BEARING	LINK BELL	7	140
6	BULKHEAD/HALFWART	SHOF/LINK BELL	7	70
7	SPLINE	SALVAGE	1	10
8	REAR AXEL	JELP	1	300
9	PILLOW BLOCK/BEARING	LINK BELL/AUTO	2	40
10	PUMP OIL	CHAPMAN	2	200
11	WATER TIGHT	THE FARM	1	20
12	FLANGE	LINK BELL	1	10
13	INTERMEDIATE CONDUIT	WATSON	100 ft	325.



WIND FURNACE  
MODEL FOUR  
VERSION A

# BILL OF MATERIAL — WIND FURNACE IV REV. 0

WORKING GROUP WE VERSION 0 SHEET NO. 0

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST 1000
1	REAR AXLE	SPICER	1	380.
2	U JOINT	SPICER	2	17.
3	YOKE	SPICER / ALVAGE	2	17.
4	SHOCK SPRING	HAMPDEN SPRING	4'	12.00
5			1	
6	WATER BRAKE / THERMOSTAT	GENERAL	1	275.
7	12" CPVC	MAG. SUPPLY	100 ft	250.
8	3" CPVC	MAG. SUPPLY	100 ft	400.
9	INSULATION		AIR	100
10	PIPE FITTING		AIR	10
11	CHASSIS	TRUCK	1	100
12	WHEEL / TIRE	TRUCK	1	200
			TOTAL	2417

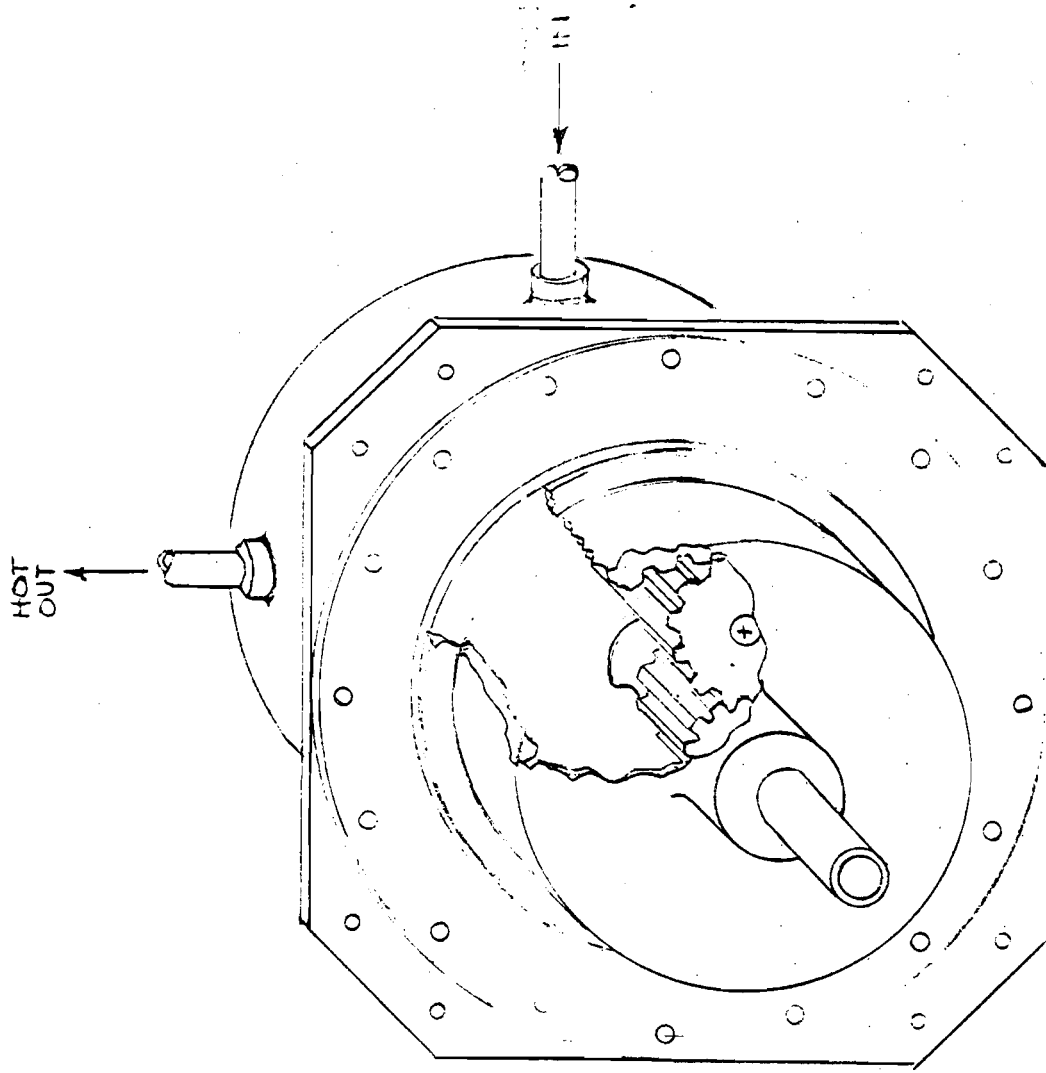
# BILL OF MATERIAL — WIND FURNACE

REV.

WORKING GROUP W F

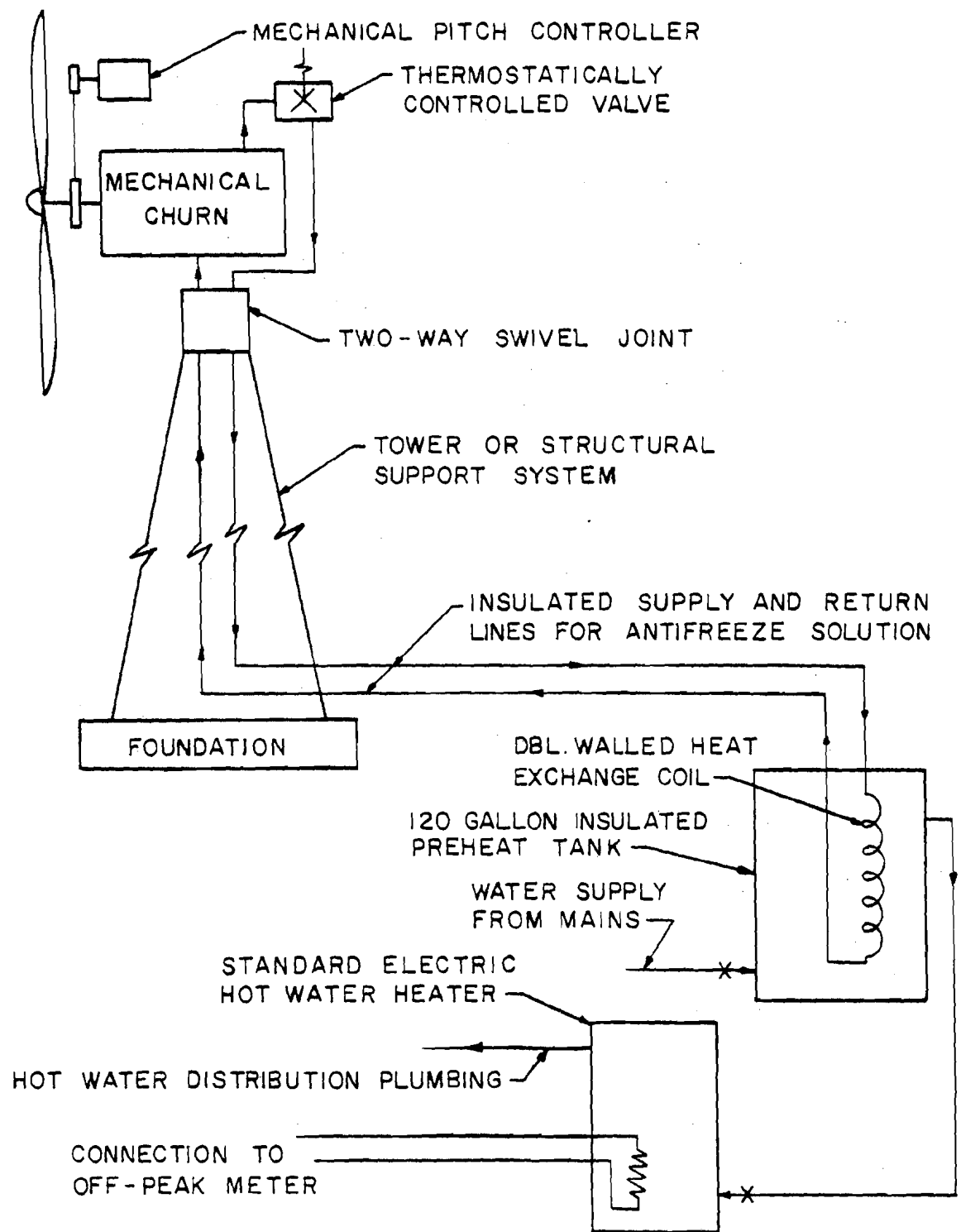
SHEET NO.

[illegible]



WATER TWISTER<sup>R</sup> WITH PLANETARY GEAR SYSTEM

M.R.



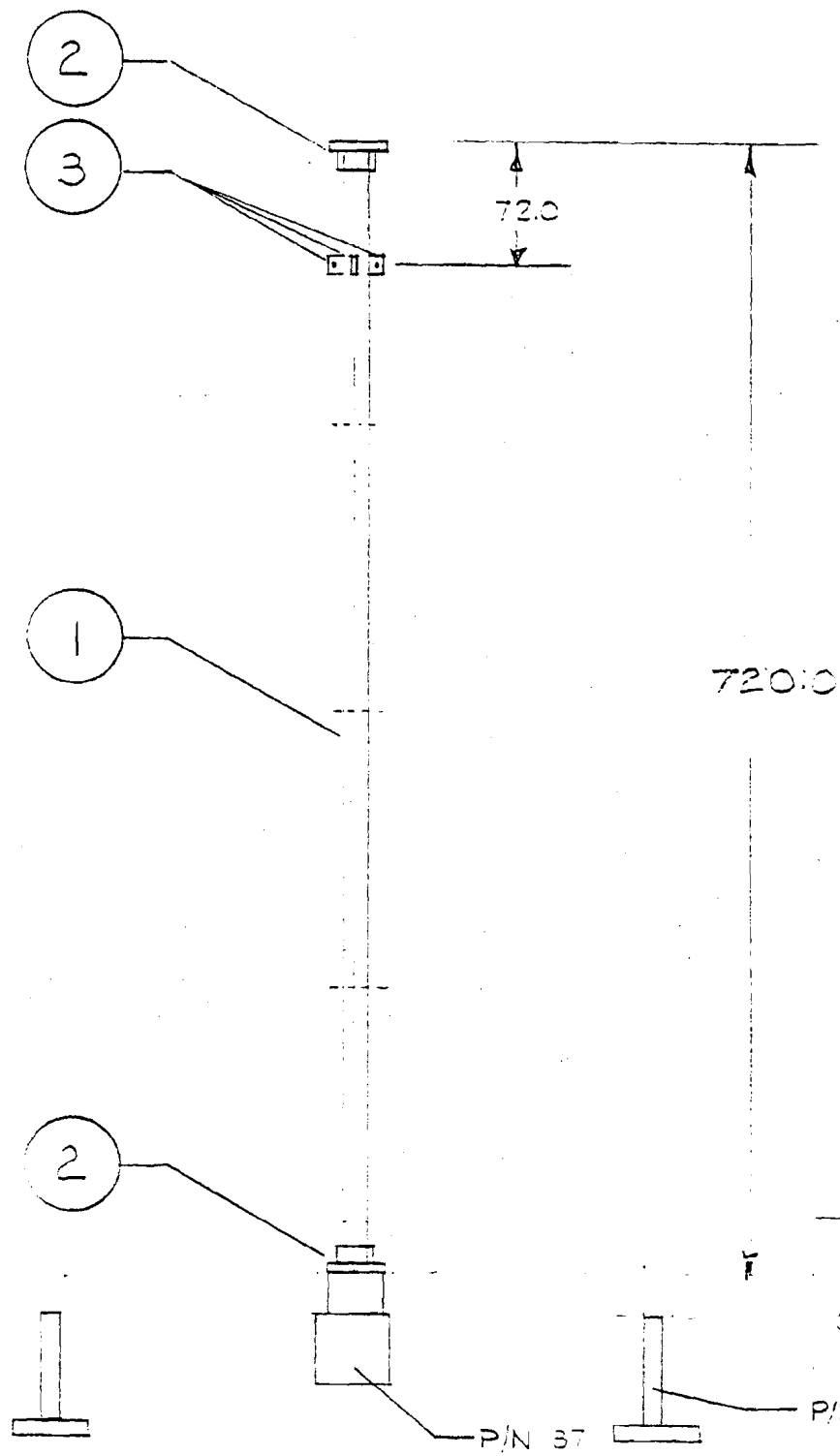
DIAGRAM, PROPOSED WIND ASSISTED ELECTRIC HOT WATER SYSTEM

## REV. \_\_\_\_\_

WORKING GROUP TOWARD AN AGREEMENT

SHEET NO. 14

[illegible]



ENERGY ALTERNATIVE  
PROGRAM  
TOWER ASSEMBLY  
02.1  
NOM 1-21-77 156

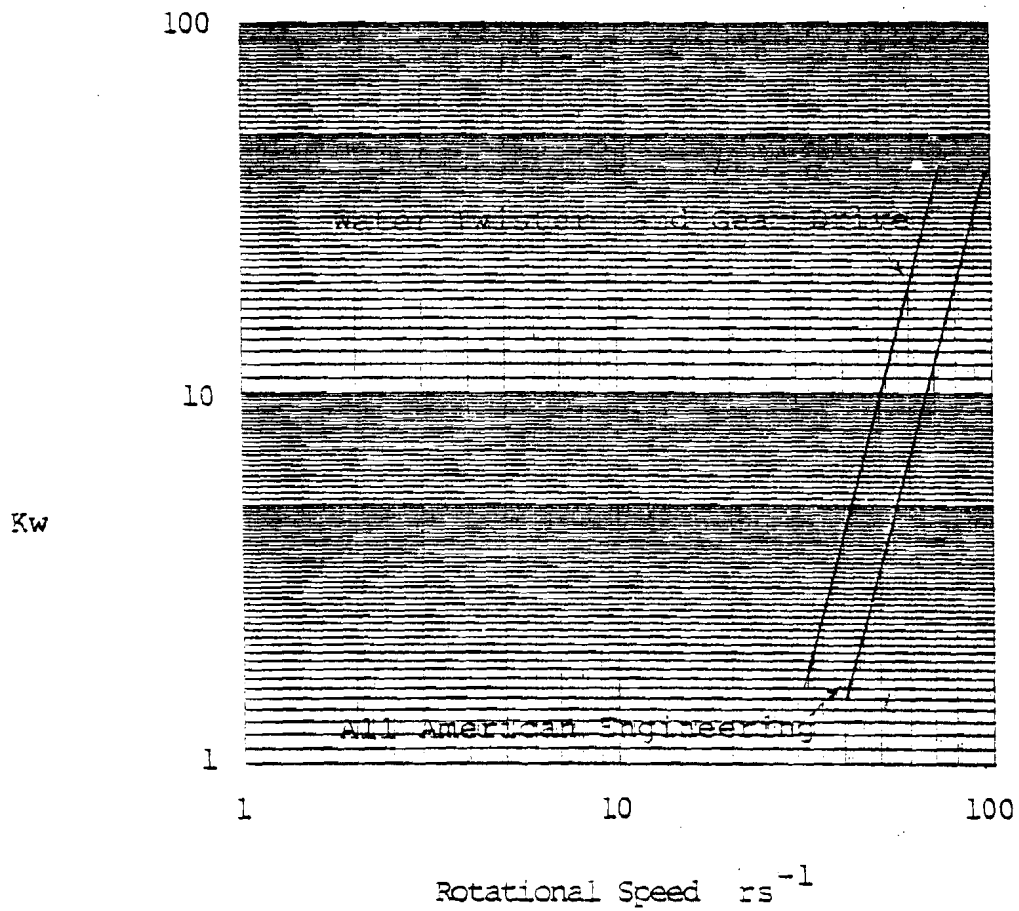


APPENDIX III

WATER TWISTER®

A. Comparison of Test Data For Two Devices of Identical Internal Dimensions.

Water Twister Model 12 Power Curves  
Developed From University of Massachusetts  
Tests and All American Engineering Company  
Published Data



MGR 7-11-79

# APPENDIX IV

## THERMAL EFFICIENCY OF COAXIAL PIPE FLUID CIRCUIT

### A. Conductivity

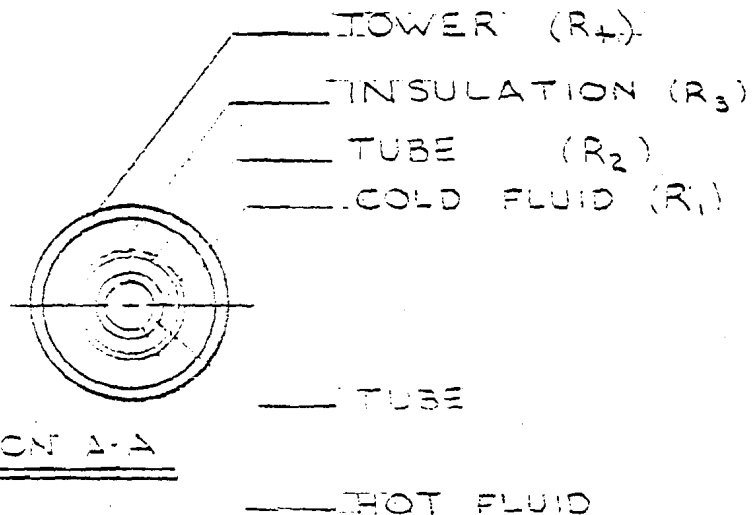
```

00100 PROGRAM MAX(INPUT,OUTPUT)
00110 PRINT 1000
00112 PRINT 2000
00120 DO 10 I=1,35,5
00140 R=(I-1)*.1+1.7
00160 R1=.012/5.38
00180 R2=10*ALOG(1.7/1.5)
00200 R3=50*ALOG(R/1.7)
00220 R4=.5*12/5.38+1/1.35*ALOG(3.0/R)
00240 R5=1/22.6*ALOG(5.38/5.)+12*.25/5.38
00260 U=1/(R1+R2+R3+R4+R5)
00295 PRINT 3000,R,U
00300 10 CONTINUE
00310 1000 FORMAT(5X,*APPROXIMATE VALUES OF CONDUCTIVITY*)
00311 2000 FORMAT(/A,10X,*INSULATION*,10X,*THERMAL*,/12X,*RADIUS*
00312+10X,*CONDUCTIVITY*)
00313 3000 FORMAT(12X,F4.1,12X,F4.3)
00320 STOP
00340 END

```

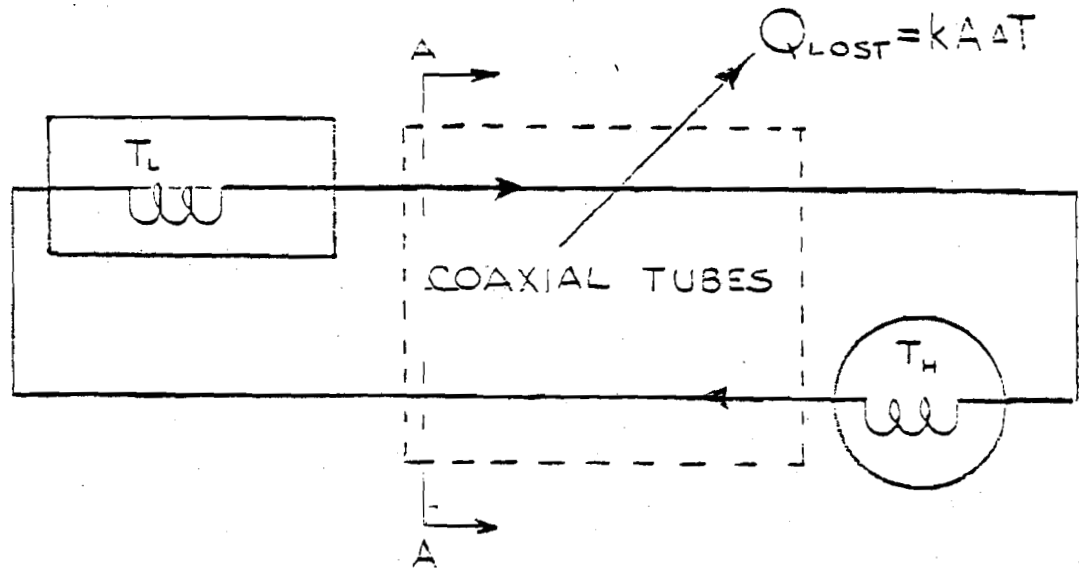
### APPROXIMATE VALUES OF CONDUCTIVITY

INSULATION RADIUS	THERMAL CONDUCTIVITY
1.7	.268
2.2	.061
2.7	.038
3.2	.029
3.7	.024
4.2	.021
4.7	.019



SECTION A-A

B. Thermal Loss In Coaxial Tubing Modeled As A Counter Flow Heat Exchanger



```

00100 PROGRAM WFTHREE (INPUT, OUTPUT)
00102 500 FORMAT (21X, KUX, 8X, KOUTSIDE TEMPX)
00103 1000 FORMAT (18X, F6.4, 8X, F5.1, ///)
00104 2000 FORMAT (12X, *LBM/HR*, 8X, *FINAL TEMP*, 8X, *BTU LOST*, 10X, *%L
00105 3000 FORMAT (12X, F4.0, 10X, F6.1, 12X, F5.0, 11X, F5.1)
00106 M=1
00110 REAL MDOT, L
00120 CP=1.002
00130 PI=3.14159
00140 TA=0.0
00150 TH=200.0
00160 TC=180.0
00180 L=60.0
00190 U=.296
00210 UOUT=.068
00235 PRINT 500
00236 PRINT 1000, UOUT, TA
00237 PRINT 2000

```

(continued)

```

00240 DO 10 I=1,2
00250 DO 20 J=1,2
00260 DO 30 K=2,10,2
00270 MDOT=K*K*K*10.0
00280 IF(MDOT.GT.3000)GO TO 10
00290 E=(2*PI*K*L*U/MDOT/CP)/(2*PI*K*L*U/MDOT/CP+1)
00300 Q=E*CP*MDOT*(TH-TC)
00310 DELTA =Q/MDOT/CP
00320 TDN=TH-DELTA
00330 TUP=TC+DELTA
00340 TLN=((TC-TH)-(TUP+TA))/(ALOG((TC-TH)/(TUP-TH)))
00350 QLOST=UOUT*2*PI*K*L*TLN
00355 CALL PERCENT(MDOT,TH,QLOST,PER,CP)
00357 IF(PER.EQ.100)GO TO 30
00360 PRINT 3000,MDOT,TDN,QLOST,PER
00370 30 CONTINUE
00380 M=10
00390 20 CONTINUE
00400 M=100
00410 10 CONTINUE
00420 STOP
00430 END
01000 SUBROUTINE PERCENT(FLOW,THIGH,QLOST,PER,CP)
01010 Q=CP*FLOW*THIGH
01020 PER =QLOST/Q*100.
01030 IF (PER.GT.100.)PER=100.0
01040 RETURN
01050 END

```

HEAT LOSS DUE TO CONDUCTION FOR WIND  
FURNACE FOUR,  $R = 15$ ,  $T_1 = 119^\circ\text{C}$  AMBIENT,  $T_L = 60^\circ\text{C}$

