SUMMARY

An airscrew windmill has been designed for use in the north east trade wind regime of the Caribbean, which is shown to be particularly favourable for agricultural irrigation purposes.

The high-speed fixed-pitch airscrew incorporates the most recent aerodynamic theory and is manufactured from fibre-glass reinforced epoxy resin. To compete with conventional prime movers high efficiency, low cost and long life are maintained by the use of standard industrial and automotive transmission components.

The strength of the blades and supporting structure has been confirmed by structural proof tests. Field tests, in which the windmill was connected to a standard deep-well turbine pump for sprinkler irrigation, have been made to determine the aerodynamic integrity of the airscrew design.

The detailed performance characteristics obtained over an extended operating range provide a guide for wind pumping performance prediction purposes in water resource development programmes in the Caribbean and elsewhere.

R.E. Chilcott

Master of Science

Agricultural Engineering

SHORT TITLE

THE DESIGN DEVELOPMENT AND TESTING

OF A LOW-COST 10 HP WINDMILL

R.E. CHILCOTT M.Sc.

THE DESIGN DEVELOPMENT AND TESTING OF A LOW-COST 10 HP WINDMILL PRIME MOVER

by

Richard Edward Chilcott

Thesis Submitted to the
Graduate Faculty in Partial Fulfillment of
The Requirements for the Degree of
MASTER OF SCIENCE

Department of Agricultural Engineering
McGill University

July 1969

SUMMARY

An airscrew windmill has been designed for use in the north east trade wind regime of the Caribbean, which is shown to be particularly favourable for agricultural irrigation purposes.

The high-speed fixed-pitch airscrew incorporates the most recent aerodynamic theory and is manufactured from fibre-glass reinforced epoxy resin. To compete with conventional prime movers high efficiency, low cost and long life are maintained by the use of standard industrial and automotive transmission components.

The strength of the blades and supporting structure has been confirmed by structural proof tests. Field tests, in which the windmill was connected to a standard deep-well turbine pump for sprinkler irrigation, have been made to determine the aerodynamic integrity of the airscrew design.

The detailed performance characteristics obtained over an extended operating range provide a guide for wind pumping performance prediction purposes in water resource development programmes in the Caribbean and elsewhere.

RESUME

Une éolienne a été conçue pour être utilisée en fonction des vents alizés du nord-est pour fins d'irrigation aux Antilles.

L'hélice est du type à pales fixes et à haute vitesse de rotation. Sa conception est basée sur les théories aérodynamiques les plus récentes et elle est fabriquée en fibre de verre renforcé d'une résine époxy. Afin de pouvoir faire concurrence aux autres formes conventionnelles d'énergie, le mécanisme de transmission utilise les éléments d'une transmission de camion.

La résistance des pales de l'hélice et de la tour qui la supporte a fait l'objet de maints essais. Des épreuves sur le champ, alors que l' éolienne était raccordée à une pompe à turbine pour puits profonds utilisée pour fins d'irrigation, ont été effectuées afin de vérifier le rendement actuel de l'éolienne.

La performance obtenue sous différents régimes éoliens sert de guide à l'utilisation de cette machine lors du développement des ressources hydrauliques aux Antilles et ailleurs.

ACKNOWLEDGEMENTS

The major part of the work described in this thesis was carried out in Barbados, West Indies, in an effort to develop a low-cost power source for developing areas as part of the wind power utilization research and development programme of the Brace Research Institute of McGill University.

During the design, development and testing phases of the project the author assisted his Research Supervisor, Dr. G.T. Ward, Director, Brace Research Institute, and was responsible for engineering the prototype airscrew windmill pumping system. The author was subsequently assisted by Graduate Student John Ionson and his Research Supervisor Professor R. S. Broughton, Chairman, Department of Agricultural Engineering of McGill University. Thanks are due Dr. Ward, Professor Broughton and John Ionson for their advice, enouragement and assistance. The author is also indebted to many Barbadians for their enthusiastic support and would particularly like to thank Allan Robinson, Davidson Ward and Eric Foster, of Brace Experiment Station workshop staff, Wendel Cummins of the McGill High Altitude Research Project, Christ Church, and Edward Cumberbatch of the Barbados Ministry of Agriculture, Bridgetown.

In the preparation of this thesis the author would like to record his appreciation of the assistance of Brace and Agricultural Engineering Staff and Graduate Students at Macdonald College of McGill University. Particular thanks are due to Robert Weyts who was responsible for the complete set of drawings for the project. The author would also like to thank Edward Lake for photographic assistance, Saeed Kadivar for computing assistance, Mrs. Jackie Morrow and Miss Rona Diamond for typing the manuscript, and William Alleyne and Rex Wailes for Figures 2 and 3.

CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

$C_L = C_L(\alpha, Re) = \dfrac{L}{\frac{1}{2}\rho V^2 A}$	Lift coefficient.
$C_D = C_D(\alpha, Re) = \dfrac{D}{\frac{1}{2}\rho V^2 A}$	Drag coefficient.
$C_{MO} = C_{MO}(\alpha, Re) = \dfrac{MO}{\frac{1}{2}\rho V^2 A . d}$	Pitching moment coefficient about aerodynamic centre.
$Cp = Cp(\mu) = \dfrac{P}{\frac{1}{2}\rho V^3 S} = \mu Cq$	Power coefficient.
$Cq = Cq(\mu) = \dfrac{Q}{\frac{1}{2}\rho V^2 S R}$	Torque coefficient.
$\mu = \mu(\sigma, \beta) = \dfrac{R\Omega}{V} = \dfrac{2\pi Rn}{60V}$	Tip-speed ratio.
$v = \dfrac{V}{\bar{V}}$	Normalised wind speed.

t	Per cent time.
L	Aerodynamic lift.
D	Aerodynamic drag.
MO	Pitching moment about aerodynamic centre.
P	Windshaft or airscrew shaft power.
Q	Windshaft or airscrew shaft torque.
ρ	Atmospheric density.
V	Wind speed.
$\bar{V}$	Mean wind speed.

A	Blade area.
S	Swept area.
c	Blade chord.
x	Chordwise centre of gravity position per cent.
Ω	Rotor shaft speed, radians per second.
n	Pump shaft speed.
G	Overall gear ratio.
B	Number of blades.
E	Young's modulus of elasticity.
I	Least inertia of blade section.
Re	Reynolds number.
$St = St\ (\alpha, Re)$	Strouhal number.
α	Angle of incidence of blade section.
β	Blade setting angle at 0.7R.
γ	Cone angle of airscrew.
η	Efficiency.
$\sigma = \frac{A}{S}$	Solidity.

<u>Subscripts</u>

C	Cut-in.
D	Design.
E	Maximum power-frequency.
G	Gust.
I	Irrigation. Ideal.
M	Month.

P	Pump.
R	Rated.
T	Transmission.
Y	Year.

THE DESIGN DEVELOPMENT AND TESTING OF A LOW-COST 10 HP WINDMILL PRIME MOVER

1. INTRODUCTION

1. 1. General Design Requirements

In many parts of the world, especially in small communities in isolated rural areas, development is often limited by the cost of conventional sources of power for water lifting and the production of fresh water, Golding (1). Often the pumping power requirement can be met by the use of a prime mover such as a diesel engine or electric motor producing about 10 horsepower. However, the diesel engine has certain disadvantages as a prime mover in developing areas where the recurrent costs of maintenance, fuel and lubricant are high and where there is generally a lack of sufficient qualified technicians. These limiting factors often imply that the supply of electricity is both expensive and unreliable. A close study by Ward (2) of alternative energy sources shows that in many areas special local conditions can often combine to make the use of a locally available source of energy, such as the sun or wind, technically feasible and economically competitive with more conventional sources of power. In certain areas the utilization of wind power can be regarded as a relatively convenient indirect use of solar energy.

The two inherent characteristics which dominate the problem of wind power utilization are the low density of air - about 800 times less dense than water - and the unpredictability of wind. Vadot (3) states that this implies that wind machines must be of a relatively large size for a given power capacity, making design economy of major importance since, although wind is freely available, the cost of generating useful power depends very much on the cost of the machine, its maintenance and its life. Also, wind power is unsuited to the supply of firm power for general purposes unless arrangements are made for energy storage. This is an expensive business, so that the most acceptable types of utilization in agriculture are those adaptable to the irregularity of energy production. Among such loads capable of being met in this way are water pumping and desalination, certain types of agricultural processing and small scale electricity generation. Caribbean, English, Dutch, Danish and Russian windmills (4,5,6,7,8) provide evidence of these traditional uses of wind power and illustrate the potential for development when appropriate conditions exist. At certain sites in many developing areas of the world favourable conditions for wind power utilization exist, Tagg (9), provided that the initial cost of the wind machine is sufficiently low, Lawand (10).

In order to compete with conventional sources of power a windmill should therefore be designed to be built as cheaply and simply as possible, consistent with strength and reliability, work at maximum efficiency and, if possible, be located to take advantage of local topography in order to make maximum use of the wind power available. In an effort to satisfy these requirements and meet the need for a rugged 10 hp power source for developing areas with suitable wind regimes, a low-cost windmill which makes use of recent aerodynamic theory and modern materials has been designed, constructed and tested.

1. 2 . Prototype Brace Airscrew Windmill Pumping System

The prototype airscrew windmill, Fig. 1, has been designed primarily for use in the north east trade wind regime of the Caribbean which is shown to be particularly favourable for water lifting and agricultural irrigation purposes. The fixed-pitch airscrew design incorporates the most recent aerodynamic theory and high efficiency is obtained by forming the twisted, variable-chord aerofoil-section blade shape from fibre-glass reinforced epoxy resin, a manufacturing method which gives scope for local production in developing areas at a later stage. To maintain high efficiency at low cost the shaft drive down the tower incorporates standard industrial and automotive transmission components, which have long life, require minimal maintenance and are available cheaply in many countries. Similarly, the supporting structure utilizes locally available standard steel sections.

The strengths of the blades and supporting tower have been confirmed by structural test and field tests have been made to determine the aerodynamic integrity of the airscrew design, the detailed performance characteristics having been obtained over an extended operating range. For the purposes of these tests the windmill was connected to a commercially available deep-well turbine-type pump for agricultural irrigation in a cooperative programme with the Barbados government. A technical note by the author (11) gives a brief outline of the scheme, and long-term tests to prove the technical and economic reliability of operation of the windmill as a power source for sprinkler irrigation water pumping are described by Ionson (20). The results of these complementary research projects provide a guide for wind turbine design and performance prediction purposes in water resource development programmes in the Caribbean and elsewhere.

1. 3. Extension Programme

The basic design, with several pump and load options, and detail revisions for batch production in Canada, is intended to power demonstration wind-pumping water development projects in chronic-need locations, financial aid being provided by a funding agency such as Canadian International Development Agency (12). In the next stage of development such windmills will be manufactured and assembled locally in small agricultural engineering workshops, initial capital and technical assistance being provided by an appropriate agency. In the Caribbean such an embryo specialist agricultural machine industry would be based on the immediate local demand for low-cost power and on export opportunities for wind turbines in Central and South America (13). A design study of a version of the basic design suitable for temperate and arctic wind regimes is at present underway.

2. RATED AND DESIGN WIND SPEEDS FOR THE CARIBBEAN TRADE WIND REGIME

2. 1. General

The towers of the old Dutch-type sugar cane crushing mills in several of the Windward and Leeward islands, such as Barbados, Fig. 2, Nevis and Antigua provide ample evidence of the prosperity of the windmill era of the 18th and 19th centuries before the introduction of steam power led to the eventual decline of the old cane mills during the first half of the present century. The turn of the century marked the introduction of the multi-bladed wind pump for water lifting, replacing the traditional windmill used to pump water for the estates and sugar factories, Fig. 3.

Slow running wind wheels are a familiar feature of the West Indian landscape and commonly used for domestic water lifting, overhead irrigation of small vegetable plots and for stock watering. Such windmills, typically 8 to 16 feet in diameter and coupled to deep well piston pumps, are characterized by a rugged reliability. With a high starting torque they provide relatively low but fairly firm power in gentle breezes of about 5 to 15 mph, with corresponding low overall efficiency at high windspeeds. Although fan mills are no longer manufactured in Canada, they are still made in the U. K. and the U. S. A. Their design, performance and economics have been thoroughly analyzed by Golding (14) and Vadot (15).

In the past few years relatively high-speed small-scale airscrew windmills of about 1 kw capacity have been used for the generation of low-cost electrical power in isolated areas. More recently larger higher-powered three-bladed airscrew windmills have been introduced such as the Russian designed 15 kw Sokol machine in Cuba and the Brace 10 hp airscrew windmill pumping system in Barbados. These larger machines run at high speed to drive machines with relatively low torque, and high shaft-speed requirements, with a minimum of step-up gearing. They have a much larger scale, higher efficiency and greater capacity than the conventional water pumping wind wheel, and in contrast they are designed to utilize the upper part of the wind power spectrum and work at optimum overall efficiency. The justication for the choice of rated wind speed and the more critical design wind speed for such wind energy converters is outlined in the following historical and climatological reviews.

2. 2. Molinological Basis for Rated Wind speed

2. 2. 1. Georgraphy of Barbados

Barbados is a small West Indian island, Fig. 4, where the main industries are sugar, rum and tourism. The coral island is pear shaped, about 24 miles long and 18 miles wide, with a variety of scenery ranging from the hilly Scotland district in the north-east to low rolling countryside reminiscent of southern England around Bridgetown, the capital, in the south-west. The maximum elevation is 1,100 feet and about one third of the area is a smooth upland plateau above 600 feet. Tilting and erosion have given rise to a scarp ridge around the Scotland district, a rugged east coast pounded by Atlantic rollers and fine coral sand beaches with relatively calm seas on the Caribbean west coast— a contrast emphasized by the prevailing north-east trade winds which provide an equable 80^{o}F climate with inherent air conditioning. The island is covered with a thick green carpet of ripening sugar cane cut into a patchwork by the grey asphalt road system and peppered with old windmill towers, sugar factories and plantation houses.

In view of its recent windmill tradition the island offers ideal conditions for wind power plant testing and the evaluation of wind as a renewable resource in competition with more conventional sources of power by which wind power has been largely superseded.

2. 2. 2. Barbadian Cane Crushing Windmills

With its favourable topography, dependable wind regime and the independent outlook of the local planter, Barbados remained a windmill stronghold long after other islands converted to more modern plant. In particular, the 19th century was a period of intense windmill activity during which millwrighting became a highly developed art. The last traditional cane crushing windmill to cease production was Morgan Lewis in 1947, while the Kirton water lifting windmill finally stopped in 1954, Fig. 1 and 2.

When contemplating the design of modern windmills for the Caribbean trade wind regime, it is pertinent to consider some design and performance features of the traditional Barbadian windmills. Dash has outlined the evolution of the windmill in Barbados from the introduction of mills with vertical crushing rolls in the second half of the 17th century to the larger horizontal-roll mills, the remains of which can be seen today and which were the forerunners of the modern multiple-roll sugar-factory plant. The height of the coral stone and rubble windmill towers varied between 50 and 30 feet and the point or sail length generally varied from about 43 feet to 25 feet; the sail width being about one fifth of the point length. In order to extract more power, tower height and sail diameter were increased; Drax Hall mill, with a sail diameter of 93 feet and a tower height of 55 feet, is an example of the larger 19th century windmills. It is interesting to note that performance was sometimes inadvertently handicapped by hills to the east of a mill and that very long points had a tendency to whip at the ends rendering them liable to damage—at Alleynedale, sails had to be shortened and an access mound made to windward, while at Portland, an excavation was made to allow clearance for lengthened sails. In general, clear, unobstructed sweeps of open country were required for good performance.

Power was transmitted from the windshaft to the set of three horizontal cane-crushing rollers by means of a vertical shaft and step-down crown wheel and pinion pairs at the top and bottom of the tower. Control was achieved by adjusting the cane load on the rolls and by turning the points in and out of wind by hand from the ground by means of a tail-tree projecting from the roundhouse or cap mounted on a collar at the top of the tower. Braking was achieved by choking the rollers with cane. The crop season started with the beginning of reaping in December or January and lasted until the end of the dry season in June or early July. There is evidence that mills were used at night although generally a 12 hour day shift was worked. The design of various component parts had to be coordinated strength and weight-wise as exemplified by high-torque transmission shafting and the static balance required between sails and tail-tree.

Considering the results obtained by these windmills, Dash (4) calculates that the average crushing capacity of a good working mill in a favourable wind was 4 to 5 tons of cane per hour. In order to determine the horsepower required, Longmuir (16) of Fletcher and Stewart, Derby, a firm which used to make windmill powered cane crushing plant, suggests the use of Parr's formula:

$$IHP = 0.3 \cdot f \cdot R^{0.45} \cdot TCH,$$

where IHP = Input horsepower at tail bar to top roller,

f = Per cent fibre on cane, normal range 11-17 per cent,

R = Number of mill rollers,

TCH = Tons of cane per hour.

Assuming a mean value of fibre on cane to be 14 per cent and a three-roll mill, IHP = 6.9 TCH; the shaft speed, depending on roller diameter, would be of the order of 1 rpm. The input power required for a maximum capacity of 5 TCH is therefore about 34.5 hp. Considering the probability of incomplete crushing to avoid choking the mill and stalling the sails, a more general input horsepower requirement might have been 20 to 25 hp.

Considering the class of equipment and the probable poor operating conditions with respect to lubrication the power-transmission losses between the points and top roller tail bar would probably have been quite high. Smith (17) suggests losses of 10 to 15 per cent would be expected for each pair of bevel gears and approximately 3 per cent per bearing. This would indicate a gearing efficiency of 80 per cent and a bearing efficiency of 75 per cent, giving a transmission efficiency of about 60 per cent. The windshaft horsepower required for 5 TCH would therefore be about 57.5 hp. The larger mills such as Drax Hall, Alleynedale and Portland were quite likely capable of meeting this power requirement, although the average windmill capacity may well have been somewhat lower, say 30 to 40 hp.

2. 2. 3. Dutch Drainage Mills

The power required by the Barbadian mills appears to be similar to windshaft-power measurements made on Dutch drainage mills by Von Baumhauer (1926) and Hoekstra (1929) who, by measuring brakewheel tooth pressure, found half hourly windshaft powers in the range of 20-50 hp.

The 'Prinsenmolen' Committee has published the records of these and other thorough field performance tests of Dutch drainage mills coupled to scoop wheels and Archemedian screw pumps (18). Probably the most significant of these tests are those carried out on the windmills of the Overwaard drainage district and Havinga's 1931 measurements (19) of scoop wheel efficiency, which was found to be about 0.50. The group of eight Overwaard mills were of 28.5 meter (93.5 ft) diameter and coupled to scoop wheels through step-up gearing of about 1:1.8. During long-term tests in 1937, it was observed that the millers thought it advisable to partially furl the sails when the average wind speed at an anemometer height of 20 meters exceeded 8.5 m/s (19 mph), a figure which gives a guide to the maximum acceptable power or rated wind speed. They preferred to stop the mills when the average wind speed approached 10 m/s

(22 mph) and dark skies threatened showers with heavy gusts. It was also considered unsafe to run at an average wind speed greater than 11 m/s (25 mph), especially at night. The normal operating windshaft speed varied between limits of 15 and 25 rpm, with an average of about 18 or 19 rpm, and an extremely violent never-exceed speed of 30 rpm in squalls.

Further information on the capacity and operating characteristics of Dutch drainage windmills with scoop wheels is provided by Stokhuyzen (5) who states that in full sail a mill would generate about 50 hp at between 20 and 22.5 rpm. The maximum rpm was generally limited by furling to about 25 rpm, above this speed there was a risk of the mill running away if the water supply failed and the scoop wheel was unable to load the mill sufficiently. In industrial mills it was usually possible to increase the load to allow occasional peak powers of between 75 and 100 hp. Referring to the Beaufort scale (7) it is concluded that these windmills would start in a gentle to moderate breeze above 10 mph and were designed to run in moderate to fresh breezes between 13 and 24 mph, the mean rated wind speed being about 18.5 mph. Typical operating characteristics of mills with sail span between 75 and 95 feet are quoted by Stokhuyzen and summarized in Table 1.

In 1939 the Prinsenmolen Committee carried out some wind gradient measurements at the original Overwaard anemometer site and found $V_6 = 0.82\ V_{20}$ which, assuming a 1/nth power relationship between wind speed and height, gives $n = 6.05$. This is more nearly a 1/6th power relation than a 1/7th and enables the 20 meter wind speed results of 1937 to be referred to a standard height of 10 meters, $V_{10} = 0.892\ V_{20}$,

TABLE 1. Operating characteristics of Dutch drainage windmills

Characteristic wind speed	Typical limits of average wind speed (approx.)		Beaufort scale of wind force	
	m/s	mph	Number	Description
Starting or cut-in	5-6	11-13.5	3-4	Gentle-moderate breeze
Normal operating or rated	8-8.5	18-19	4-5	Moderate-fresh breeze
Partial furling or reefing	10	22.5	5	Fresh breeze
Complete furling or bare poles	12	27	6	Strong breeze
Shut down	Miller's discretion		7	Moderate gale

for the estimation of the rated wind speed and the effective aerodynamic efficiency, or power coefficient, of the Overwaard windmills, the 10 meter rated wind speed then becomes 17 mph. Table 2 shows that the windshaft power is in the range 25 to 45 hp, and that the mean three-hourly value of Cp_{10} is about 0.20. These results are in accord with Von Baumhauer's full scale and model tests of 1926 and Smeaton's model tests of 1759 cited by the 'Prinsenmolen' Committee (18). They indicate that in the case of these mills, variations of wind speed with time tend to balance the variation of wind speed with height and justify, for power prediction purposes, the use of steady state efficiencies and hourly mean wind speeds at some reference height. It is therefore apparent that wind speed should be referred to rotor centreline height or more conveniently, taken as the standard meteorological anemometer wind record height of 10 meters, at least when the centre-line height is within about one meter of the standard anemometer height.

2. 2. 4. Molinological Rated Wind speed

In order to estimate the rated wind speed of the Barbados cane mills the similarity of windshaft power and sail scale and geometry suggests that the aerodynamic efficiency would be the same as that of the Dutch mills. Taking a power coefficient Cp of 0.2, Drax Hall mill with a diameter of 93 feet developing 57.5 windshaft hp at an atmospheric density of 0.002285 slugs/ft^3, would require a mean rated wind speed of about 18.5 mph. This 10 per cent increase over the 17.0 mph calculated for the Overwaard mills might be considered due to the relative smoothness of the more favourable trade wind regime, which would

TABLE 2. Performance tests on Dutch drainage windmills, Overwaard, 1937.

No.	Run time hrs	Average wind speed at 20 m V_{20} m/s	Water horsepower WHP	Estimated wind speed at 10 m V_{10} m/s	$\frac{WHP}{V_{10}^3}$	Estimated windshaft power HP	Estimated power coefficient Cp_{10}
16	3	8.7	13.3	7.76	0.0285	44.3	0.183
17	3	7.6	9.8	6.78	0.0316	32.6	0.203
18	3	6.9	7.0	6.15	0.0300	23.3	0.193
19	3	7.5	10.6	6.69	0.0355	35.2	0.228

Assumptions: $V_{10} = 0.892\ V_{20}$,

Scoop wheel efficiency 0.50,

Transmission efficiency 0.60,

Atmospheric density 0.002378 slugs/ft^3,

Sail span 28.5 m (93.5 ft).

allow the Barbadian mills to run very well in moderate and fresh breezes of the Beaufort scale. Most windmills were in fact built in open situations above 200 feet where the annual average wind speed would probably be in the range 12.5 to 15.0 mph, so that the rated wind speed and power would generally be significantly greater than the annual average windspeed and the power corresponding to the average wind speed. At exceptionally good sites, such as the scarp rimming the Scotland district along which the mean annual wind speed approaches 20 mph at Springhead smaller diameter sails were used for the same power; the strenth of the materials available, crushing equipment size and capacity and availability of cane all being limiting factors.

2. 3. Climatological Basis for Rated and Design Wind Speed

2. 3. 1. General

When designing a modern medium-power wind turbine for intermittent agricultural duties it seems reasonable, in the absence of more reliable information, to rate the machine on the historical basis of earlier successful operation. A more rigourous approach to rated wind speed is through the analysis of relevant climatological data at any given site. In the case of a standard windmill prime mover to be used for a variety of loads at different sites the useful power output will be a function of several variables, such as seasonal wind strength and frequency and agricultural irrigation, drainage or processing requirements. A design wind speed is therefore introduced at which the prime mover, driven machine and load are best matched for a particular requirement. Generally the overall efficiency of the system will be at or near a maximum at the design wind speed. With careful load and component matching and selection a high over-all efficiency can be maintained over the working wind speed range, although slight reduction in efficiency is inevitable at the extremes of the range. Analysis of the seasonal wind power available and its frequency in relation to the load requirements allows the rated wind speed and design envelope to be defined and also enables the design wind speed to be selected for a given application.

2. 3. 2. Barbados Weather

Skeete (21) provides a vivid account of the weather in Barbados from 1901 to 1960, giving month by month descriptions of the wind and rainfall patterns to be anticipated in this and similar West Indian islands. Discussing long term climatic changes Lamb (22) points out that strengths of the trade winds over the last 100 years show a climax of vigour within the first 40 years of this century, which might be argued to have retarded the demise of the traditional cane crushing mill and aided the introduction of the wind wheel water pump in the area. Reliable long term forecasts are unavailable, due to a lack of knowledge of the radiation absorbing properties of atmospheric pollutants and other unpredictable man-made changes. It is therefore assumed that the trends indicated by Skeete are generally applicable to the more recent data available from such sources as Seawell Airport, Springhead, Greenland and the McGill Climatology Programme in Barbados (23).

Skeete classifies the weather of Barbados as "seasonal" with broad divisions into "dry" and "wet" seasons and "cool" and "warm" seasons. The cool season normally lasts from December to February while the warm season varies from year to year, the warmest months being those with low rainfall and high sunshine. The annual average rainfall for the island is about 60 inches, 23 per cent of this total falls during the dry season from the middle of December to the end of May while 77 per cent falls during the wet season from June to the middle of December. The annual average wind speed for Codrington, Fig. 5, is about 10 mph, Table 3. Comparison of mean monthly wind speeds at Codrington and mean monthly rainfall for the island shows that the dry season can be regarded as relatively windy, while the wet season is relatively calm Table 3. The wind power-rainfall ratio is shown graphically in Fig. 7, together with the wind power-rainfall deficit ratio for the months of February, March, April and May during which irrigation is most needed. Although this relationship needs refining to take into account local wind speed, rainfall and evapotranspiration, the March minimum serves to emphasize that, although wind speed and rainfall are broadly complementary on a seasonal basis, it is necessary to take account of the crucial power-deficit month and to maximize the power available by taking advantage of local topography if at all possible.

A review of the lowest recorded average for each month for the Codrington 58-year monthly averages shows that the normalized mean March wind speed of $v = 1.14$ is relatively reliable. Skeete points out, however, that from the middle of December to the end of March there are often one or two short periods of two or three days during which there are calms or

TABLE 3 Barbados wind speed and rainfall

Sources: Skeete (21), Best (27), Ionson (20).

Dry season month	Jan	Feb	Mar	Apr	May	June
Codrington wind speed (mph)	11.0	11.3	11.6	11.4	11.8	12.6
Annual average = 10.2 mph						
Island rainfall (inches)	3.37	2.20	1.93	2.34	3.21	5.41
Annual total = 60.08 inches						
Seawell wind speed (mph)	14.4	15.1	14.25	15.6	15.2	16.4
Annual average = 14.0 mph						
Seawell rainfall (inches)	2.46	1.86	1.34	1.83	2.73	4.24
Annual total = 48.85 inches						
Greenland wind speed (mph)	12.8	12.25	10.0	9.7	15.1	13.7
Annual average 1967 = 11.3 mph						
Greenland rainfall (inches)	2.74	2.00	1.16	2.43	3.12	4.98
Annual total = 60.19 inches						

Wet season month	July	Aug	Sept	Oct	Nov	Dec
Codrington wind speed (mph)	11.3	8.8	7.5	7.3	8.1	9.9
58-year period 1903-1960						
Island rainfall (inches)	6.26	7.31	7.65	7.89	7.74	4.77
114-year period 1847-1960						
Seawell wind speed (mph)	15.0	12.75	11.4	12.3	11.8	13.6
10-year period 1953-1962						
Seawell rainfall (inches)	5.86	6.00	6.37	6.64	5.61	3.91
25-year period 1943-1967						
Greenland wind speed (mph)	13.3	12.5	6.6	8.3	9.7	10.8
1-year period 1967						
Greenland rainfall (inches)	6.80	6.07	6.34	7.16	6.61	4.70
11-year period 1955-1966						

winds of low velocity and variable direction. This, together with the inevitable occurrence of below-average monthly wind speed and rainfall, implies that careful management and provision of emergency auxillary power input are essential for a viable wind pumping irrigation system.

On exceptional days, the wind speed between 6 a.m. and 6 p.m. averages 20 to 24 mph and between 6 p.m. and 6 a.m. 15 mph; during these days gusts of up to 30 mph are experienced. The role of momentum exchange in producing such variations in wind speed over a heated island has been reviewed by Garstang (24) who presents diurnal variations of summer wind speed for Seawell, Springhead and Brace; similar variations are evident in Nevis (25). The influence of convection on wind frequency distribution is more pronounced the lower the base wind speed, as indicated by the extreme diurnal variation at Piarco Airport, Trinidad and evident in the early dry season wind records for the windmill test site at Greenland, Barbados Fig. 5.

Reviewing Skeete's maximum anticipated wind speeds and prevailing wind directions; Maximum Wind Speeds - Gusts of 35 to 50 mph are experienced occasionally, while gusts of 50 to 60 mph are infrequent although hurricane force winds in excess of 100 mph are a distinct possibility. During hurricane Janet in 1955 wind speeds of 110 mph were estimated in southern coastal districts of Barbados. Prevailing Wind Directions - Throughout the dry season, the period from the middle of December to the end of March, the wind is regularly from approximately ENE; veering from a NE tendency in January more towards E in March. In April and May the direction is mainly E, shifting frequently to ESE;

while in June and July the direction usually varies between ENE and ESE.

2. 3. 3. Annual and Monthly Wind Speeds

In studying wind data in relation to the generation of electricity by wind power Tagg (9) presents daily, monthly and annual average wind speeds from a variety of sources for a world wide selection of stations and emphasizes that considerable care is required in the assessment of local wind power potential from existing long term wind records taken from airport and agrometeorological stations. This is especially true in marginal and developing areas where topography, building and plant growth often exert a significant influence on wind records which are in any case not necessarily representative for wind power prediction purposes at sites where, say, irrigation is required. The only completely safe and reliable way to eliminate such uncertainty at a given site is to obtain intensive short term wind records at a standard anemometer height of 10 meters, or at the proposed airscrew centreline height, and correlate local wind speed measurements with synoptic and relatively long term measurements at a nearby reference, such as an airport weather station.

For Barbados, Fig. 5, Seawell Airport represents a convenient reference station which has the advantage of being a representative wind power utilization site. Table 3 gives the mean monthly 10 meter wind speed for Seawell taken over a period of 10 years, 1953-62. The results are derived from spot hourly wind speeds taken throughout 24 hour periods from a Bendix-Friez Aerovane airscrew type anemometer. Anderson (26) suggests that when fairly strong winds and relatively few calms occur at a station visual estimation of indicated wind speed by an observer may show a greater monthly total wind mileage than that recorded by a cup counting or contact anemometer. No account has been taken of this when obtaining annual average wind speeds from the Seawell data (27). Analysis of the 15-year period 1953-67 shows a progressive decline in annual average wind speed which is particularly marked during the last 5-year period, 1963-67, and therefore assumed to be caused by sheltering of the anemometer due to additional buildings at the airport. The 10-year, 1953-62, wind speed of 14 mph, has therefore been taken as representative of the site, although the more recent data are probably consistent and can be used for correlation purposes.

Comparison of wind speed and rainfall at Seawell shows that March is a crucial month, however the relatively low wind speed for March is at variance with Codrington when allowance is made for the relative exposure of Seawell. This might be attributed to the layout of the buildings surrounding the Seawell anemometer site, in which case a value of $v = 1.10$ might be more appropriate for the normalized March monthly wind speed at Seawell.

2. 3. 4. Velocity-Duration and Velocity-Frequency Distributions

The velocity-duration or cumulative-velocity-frequency curves for a given site give some idea of the regularity of a wind regime. The characteristic features of the trade wind regime are shown by the annual and monthly velocity-duration curves for Springhead, Barbados, a very exposed site at 800 feet with an annual wind speed of about 20 mph, Figs. 4 and 8. Four-weekly periods have been chosen to represent the best and worst dry season months and the worst wet season month of 1962. The latter curve is almost identical to the mean for the year, and Fig. 8 shows that an envelope is defined by the best dry and worst wet monthly curves. The monthly curves are similar in shape although separated by a shear displacement. The central portions of the annual distribution and the monthly distributions have been linearized and, following Tagg's approach (9), the curves can be analysed by considering each of the component parts listed in Table 4.

Using a standard FORTRAN polynomial regression subroutine (28), second and third degree polynomials have been fitted to the Springhead velocity-duration curves by considering time as the dependent variable over the component per cent time ranges 0 to 70 and 70 to 100 respectively. This analytical model allows the velocity frequency curve to be conveniently derived from the negative inverse slope of the velocity duration curve, a method mentioned by Vadot (15). The power-frequency curve can be found by multiplying the velocity-frequency curves by the cube of the velocity, which enables v_E the normalized velocity for maximum power frequency to be obtained by differentiation, as shown below and in Table 5. The results are similar to those presented by Betz (32) and indicate that in a trade wind regime most of the wind energy is available at only slightly above-average wind speeds, a characteristic which is particularly favourable for efficient, low-cost wind power utilization by fixed pitch windmills.

$$\text{For } 0<t<70: \quad t = t\,(V),$$

$$= a_o + a_1V + a_2V^2,$$

$$\frac{dt}{dV} = a_1 + 2a_2V,$$

$$V^3\frac{dt}{dV} = a_1V^3 + 2a_2V^4,$$

$$\frac{d}{dV}V^3\frac{dt}{dV} = 3a_1V^2 + 8a_2V^3,$$

$$V_E = -\frac{3}{8}\frac{a_1}{a_2} \quad .$$

Viewing the year as a whole, it can be seen from Fig. 8 that the most frequent wind speeds are distributed in a narrow band about the mean annual wind speed, with an almost symmetrical distribution of higher and lower wind speeds about the modal plateau. The velocity frequency distribution of above-average wind speed for $0<t<30$ is linear while the distribution of below-average wind speed for $70<t<100$ is parabolic. It is therefore evident that for efficient trade wind power utilization, the design wind speed for a particular requirement should be selected to be at or near the relevant wind speed for maximum power frequency, always assuming that the frequency is suitable for the load. By using the polynomial approach the wind power hours between various velocity limits can easily be obtained analytically by integration as shown below:

$$\text{For } 0<t<70: \quad \int V^3 dt = 0.25a_1V^4 + 0.40a_2V^5 \quad .$$

This relationship illustrates how strongly wind power available is dependent on wind velocity, or site selection, and on number of hours of operation, or wind velocity range.

TABLE 4 Trade wind velocity-duration characteristics

Percent time t	Velocity characteristic V	Polynomial degree t = t(V)
0-30	Occasional periods of high wind	2
30-70	Long periods of very steady wind	1 or 2
70-100	Calms occuring only infrequently	3

TABLE 5 Springhead velocity duration model.

$t = t(V)$ for $0 < t < 70$.

Period 1962	Polynomial Coefficients: Intercept a_0	Regression Coeff. a_1	a_2	$v_E = -\frac{3}{8}\frac{a1}{a2}\frac{1}{\bar{V}}$
Year	362.0	-13.98	0.1347	1.20
Max dry month	667.3	-23.46	0.2063	1.31
Min dry month	495.3	-20.18	0.2054	1.14
Min wet month	351.3	-16.71	0.1985	0.97

$\bar{V}$ = 32.5 km per hour

Referring to Table 5 it is noted that the annual velocity duration curve gives a fair value of $v_E = 1.20$, which might be used if more precise monthly data were unavailable. The best dry month value of v_E is 1.31 which can be regarded as the maximum anticipated design wind speed or used to define v_R the normalized rated wind speed. The worst dry month value of v is 1.14 which can be used to define v_{DI} a normalized irrigation design wind speed. This design wind speed occurs at 30 per cent time which appears ideal for 8 to 12 hour daytime shift operation of sprinkler irrigation systems bearing in mind the diurnal variation of wind speed to be expected. The difference between rated and irrigation design wind speeds suggests that provision for multiple agricultural loads should be made to utilize seasonal peak powers. The worst wet month might be used to define a year round normalized design wind speed v_{DY}, somewhat less than the annual wind speed for low-power wind pumping applications utilizing wind wheels for general purpose water lifting.

It can be assumed as a conservative first approximation that the Springhead rated and design wind speed results are applicable to relatively less well exposed but more typical wind power utilization sites, such as Seawell at 180 feet with a mean annual wind speed of 14 mph. In this case, the maximum dry month gives a rated wind speed of about 18.5 mph, in accord with the molinological estimate, while the minimum dry month gives an irrigation wind speed of about 16 mph. It is interesting to note here that the rated windshaft power of Russian windmills, such as the variable pitch, 12 meter (39.4 ft) diameter Sokol are quoted by Levy (29) at a design wind speed of 7 m/sec (15.7 mph), which would probably be suitable for Cuba (30) and other Caribbean islands.

2. 3. 5. Prototype Rated and Design Wind Speeds

The rated wind speed of the prototype windmill has been selected as 18.5 mph and the 32 ft diameter rotor sized to produce 10 brake horsepower at this wind speed, assuming a combined aerodynamic and transmission efficiency of 0.30. With a tip-speed ratio of 6.0 the rated windshaft speed is about 100 rpm.

The test site selected for the prototype windmill at Greenland Fig. 5, with an annual rainfall of about 60 inches and an elevation of 50 feet is sheltered from the sea by a ring of hills about 200 feet high half a mile eastward of the site. Wind records for 1967, for which year the annual average wind speed is about 11.3 mph, indicate that this causes a cut-back in wind speed so that the site is not ideally suitable. However, it can be argued that the cut-back would be expected to be more severe in the wet season when low winds would lack penetration. In this case, the dry season average wind speed of 12.5 mph might be taken as a basis for design and the irrigation design wind speed would then be about 14 mph. The design point can be adjusted further to take account of reliability required and the efficiency characteristics of the driven machine and its load, although caution is necessary here since design windshaft power is proportional to the cube of design wind speed.

Considering typical potential sites for wind powered irrigation schemes such as Seawell and Greenland shows that the hourly mean irrigation design wind speed would be expected to be somewhere in the region of 14 to 16 mph. Assuming a local minute to minute gustiness factor, $\frac{\text{gust speed minus lull speed}}{\text{average wind speed}}$, of unity (31), this means that for maximum power utilization the windmill should be allowed to operate in instantaneous wind speeds of about 7 to 24 mph. Allowance should also be made for occasional higher wind speeds and squalls of about 30 mph (21). This range of operation then enables virtually the total wind power spectrum to be tapped to provide useful shaft power.

The design wind speed for the airscrew rotor has therefore been taken as 15 mph and the aerodynamic layout based on a power coefficient Cp_D of 0.40, which guarantees a minimum of 5 brake horsepower at the design point. Normal operation is about this design point, usually from a minimum continuous running wind speed of 7 mph to an optional shut-down wind speed of between 24 and 30 mph. The cut-in wind speed, above which useful power is generated, depends on transmission, pump and load characteristics and would be expected to be in the range 10 to 12.5 mph. Manual starting is necessary at normal wind speeds due to the negligible starting torque of the high-speed fixed-pitch windmill blades. Provision should also be made for tractor power take-off drive during extended periods of low wind speed or for emergency use during prolonged periods of drought.

The design and operating envelope is completed by allowing for a stalling gust at 24 mph, normal operation up to 30 mph, and a 100 mph gale with the airscrew stationary. For long thin flexible blades static proof loading tests tend to give generous safety margins due to the increase of blade stiffness with shaft speed and load alleviation due to large aeroelastic deflections in the 100 mph gale case. Flexible blades also have the advantage of absorbing gusts by elastic recoil followed by gradual dissipation of energy by vibration. Of these loadings, the 100 mph gale is the most severe for a fixed pitch windmill so that provision is made for lowering the mill in case of hurricane force winds in which flying debris are also a hazard.

3. WINDMILL PUMPING SYSTEM DESIGN AND CONSTRUCTION

3. 1. Load Matching Technique

The philosphy behind the connection of a suitable load to a fixed-pitch, variable-speed windmill airscrew is based on the concept of a wind turbine as a piece of rotodynamic fluid machinery, the output of which varies roughly as the cube of the shaft speed when operating at its optimum power coefficient. The ideal load would have a shaft power-speed characteristic identical to that of the windmill airscrew; the two components can then be connected by suitable gearing so that they both operate in the most efficient parts of their ranges, regardless of wind speed variation over a wide range. This is important as the whole assembly has to be designed keeping in mind that the shaft speed will be continually varying between limits as the wind speed varies about the design speed.

In order to determine the useful power output from a given windmill prime mover the variation with wind speed of the overall efficiency of the machine as an energy converter is required. Ideally the overall efficiency should remain as high as possible over an extended operating wind speed range. Often the useful power output, in terms of kilowatt hours, long term water pumped or water horsepower hours is more nearly a linear or parabolic function of wind speed, rather than a cubic relationship. This is due to variation of component aerodynamic, transmission and pump efficiencies with wind speed and load. This gives rise to useful power characteristics well below the theoretical wind power available, especially at starting and at cut-in wind speeds. For optimum performance it is therefore essential to match airscrew, transmission and pump efficiencies and load at the design wind speed.

For agricultural irrigation a reservoir and centrifugal pump connected to a sprinkler irrigation system represents an ideal load for a fixed-pitch windmill, providing the network can be adjusted by altering the number of laterals to suit the wind speed, either automatically or manually, and accepting a limited pressure range at the nozzles. The pump is therefore selected to give the required head at its design point and match the design power available from the windmill when allowance has been made for transmission losses of gearing and bearings in the case of a mechanical shaft drive. The overall gear ratio then required is obtained from the design point shaft-speed ratio.

Performance tests on windmills coupled to rotodynamic pumps carried out in 1927 and 1928 by Von Baumhauer and Hoekstra are cited by the Prinsenmolen Committee (18). In these tests an improved Dutch windmill was fitted with three axial flow turbine pumps with friction couplings; however the results were inconclusive due to the anemometer and water supply difficulties. The results of Hoekstra's 1929 experiments (18) on an axial flow turbine pump driven through 14:1 ratio step-up gearing by a mill with improved sails indicate an increase in overall efficiency attributed mainly to improved aerodynamic efficiency which was partially offset by the low pump efficiency due to flow friction and fluctuating speed of the turbine pump. It is concluded that in the case of the Dutch windmills any improvement in performance with a medium efficiency turbine pump is probably marginal in view of the initial low transmission efficiency and the additional gearing required. More recently Vadot (15) has used a high speed variable pitch airscrew windmill to drive a low lift propeller type pump for irrigation purposes, but concludes that for optimum efficiency a variable pitch pump is required.

For pumping at higher lifts, wind powered electrical submersible deep-well pumps are currently in use for agricultural irrigation in the Soviet Union, where wind pumping systems using a Mono progressive cavity screw type pump connected to an airscrew windmill through a centrifugal clutch have recently been investigated by Levy (29).

It is apparent that appropriately balanced values of aerodynamic, transmission and pump efficiency are desirable for a successful design. Also, although the centrifugal pump is ideally suited to the present pond-sprinkler irrigation load, Fig. 1, propeller and simple rotary positive displacement pumps, such as the Mono or Archemedian screw, should be considered as potential load options in design studies for other pumping applications.

3. 2. Windmill Rotor Design

3. 2. 1. General Arrangement

The aerodynamic design of the 32 foot diameter airscrew is according to Theodorsen's propeller theory and design procedure (33), taking account of the large wake expansion and relatively high tip-speed ratio for a windmill rotor. The windmill blade layout, shown in Fig. 9, is based on a power coefficient Cp of 0.40 and a tip-speed ratio μ of 5.95, in order to provide a design load shaft speed N of 2,500 rpm with an overall gear ratio G of 32:1 at a wind speed of 15 mph. For low cost and simplicity, the airscrew is of fixed pitch and consists of three fibre-glass reinforced epoxy resin blades of aerofoil section NACA 4415 bonded to high tensile steel stub shafts, which are mounted in lugs in a fabricated steel hub to give a coning angle of 10 degrees. The airscrew is mounted downwind of the tower at a centre-line height of 30 feet on the wheel hub of an Austin 5-ton truck rear axle at the top of the tower, as shown in Fig. 1.

3. 2. 2. Aerodynamic Design Considerations

The choice of a power coefficient Cp of 0.4 at a tip-speed ratio of about 6.0 represents a practical compromise between a value for power coefficient of 0.2 to 0.3 for a slow-running multibladed windmill rotor, running at a tip-speed ratio of the order of 1.0 or 2.0, and an extremely high speed airscrew type windmill rotor, with two or three fine, low-drag blades running at a tip-speed ratio of the order of 10to15, for which values of power coefficient might be in the range of 0.4 to 0.5, Hutter (34) and Levy (29).

Glauert (35) provides an introduction to propeller, windmill and fan theory and discusses windmill characteristics in relation to design requirements, emphasizing that the rotor starting torque has to overcome internal friction of the system to allow self starting. This is an essential feature of the unattended slow running multibladed windmill which is particularly suitable for light and variable winds. For a fixed-pitch high-speed airscrew windmill starting torque is negligible so that, even with the provision of a centrifugal clutch con't ....

to reduce starting load, manual, auxiliary or grid starting are likely to be required. Generally, the high speed rotor has the advantage of requiring less gearing although the friction drag of the blades is relatively high and the blades are subject to a large centrifugal force so that careful aerodynamic and structural design are required.

Glauert determines the radial variation of windmill blade angle and chord for optimum performance by using simple axial momentum theory to derive an expression for σC_L to characterize windmill rotor blade layout as a function of tip-speed ratio. This relationship shows that for a high-speed windmill blade at constant lift coefficient C_L the chord will taper towards the tip so that the solidity σ of the rotor disc will be low, in contrast to the low-speed multi-bladed rotor. A more precise generalized momentum theory is discussed by Lilley and Rainbird (36) who use Theodorsen's propeller theory and introduce a vortex windmill theory in which the mean values of the periodic induced velocities at the blade elements are considered in relation to ducted windmill design. The theory indicates the importance of slipstream expansion in both ducted and free stream windmills. In the present design the tower height has been taken as 30ft to allow full wake expansion and give reasonable ground clearance for the 16ft radius blades.

A critical review of the present airscrew design by Nevot (37) indicates that the blade layout based on Theodorsen's theory compares well with the predictions of theories due to Lock and Yeatman (38), Iwasaki (39), Lock (40) and a modified Sabinin theory cited by Nevot. Slight discrepancies in the predicted induced velocities suggest that, with the design geometry and aerofoil section adapted, the airscrew should run at a power coefficient slightly less than 0.4 and at a tip-speed ratio somewhat greater than 6.0, a result which has been confirmed by field performance tests.

A significant shortcoming of existing propeller design theories is their limitation to steady uniform flow, which conditions are unusual for most windmills. In general, the windmill rotor will operate in a non-uniform, unsteady, turbulent flow within the earth boundary layer, in which it will be subject to large angles of yaw, distorted inflow and gusts. In attempting to meet low cost design requirements and achieve reasonable performance, some account needs to be taken of the practical influence of these limiting but often unpredictable atmospheric environmental conditions.

The choice of aerofoil section represents a typical compromise governed by aerodynamic and structural considerations. At the design point, a high lift-drag ratio aerofoil section is desirable for high efficiency, while a high value of section maximum lift coefficient and a gentle stall with a stable, nose-down, pitching moment break is generally desirable for stable off-design operation. The 15 per cent thick NACA 4415 aerofoil section, Abbot and Von Doenhoff (41), chosen for the blades satisfies these requirements with a design lift coefficient of about 1.0, a conservative maximum lift coefficient of 1.275 and adequate thickness for strength.

Considering gust conditions, Hutter (42) allows for variable pitch operation with normalized gust velicities v_G in the range 0.5 to

1.5, which is equivalent to the gustiness factor of unity previously assumed to define the minute-to-minute design wind speed envelope for the present design. For operation of fixed pitch airscrew windmill blades in unsteady flow, some degree of shaft speed control in gusts can be achieved by progresssive stalling, a method mentioned by Vadot (15). Some indication of the anticipated aerodynamic response of the present blade design to gust conditions is shown in Table 6, which gives the approximate sharp edged axial gust velocity to partially stall the airscrew when operating at the design-point power coefficient and tip-speed ratio in a steady wind of 24 mph. It is seen that at the design point normalized gusts v_G greater than 1.25 will tend to stall the blades and give some smoothing of shaft torque and speed variation. Although some load relief occurs, as the stalling gust is unlikely to be a step function, the airscrew has considerable rotational inertia and complete stalling at the design-point represents a severe blade loading and large deflection condition. This is only partially covered by the 100 mph gale with airscrew stationary loading case so that the static proof test loading distribution envelops both loads.

In transient conditions it can be assumed within certain limits that the inertia of the windmill rotor and brake load tend to give relatively constant shaft speed so that tip-speed ratio is inversely proportional to wind speed. This is certainly the case for a constant-speed variable pitch machine Hutter (42) and is a useful concept for analyzing the response to wind speed variation of a fixed pitch variable speed machine, the limiting factors being design lift coefficient,

TABLE 6 Sharp edged gust speeds for progressive blade stalling.

Stalled blade radius per cent	Sharp edged gust speed mph	Normalized gust speed v_G
10	30	1.25
70	35	1.50
100	40	1.65

Wind speed	24 mph
$C_{L\ DESIGN}$	1.0
$C_{L\ MAX}$	1.275

gustiness and inertia. It is apparent that for efficient operation of a fixed pitch windmill a nice balance is required at the operating point between a low lift coefficient with high tip-speed ratio, low power coefficient and rapid response to gusts; and a high lift coefficient with the possibility of incipient stalling and power loss due to lack of response to normal wind speed variation. For a given fixed pitch blade design, the blade setting angle can be adjusted slightly to take account of local conditions and special load requirements can be met by the selection of appropriate gearing.

3. 2. 3. Aeroelastic and Structural Considerations.

In the present airscrew design large bending stresses are avoided by coning the blades 10 degrees down wind to allow the bending moment due to aerodynamic lift to balance the bending moment due to centrifugal force. Fung (43) discusses the twisting moment acting on a blade section due to centrifugal force for the general case in which aerodynamic centre, elastic axis and centre of mass do not coincide. For torsional equilibrium of a flexible blade the moment due to centrifugal force should just balance the aerodynamic pitching moment about the aerodynamic centre. This implies that the blade will not twist if the design, or ideal, lift coefficient is given by $C_{LI} = \frac{-C_{MO}}{x-0.25} = 0.6$ approximately for NACA 4415. Operating below the ideal lift coefficent reduces the chance of stalling and thus raises the flutter speed. The closeness of the elastic axis, centre of gravity and aerodynamic centre generally make normal propeller and helicopter blades inherently strong against classical flutter. However, some care is required to prevent stall flutter, with consequent vibration and low fatigue life, by working at an angle of attack below the stalling angle and ensuring that the blades do not twist aeroelastically to the stalling angle, at which flutter starts, by appropriate choice of design lift coefficient, mass and stiffness distributions. The phenomenon of propeller stall flutter is characterized by a peculiar buffeting noise, tip weave and torsional oscillation of the blades. For long thin flexible blades the natural torsional oscillation frequency is very much higher than the flexural oscillation frequency and in this case relatively stable long-period transitional oscillation may dominate the motion. The blades may therefore tend to induce a stall flutter when vibrating at their natural flexural frequency, which will be a function of shaft speed, Ker Wilson (44). It is therefore desirable to build high structural damping into the blades by the choice of suitable material and construction, and also to ensure that the transmission system and supporting structure are capable of absorbing induced and residual vibration without resonance.

In order to achieve a high tip-speed ratio with reasonable inertia distribution and reduce gyroscopic moments and possible vibration problems in the case of automatic orientation, a three bladed airscrew is used. The blade structural requirements are met by the use of fibre-glass reinforced epoxy resin which has been used as a structural material in the aircraft industry (45), in jet lift engines (46) and for large hovercraft propellers (47). Hutter has successfuly used the material to form airscrew windmill blades (45), for which purpose the strength, flexibility and ease of forming twisted, variable chord, aerofoil section blades are ideally suited for batch production.

The airscrew rotor is supported downwind of a guyed, square hollow-section, space-frame tower which allows clearance for large tip deflections of the airscrew blades. The wide space-frame wake (48) at the airscrew disc is considered less likely to produce tower shadow interference with blade performance than the wake due to an equivalent unfaired circular cylinder, thus giving reasonably uniform flow conditions essential for prototype airscrew performance tests. The wake just down-stream of a circular cylinder is particularly turbulent and the velocity deficit in the wake near the blade disc remains relatively intense (49). For production windmills reinforced concrete or faired steel tube towers, which might offer water storage capacity, should be considered.

Aeroelastic deflection will cause a significant reduction in the static loading due to a 100 mph gale and give a generous safety margin which can only really be checked by wind tunnel testing (50). In the prototype design, allowance has been made for lowering the tower in case of hurricane force winds, although for production mills adjustable blades might be feathered or the mill head assembly pivoted at the top of the tower to allow the blade disc to be locked in a horizontal position.

3. 3. Power Transmission

The elements of the prototype power-transmission system are shown in Figs. 10 to 15, Fig. 22 and Figs. 31 to 33. The airscrew is mounted downwind of the tower at a centre-line height of 30 feet on the wheel hub of an Austin 5-ton truck rear-axle at the top of the tower. Airscrew shaft power is transmitted through the truck half-shaft to the differential which is locked by welding the differential pinions to the differential spider in order to provide a right-angle drive with a step-up gear ratio of 7.2:1 through the spiral-bevel crown wheel and pinion. The pinion drives a vertical power transmission shaft through a flexible coupling. The vertical shaft is supported by standard self-aligning bearings attached to the tower and spaced to prevent shaft whirl.

For test and experimental purposes power from the vertical transmission shaft is transmitted to the pump through a spiral-mitre gearbox, a brake, a centrifugal clutch and an eight-speed Land Rover transfer and gearbox. These items are mounted on a separate machinery pad and connections to mill and pump-head gearbox are made by Land Rover Hardy-Spicer Shafts, Fig. 22. The prototype load is a standard water-lubricated 10-stage Wade Rain 6JC3 deep-well turbine pump with a right-angle drive gearbox providing a step-up gear ratio of 1.333:1 with an optional alternative ratio of 0.75:1. The pump delivers water from an offset pond to a 10-acre sprinkler irrigation system described by Ionson (20). The Land Rover gearbox provides the additional step-up gearing required to allow both the airscrew and the pump to work at optimum efficiency and also enables the detailed airscrew performance characteristics to be determined by using the pump as a hydraulic brake. Various hydraulic circuit settings and the wide range of gear ratios available allow the system characteristics to be determined over an extended operating range.

This transmission arrangement enables overall gear ratios between 6.2:1 and 51.1:1 to be selected for test purposes. Allowing 0.90 for the Land Rover unit transmission efficiency, based on manufacturers figures and 0.90 for the combined mitre-gearbox and rear-axle efficiency and 0.01 for bearing losses, the overall transmission efficiency is about 0.80. With a vertical pinion shaft the transmission efficiency of the rear-axle is likely to be significantly less than that for normal vehicle operation, due to oil churning losses, especially at low power transmission levels when the overall transmission efficiency tends to be low because power losses in oil turbulence and sealing remain substantially the same however small the load, Tuplin (51). Reduction in gear oil quantity and lowering the viscosity from 140 to 90 SAE tends to improve efficiency. On the prototype mill-head rear-axle, a single pinion seal works well provided the axle is well ventilated, however two well-lubricated seals, as recommended by the manufacturers, would probably give longer seal wear life. Without the Land Rover unit, a reasonable value for the transmission efficiency of a production windmill would be 0.90 for a horizontal power output shaft.

The prototype airscrew is started manually using a bicycle transmission system at the tower base, an arrangement which was found to be very convenient for the frequent starting required during performance testing. In normal winds the airscrew is easily spun up to about 35 rpm at which speed the airscrew unstalls, accelerates and continues to run at wind speeds above 7 mph. On a production windmill this starting arrangement would be simply replaced by a diesel-engine hand crank. An essential feature of the starting and transmission system is the spring-controlled centrifugal clutch which allows manual starting at virtually no load. Centrifugal clutch engagement at an airscrew shaft speed of about 50 rpm was found to give satisfactory acceleration of the turbine pump load on the prototype windmill.

For normal braking, it is preferable to use the original truck hand-brake system with the hand-brake lever attached to the lower part of the tower and with the brake adjacent to the airscrew. The transmission machinery pad brake was used only for shaft torquemeter calibrations having been found to induce undesirable torsional vibration and overload of the transmission shafting, due to its position at the remote end of an effectively very long small diameter shaft attached to a high inertia rotor. Although truck brakes have high utilization, energy dissipation and wear life (52), braking is best carried out during lulls to prevent excessive wear. In other applications, an emergency "once-only" operation, overspeed centrifugal trip outlined by Gillespie (53), should be considered as possible means of braking, together with commercial vehicle auxiliary brakes such as hydraulic retarders and eddy current brakes such as those described by Newcomb and Spurr (54).

For auxiliary drive and comparative irrigation performance tests provision was made to easily disconnect the vertical transmission shaft Hardy Spicer shaft and couple a stationary 10 hp diesel engine to the horizontal shaft of the tower-foot mitre gearbox.

3. 4. Supporting Structure

The supporting structure for the prototype windmill was designed for minimal airscrew-disc tower-wake interference for airscrew performance test purposes. It was also designed for ease of manufacture by the Brace Experiment Station workshop in Barbados. The resulting structure is inherently simple to erect, maintain and operate. The general rules outlined by Pugsley (55) for loading and strength were used for the structural design and generous safety margins were allowed for unpredictable and transient loads. The basic welded-steel space-frame structure, shown in Fig. 1, Figs. 15 to 23 and Figs. 29 to 34, consists of a guyed square-hollow-section tower, a tower base, and a foundation. The 2 ft square tower is mounted on a wheel hub, bearing and axle cut from the mill-head rear-axle. The axle is built into a tower yoke assembly which is supported by the tower base. The tower itself is supported by four wire-rope guys attached to a fabricated 4 ft diameter bearing at a height of 19ft, the downwind guys being set to allow adequate clearance for blade tip deflection. The fabricated bearing comprises an inner ring, sandwiched between upper and lower halves of the tower, on which are mounted two sets of double row angular contact ball bearings to take the horizontal and vertical loads imposed by the mating, guyed outer ring. This arrangement allows the tower and blade disc to be manually oriented with mean wind direction by means of a handwheel and truck crown wheel and pinion at the tower base.

The tower base is attached to the foundation which consists of a 2ft thick reinforced concrete cross into which the four guy pick-up points are built. For erection, the tower yoke is pivoted at its attachment to the tower base and this, together with four side stays, a gin pole, pulley and winch arrangement allows the windmill to be manually raised and, when necessary, lowered by two men in half an hour.

3. 5. Prototype Airscrew Windmill Costs

The cost breakdown of the prototype airscrew windmill pumping system as built in Barbados is shown in Table 7. The cost of the windmill as a prime mover can be obtained from the total cost by subtracting the cost of the Land Rover gearbox, used only for test purposes, and the cost of the pump, together with the labour involved in handling these items. This gives the cost of the windmill alone as Canadian $10,000, a figure which includes Canadian $980 for the cost of the airscrew moulds which are good for 100 sets of blades.

Recent firm quotations for production of windmills in Canada to the prototype specification give some indication of the variation of unit price with quantity, Table 8. The estimates for the blades were obtained from the aircraft production oriented, custom reinforced fibreglass fabricators, who made the prototype blades (56). The steel work estimates in response to the tender outlined in Table 9 are from an engineering company which specializes in aerospace subcontracts (57). Assuming a quantity of 10 windmills gives a unit cost of Canadian

TABLE 7. Cost breakdown of prototype windmill pumping system as built in Barbados.

Item	Cost East Caribbean $	Notes
Airscrew blades	$ 4,500.00	Set of 3 plus mould good for 100 sets.
Pump	2,185.00	Wade Rain 6JC3.
Foundation	1,210.00	Includes reinforcing .
Machinery pad	1,695.00	Includes Land Rover gearbox for test purposes.
Tower structure	1,970.00	Steelwork; rear axle and vertical transmission shafting.
Tower bearing	1,170.00	Fabricated, for orientation.
Erecting gear	1,310.00	Includes guy wires.
Total material	EC $14,040.00	Includes welding services
Labour	EC 7,200.00	Workshop wages 3 men 1 year
Total	EC $21,240.00	
1966 Exchange rate		1 Canadian dollar = $1.60 EC approx.

TABLE 8. Variation of unit price with quantity for manufacture of airscrew windmills in Canada.

Quotation date	Drawing No.	Description	Quantity	Unit price Canadian $
31.3.69	F-3-001	Airscrew Blade.	1	$1,475.00
		Set of 3 with	10	1,430.00
		Root, Item No. 1,	50	1,125.00
		supplied by customer.	100	930.00
31.3.69	F-3-001	Airscrew Blade.	1	$1,600.00
		Set of 3 with	10	1,518.00
		Root, Item No. 1,	50	1,200.00
		included in quotation.	100	1,000.00
29.5.69	B-3-041	Windmill Steelwork.	1	$8,800.00
		Fabrication, hot-	2	8,550.00
		dip galvanizing	3	8,387.00
		and crating.	4	8,223.00
			5	7,975.00
			6-10	7,562.00
			11-25	7,350.00

Notes: 1. Unit price quoted F.O.B. Montreal plant
2. Airscrew Blade Root, Drawing F-3-001 Item No.1, supply by customer, steelwork manufacturer or blade manufacturer optional.

TABLE 9. Costing notes on manufacture of steelwork to airscrew windmill prototype specification

Assembly Drawing No.	B-3-041 Item No.	Description	Notes on items required
D-3-011	2	Airscrew Blade Root	Fibreglass blade is moulded around airscrew blade root. Two balance weights required per set of blades.
D-3-005	3	Airscrew Hub	Airscrew blades and hub need to be balanced as an assembly.
D-3-001	4	Truck Rear Axle	Eaton, Rockwell, G.M., Ford or similar high efficiency substitute acceptable. Differencential locked by welding differential pinions to differential spider. Left hand hub and brake used for airscrew. Hand-brake system as on truck.
E-3-004	5	Tower	Drawn as for prototype, may be modified if required.
E-3-002/2	6	Tower Bearing	Rotek bearing replaces fabricated bearing on prototype.
C-3-012	7	Guys	Drawn as made up for prototype.
D-3-004	8	Tower Yoke	Uses right hand hub of the truck rear axle and allows tower to be oriented by hand and pivot for raising or lowering.

con't .....

Table 9 con't .....

D-3-007	9	Tower Base	Drawn as built, may be modified if required.
D-3-008	10	Foundation	Rough estimate of reinforcing material cost required.
E-3-003	11	Transmission Machinery Pad and Pump	Quote only for parts up to and including centrifugal clutch as pump and additional gearing depend on application.
D-3-006	12	Erecting Gear	Include for hand winch. Pulleys drawn as built.
A-3-008	13	Parts List for One Set of Blades	Mainly a check list for packing blades. Quote for items listed.
F-3-002	14	Blade Shipping and Storage Container	To suit set of three airscrew blades and vertical shafting.

<u>Additional Requirements</u>:

1.	Lubrication	Tropical grade oil and grease required for Caribbean applications. Appropriate grades for applications elsewhere.
2.	Finish on metal parts	Thorough surface preparation, etch prime, red oxide prime plus two coats aluminum paint. Alternatively hot-dip galvanize.
3.	Crating and packing	All parts should be prepared for shipment bearing in mind the primitive conditions under which this type of prime mover is likely to be set up and operated.
4.	Shipping	Quote for most appropriate rate: a) F.O.B. local plant, b) F.O.B. Montreal, c) C.I.F. Caribbean port, e.g. Bridgetown, Barbados.

$9,000, F.O.B. Montreal plant. Allowing Canadian $1,000 for shipping, foundation and erection indicates that the installed cost of the prototype windmill would again be about Canadian $10,000.

The power cost has been estimated on an equivalent annual cost basis by using the installed capital cost figure of Canadian $10,000 and 50,000 hours of operation at a mean rated power of 10 horsepower. This duty is equivalent to 2,500 hours of operation per year at 10 horsepower spread over a life 20 years and represents an annual power output of 25,000 horsepower hours obtained by high utilization at a good wind-power site. Allowing for amortization of the capital cost over 20 years at an interest rate of 8 per cent requires Canadian $1,018.5 per year (58). For Caribbean operation, the services of an operator at Canadian 50 cents per hour for 10 per cent of the operating time require Canadian $125 per year. For lubrication and maintenance, including greasing, annual oil change and occasional painting, a further Canadian $110 per year are required. On this basis, the equivalent annual cost of the prototype design is Canadian $1,250 and the power cost is Canadian 5.0 cents per horsepower hour. The equivalent diesel power cost is about Canadian 4.0 cents per horsepower hour (10). The major items in case of the diesel engine are the recurrent fuel and maintenance costs, in contrast with the low running cost and high capital cost of the windmill prime mover. It is therefore apparent that in relatively remote areas where good wind-power sites combine with high fuel cost, the windmill can be competitive with the diesel engine, especially when such factors as the regularity of fuel supplies and availability of skilled maintenance are taken into account. In order to be more competitive with an equivalent diesel engine some development, including appropriate modification, substitution and streamlining, of the prototype windmill design is required for really low-cost Caribbean or Canadian production. On the basis of relatively low recurrent operating cost a strong case can then be made for capital assistance to aid development of water supplies in isolated rural areas by means of efficient low-cost wind power system developed from the prototype airscrew windmill.

The prototype cost breakdown shown in Table 7, shows that considerable savings can result from reduction in cost of orienting and erection gear. An example of effective innovation is shown in Figs. 16 and 17 in which an industrial turntable bearing has been substituted for the fabricated prototype tower bearing. Further economy would result from using this type of bearing at the mill head and orienting the mill from the ground by means of the vertical transmission shafting. Similarly, cost reductions could be effected by using the tower guys as stays during erection, a method which was not feasible on the prototype due to site limitations. In view of the 100 mph static gale proof loading, high erecting loads and hurricane possibilities, a self slewing turntable with manually adjustable pitch blades or manually coned blades, or rotating the blade disc about a horizontal axis at the top of the tower for the hurricane case and erection, should be considered as possible means to reduce supporting structure and total costs. For water pumping and desalination loads the possibilities of hollow steel and aluminum section towers, which offer useful water capacity, should be considered, together with the possible use of reinforced concrete.

Such system design optimization offers plenty of scope for ingenuity, and a detailed design and value engineering study is at present under way in cooperation with the industrial products division of a large engineering company (59). With emphasis on the power-transmission system and structural engineering design the aim is to cut the installed unit cost for quantities of 10 windmills to less than Canadian $5,000.

4. PROTOTYPE AIRSCREW WINDMILL STRUCTURAL AND PERFORMANCE TESTS.

4. 1. General Review

In order to check the strength and flexibility of the prototype airscrew windmill blades load-deflection and static proof-load tests were carried out. The airscrew rotor was then balanced as an assembly, and the tower and erecting gear were proof tested by hydraulic loading, before final erection of the windmill at the Greenland test site, Figs. 5 and 29.

Preliminary trials were undertaken to check the safety and reliability of the airscrew, power-transmission equipment and supporting structure. During initial runs several minor modifications were found necessary. These included the bicycle self-starter, repositioning the downwind guy wires to avoid blade tip interference, the addition of a new tower-base brake to prevent tower torsional vibration; together with the installation of additional vertical transmission shaft bearings and flexible couplings to reduce shaft vibration. The completed airscrew windmill pumping system was first run in November 1966 and to date the only replacement parts required have been a cartridge bearing for the vertical transmission shaft and a pinion oil seal for the mill-head rear axle.

An initial series of aerodynamic performance tests was carried out on the windmill pumping system in order to check the validity of the airscrew blade design procedure. The Greenland test site wind records and the windmill power-speed characteristics were then used to design and predict the performance of a 10-acre sprinkler irrigation system, the field performance of which has been evaluated by Ionson (20). Check aerodynamic performance tests were made in order to confirm the initial results over an extended range of operation.

4. 2. Airscrew blade structural tests

4. 2. 1. Blade flexural characteristics

A structural test rig was built for static load-deflection and proof tests on the airscrew blades. Fig. 24 shows the general layout of the rig, complete with blade test frame, loading arrangement and prop used to prevent excessive deflection during load-deflection tests to determine EI, the flexural rigidity distribution for each blade. The slope and deflection due to load distributions of about 0.25, 0.50 and 0.75 times the proof loading were measured by means of three dial test indicators and an indicator bridge. Using this arrangement the mean value of EI was found to be $212.4\ c^4$ lb in^2, where c is the local blade chord. For the NACA aerofoil section blade 4415 $I=1.555\ 10^4\ c^4$, so that by assuming a solid homogeneous blade section $E=1.366\ 10^6$ lb per in^2, in accord with normally accepted values for modulus of elasticity of fibreglass cloth and woven laminae.

Time did not allow the prototype blade torsional characteristics to be determined and it is therefore suggested that the flexural axis, or shear centre, be determined and that the flexural and torsional characteristics be measured on the pre-production blades in order to evaluate the aeroelastic characteristics of the airscrew.

4. 2. 2. Blade static proof load tests

A graphic indication of the large deflections to be expected with flexible fibreglass blades is given by the 4ft tip deflection under the blade static proof test load shown in Fig. 24. The static proof loading envelops the 100 mph gale airscrew stationary and the 24 mph stalling gust load distributions. Initially hollow, the blades failed by buckling under proof test loading. They were therefore injected with resin filler and on subsequent retest were found to withstand the static proof load specified.

In practice the 100 mph gale load is relieved aeroelastically by the blade flexibility, while the increased stiffness due to centrifugal force tends to offset the severity of the 24 mph stalling gust load, an effect which can be increased by use of a larger static blade coning angle. However, large deflections should be anticipated in the transient conditions likely to be met during normal operation and it is therefore evident that an absolute minimum of 2ft static clearance between blade tip and tower or guy is essential for safe operation.

4. 3. Airscrew performance tests

4. 3. 1. Test equipment and instrumentation

4. 3. 1. 1. General

In order to determine the aerodynamic efficiency of the windmill rotor, airscrew torque coefficients Cq, power coefficients Cp, and tip-speed ratios μ were obtained from high-speed records of wind speed, and airscrew shaft torque and speed, Fig. 35. The test load equipment, comprised the 6JC3 deep well turbine pump, a variable hydraulic load provided by the pumping system hydraulic circuit, and a variable speed drive provided by the Land Rover gear box, Figs. 27 and 31. The test instrumentation diagram Fig. 28, shows the layout of the wind and water-side instrumentation, consisting of sensors, indicators and recorders, described below.

4. 3. 1. 2 Wind speed and direction measurement

Wind speed and direction were measured at airscrew centre-line height about two airscrew diameters upwind of the airscrew disc, as recommended by Villers (60). The anemometer tower was a standard 30ft guyed domestic television antenna tower sited 70ft-3in N66^{o}E from the tower centre line, the axis of symmetry of the windmill base and foundation being at N65^{o}E.

For test purposes the airscrew was manually oriented with the minute-to-minute mean wind direction by using the tower yoke hand-wheel, and then clamped for the duration of the test by means of the tower yoke brake. For reliable results the mean wind direction was preferably in the range N50^{o}E to N80^{o}E, indicted by a highly damped electrical wind direction transmitter mounted on the anemometer tower centreline.

For wind speed measurement the wind speed output signal from a cup anemometer wind speed transmitter was fed to an indicator and a twin-channel high-speed recorder. A similar anemometer unit was used for wind speed checking purposes. Long-term hourly-mean site wind records were obtained from a wind speed and direction recorder and a standard cup counter anemometer.

4. 3. 1. 3. Airscrew shaft power measurement

Airscrew shaft speed was recorded by sensing the horizontal transmission shaft speed at the tower base by using a tacho generator. The output signal was fed through a potential divider to a high-speed single-channel recorder calibrated in airscrew rpm.

Airscrew shaft torque was measureed by a torquemeter comprising four active torque-sensing foil strain gauges connected in a Wheatsone bridge network on the airscrew shaft as shown in Figs. 25 and 33. The input and output signals were transmitted by four insulated silver slip rings and eight carbon brushes, supplied in kit form by Bean (61). In later tests a commercially available torquemeter was fitted to the vertical transmission shaft, as shown in Figs. 26 and 30. Both torquemeters were connected to the high-speed twin-channel recorder and statically field calibrated.

The airscrew shaft torquemeter was calibrated by locking the transmission pad brake and loading up to four post-weighed people on a wooden platform suspended 3ft above ground level from a horizontal airscrew blade root. The vertical shaft torquementer was calibrated with the tower horizontal and the mill-head brake locked, by hanging known weights from a horizontal torque arm bolted to the lowest vertical-shaft flanged coupling. During calibration and test runs considerable difficulty was experienced with drift on both torque measurement systems, so that frequent zeroing by spinning up the airscrew and allowing it to freewheel to a stop was required in order to obtain reliable results.

In an effort to directly measure mill-head rear axle transmission efficiency the two torquemeters were connected simultaneously to the high-speed strip-chart recorder; however, the drift proved excessive and this test was inconclusive. The results of a current no-load laboratory test in which a similar rear axle driven at variable speed by a calibrated direct-current motor should allow a value of the mill-head transmission efficiency to be obtained in order to refer the vertical shaft torquemeter results to the airscrew torque through the rear axle losses.

4. 3. 1. 4. Water-side intrumentation

On the water side records of pump pressure and flow could be obtained from a pressure transducer and a 3 inch diameter turbine flowmeter in the main water line just downstream of the pump outlet. Long-term records of water pumped could be read from a 4-inch diameter turbine flowmeter.

4.4 Performance Test Results

In order to predict the airscrew performance over its operating range the non-dimensional characteristic curves of torque coefficient Cq and power coefficient Cp against μ were derived for the airscrew from typical high-speed chart traces of the variation of wind speed V, shaft torque Q and shaft speed n, with time, such as those shown in Fig. 35. Airscrew shaft power and speed, for selected gear ratios and hydraulic circuit settings, over the range of wind speeds to be expected during normal operation were plotted as shown in Fig. 36. In this figure, the centrifugal clutch and starting characteristics are also shown, together with rated power and speed limits. The airscrew was run unloaded, lightly loaded and through optimum loading to stalling loads,

TABLE 10. Typical prototype airscrew windmill performance test results, Greenland, Barbados.

Airscrew shaft horsepower hp	Airscrew shaft speed rpm	Upstream anemometer indicated wind speed (5 sec. approx.) mph	Power coefficient Cp	Tip-speed ratio μ
3.46	83.9	14.0	0.2395	6.84
3.58	85.5	12.0	0.394	8.14
4.80	96.7	12.7	0.445	8.70
4.80	90.4	14.8	0.281	6.95
5.95	100	15.4	0.309	7.42
6.59	110.2	12.8	0.594	9.84
7.31	113	15.0	0.412	8.60
8.36	123.5	16.0	0.387	8.81
7.71	128.5	15.7	0.381	9.35
10.03	132	20	0.239	7.54
11.7	138.6	19.3	0.310	8.20
12.25	140	17.3	0.451	9.25
13.3	144	18.0	0.433	9.15
13.2	147	21.9	0.239	7.66
16.7	161.3	19.4	0.436	9.50
19.41	170.4	22.6	0.32	8.61

Date	1st February, 1967,
Airscrew pitch angle at 0.7 radius	Design blade angle,
Tower setting angle	N70°E wind direction,
Gear ratio H3	16.65:1 overall ratio,
Pond level	6'-11.25",
Atmospheric density	0.002285 slugs/ft^3.

TABLE 11. Predicted performance of Brace airscrew windmill

Wind speed (mph)	10	15	20	25	30
Airscrew shaft speed (rpm)	61.3	92.0	122.6	153.3	183.9
Output shaft speed (rpm)	441	662	882	1103	1323
Airscrew shaft horsepower	1.845	6.24	14.76	28.8	49.8
Airscrew windmill brake horsepower	1.66	5.61	13.27	25.9	44.7
Pump water horsepower	1.16	3.93	9.28	18.1	31.3
Water delivery at 100 ft head (Igpm)	38.4	130	307	598	1035

Airscrew power coefficient	0.35
Airscrew tip-speed ratio	7.0
Atmospheric density	0.002285 slugs/ft^3
Transmission efficiency	0.90
Pump efficiency	0.70
Mill head gear ratio	7.2:1
Airscrew diameter	32 ft

For water delivery at total heads other then 100ft, obtain by inverse proportion.

up to a never-exceed speed of 200 rpm and a maximum power of 20 hp, at wind speeds up to about 24 mph.

The general trends of wind speed, shaft torque and shaft speed of Fig. 35 indicate that all three variables can be related in time intervals during which the wind direction is such that a response to wind speed is reflected in the shaft torque and speed traces. As wind speed, as well as direction, is varying, a variable lag in response to wind speed is evident. Significant smoothing of torque and speed are apparent and, although wind direction variation partially accounts for this, wind speed variation appears to be damped mainly by the combination of rotor and load inertia and progressive blade stalling in sudden gusts, Table 6.

Appropriate wind speeds were used to evaluate power coefficients Cp ($= \mu$ Cq) and tip speed ratios μ , as shown in Table 10. In order to determine the maximum value of power coefficient and the associated tip-speed ratio, a composite "linearized" plot of torque coefficient against tip-speed ratio, Fig. 37, was used to obtain the smoothed "parabolic" variation of power coefficient with tip-speed ratio shown in Fig. 38. The predicted performance of the airscrew windmill at a maximum power coefficient of $Cp_{MAX} = 0.35$ is shown in Table 11.

Considering the chart traces of Fig. 35 in detail, the ripple on the anemometer trace is due to anemometer cup interference giving rise to uneven no-load torque and speed which is reflected in the output signal. In view of this, mean values of wind speed have been used which were taken over periods of about 5 seconds, during which there was quasi-steady state response of airscrew shaft torque and speed to wind speed. Such wind speeds are regarded as "instantaneous" in contrast to "integrated" hourly mean wind speeds for which Ionson (20) has found the characteristic performance of the windmill pumping system.

In a turbulent environment windmill power output would be expected to be a function of measurement time interval, system operating point and inertia, Clausnizer (69). For a fixed pitch windmill system, the rotor inertia, load inertia and aerodynamic response characteristics, such as progressive blade stalling, tend to balance the effect of energy pattern factor, Hutter (66), so that a careful choice of operating point is required for optimum operation, taking into account the maximum acceptable shaft speed variation. The optimum operating point on the power coefficient against tip-speed ratio curve is therefore a function of site gustiness and load which, in the case of the prototype pumping system, was found to be at a tip-speed greater than that for maximum power coefficient, Fig. 38.

The airscrew shaft torque chart trace, Fig. 35, and the full-scale shaft speed trace both show an almost constant frequency ripple which is attributed to blade "nodding" rather than to tower shadow interference. The torsional vibration of the shaft appears to be caused by the variation in aerodynamic loading on the blades due to change of incidence caused by blade flexural vibration induced by atmospheric turbulence. Due to the effect of centrifugal force on stiffness the frequency of blade vibration will be a function of airscrew shaft speed, Ker Wilson (44). It is therefore apparent that in the system design estimates of blade frequency variation with shaft speed are required in order to avoid torsional resonance of the transmission system. Although the oscillations are stable, as pointed out by Hutter (68), further study of this aspect of blade vibration together with an investigation of the various system vibration modes likely to be encountered would be very useful.

A large amplitude vibration phenomenon appears to occur in response to sudden gusts which are sufficiently strong to cause the blade tips to stall and deflect downwind. During the ensuing blade vibration towards a new equilibrium position the blade incidence may oscillate about the stall angle and give rise to stall flutter, which is characterized by a buffeting noise and tip weave, as described by Fung (43). This is in accord with subjective observations on the prototype airscrew from which occasional buffeting noises were heard and bursts of shaft torsional vibration noted on chart traces. Fortunately, these energy absorbing and dissipating oscillations are stable, Fung (43). However, possible aero-elastic means to limit such disturbances and the effect on fatigue life should be considered, especially when progressive blade stalling is used for shaft speed control in gusts. Here there are probably similarities with helicopter blade vibrations, and ciné and sound records of the prototype should be used to confirm predictions of stall flutter phenomena.

The spot "instantaneous" wind speeds V, inserted on the airscrew shaft power against airscrew shaft speed plot, Fig. 36, give some idea of the scatter to be expected in evaluating non-dimensional coefficients based on V, V^2 and V^3. In Table 10, airscrew shaft powers have been grouped above and below 10 hp, the rated horsepower, and it is seen that this tends to group the 5-second wind speed broadly above and below the rated speed of 18.5 mph. When plotting the non-dimensional coefficients a large scatter is inevitable in view of the unsteady atmospheric motion and the difficulty of determining truly representative wind speeds. This has been minimized in Fig. 37, by plotting Cq ($= \frac{Cp}{\mu}$) against μ and inserting rectangular hyperbolae for constant Cp in order to determine the variation of Cp with μ through the definition of Cq. The straight lines used to help define the variation of Cq are taken from Sabinin (62), cited by Hutter (42). The light load, optimum load and stall are well defined in Fig. 37, and it is seen that at optimum load the points cluster about the power coefficient band of Cp between 0.3 and 0.4. A value of maximum power coefficient has therefore been taken as $Cp_{MAX} = 0.35$, which has a tangent to the Cq against μ plot at a tip-speed ratio of $\mu = 7.0$. Transformation to the Cp against μ plot, Fig. 38, gives the characteristic "cubic parabola" shape typical of high-speed windmill rotors. The envelope for optimum operation emphasizes both the sensitivity of Cp to V and the distortion which takes place during transformation.

The experimental value of Cp = 0.35 is less than that of 0.40 assumed for design, while the tip-speed ratio $\mu = 7.0$ for Cp_{MAX} is greater than that of 5.95 assumed for design. For optimum operation with the present 6JC3 load at Greenland, Barbados, the optimum tip-speed ratio has been confirmed to be between 7.0 and 9.0. In view of the uniform, steady state flow assumptions involved in the airscrew design and the non-uniform, unsteady, variable direction, flow in which the experiments were performed, the agreement with theoretical predictions is remarkably good.

The major factors reducing the design Cp by about 12.5 per cent are the fixed orientation of the blade disc, the downwind blade coning angle and additional blade profile drag, together with some inevitable power loss in the airscrew shaft bearing. The effect of fixed orientation can be regarded as reducing the effective wind speed due to yaw, or reducing the effective disc area. It is likely that with downwind blade coning the disc area is also effectively reduced since the flow will have a greater radial component and therefore tend to be more three dimensional and less effective; it could be argued that the reverse would hold for upwind coning. The effect of finite trailing edge thickness, which occurs in most practical blade sections would be to increase profile drag, Hoerner (63), and make careful setting of blade angle essential. These losses may be partially offset by the energy pattern factor, the variation of wind speed with height and by running at a slightly higher tip-speed ratio. The experimental value of tip-speed ratio for maximum power coefficient was higher than the design ratio, in agreement with Nevot's (37) predicted trend for the design solidity and possibly partly accounted for by a nose-up aeroelastic twist. For variable wind speed and the given design lift coefficient, the operating tip-speed ratio can be chosen between $\mu = 5.0$, for a very steady shaft speed, and $\mu = 7.0$ to 9.0 for greater response to wind speed. Alternatively, the blade setting angle can be adjusted to suit site and load.

The scatter evident in the non-dimensional plots indicates that short term field tests to optimize blade angle are not possible without exceptionally good site conditions and continuously adjustable blade setting angle. However, large, modern wind tunnels are available in which variation of performance with blade setting angle could be used to determine optimum setting angle and variable pitch characteristics. The effect of blade coning angle, disc yaw and checks on static gale loading in various configurations could also be made in this manner. The return circuit of the NRC 30 ft. V/STOL wind tunnel (64) should be sufficiently large to allow these tests on full scale blades with minimal wake expansion corrections.

5. CONCLUSIONS

The results of field performance tests on the windmill airscrew allow the performance of the airscrew to be predicted over its range of operation. Together with Ionson's hourly mean windmill performance data (20), the present results can be used to design and predict the performance of medium power, fixed pitch windmill pumping systems with various loads.

The actual shaft power output is slightly less than that predicted from airscrew design theory and the optimum operating point is at a higher shaft speed than predicted. This is due mainly to the limitations of the design theories, to rigid blades operating in steady, uniform, uni-directional flow, so that discrepancies are to be expected with fixed orientation and fixed pitch operation of flexible blades. This mode of operation is acceptable for the type of trade wind regime utilization considered since the inherent economy and simplicity of design, operation and maintenance largely outweigh slight losses in efficiency. Although modern materials and techniques are used in the design, the method of working remains broadly similar to that used for traditional Dutch and Barbadian windmills and has few disadvantages for the type of utilization intended.

Rigorous proof and field tests have shown the design and method of construction of the blades to be sound, although some development and wind tunnel optimization should allow a maximum power coefficient of 0.4 to be realized. Wind tunnel tests would also enable the influence on performance of unsteady, non-uniform flow and various blade layout parameters to be quantified and subsequently used in the airscrew design procedure. The practical limitations of shaft speed and power control by progressive blade stalling, together with blade aero-elastic response to gusts are interesting separated and unsteady flow phenomena which deserve further investigation. Initially, static and dynamic flexural and torsional characteristics of the pre-production blades are required, followed by field performance tests using a reliable high-speed multi-channel recorder to confirm aeroelastic gust response predictions.

When predicting windmill performance some care is required in order not to be over optimistic in estimating wind power available. In this respect existing wind records are often misleading, Rudder (70), and it is concluded that permanent 10 meter reference anemometers should be set up at well attended agrometeorological or airport weather stations. Waterford, Barbados, and Coolidge Field Airport, Antigua, are ideal sites representative of the Windward and Leeward Islands and it is suggested that Department of Transport 45 B anemometers be installed at these reference stations. Short-term synoptic 10 meter wind records at potential sites can then be reliably correlated and spot checks made to determine variation of wind speed with height above typical surface roughnesses.

Relatively long-term 10 meter wind records should be used to check the validity of the polynomial trade wind frequency model and to confirm normalized design wind speeds obtained from wind power frequency maxima. The polynomial approach should be compared with Wenk's wind frequency analysis for temperate wind regimes (65). The monthly wind power-rainfall deficit method appears to be a sound basis for design when wind powered irrigation loads are considered and the method should prove useful for desalination system design load estimation. Where topography at marginal wind powered irrigation sites is a limiting factor the possibility of reducing evapotranspiration and enhancing wind speed in the vicinity of the windmill by suitable choice of shelter-belt gap arrangement, Caborn (67), should not be overlooked. Wind tunnel tests could be made to check this alternative to remote siting and electricity generation.

The prototype windmill has proved itself rugged and reliable throughout two years of commercial irrigation operation at the Greenland test site in Barbados. The test results confirm that at below average wind power sites a minimum of 5 brake horsepower can be guaranteed at a wind speed of 15 mph. Cost estimates indicate that production models developed from the prototype should be fully competitive with diesels on an equivalent cost per horsepower hour basis.

In view of the immediate social and economic need for low-cost power for water lifting and desalination in many small Caribbean communities, and the special local conditions which exist, medium power windmills should be seriously considered as potential power sources. Since windmill prime movers are characterized by a relatively high capital cost and low recurrent operating costs they offer ideal power units for grant aided water development projects. They have the additional advantage of scope for partial and complete local assembly and manufacture at later stages of development.

REFERENCES

1. GOLDING, E.W., Power Supplies, An Overseas Development Institute Development Pamphlet, London. ca. 1962.

2. WARD, G.T., Energy as a Major Factor in Man's Development, Brace Research Institute of McGill University, Technical Report No. T.11., 1964.

3. VADOT, L., General Report: The Design and Testing of Wind Power Plants, U.N. Conference on New Sources of Energy, Rome, E/CONF. 35/GR/7(W), 1961.

4. DASH, J., The Windmills and Copper Walls of Barbados, Journal of the Barbados Museum and Historical Society 31, No. 2: 43-60, 1965.

5. WAILES, R., Windmills-Their Rise and Decline, In Engineering Heritage, Vol. I, The Institution of Mechanical Engineers, London, pp. 10-17, 1963.

6. STOKUYZEN, E., The Dutch Windmill, 2nd ed., Bossum, Holland, C.A.J. van Dishoeck, 1965.

7. GOLDING, E.W., The Generation of Electricity by Wind Power, London, E. and F.N. Spon Limited, 1955.

8. FATEEV, E.M., Wind Turbines and Their Application to Agriculture. (Translated title), Moscow, State Scientific and Technical Publishing House of Mechanical Engineering Literature, 1962.

9. TAGG, J.R., Wind Data Related to the Generation of Electricity by Wind Power. The British Electrical and Allied Industries Research Association, Technical Report C/T115, 1957.

10. LAWAND, T.A., The Economics of Wind Powered Desalination Systems, Brace Research Institute of McGill University, Technical Report No. T.36, 1967.

11. CHILCOTT, R.E., Notes on the Development of the Brace Airscrew Windmill as a Prime Mover, The Aeronautical Journal of the Royal Aeronautical Society 73:333-334, 1969.

12. CHILCOTT, R.E., Report on Visit to St. Kitts-Nevis, February 12-17, 1968, Brace Research Institute of McGill University, Internal Report No. I.40, 1968.

13. CHILCOTT, R.E., Implications of the Utilization of Wind Power for the Development of Small Caribbean Communities, Proceedings of the Conference on Technology and Development in Small Transitional

Societies, University of the West Indies, St. Augustine, Trinidad, West Indies, February 1968.

14. GOLDING, E.W., Windmills for Water Lifting and the Generation of Electricity on the Farm, F.A.O. Farm Power Informal Working Bulletin No. 17, ca. 1961.

15. VADOT, L., Le Pompage de l'Eau par Eoliennes, (Water Pumping by Windmills), La Houille Blanche No. 4; 496-535, September 1957.

16. LONGMUIR, J.C., Derby, England. Traditional cane crushing windmill transmission efficiency and power requirements, Private Communication, June 1969.

17. SMITH, A.A., Bromley, England. Traditional cane crushing windmill transmission efficiency and power requirements, Private Communication, June 1969.

18. 'PRINSENMOLEN' -COMMITTEE, Research Inspired by the Dutch Windmills, Wageningen, Holland, H. Veenman en Zonen, ca. 1960.

19. HAVINGA, A., Experiments with a Model of a Scoop Wheel. (Translated title) The Hague, Koninklijk Institut van Ingenieurs, De Ingenieur, 8: February 1934.

20. IONSON, J.M., The Field Performance of a Windmill Powered Sprinkler Irrigation System. Unpublished M.Sc. Thesis, Agricultural Engineering Department, Macdonald College of McGill University, 1969.

21. SKEETE, C.C., A Description of the Weather of the Island of Barbados, W.I. Unpublished Manuscript, Woods Hole Oceanographic Institution, Woods Hole Massachusetts, Reference No. 63-19, 1963.

22. LAMB, H.H., Britain's Climate, The Chartered Mechanical Engineer 15: 2-6, 1968.

23. GARNIER, B.J., The McGill University Climatology Programme in Barbados, Bulletin of the American Meteorological Society 49: 636-639, 1968.

24. GARSTANG, MICHAEL, The Role of Momentum Exchange in Flow Over a Heated Island, Internal Report, Florida State University, Tallahassee, Florida, 1967.

25. CHILCOTT, R.E., and LAKE, E.B., Proposal for the Establishment of a 10 hp Windmill Water Pumping Pilot Plant in Nevis, West Indies. Brace Research Institute of McGill University, Internal Report No. I.45, 1968.

26. ANDERSON, S.R., A Comparison of Monthly Wind Mileages from 45B and U2A Anemometers, Mimeo. Canada Department of Transport, Toronto, ca. 1965.

27. BEST, D.F., Seawell Airport, Barbados. Wind speed and rainfall data for Seawell, Private Communication, June 1969.

28. ANON, Fortran Subroutine for Polynomial Regression IBM System 360 Scientific Subroutine Package (360A-CM-03X) Version III Programmer's Mannual.

29. LEVY, N., Current State of Windpower Research in the Soviet Union, Edited by G. T. Ward, Brace Research Institute of McGill University, Technical Report No. T.56., 1968.

30. ANON, "SOKOL" 15 kw Wind Turbines Destined for Cuba, Izvestia, Moscow, August 23, 1966.

31. LAMB, H.H., The English Climate, 2nd ed. London, The English Universities Press, 1964.

32. BETZ, A., Wind Energy and its Utilization by Windmills. (Translated title), Wind-Energie und ihre Austnutzung durch Windmuhlen, Gottingen, Bandenhoek and Ruprecht, 1926.

33. THEODORSEN, T., Theory of Propellers, New York, McGraw-Hill, 1948.

34. HUTTER, U., Betriebserfahrungen mit einer Windkraftanlage von 100kw. VDI-Verlag GmbH. Dusseldorf, Sonderdruck aus Brennstoff-Warme-Kraft 16, No. 7: 333-340, July 1964.

35. GLAUERT, H., Windmills and Fans, In Aerodynamic Theory, Vol. 1V: 324-332, William Frederick Durand ed., New York, Dover Publications, Inc., 1963.

36. LILLEY, G.M. and RAINBIRD, W.J., A Preliminary Report on the Design and Performance of Ducted Windmills. The British Electrical and Allied Industries Research Association Technical Report No. C/T119, 1957.

37. NEVOT, M.A., Comparacion de Differentes Teorias Para el Calculo de la Performence de Molinos, Brace Research Institute of McGill University Report No. DT. 7, August 1966.

38. LOCK, C.N.H. and YEATMAN, D., Tables for Use in an Improved Method of Airscrew Strip Theory Calculation, Aeronautical Research Committee Reports and Memoranda No. 1674, 1934.

39. IWASAKI, MATSUNOSUKE, The Experimental and Theoretical Investigation of Windmills, Japan, Reports of Research Institute for Applied Mechanics, 11, No. 8: 181-229, 1953.

40. LOCK, C.N.H., A Graphical Method of Calculating the Performance of an Airscrew, Aeronautical Research Committee Reports and Memoranda no. 1849, 1938.

41. ABBOTT, I.H., and VON DOENHOFF, A.E., Theory of Wing Sections, New York, Dover Publication, Inc., 1959.

42. HUTTER, U., The aerodynamic layout of wind blades of wind-turbines with high tip-speed ratio, United Nations Conference on New Sources of Energy, E/CONF. 35/W/31, Rome, 1961.

43. FUNG, Y.C., An Introduction to the Theory of Aeroelasticity, Galcit Aeronautical Series, New York, John Wiley and Sons, Inc., c1955.

44. KER WILSON, W., Practical Solution of Torsional Vibration Problems, New York, John Wiley and Sons, Inc. 1956.

45. HUTTER, U., Glass-fiber reinforced plastics as structural material for the aircraft industry, Society of the Plastics Industry, Inc., Technical and Management Conference Reinforced Plastics Division, 16th Annual Meeting, Section 13-A, Chicago, 1961.

46. GRESHAM, H.E., and HANNAH, C.G., Reinforced Plastics for Jet Lift Engines, Journal of the Royal Aeronautical Society 71: 355-365. May 1967.

47. KETLEY, G.R., Propellers for Hovercraft, Hawker Siddley Technical Review 5, No. 1: 20-29, 1969.

48. TAYLOR-RUSSELL, A.J., Department of Aeronautics, Imperial College, London, Wakes Downstream of Perforated Plates, Private Communication, June 1969.

49. FIEDLER, H.E., and WILLE, R., Some Measurements in the Immediate Near Wake of Blunt Bodies, American Institute of Aeronautics and Astronautics Paper No. 69-746, July 1969.

50. POPE, Alan, Wind-Tunnel Testing, 2nd ed., New York, John Wiley and Sons, Inc., 1958.

51. TUPLIN, W. A.,. Gear Design, Brighton, England, The Machinery Publishing Co., 1962.

52. THE INSTITUTION OF MECHANICAL ENGINEERS, Commercial Vehicles, London, The Institution of Mechanical Engineers, c1967.

53. GILLESPIE, S.L., Woodward Company, Fort Collins, Colorado, Overspeed trip for Brace Windmill, Private communication, July 10, 1968.

54. NEWCOMB, T.P. and SPURR, R.T., Braking of Road Vehicles, London, Chapman and Hall Ltd., 1967.

55. PUGSLEY, A., The Safety of Structures, London, Edward Arnold (Publishers) Ltd., 1966.

56. WENNINGA, E., Epothane Products Ltd., Montreal, Quotation for Fabrication of Production Brace Airscrew Windmill Blades, Private communication, March 31, 1969.

57. WILSON, W.M., Heroux Limited, Longueuil, Quebec, Quotation for Manufacture of Steelwork to Prototype Brace Airscrew Windmill Specification, Private communication, May 29, 1969.

58. ANON, Benefit-cost evaluations as applied to aid financed water or related land use projects, Supplement No. 1 to Feasibility Studies Economic and Technical Soundness Analysis Capital Projects, Office of Engineering, U.S. Department of State, Agency for International Development, May 31, 1963.

59. ANGUS, S.F., Dominion Bridge Company Limited, Montreal, Design Contract for Production Brace Windmills for manufacture in Canada and Elsewhere, Private communication, March 28, 1969.

60. VILLERS, D.E., The testing of wind-driven generators operating in parallel with a network, British Electrical and Allied Research Association Technical Report C/T116, 1957.

61. BEAN, W.T., Instructions for MAK-IT Slip Ring Kit, Detroit 23, Michigan, William T. Bean, Inc., 1953.

62. SABININ, G.H., Theorie Schnellaufender Stabilisierter Windrader, (Russian), ZAGI, S 16/20, Ausg. Buro Wissenschaftlicher Information, 1953.

63. HOERNER, SIGHARD F., Fluid Dynamic Drag, Midland Park, New Jersey, published by the author, 1958.

64. ANON, The NRC 30 Foot V/STOL Wind Tunnel, Ottawa National Research Council of Canada, June 1969.

65. WENK, F., Die analyt, Formulierung der Wind-starken-Haufigkeitskurve, Stuttgart, Windkraft-Mitt, Nr.5.

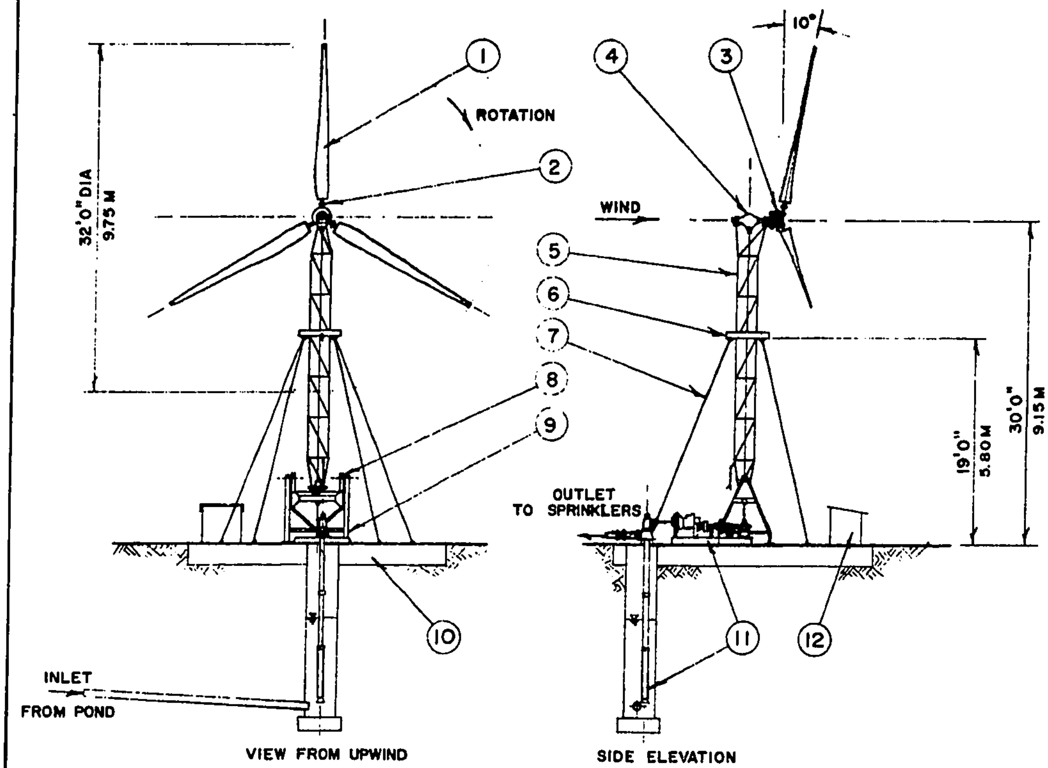
66. HUTTER, U., Wind Power Glossary, (Translated title) Windkraft in Stichworten, In Beitrag zum Lueger Lexikon der Technik, BD 6/7, Stuttgart, Deutsche Verlags-anstalt, ca., 1965.

67. CABORN, J.M., Shelterbelts and Windbreaks, London, Faber and Faber Ltd., c1965.

68. HUTTER, U., Recent Developments and Potential Improvements in Wind Power Utilization, United Nations Conference on New Sources of Energy, E/CONF.35/GR/8(W), Rome, 1961.

69. CLAUSNIZER, G., Various relationships between wind speed and power output of a wind power plant, U.N. Conference on New Sources of Energy, Rome, E/CONF. Agendy Item No. 11. B.2 (b), 1961.

70. RUDDER, G.M.D., Meteorological Office, Seaswell Airport, Barbados, Barbados Wind Records, Private Communication, July 16, 1969.



PROTOTYPE BRACE AIRSCREW WINDMILL PUMPING SYSTEM.

DRAWING NO.	ITEM NO	DESCRIPTION
F-3-002	14	BLADE SHIPPING AND STORAGE CONTAINER
A-3-008	13	PARTS LIST FOR ONE SET OF BLADES
D-3-006	12	ERECTING GEAR
E-3-003	11	TRANSMISSION MACHINERY PAD AND PUMP
D-3-008	10	FOUNDATION
D-3-007	9	TOWER BASE
D-3-004	8	TOWER YOKE
C-3-012	7	GUYS
E-3-002/2	6	TOWER BEARING
E-3-004	5	TOWER
D-3-001	4	TRUCK REAR AXLE
D-3-005	3	AIRSCREW HUB
D-3-011	2	AIRSCREW BLADE ROOT
F-3-001	1	AIRSCREW BLADE

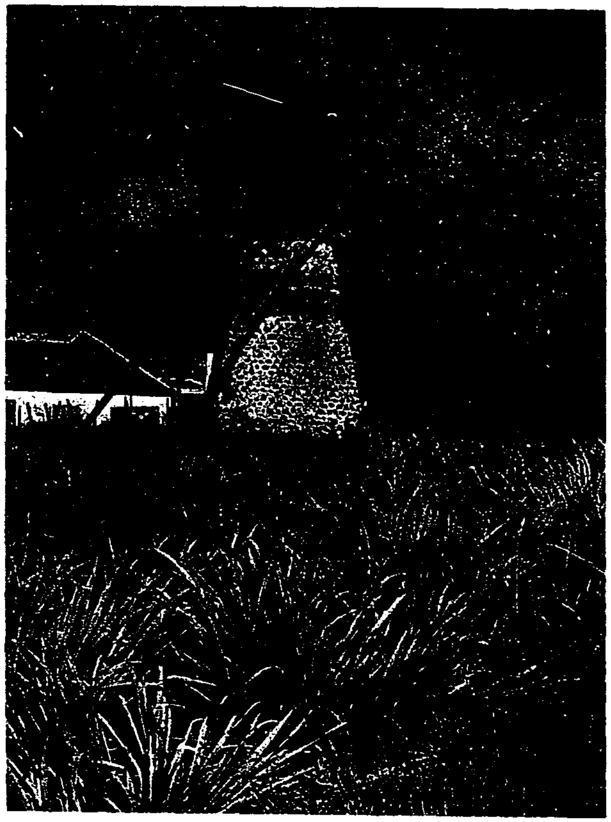
BRACE RESEARCH INSTITUTE
OF McGILL UNIVERSITY
MACDONALD COLLEGE, P.Q. CANADA

TITLE GENERAL ASSEMBLY
BRACE AIRSCREW WINDMILL

SCALE 1/100

DRAWN	R.H.WEYTS	JULY 1968
STRESSED		
CHECKED		
APPROVED	R.E.CHILCOTT	JULY 1968
ASSEMBLY		
DWG. No.	B-3-041	

Fig. 1. Prototype Brace airscrew windmill pumping system.



(a) Lowland, Christ Church.

(b) Morgan Lewis, St. Andrew.

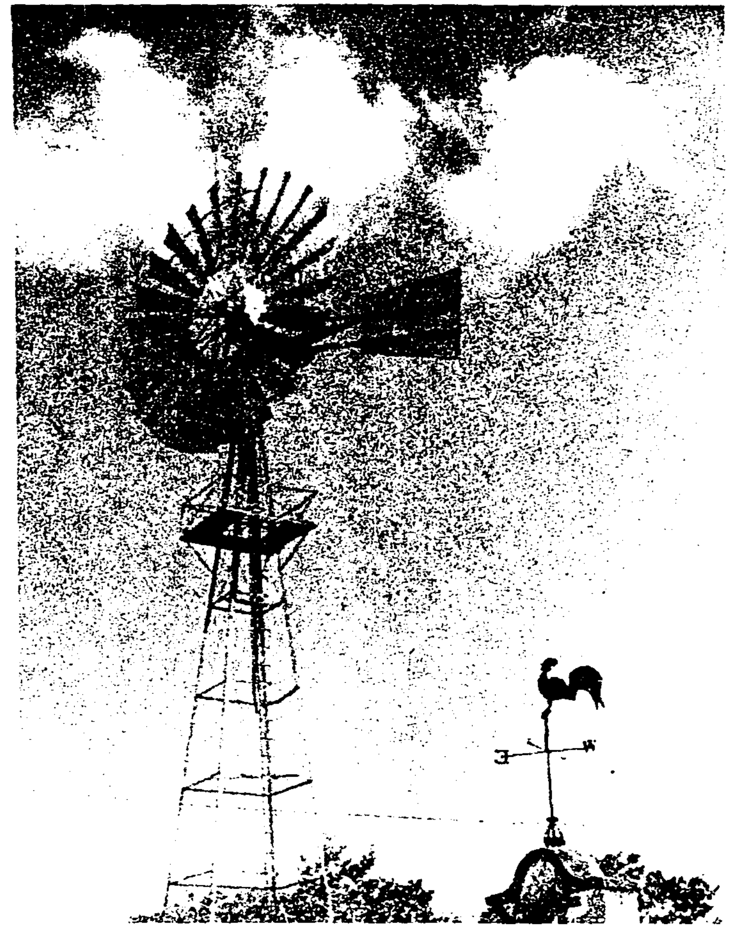
Fig. 2. Traditional cane crushing windmills, Barbados.

(a) Lowland, Christ Church.

(b) Morgan Lewis, St. Andrew.

Fig. 2. Traditional cane crushing windmills, Barbados.

(a) Kirton, St. Philip.



(b) Graeme Hall, Christ Church.

Fig. 3. Water lifting windmills, Barbados.

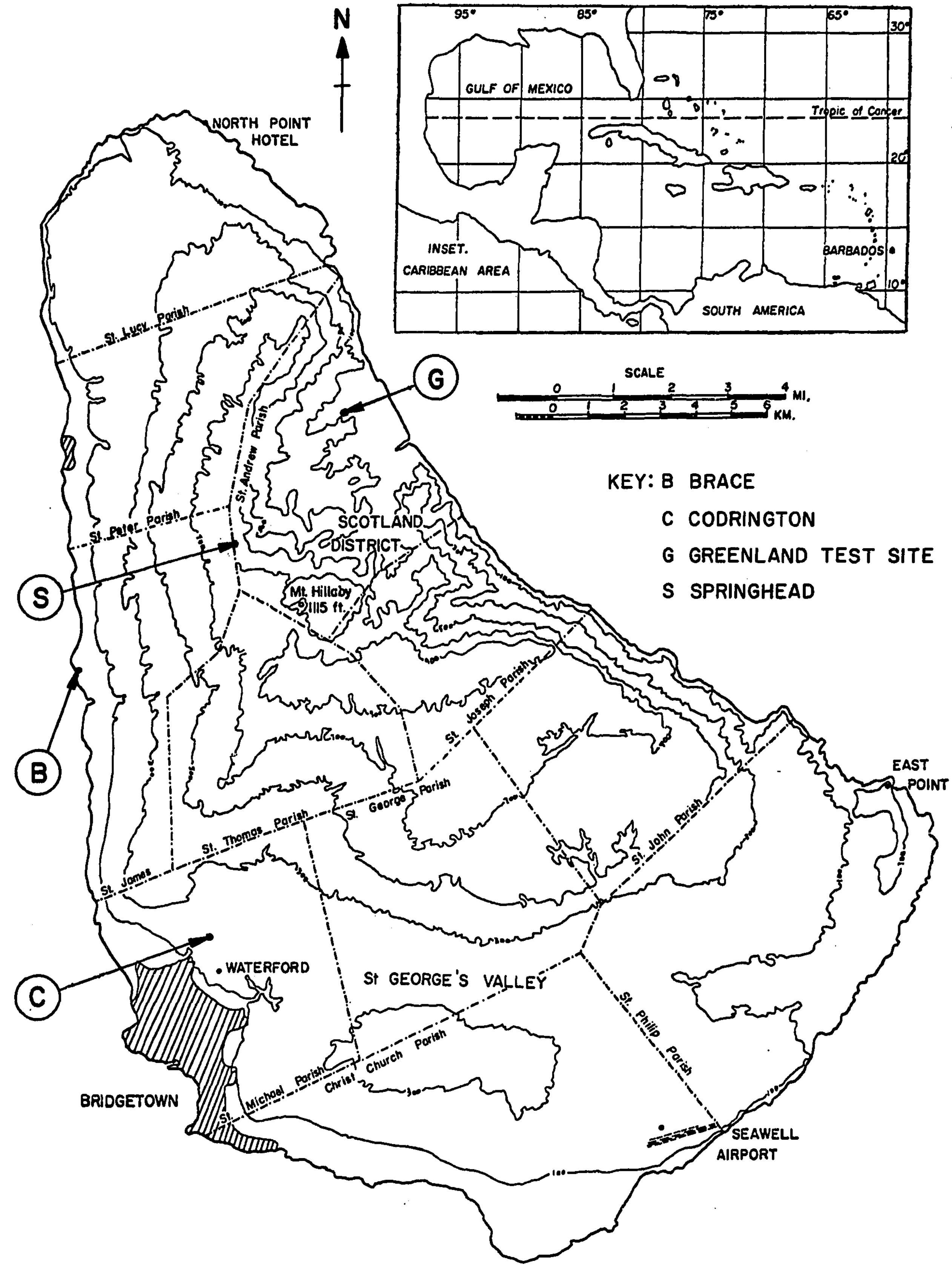


FIG. 4. BARBADOS LOCATION MAP

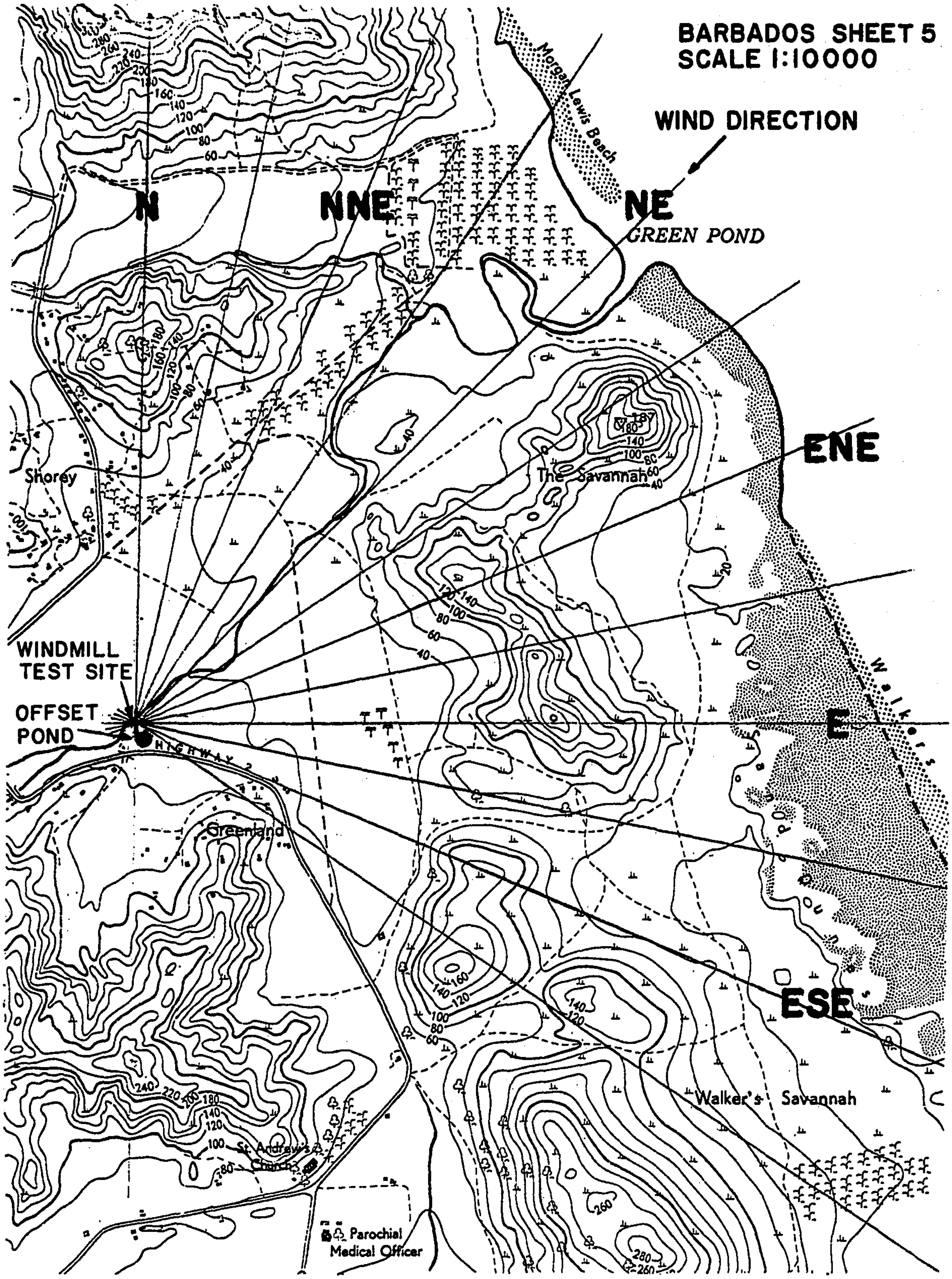


FIG.5. GREENLAND TEST SITE LOCATION.

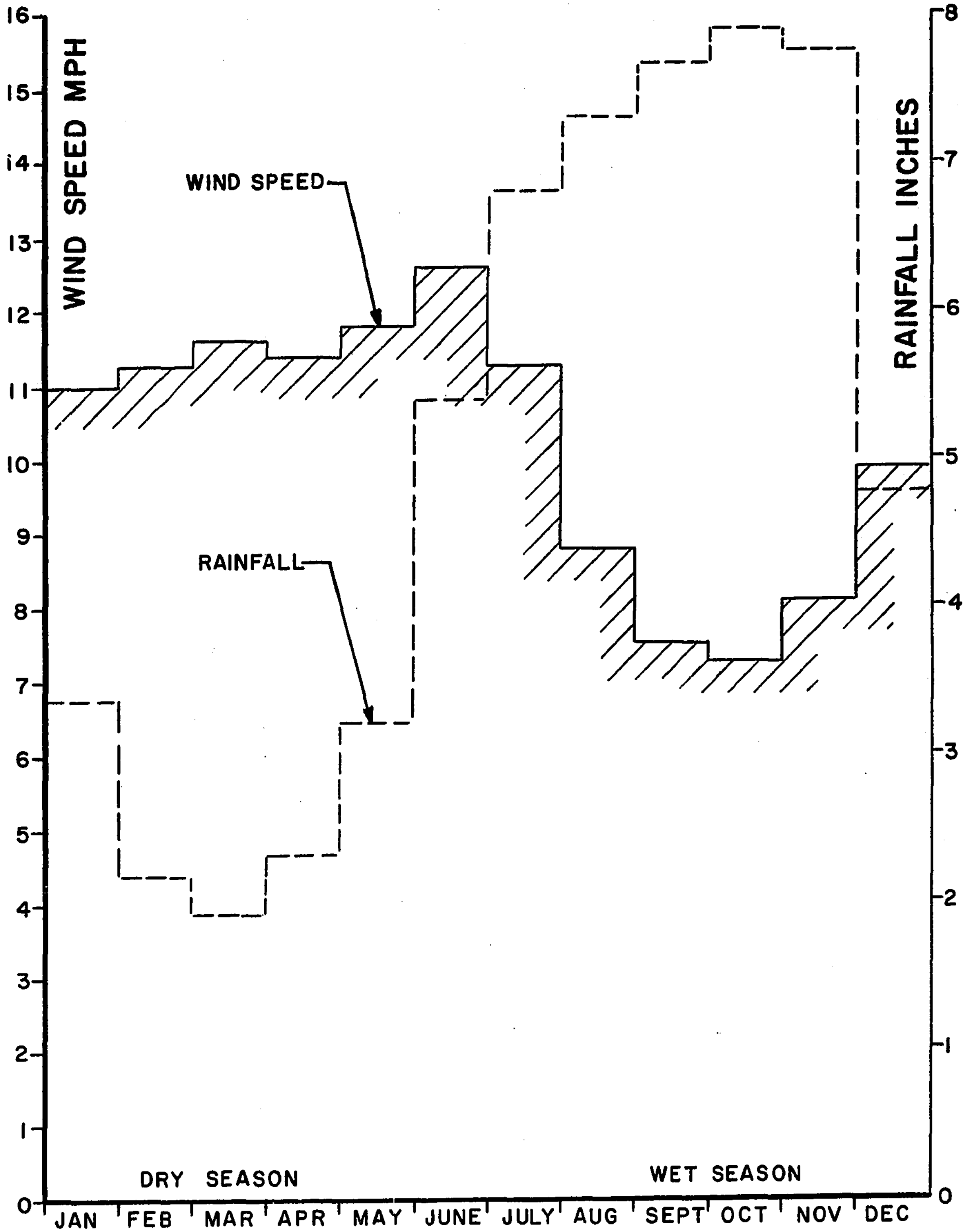


FIG. 6. CODRINGTON WIND SPEED AND ISLAND RAINFALL SEASONAL CHARACTERISTICS.

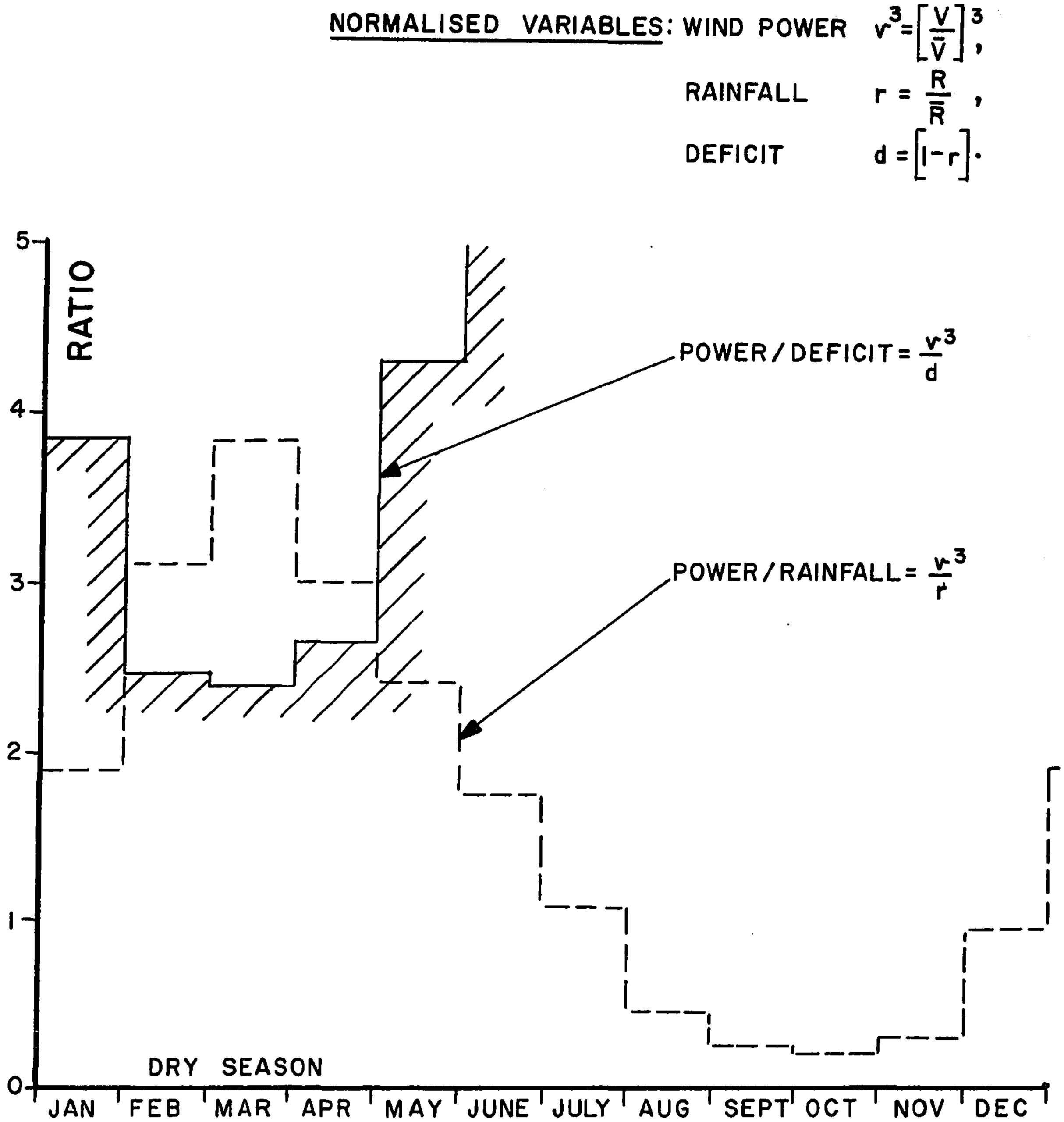


FIG. 7. CODRINGTON MONTHLY WIND POWER-ISLAND RAINFALL RELATIONSHIPS.

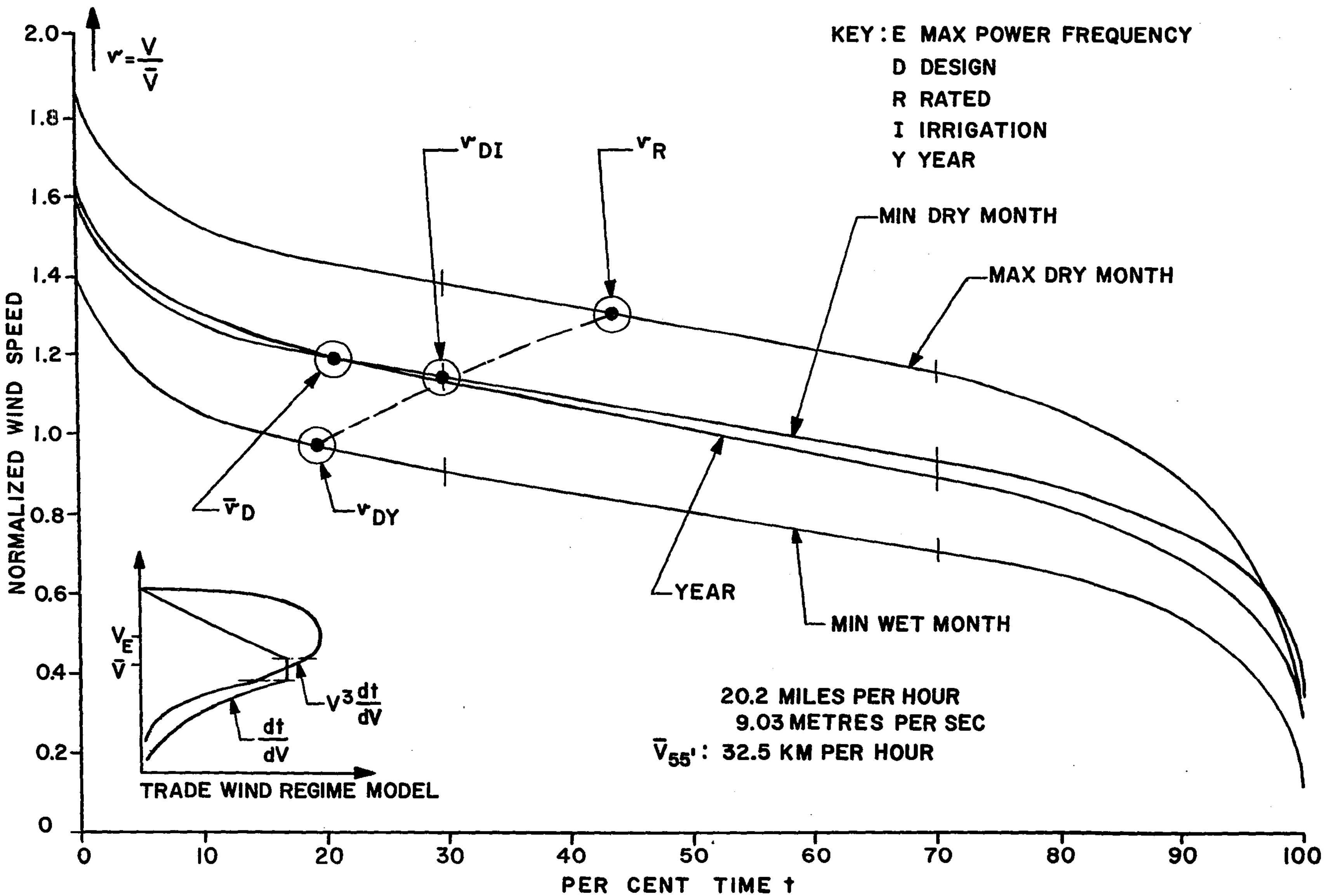


FIG.8. SPRINGHEAD VELOCITY-DURATION CURVES 1962.

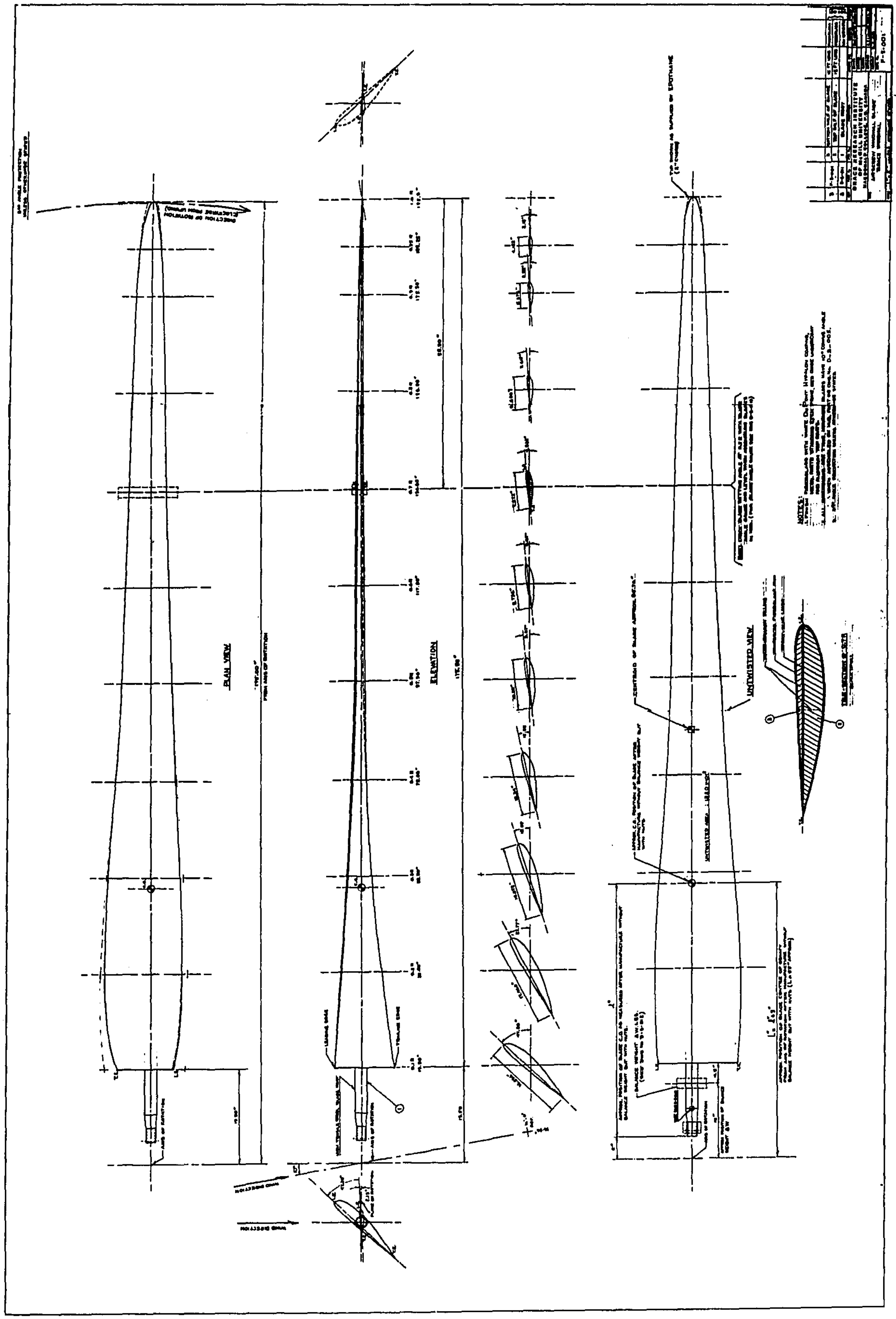


Fig. 9. Airscrew windmill blade.

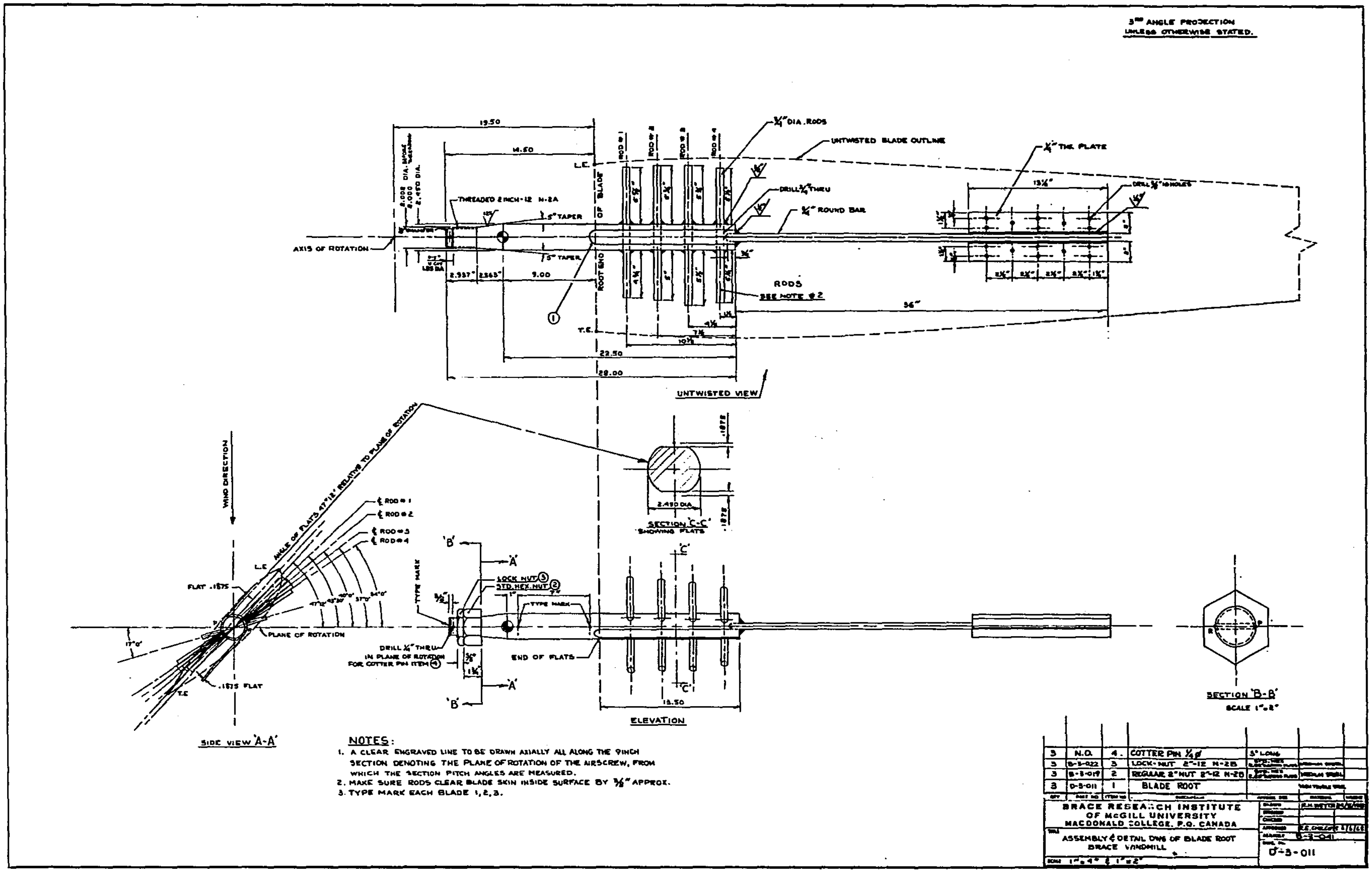


Fig. 10. Airscrew blade root.

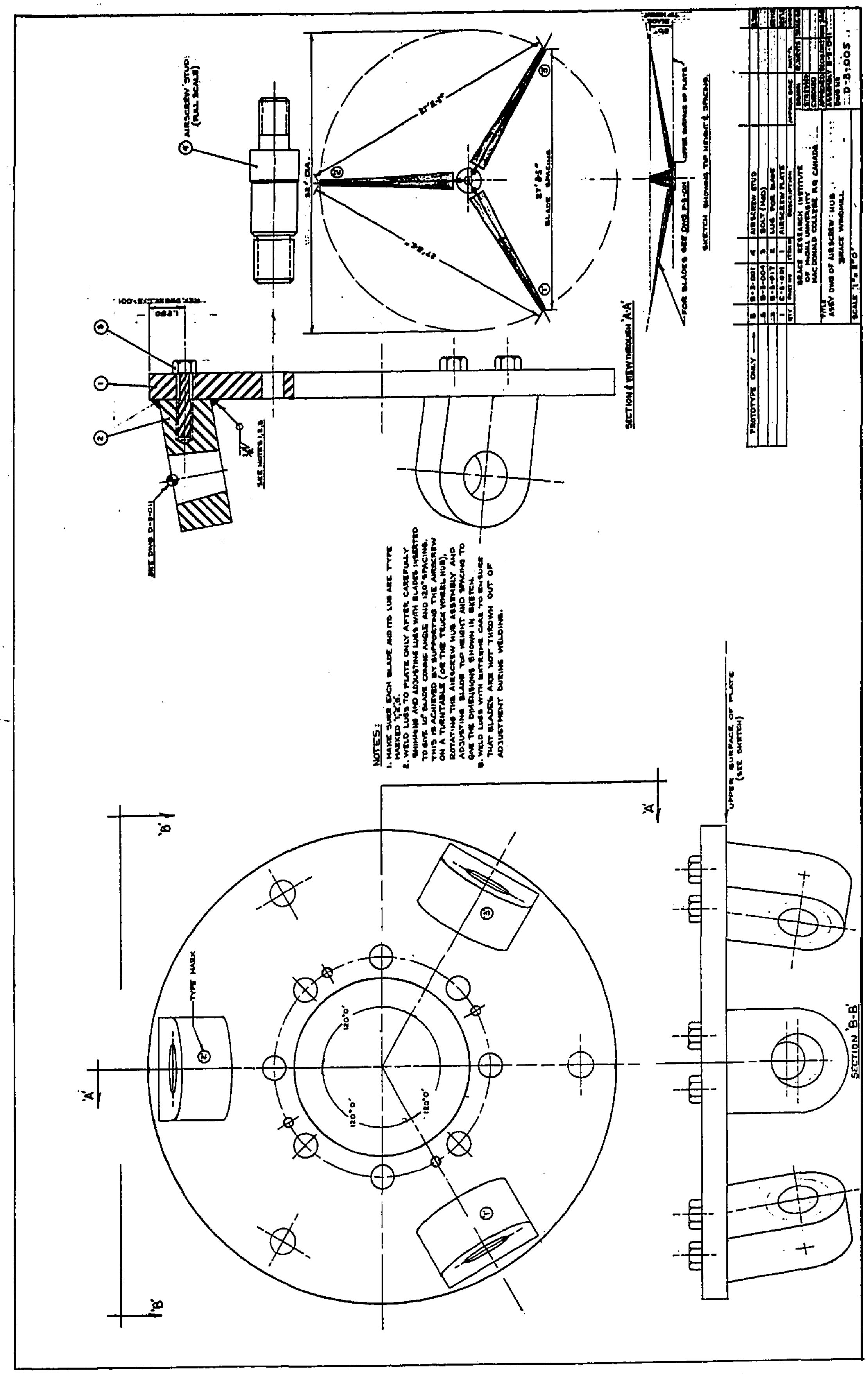
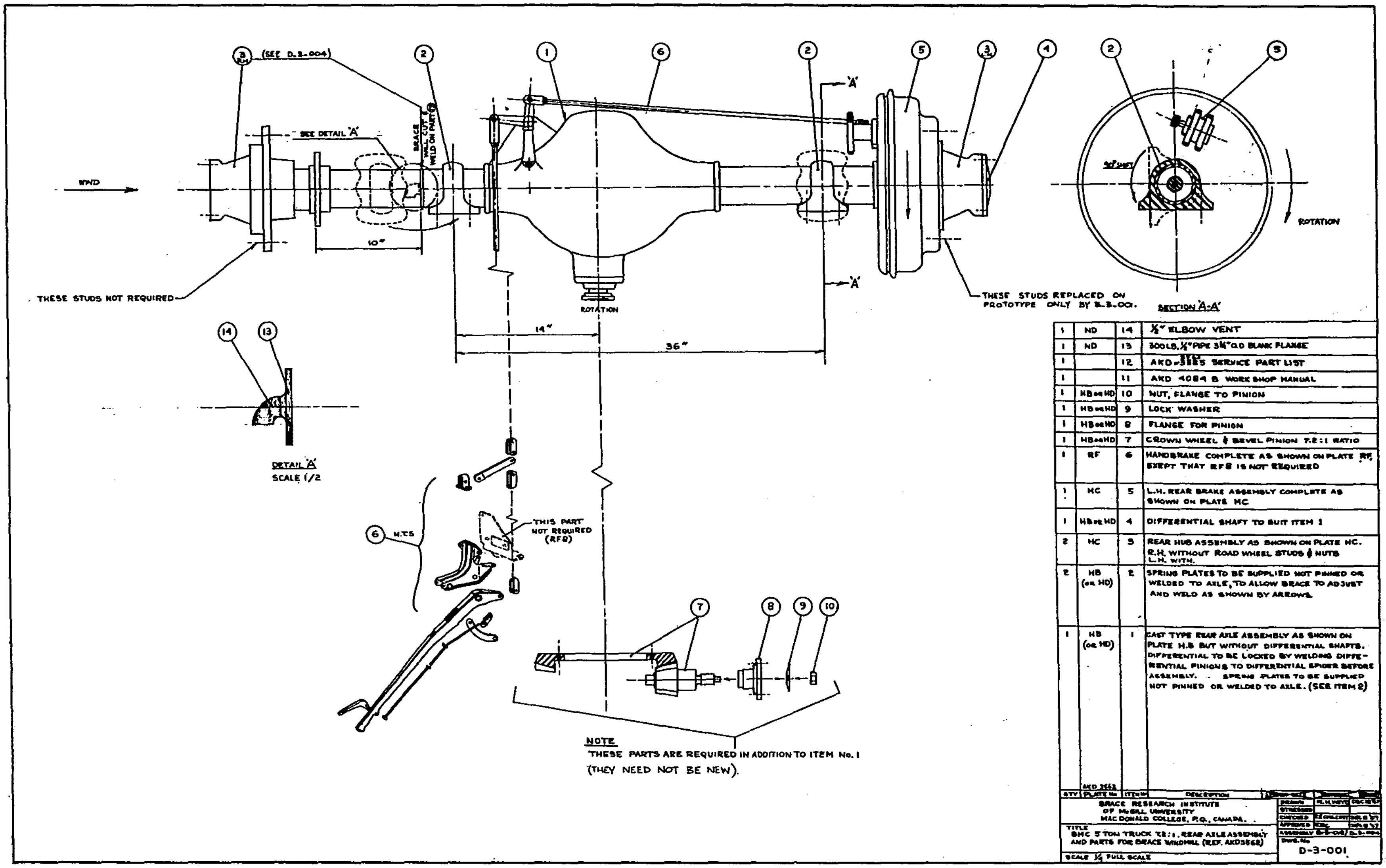


Fig. 11. Airscrew hub assembly.



QTY	PLATE No	ITEM	DESCRIPTION
1	ND	14	½" ELBOW VENT
1	ND	13	300LB, ½" PIPE 3¼" O.D BLANK FLANGE
1		12	AKD-3585 SERVICE PART LIST
1		11	AKD 4084 B WORK SHOP MANUAL
1	HB or HD	10	NUT, FLANGE TO PINION
1	HB or HD	9	LOCK WASHER
1	HB or HD	8	FLANGE FOR PINION
1	HB or HD	7	CROWN WHEEL & BEVEL PINION 7.2:1 RATIO
1	RF	6	HANDBRAKE COMPLETE AS SHOWN ON PLATE RF, EXEPT THAT RF8 IS NOT REQUIRED
1	HC	5	L.H. REAR BRAKE ASSEMBLY COMPLETE AS SHOWN ON PLATE HC
1	HB or HD	4	DIFFERENTIAL SHAFT TO SUIT ITEM 1
2	HC	3	REAR HUB ASSEMBLY AS SHOWN ON PLATE HC. R.H. WITHOUT ROAD WHEEL STUDS & NUTS L.H. WITH.
2	HB (or HD)	2	SPRING PLATES TO BE SUPPLIED NOT PINNED OR WELDED TO AXLE, TO ALLOW BRACE TO ADJUST AND WELD AS SHOWN BY ARROWS.
1	HB (or HD) AKD 3562	1	CAST TYPE REAR AXLE ASSEMBLY AS SHOWN ON PLATE H.B BUT WITHOUT DIFFERENTIAL SHAFTS. DIFFERENTIAL TO BE LOCKED BY WELDING DIFFERENTIAL PINIONS TO DIFFERENTIAL SPIDER BEFORE ASSEMBLY. SPRING PLATES TO BE SUPPLIED NOT PINNED OR WELDED TO AXLE. (SEE ITEM 2)

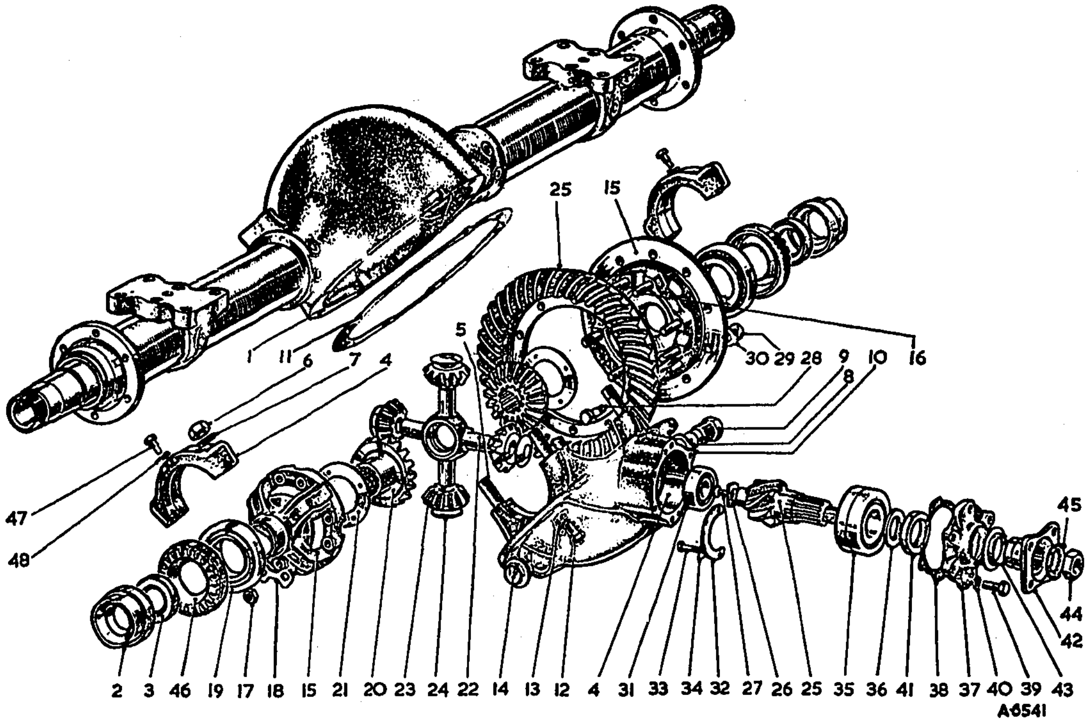
BRACE RESEARCH INSTITUTE
OF McGILL UNIVERSITY
MAC DONALD COLLEGE, P.Q., CANADA.

TITLE
BMC 5 TON TRUCK 7.2:1, REAR AXLE ASSEMBLY AND PARTS FOR BRACE WINDMILL (REF. AKD3562)

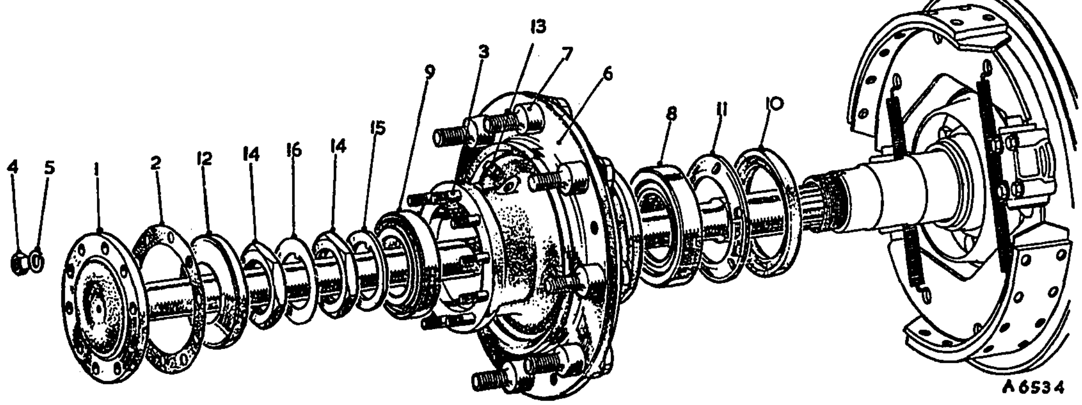
SCALE ¼ FULL SCALE

DWG. No D-3-001

Fig. 12. Truck rear axle modifications.



(a) AXLE HOUSING AND DIFFERENTIAL.



(b) REAR HUB AND HALF SHAFT.

FIG.13. TRUCK REAR AXLE TRANSMISSION DETAILS.

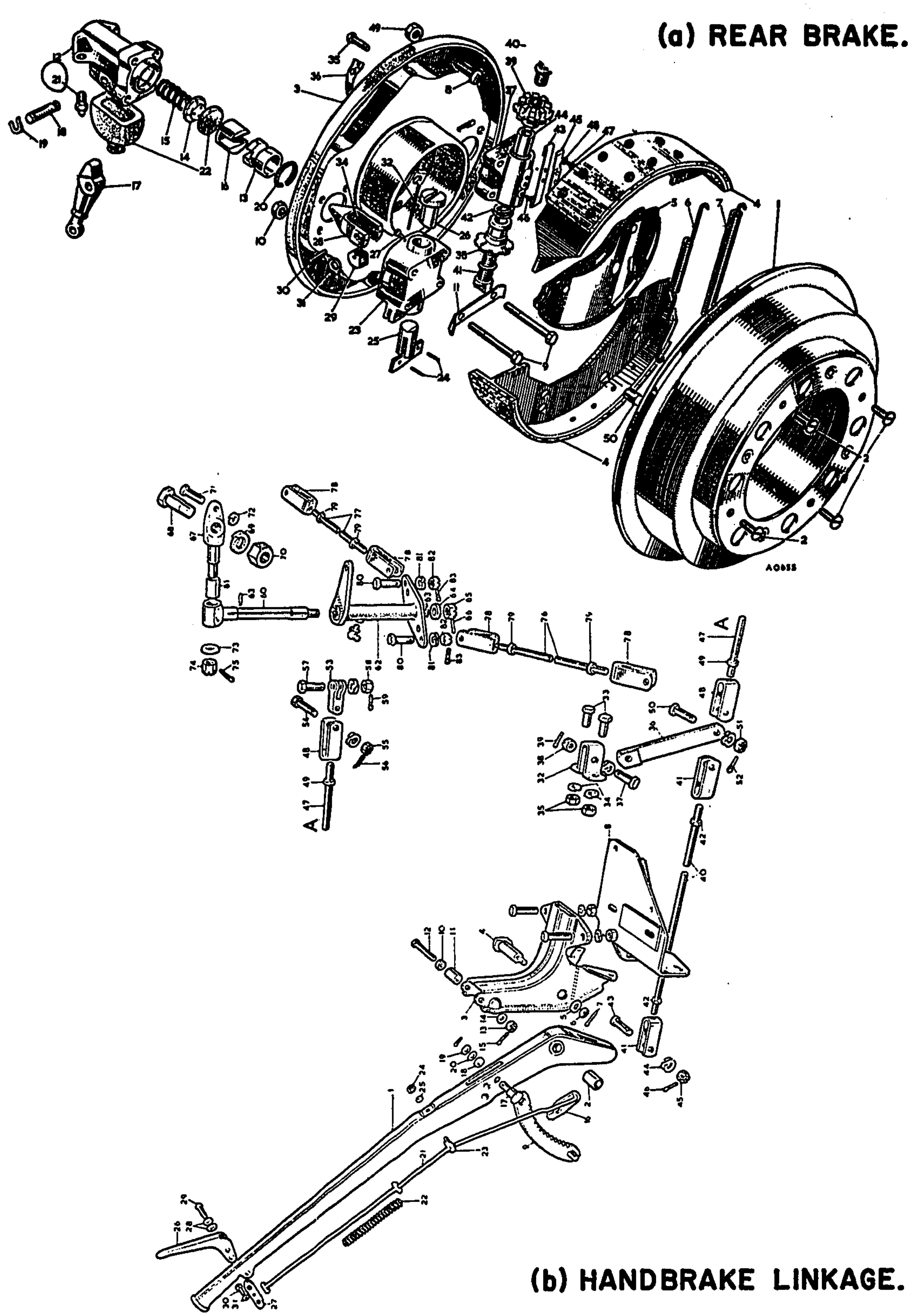


FIG.14. TRUCK HANDBRAKE SYSTEM.

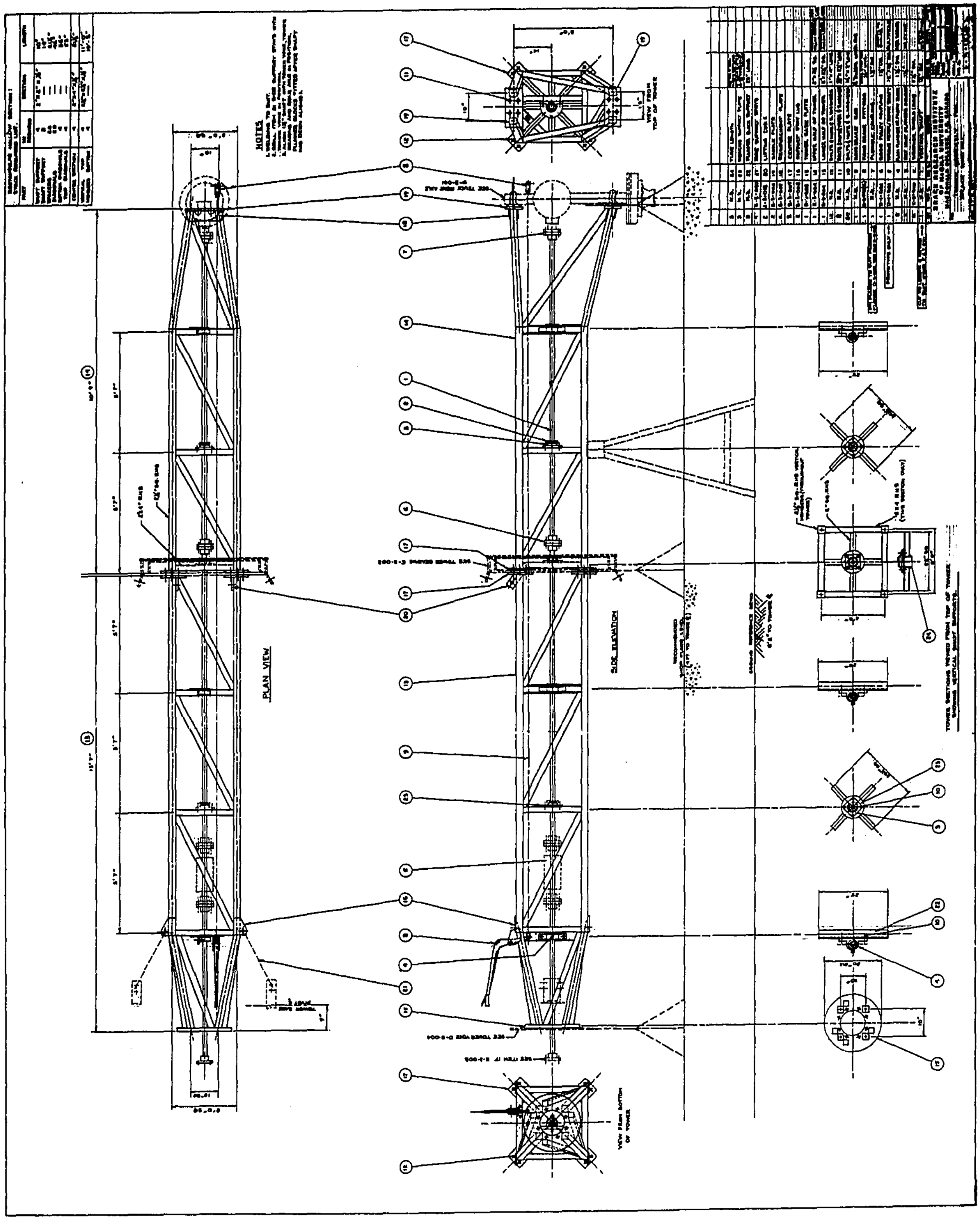
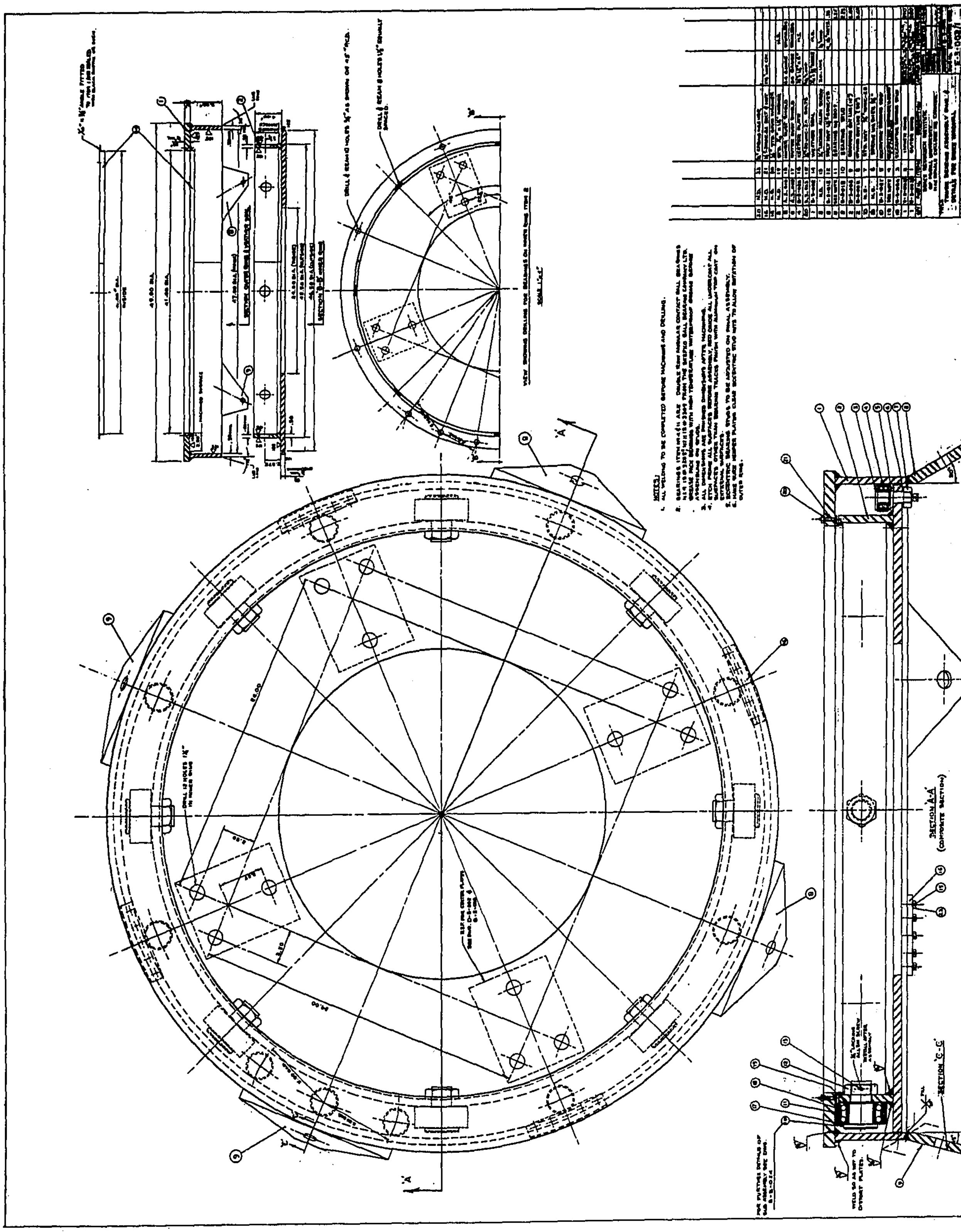


Fig. 15. Tower structure arrangement.



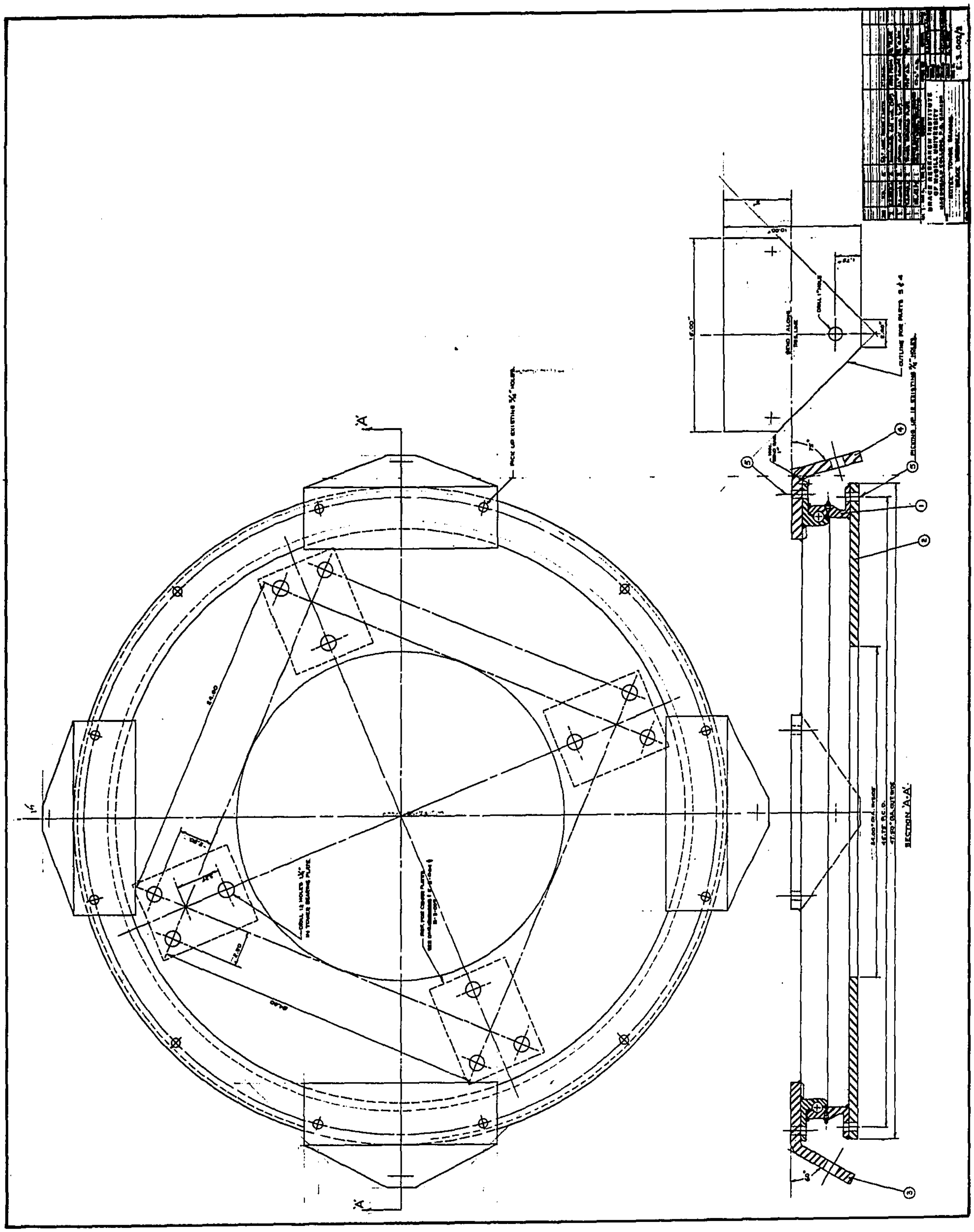


Fig. 17. Tower bearing, Rotek fifth wheel.

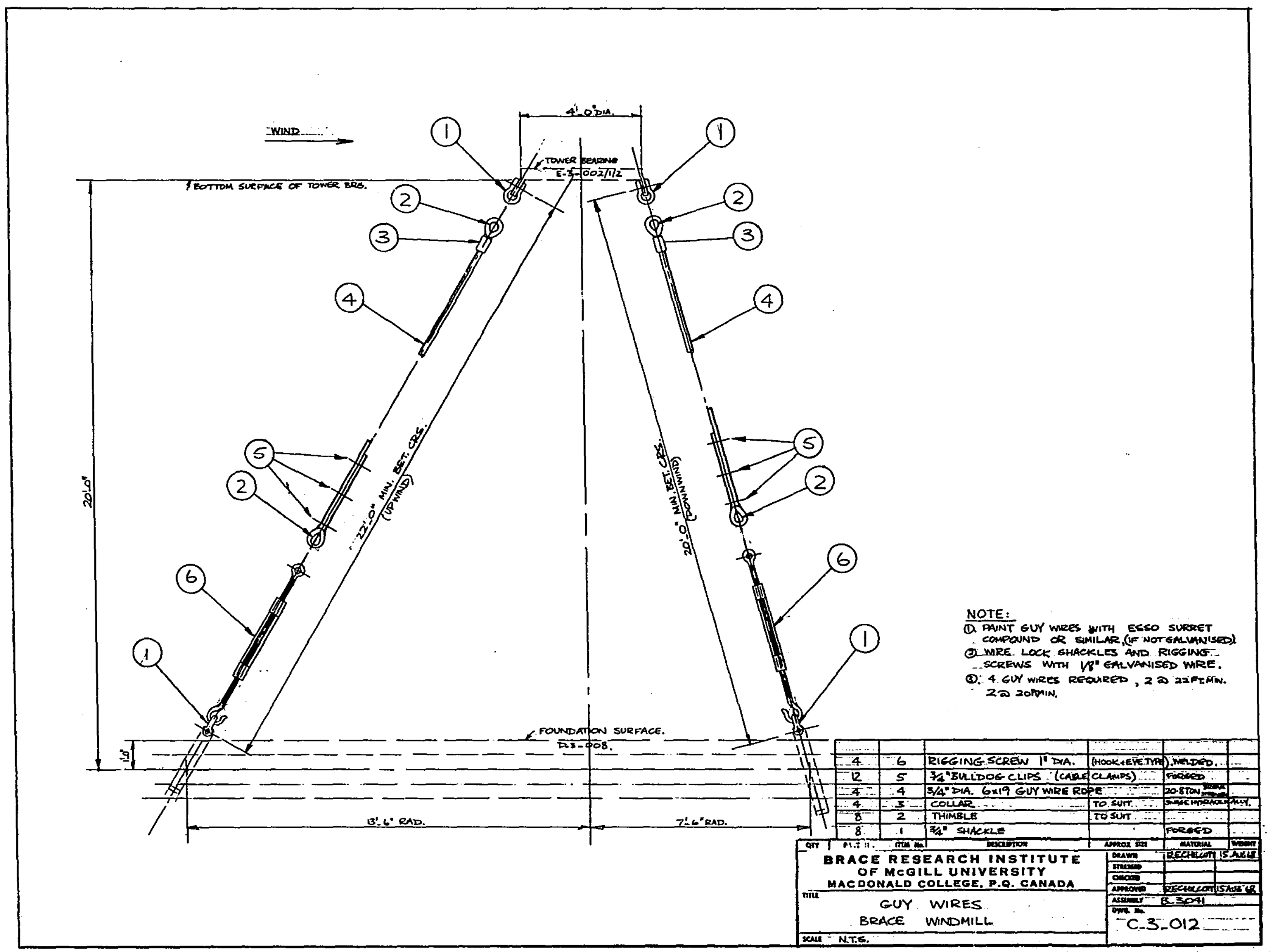


Fig. 18. Tower guy wires.

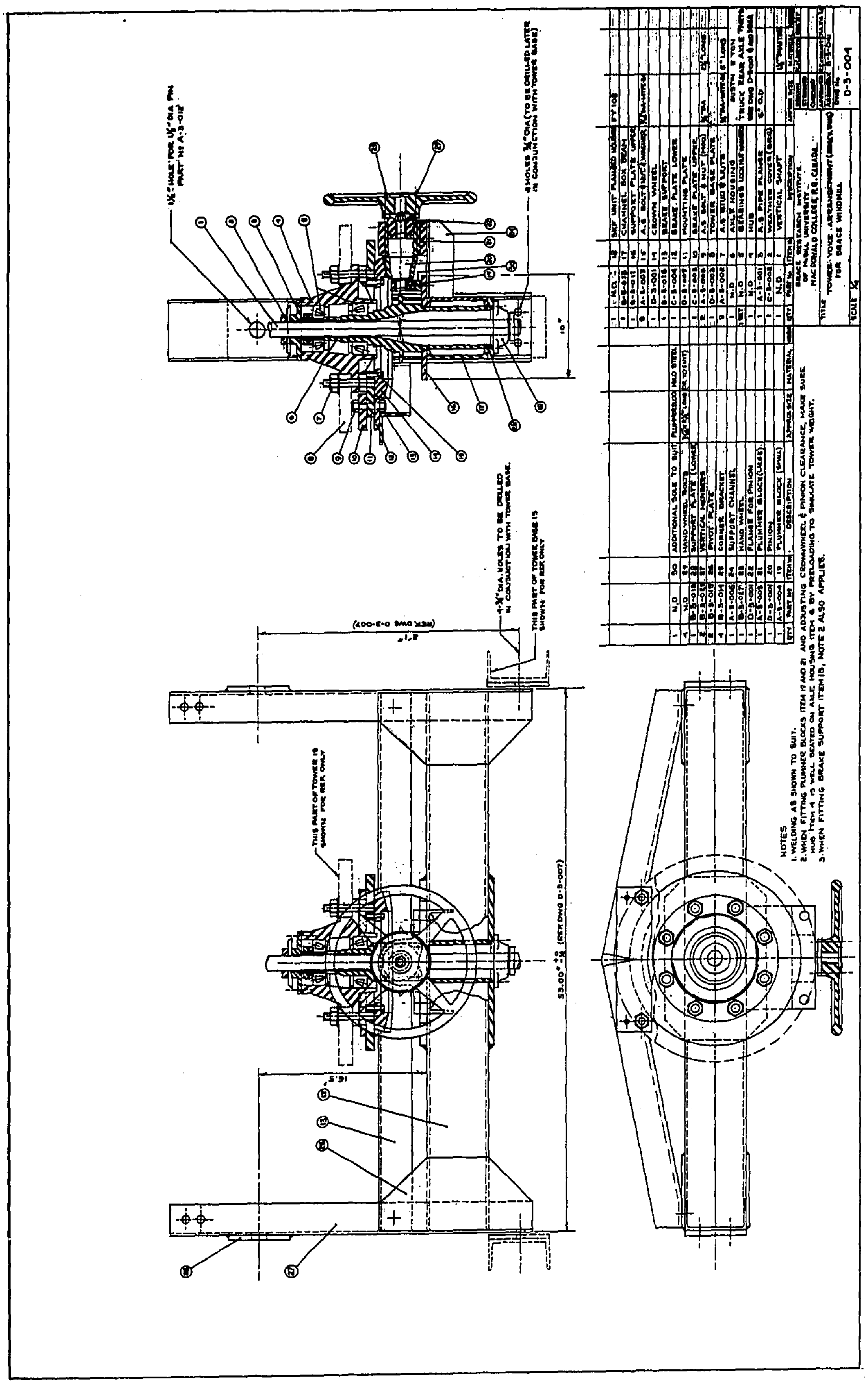


Fig. 19. Tower yoke assembly.

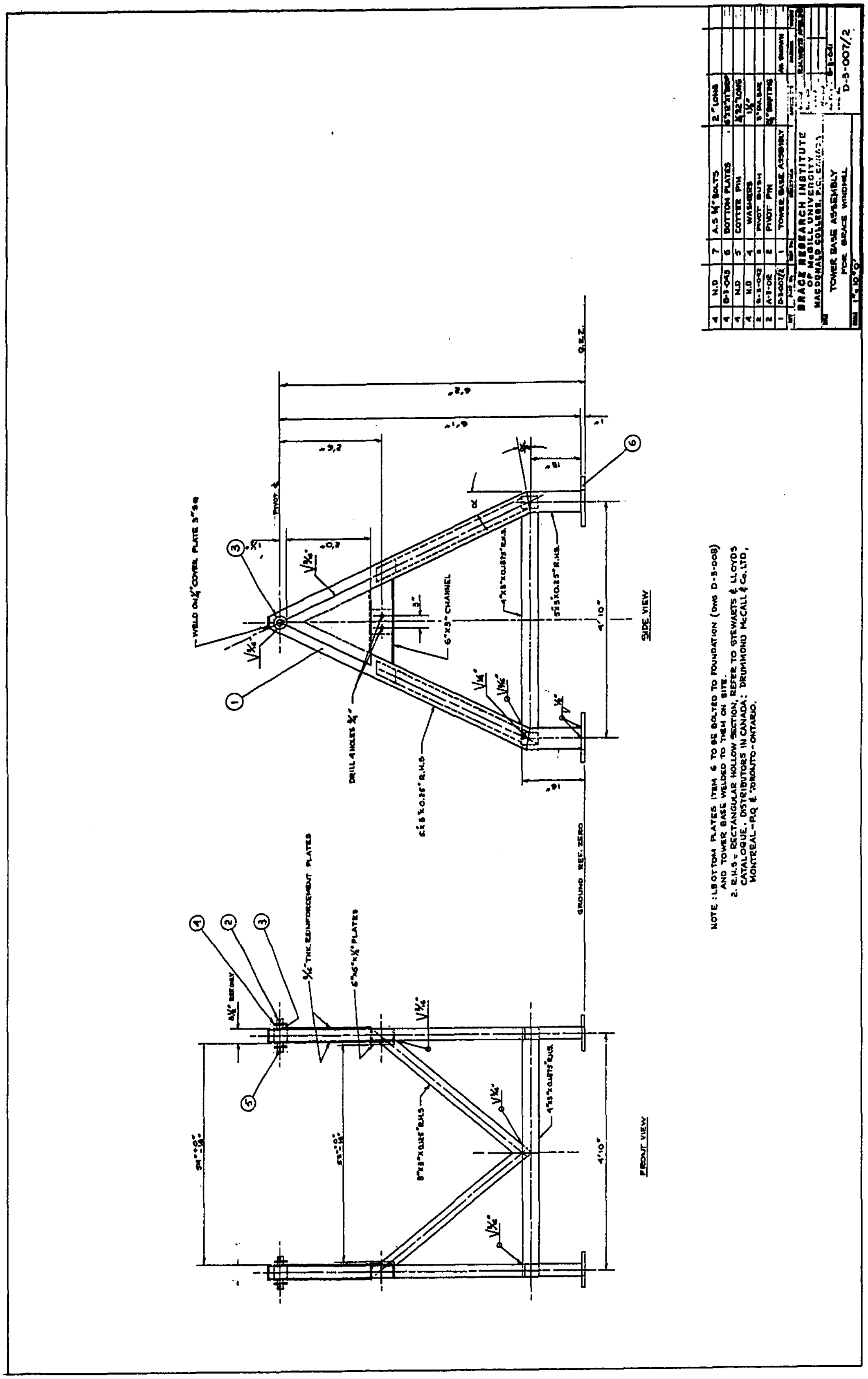


Fig. 20. Tower base assembly.

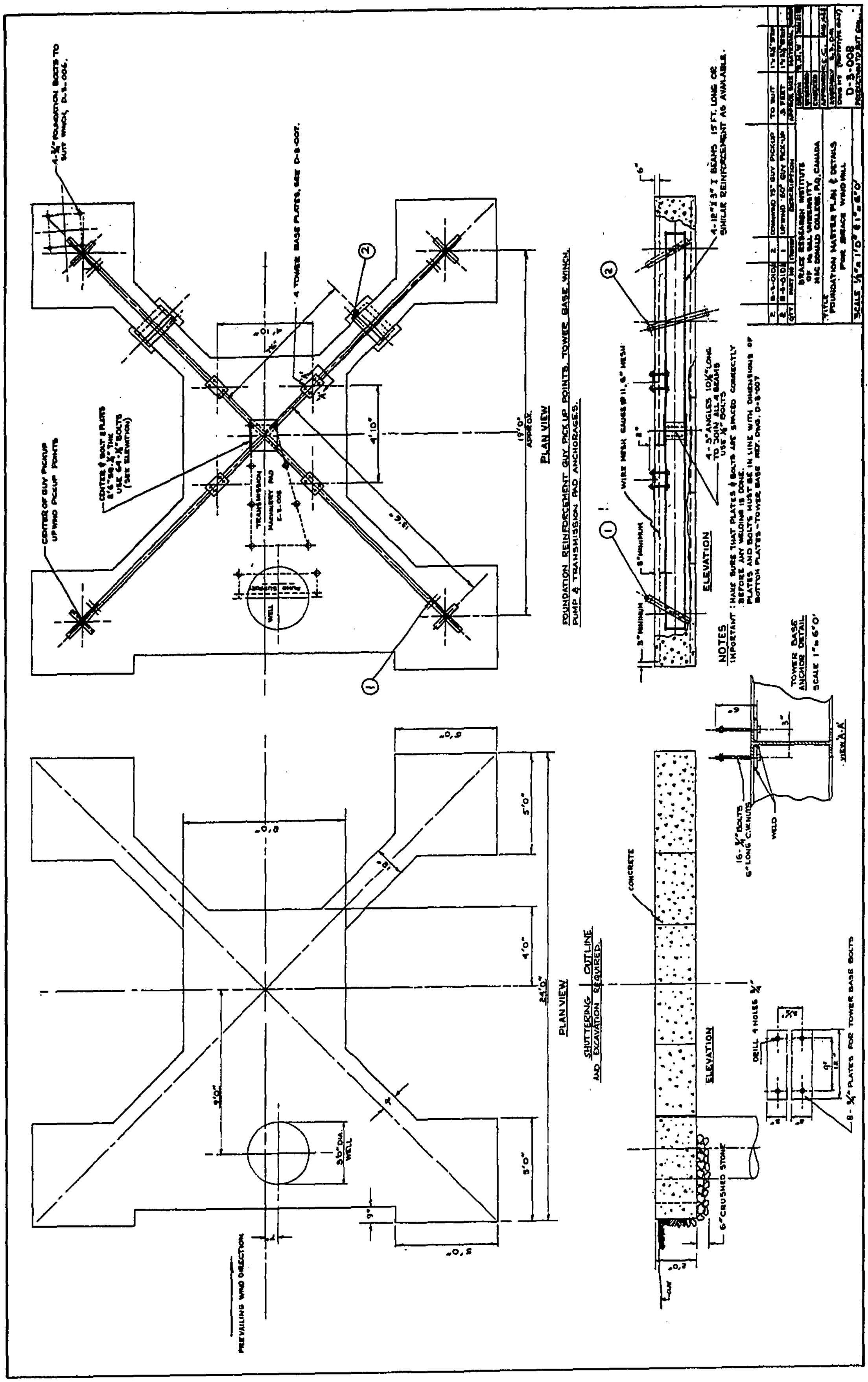


Fig. 21. Foundation arrangement and details.

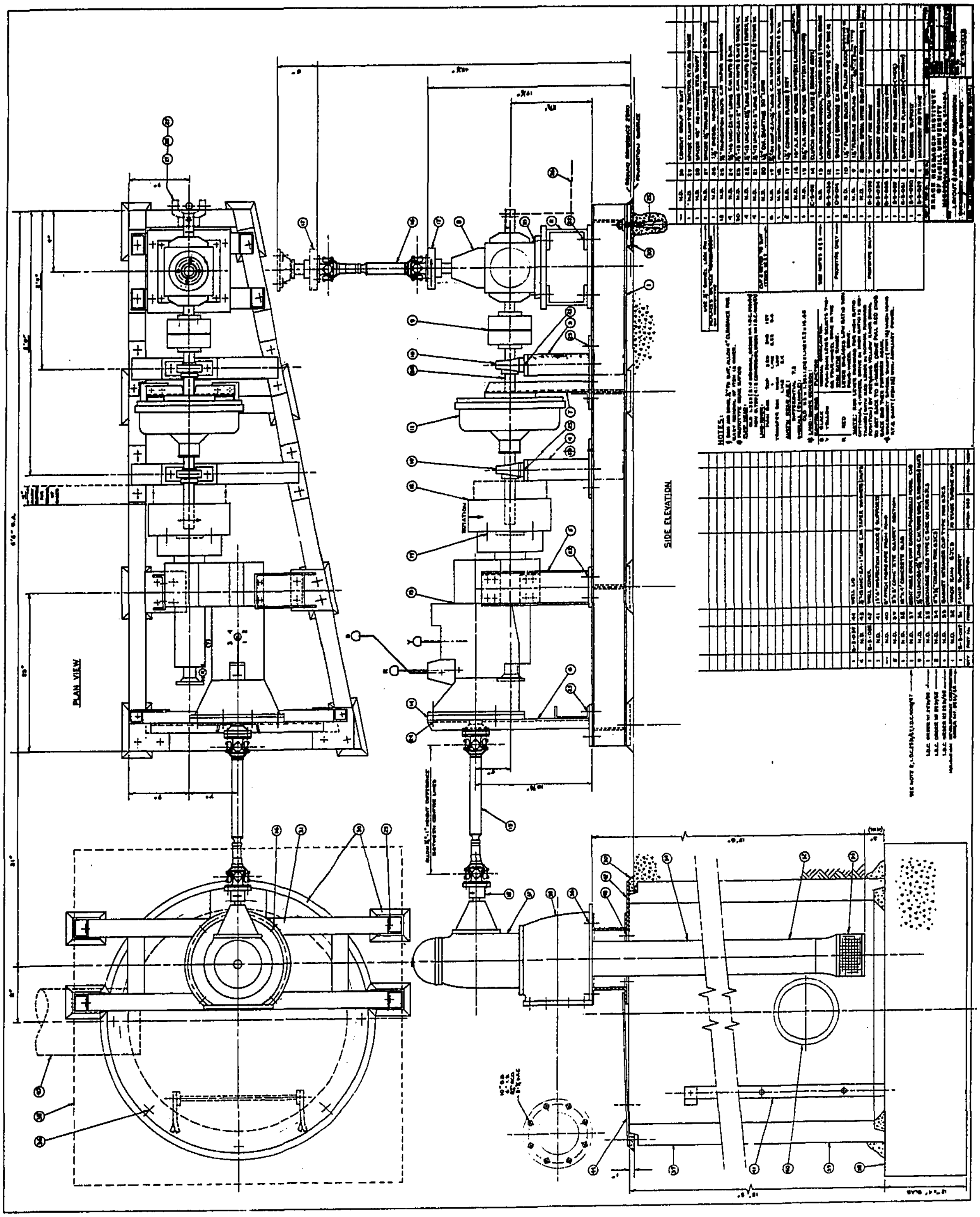


Fig. 22. Transmission machinery pad and pump arrangement.

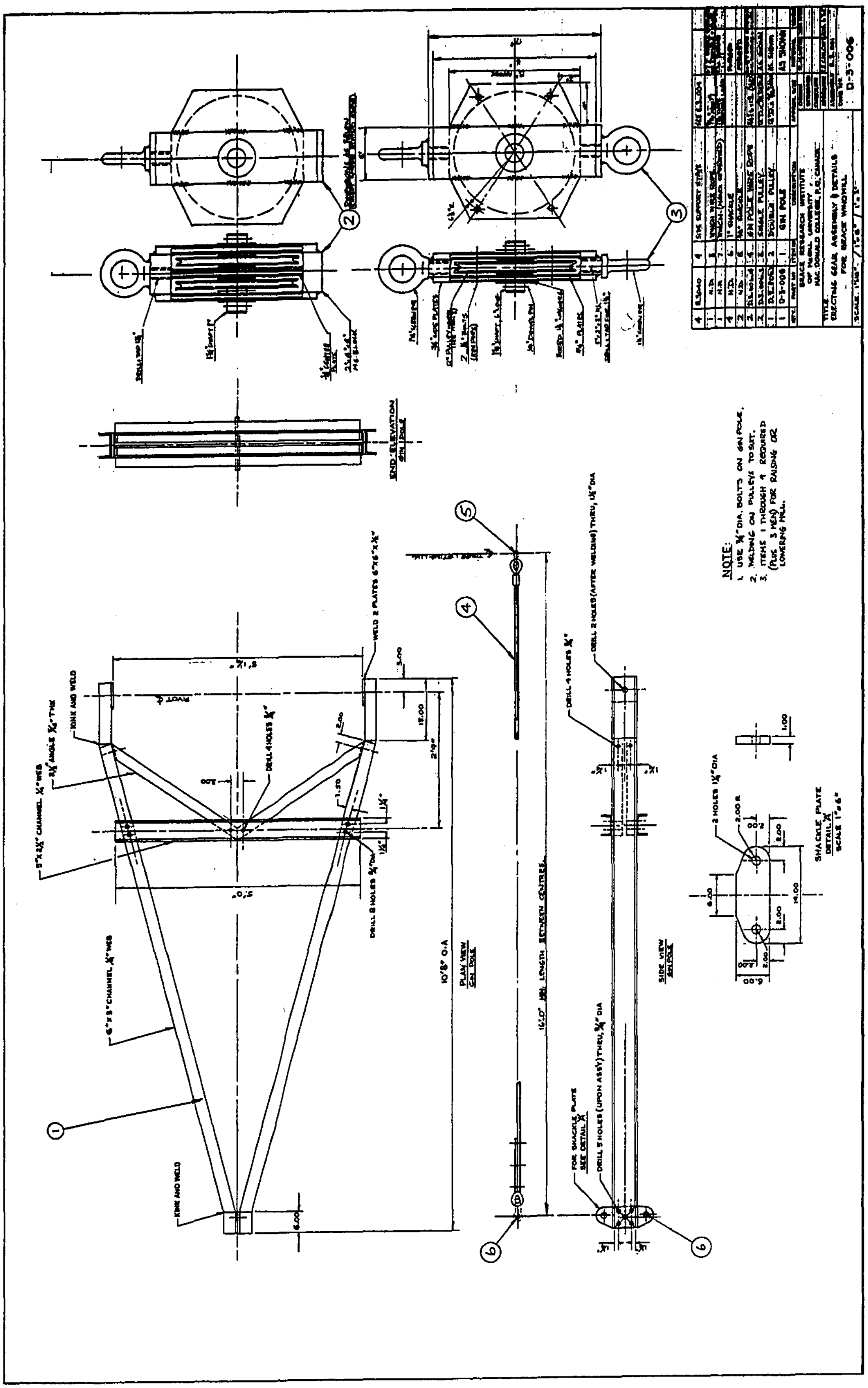


Fig. 23. Erecting gear details.

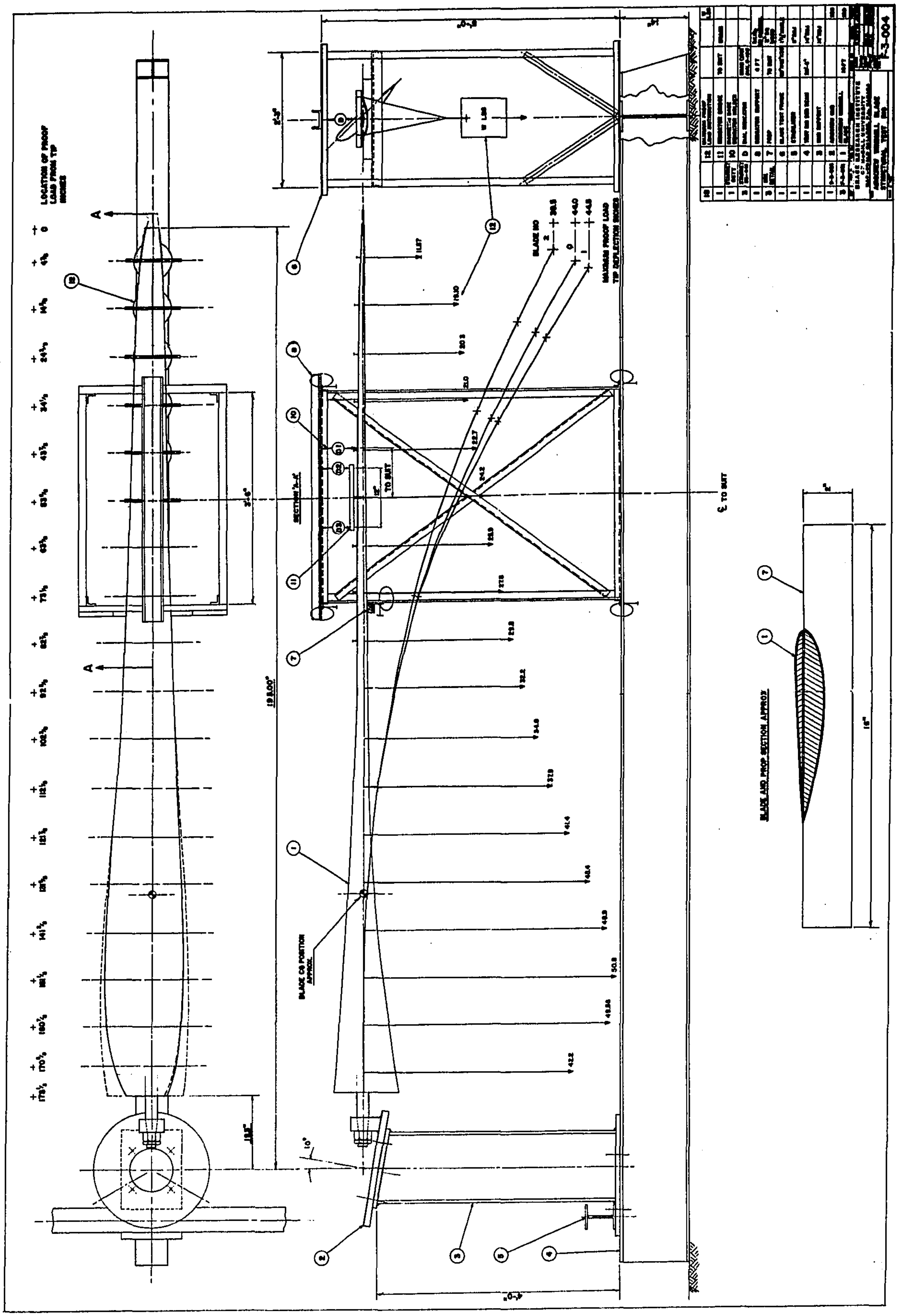


Fig. 24. Airscrew blade structural test rig.

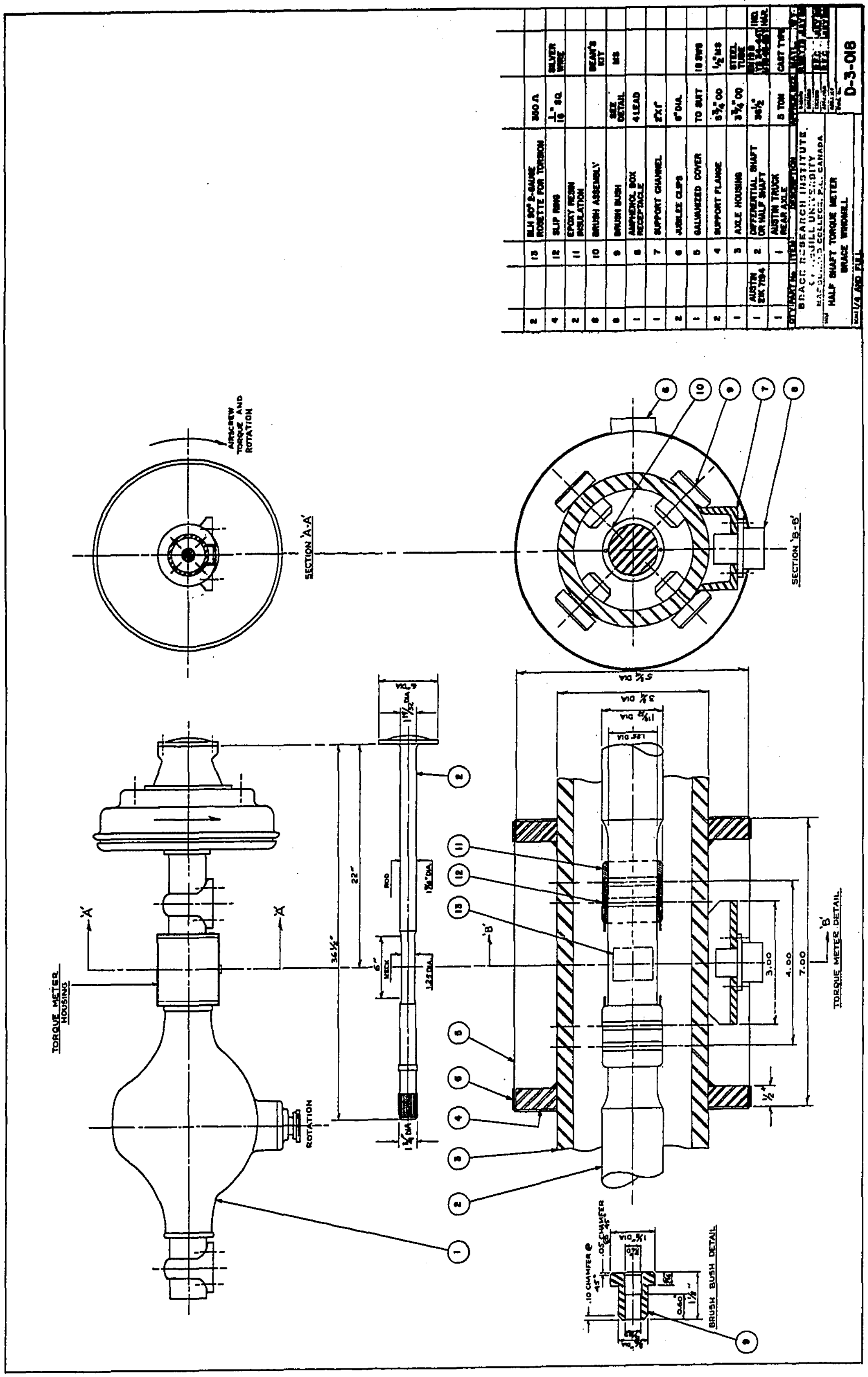


Fig. 25. Airscrew shaft torque meter.

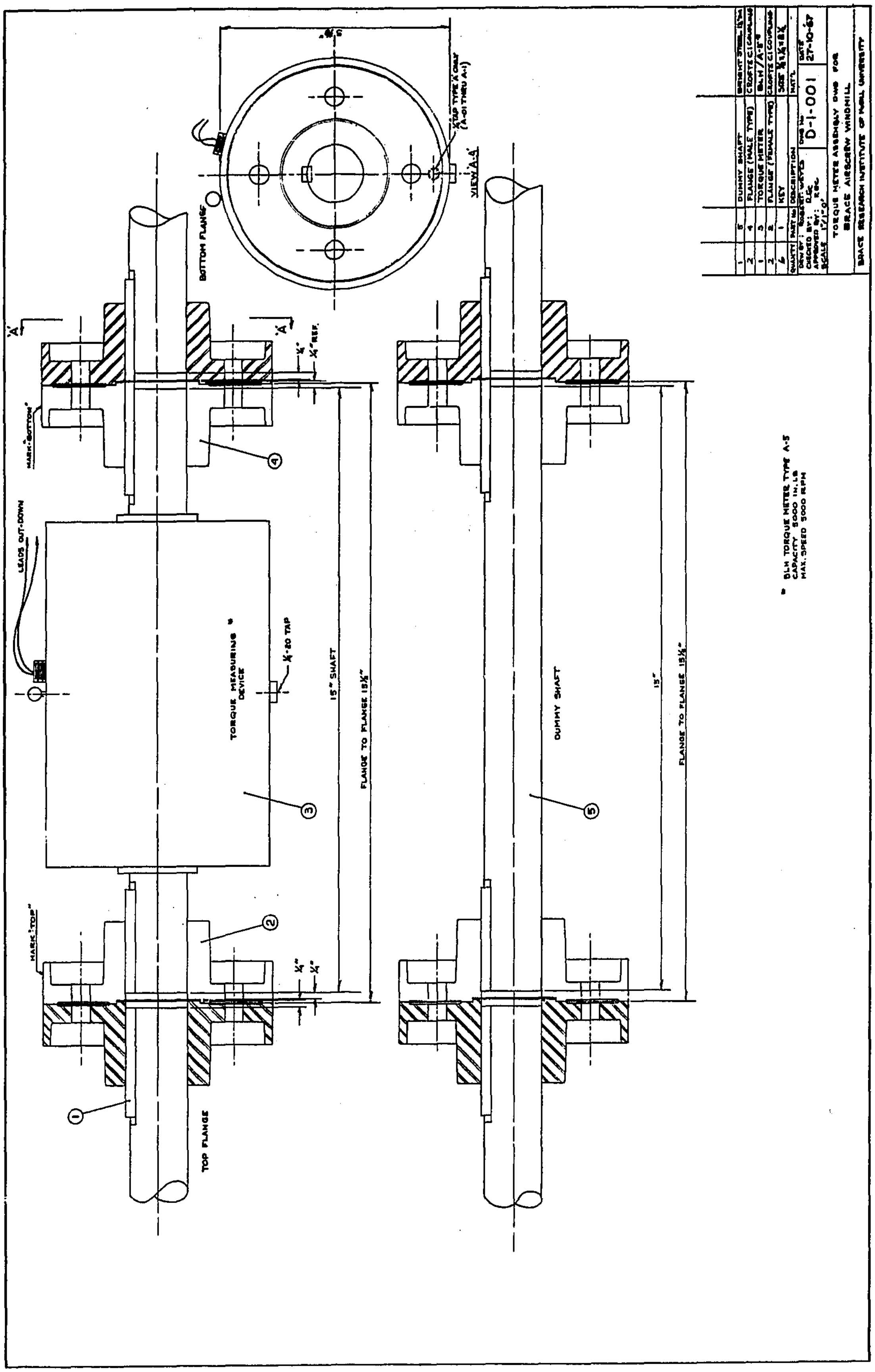
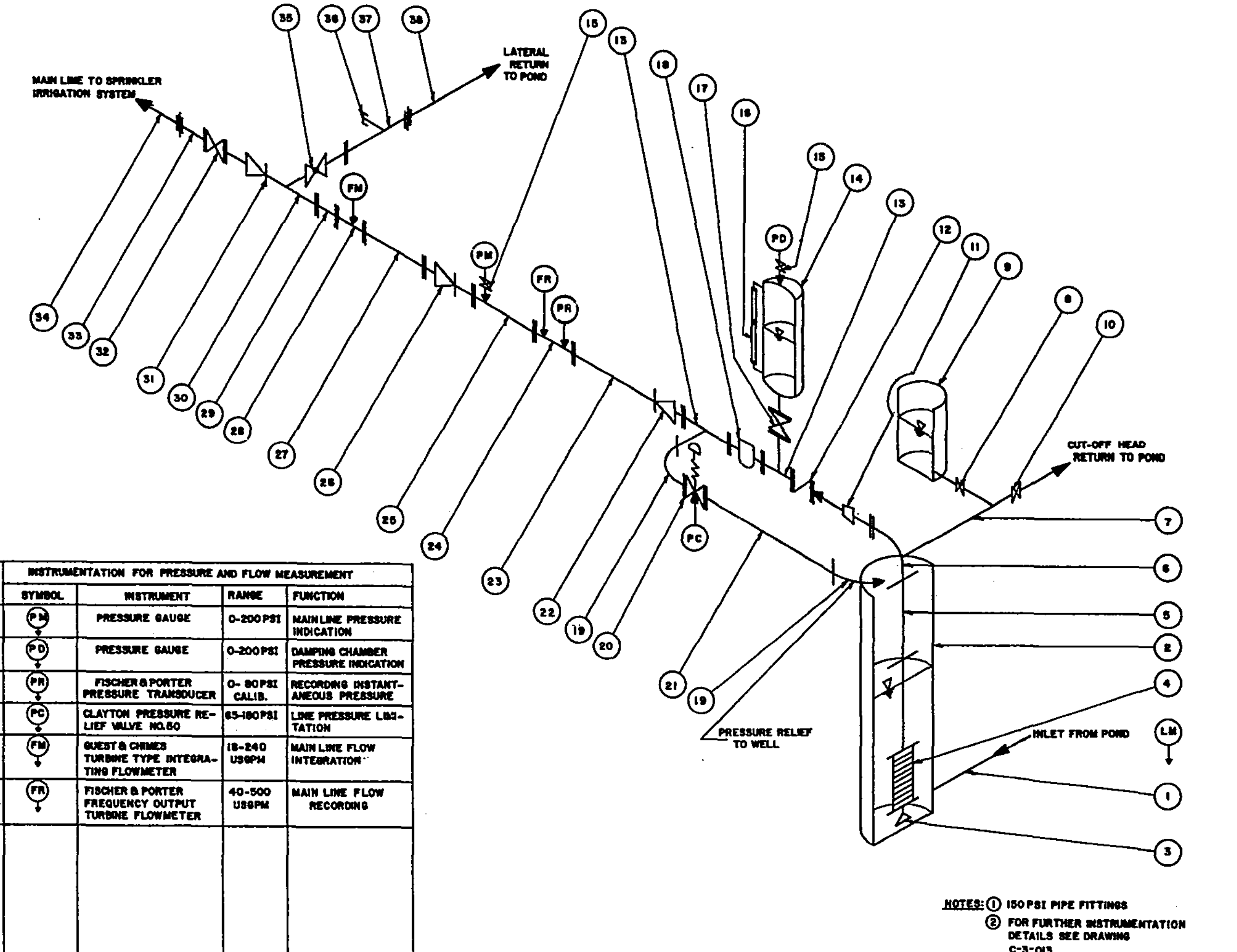


Fig. 26. Vertical shaft torque meter.



ITEM	SYMBOL	INSTRUMENT	RANGE	FUNCTION
	PM	PRESSURE GAUGE	0-200 PSI	MAIN LINE PRESSURE INDICATION
	PD	PRESSURE GAUGE	0-200 PSI	DAMPING CHAMBER PRESSURE INDICATION
	PR	FISCHER & PORTER PRESSURE TRANSDUCER	0- 80 PSI CALIB.	RECORDING INSTANTANEOUS PRESSURE
20	PC	CLAYTON PRESSURE RELIEF VALVE NO.60	65-180 PSI	LINE PRESSURE LIMITATION
28	FM	GUEST & CHIMES TURBINE TYPE INTEGRATING FLOWMETER	18-240 USGPM	MAIN LINE FLOW INTEGRATION
24	FR	FISCHER & PORTER FREQUENCY OUTPUT TURBINE FLOWMETER	40-500 USGPM	MAIN LINE FLOW RECORDING

INSTRUMENTATION FOR PRESSURE AND FLOW MEASUREMENT

NOTES: (1) 150 PSI PIPE FITTINGS
(2) FOR FURTHER INSTRUMENTATION DETAILS SEE DRAWING C-3-013

QTY	PART	ITEM	DESCRIPTION	SIZE	MATERIAL
1	LM	39	POND LEVEL INDICATOR		
1		38	AL IRRIGATION PIPE RETURN TO POND	4"X20'-0"	
1		37	PUMPFLANGE TYPE AL IRRIGATION PIPE TEE	4"X4"X4"	
1		36	AL ENDCAP	4"	
1		35	GLOBE REGULATING VALVE	4"	
1		34	IRRIGATION PIPE TO SPRINKLER SYSTEM	5"X30'-0"	
1		33	PUMP FLANGE TYPE IRRIGATION PIPE LENGTH	5"	
1		32	GATE VALVE	6"	
1		31	REDUCER	5"X4"	
1		30	TEE	4"X4"X4"	
1		29	OUTLET PIPE FROM METER	4"X16"	
1	FM	28	GUEST & CHIMES C.T. TURBINE FLOW METER	4"	FLANGET TYPE 5B
1		27	INLET PIPE TO METER	4"X6'-0"	
1		26	REDUCER	4"X3"	
1		25	OUTLET PIPE FROM METER	3"X16"	
1	FR	24	FISCHER & PORTER TURBINE FLOWMETER	3"X500 USGPM	
1		23	INLET PIPE TO METER	3"X4'-0"	
1		22	REDUCER	4"X3"	
1		21	RETURN PIPE TO WELL	3"	
1	PC	20	PRESSURE RELIEF VALVE	3"	
2		19	ELBOW	3"	
1		18	A.O.SMITH STRAINER	4"	40 MESH
1		17	GATE DAMPING VALVE	3"	
1		16	SIGHT TUBE	3/4"	
2		15	GAUGE COCK	1/2"	
1		14	PRESSURE VESSEL	18"X5'-0"	
2		13	TEE	4"X4"X3"	
1		12	GUEST & CHIMES CHECK VALVE	4"	
1		11	BUSHING	5"X4"	
1		10	CUT-OFF HEAD RELIEF VALVE	1/2"	
1		9	PRE-LUBE RESERVOIR	2"X5"	
1		8	PRE-LUBE VALVE	1/2"	
1		7	PRE-LUBE PIPE	1/2"	
1		6	PUMP HEAD	5"	
1		5	PUMP COLUMN	5"X3/4"	
1		4	WADE RAIN 6JC3 PUMP	10 STAGE	
1		3	PUMP INLET	5"	
1		2	CONCRETE CULVERT SECTION WELL	3'X15'X3'-0" DIA	
1		1	PITCH FIBRE PIPE FROM POND	6"	

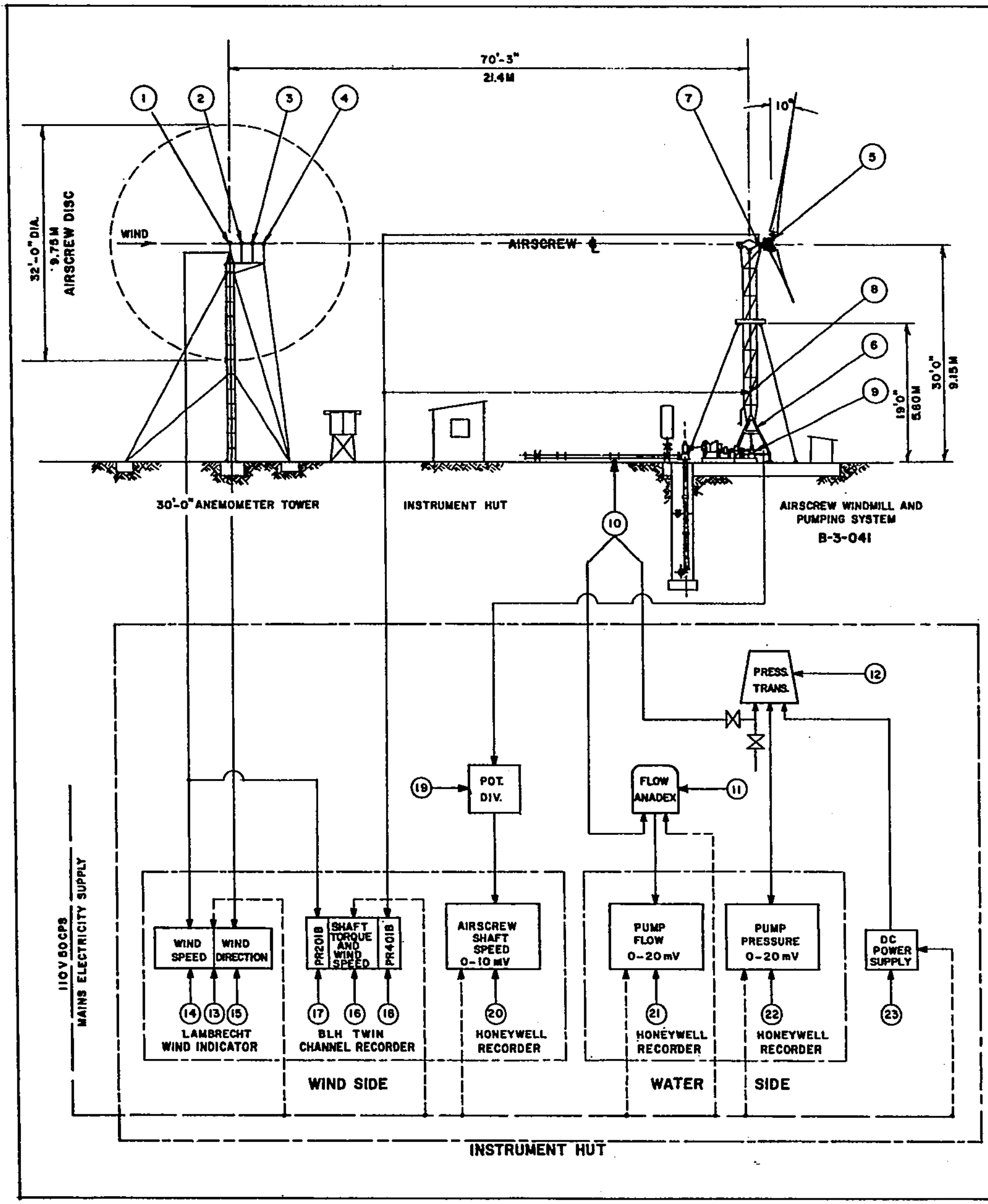
BRACE RESEARCH INSTITUTE
OF McGILL UNIVERSITY
MACDONALD COLLEGE, P.Q., CANADA

HYDRAULIC CIRCUIT

PROTOTYPE ARCHEY WINDMILL

D-3-016

Fig. 27. Prototype windmill pumping system hydraulic circuit.



NOTE FISCHER & PORTER EQUIPMENT ORDER No 66914

QTY	PART No.	ITEM	DESCRIPTION	SIZE	FUNCTION
1	55GL1155	23	DC POWER SUPPLY F&P		4-20 MA DC
1	BRACE No2	22	HONEYWELL STRIP CHART RECORDER		PUMP PRESSURE 0-20MV
1	BRACE No1	21	HONEYWELL STRIP CHART RECORDER		PUMP FLOW 0-20MV
1	BRACE No5	20	HONEYWELL STRIP CHART RECORDER		AIRSCREW RPM 0-10 MV
1	POTENTIAL DIVIDER	19	10 Ω PRECISION RESISTOR 0-1000 Ω BECKMAN HELIPOT		
1	PR401B	18	CARRIER OPERATED STRAIN GAUGE PREAMPLIFIER MODULE		SHAFT TORQUE RECORD
1	PR201B	17	MEDIUM GAIN DC PRE-AMPLIFIER MODULE		WINDSPEED RECORD
1	BSA-270BA	16	BLH TWO CHANNEL HIGH-SPEED STRIPCHART RECORDER		0.5-200 MM/SEC
1		15	WIND DIRECTION INDICATOR		
1		14	WIND SPEED INDICATOR		
1	1475	13	LAMBRECHT INDICATOR PANEL		
1	50EQ107IA	12	F&P PRESSURE TO CURRENT TRANSDUCER	1/2"	0-80 PSI CALIB.
1	PI-200PIKC	11	ANADEX FREQUENCY TO VOLTAGE TRANSDUCER		0-50 MV
1	10C1506A	10	FISCHER & PORTER TURBINE FLOWMETER	3"	40-500 USGPM
1	5A-757A-2	9	SERVOTEK TACHO GENERATOR		7 V/1000 RPM
1	SR-4-5000	8	BLH TYPE A-5 TORQUE PICKUP		5000 IN LB CAP. 5000 RPM MAX.
1	D-3-017	7	AIRSCREW SHAFT TORQUE METER		350 Ω
1	D-3-004	6	TOWER ALIGNMENT GAUGE	1°	
3	F-3-001	5	BLADE ANGLE GAUGE	1/2°	
1	W1208/1	4	CASELLA CUP COUNTER ANEMOMETER	KM	LONG TERM WIND RUN
1	1462	3	LAMBRECHT MECHANICAL WIND RECORDER	KM	HOURLY MAIN WIND-SPEED DIRECTION
2	1467G	2	LAMBRECHT WIND TRANSMITTER		WIND SPEED CHECK
1	1465G	1	LAMBRECHT COMBINED WIND TRANSMITTER		WIND SPEED AND DIRECTION INDICATION

BRACE RESEARCH INSTITUTE
OF McGILL UNIVERSITY
MACDONALD COLLEGE, P.Q. CANADA

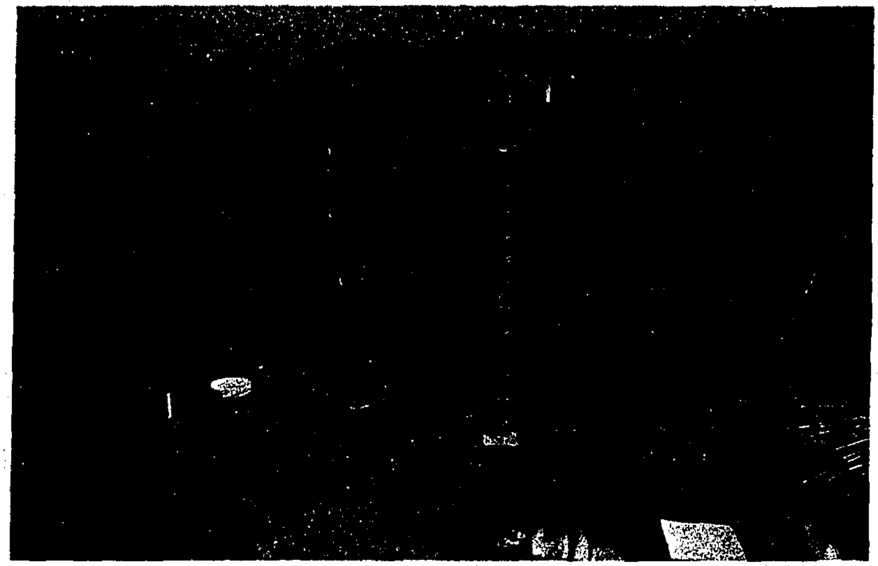
DRAWN R.WEYTS JUNE 20 69
STRESSED
CHECKED R.E.C. JUNE 23 69
APPROVED R.E.C. JUNE 23 69
ASSEMBLY

TITLE
TEST INSTRUMENTATION DIAGRAM
PROTOTYPE AIRSCREW WINDMILL

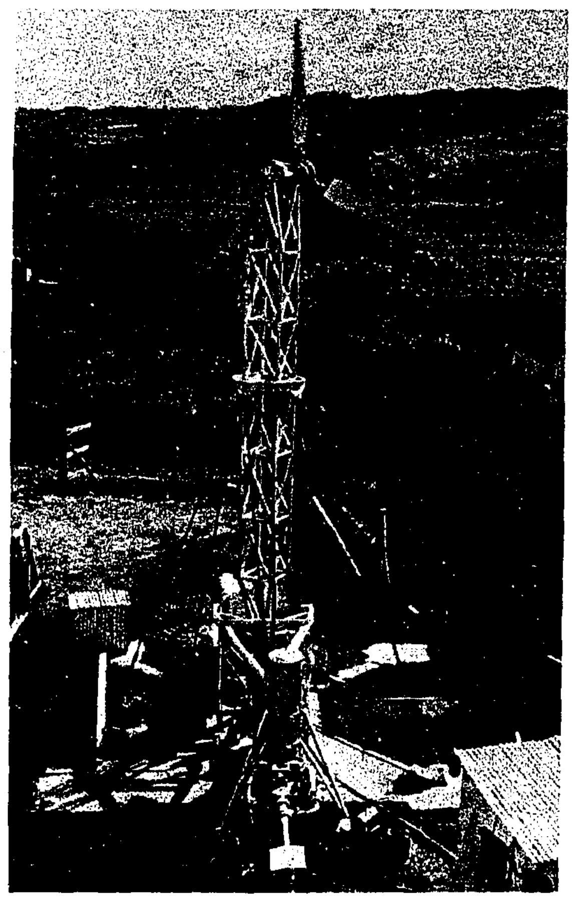
SCALE 1:50

DWG. No. C-3-013

Fig. 28. Prototype performance test instrumentation diagram.

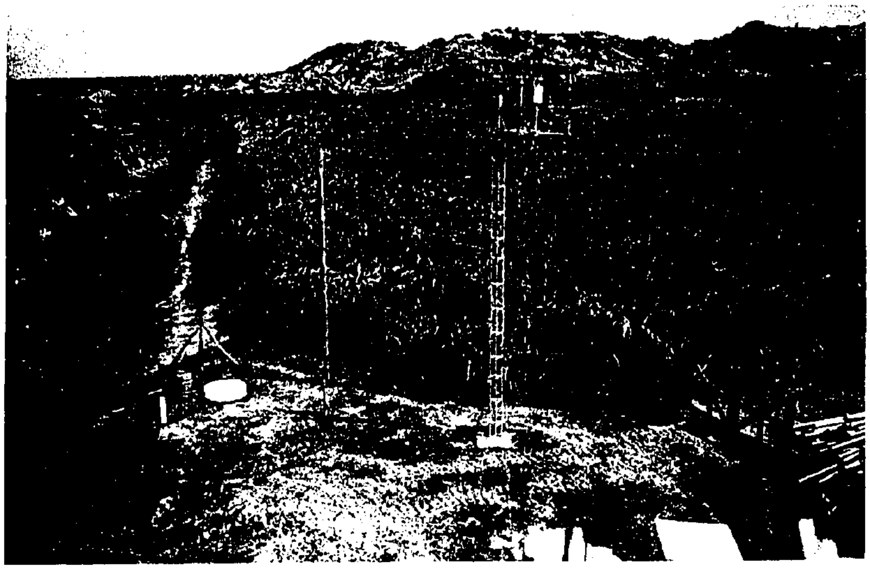


(a) View upwind from top of windmill tower.

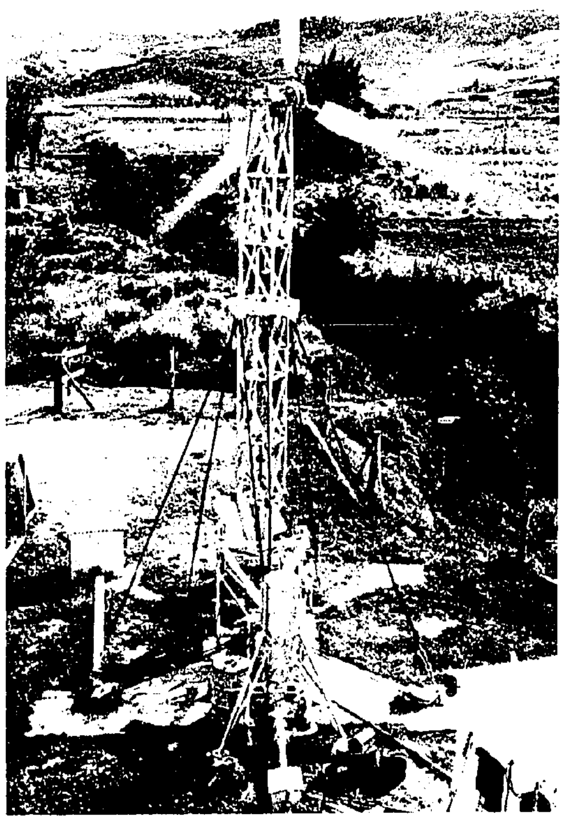


(b) View downwind from anemometer tower.

Fig. 29. Prototype windmill pumping system test site, Greenland.

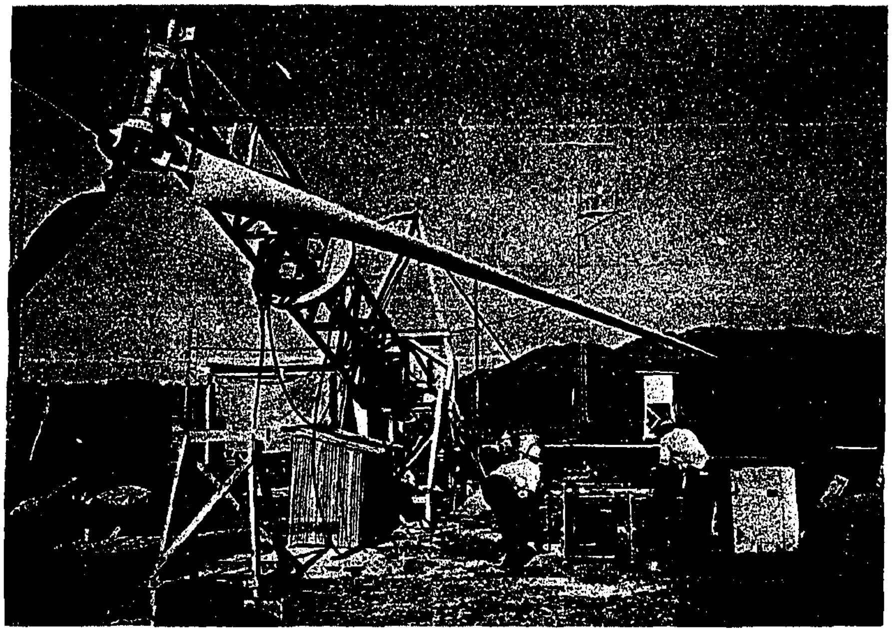


(a) View upwind from top of windmill tower.



(b) View downwind from anemometer tower.

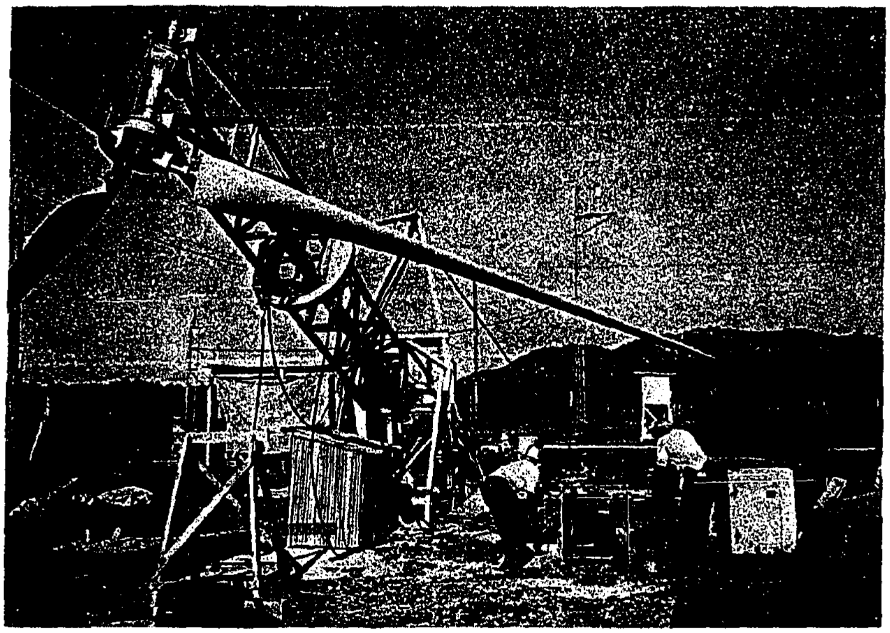
Fig. 29. Prototype windmill pumping system test site, Greenland.



(a) Erection using hand winch pulley system.

(b) Gin pole, tower yoke and pulley arrangement.

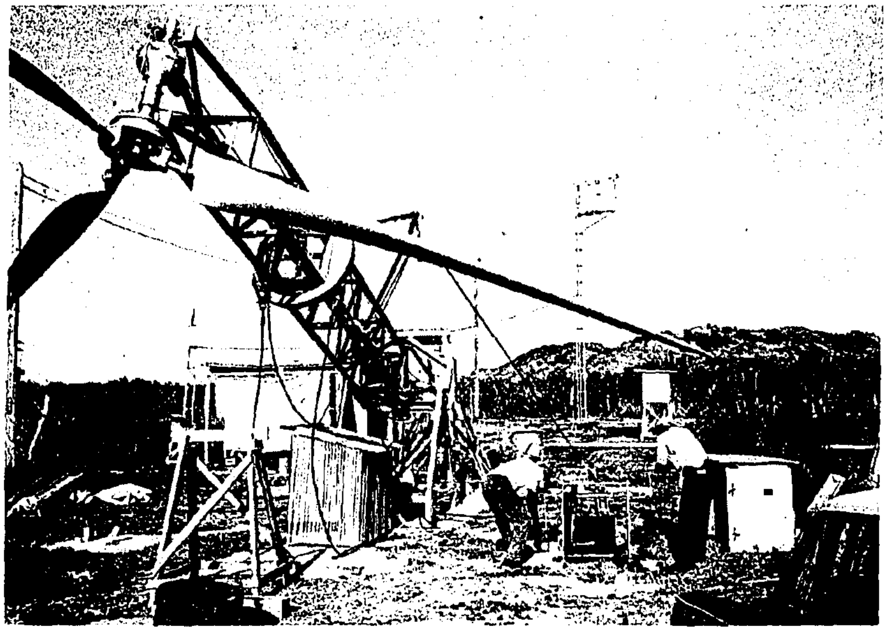
Fig. 30. Windmill erection and lowering.



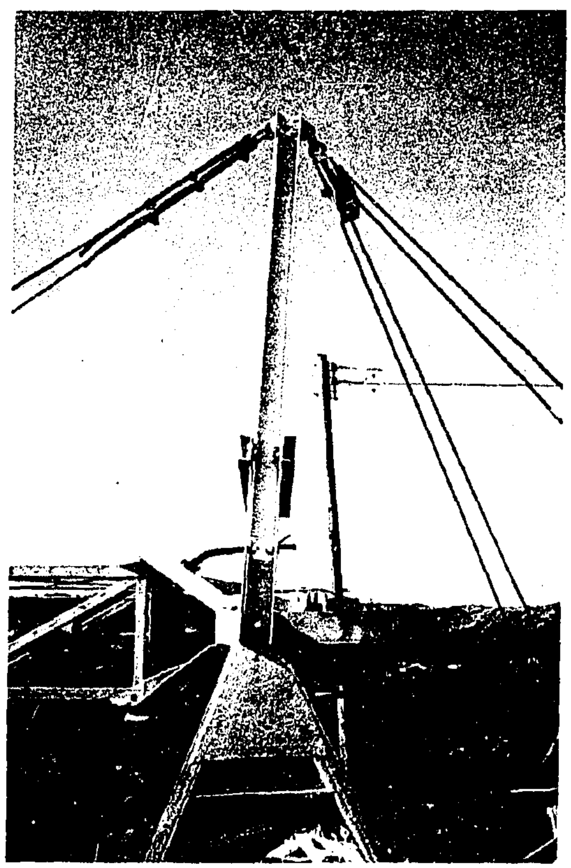
(a) Erection using hand winch pulley system.

(b) Gin pole, tower yoke and pulley arrangement.

Fig. 30. Windmill erection and lowering.

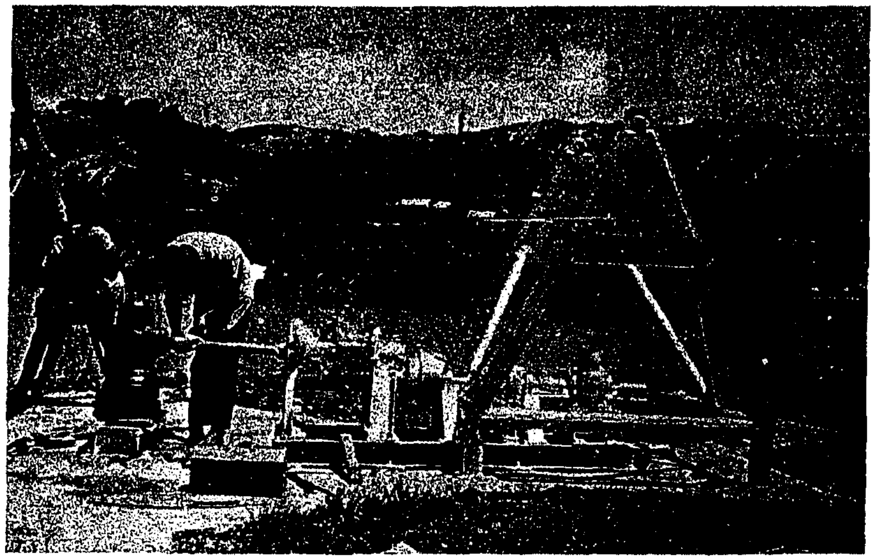


(a) Erection using hand winch pulley system.

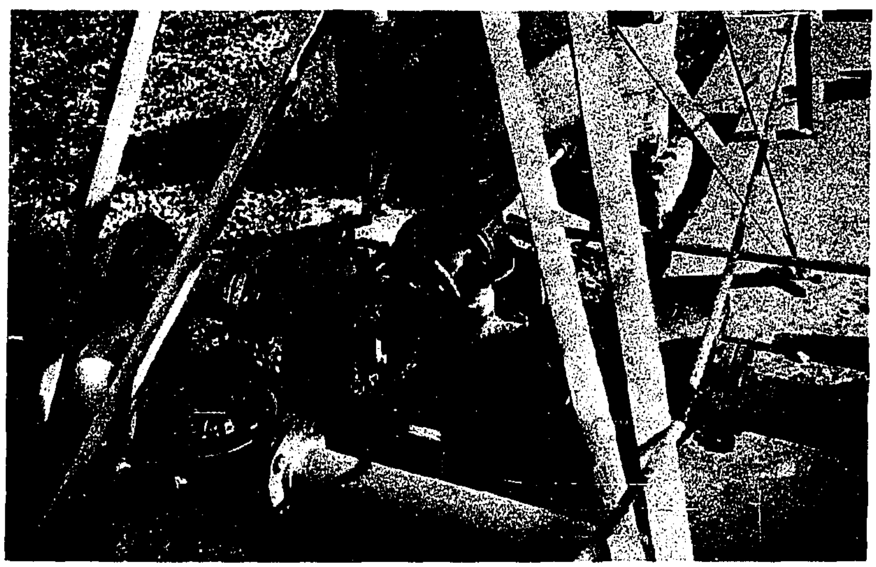


(b) Gin pole, tower yoke and pulley arrangement.

Fig. 30. Windmill erection and lowering.

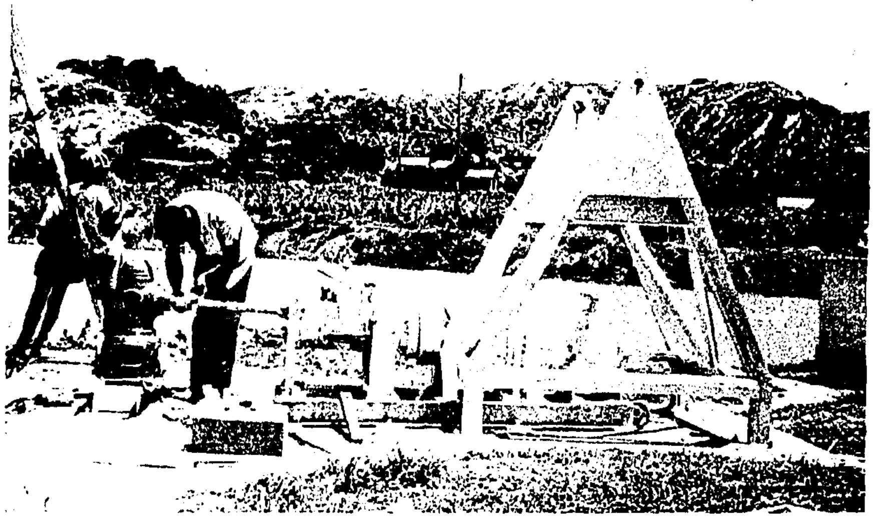


(a) Tower base, transmission machinery pad and pump head.

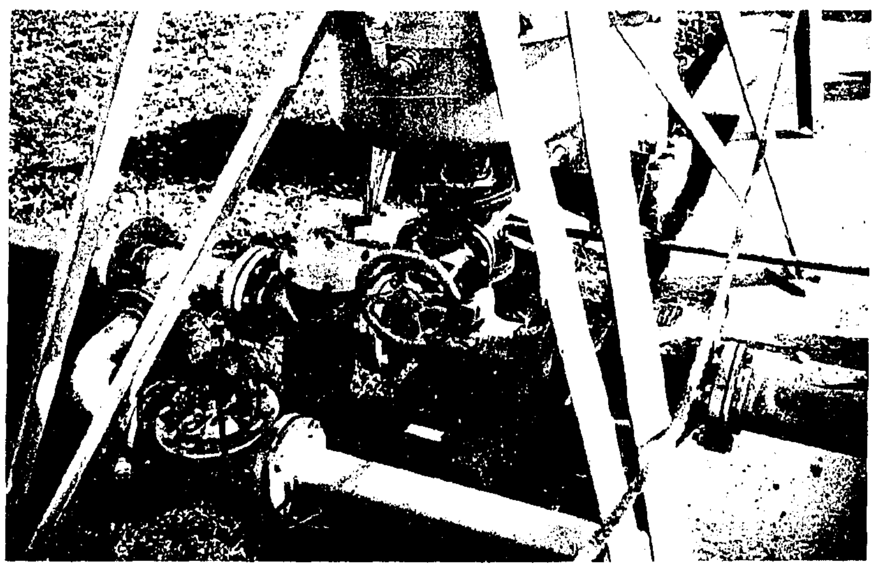


(b) Pump outlet hydraulic circuit arrangement.

Fig. 31. Tower base and test equipment.

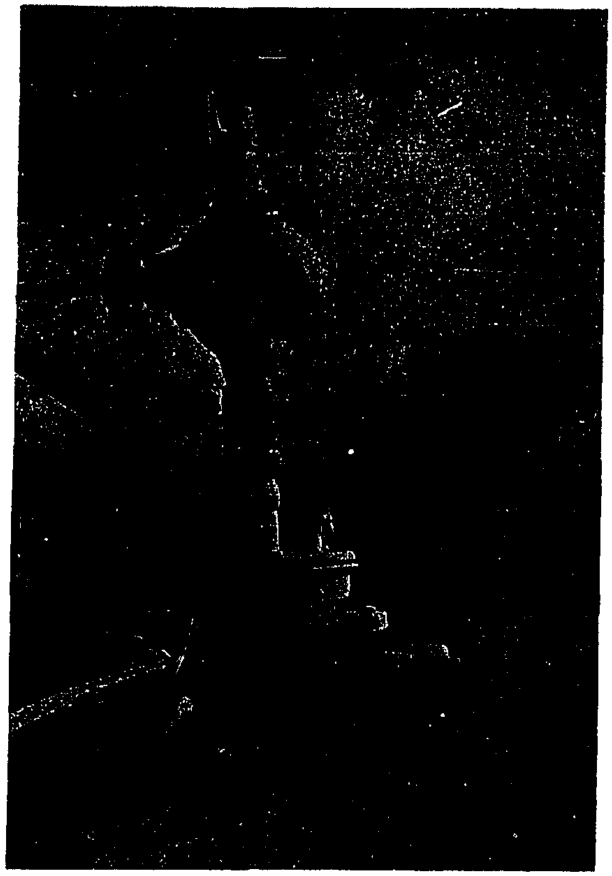


(a) Tower base, transmission machinery pad and pump head.



(b) Pump outlet hydraulic circuit arrangement.

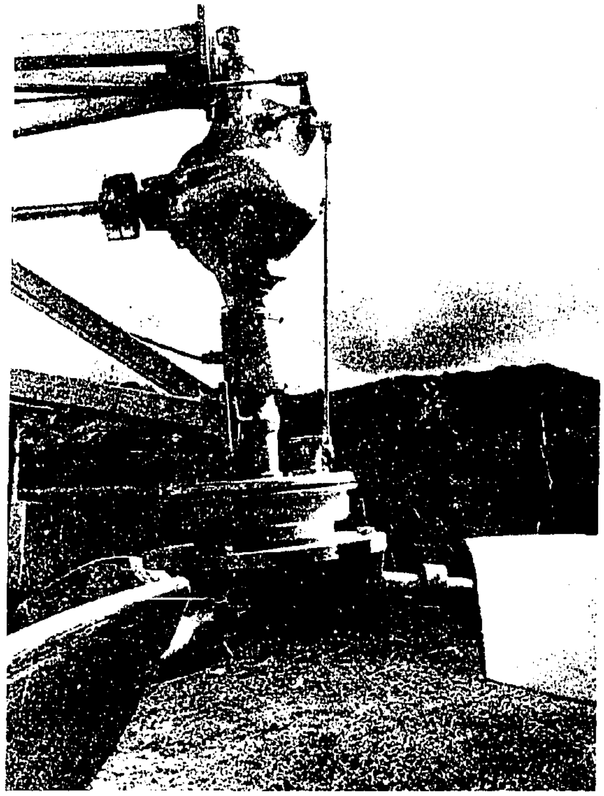
Fig. 31. Tower base and test equipment.



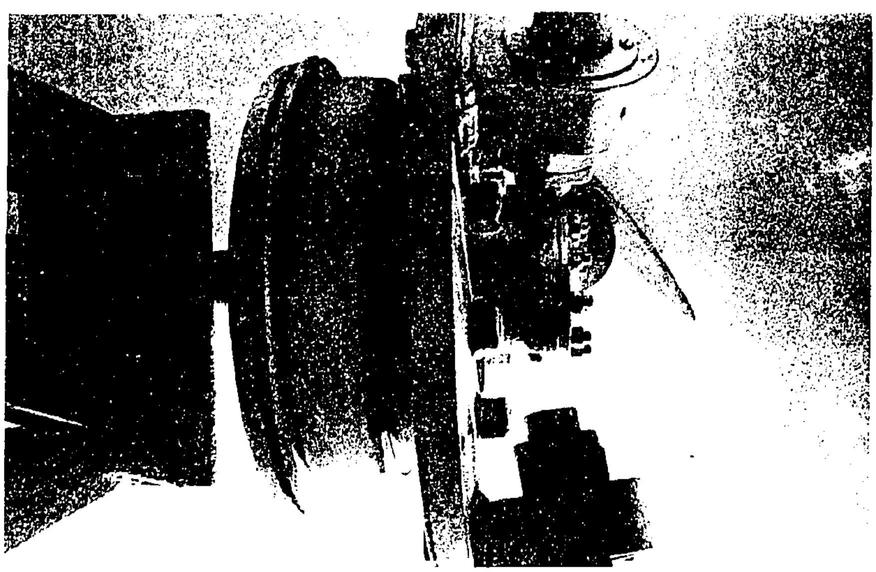
(a) Rear axle, brake and halfshaft torque meter housing.

(b) Airscrew hub and blades.

Fig. 32. Truck rear axle mill head.

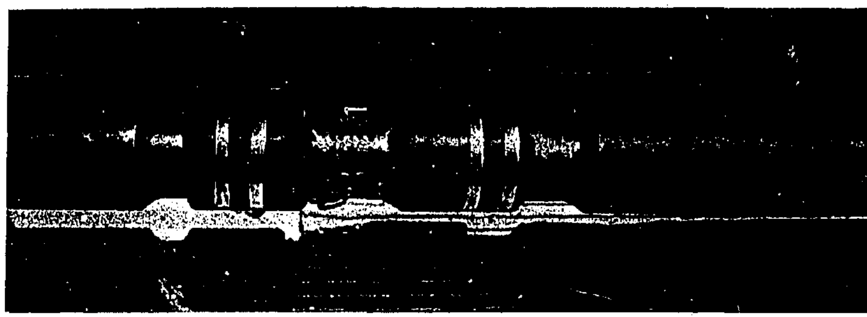


(a) Rear axle, brake and halfshaft torque meter housing.

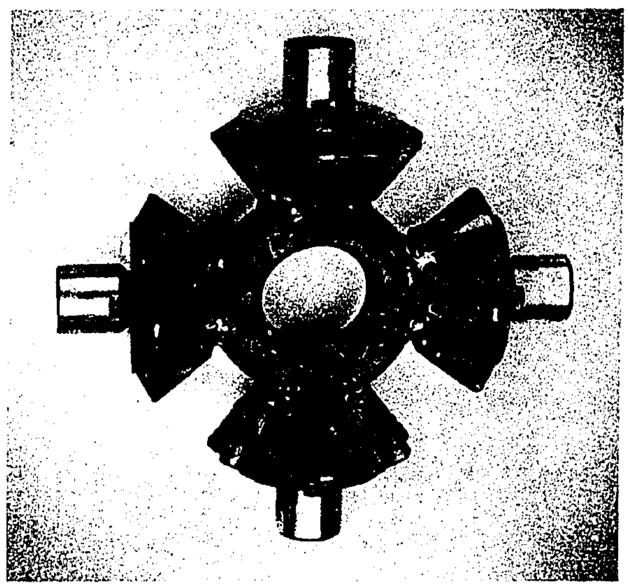


(b) Airscrew hub and blades.

Fig. 32. Truck rear axle mill head.

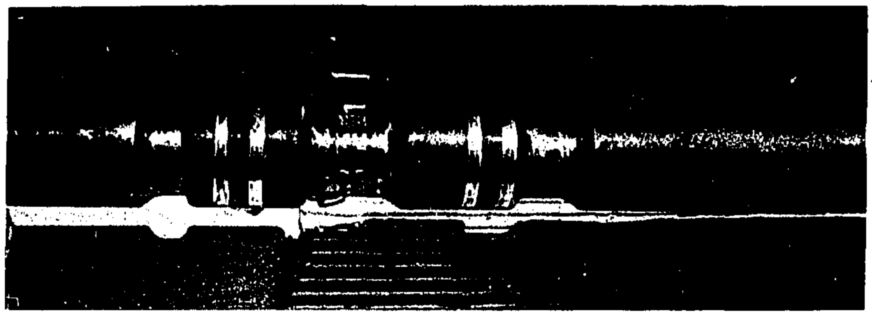


(a) Airscrew shaft torque meter strain gauges and slip rings.

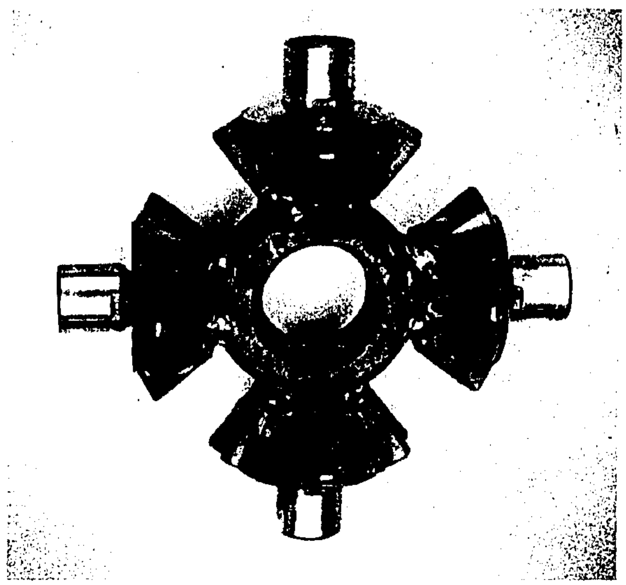


(b) Method of locking differential by welding pinions to spider.

Fig. 33. Rear axle transmission details.

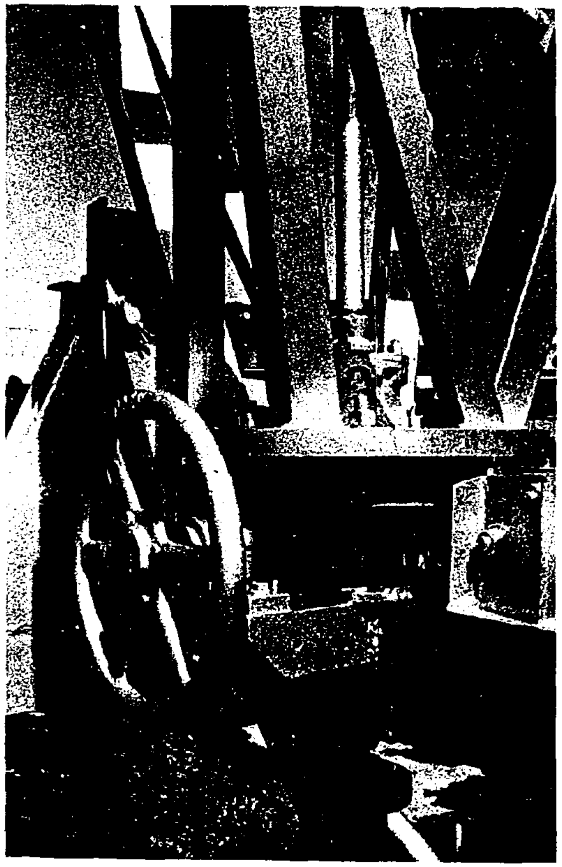


(a) Airscrew shaft torque meter strain gauges and slip rings.

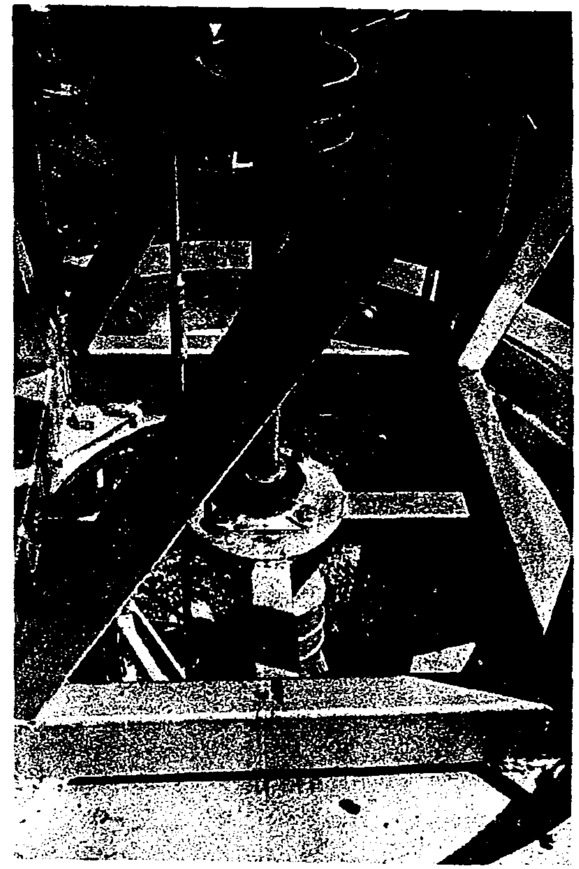


(b) Method of locking differential by welding pinions to spider.

Fig. 33. Rear axle transmission details.

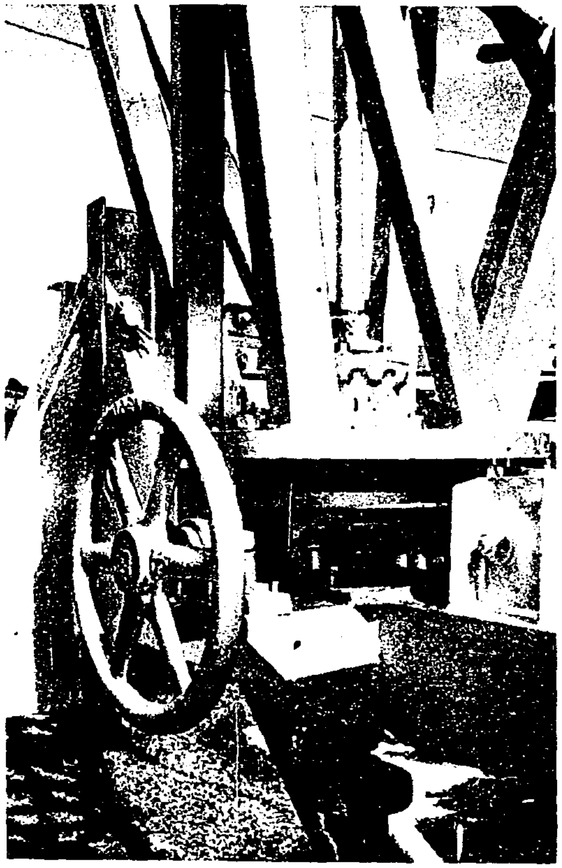


(a) Tower yoke, crown wheel, pinion and hand wheel.

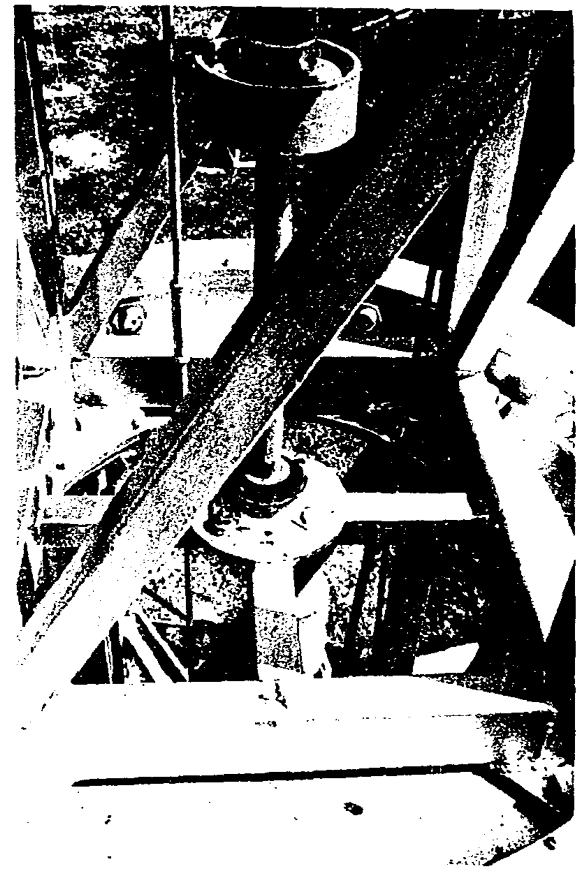


(b) Tower bearing and vertical transmission shafting.

Fig. 34. Tower orientation arrangement.



(a) Tower yoke, crown wheel, pinion and hand wheel.



(b) Tower bearing and vertical transmission shafting.

Fig. 34. Tower orientation arrangement.

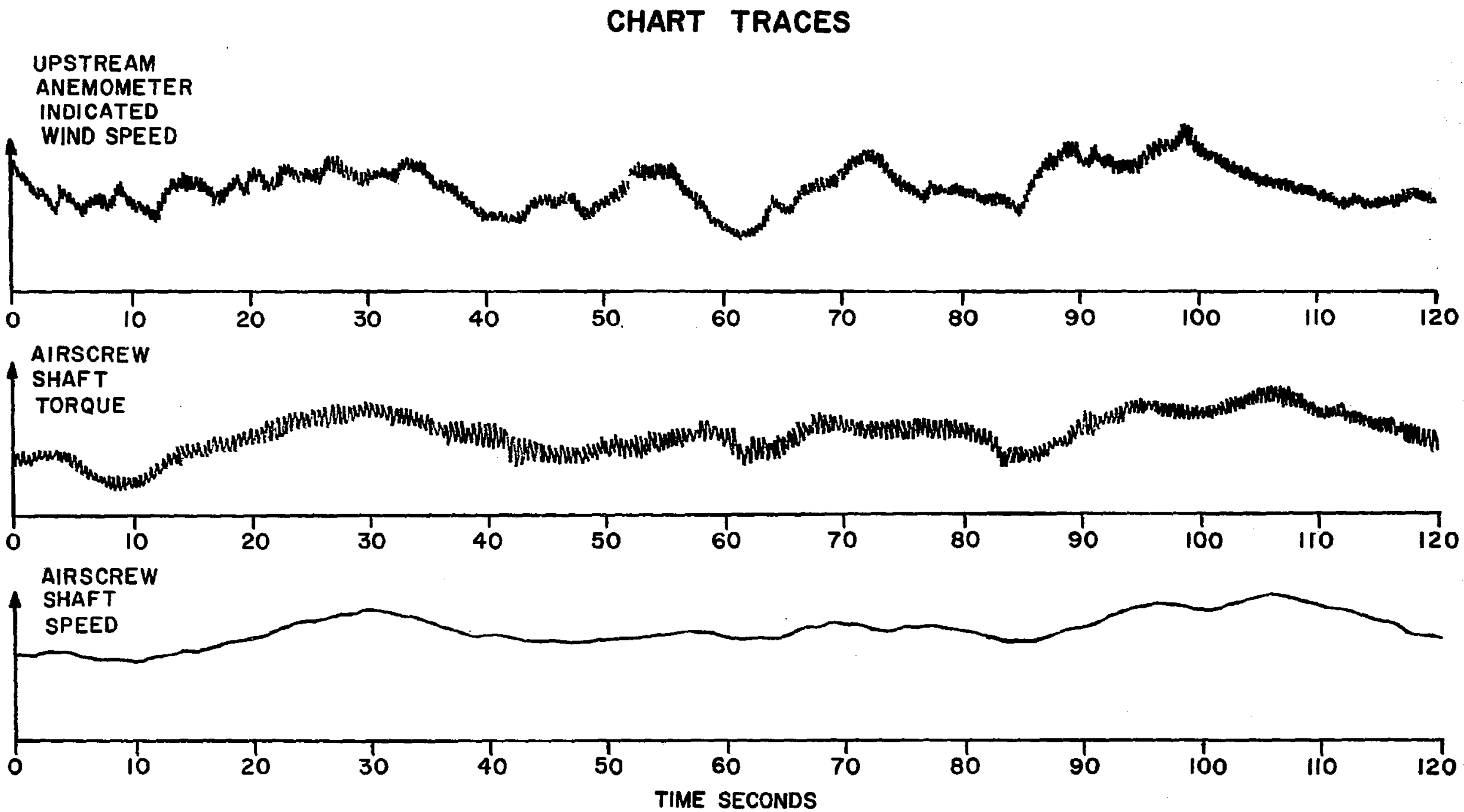


FIG.35. VARIATION OF WIND SPEED SHAFT TORQUE AND SHAFT SPEED WITH TIME.

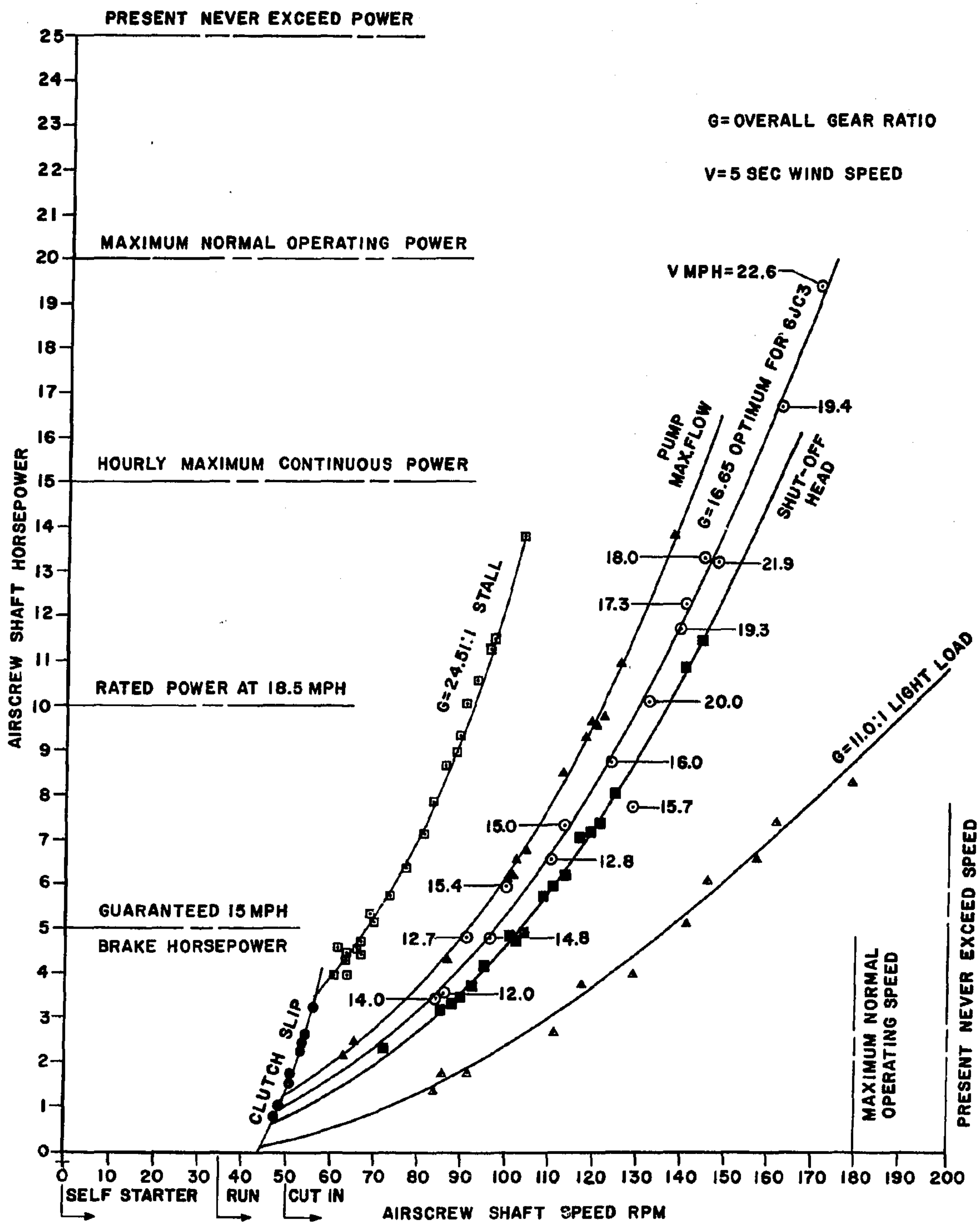


FIG.36. AIRSCREW POWER-SPEED CHARACTERISTICS FOR PROTOTYPE WINDMILL PUMPING SYSTEMS.

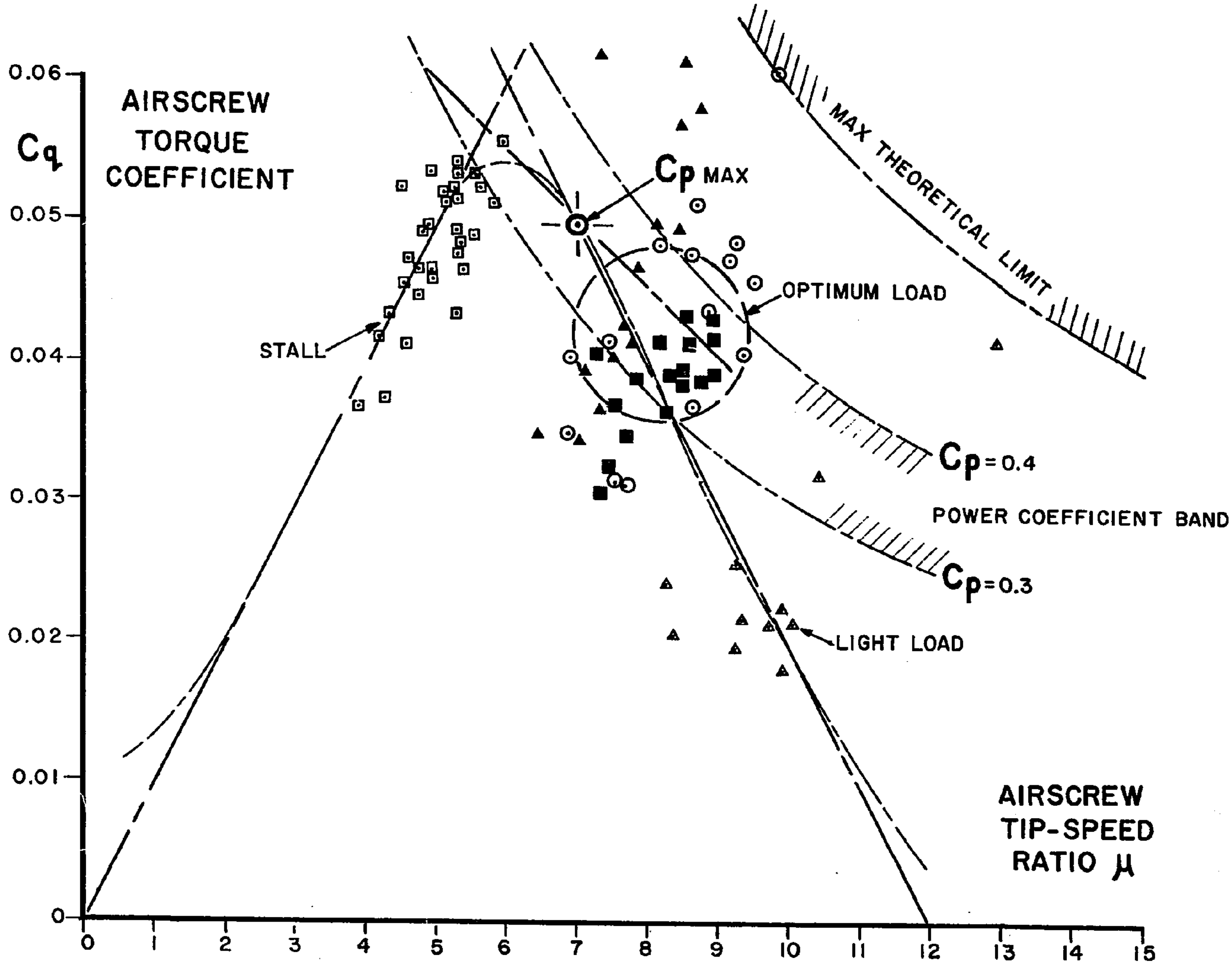


FIG. 37. EXPERIMENTAL TORQUE COEFFICIENT VERSUS TIP-SPEED RATIO.

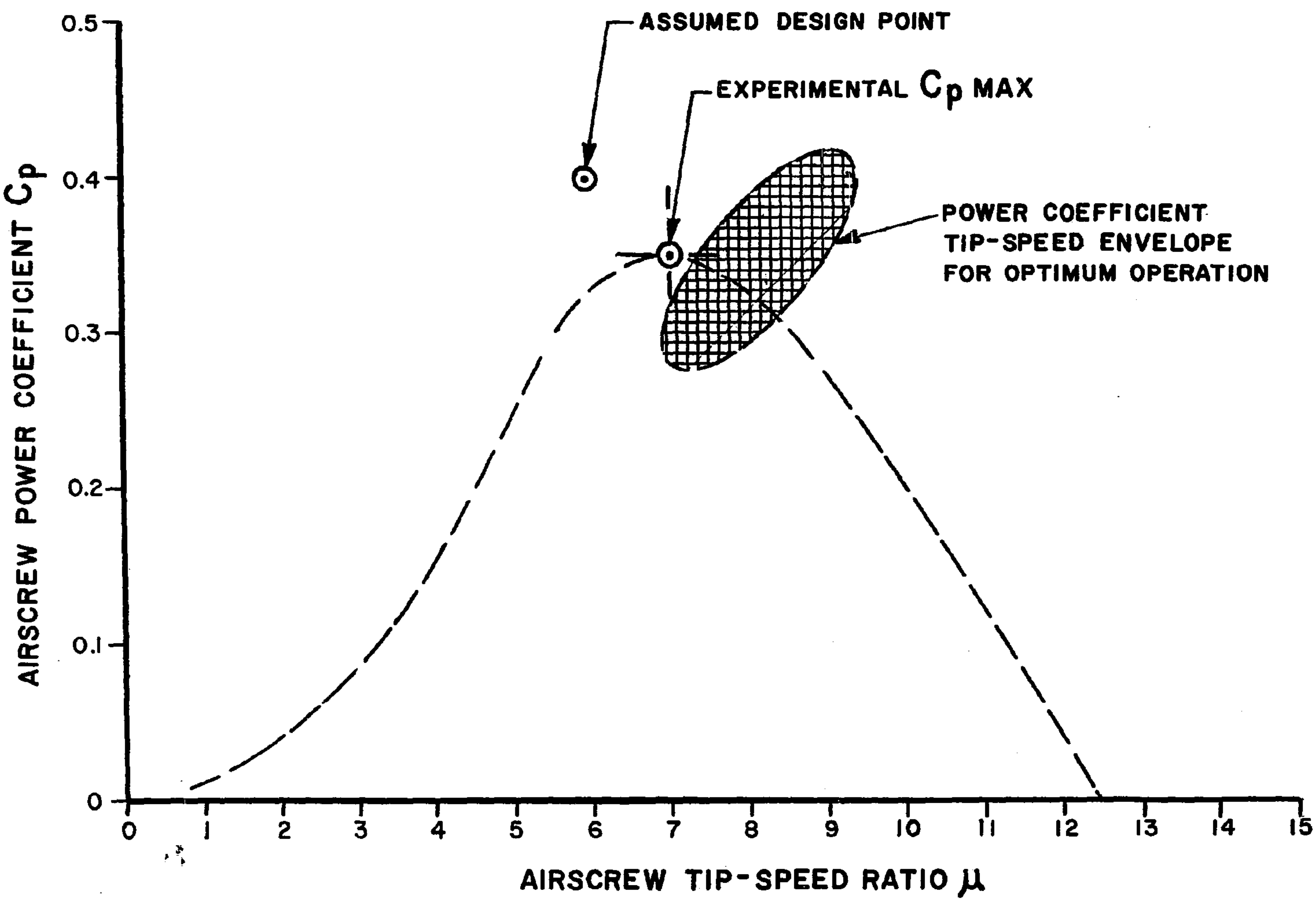


FIG. 38. SMOOTHED POWER COEFICIENT VERSUS TIP-SPEED RATIO.