Final Report

on the

FEASIBILITY OF USING WIND POWER TO PUMP IRRIGATION WATER

Report Prepared

by

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ABSTRACT

A wind power irrigation system is analyzed for two types of pumps, positive displacement and airlift. The general system parameters are for a wind rotor which could pump 200 gpm from 200 ft depth. The rotor design (20 kw at 22 mph) and operation are analyzed by using a computer program which gives the optimized power coefficient. The calculated annual efficiencies for the three modes of operation for the designed rotor are: 52% for constant power coefficient, 45% for constant rpm, and 35% for constant torque. The positive displacement and airlift pumps (constant torque mode of operation) and the problems of matching torque-rpm characteristics of loads to those of the wind rotor are analyzed. Then these two systems are modeled for performance by calculating the time correlation of available wind energy and daily water demand. Even with the variability of wind power it is technically feasible for pumping irrigation water. The cost-benefit of the two systems are compared to the conventional irrigation system. A larger initial capital investment is involved, but if the wind systems are mass produced the cost of the energy over the lifetime of the system would be comparable to electricity at $.03/kwh.

ACKNOWLEDGEMENTS

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INTRODUCTION

Practically everyone understands the importance of agriculture in the US and the Texas economy and the prominent role played by agricultural products in international trade. Although agriculture requires only about ten percent of the energy consumed in the nation, agribusiness is nevertheless heavily dependent on the availability of energy, which presently means fossil fuels. In Texas, thirty-nine percent of all the energy consumed in agriculture is used for irrigation and of that thirty-nine percent, natural gas supplies the largest amount. There are 141,000 wells in the State but the wells requiring most energy are in the High Plains where 44,000 wells use natural gas for pumping water (1).

Other users are competing more intensively for the remaining gas resources, production and reserves in the US are declining, and mounting political pressures resulting from distribution and cost problems mean that other sources of energy need to be developed in the near future. The short term outlook is bad and the long term outlook is dismal for those who now use natural gas for irrigation. Already irrigation farmers in the Pecos Region are going out of business. The options for the irrigation farmer are: (1) alternative sources of energy (wind, solar, bioconversion), (2) electricity which will be generated from coal and/or nuclear power, or (3) a return to dryland farming with a major reduction in crop yield. For the near future, they will need to obtain higher prices for their products to cover the increased cost of natural gas and/or electricity.

The obvious advantages of using wind power are: (1) it is a non-depletable and non-polluting resource, (2) zero fuel costs, and (3) its use would afford a considerable degree of independence from outside energy resources and the distribution of those resources. The major disadvantages are: (1) low energy density which means a relatively large capture area, (2) intermittent and variable behavior of the wind, and (3) the combination of (1) and (2) means a high initial capital cost. Those areas of high wind energy potential will likely provide sites where economic success is most probable, particularly if power demands in the region are compatible. Therefore the combination of the wind energy resource (2) and of the irrigation demand makes the High Plains of Texas the logical place to implement prototype systems. The overall problem is to develop a complete system to pump water in quantities significant for irrigation, which is both reliable and economic.

The kinetic energy of the wind can be converted to electrical, mechanical, thermal, and chemical energy. Most of the current efforts to use wind energy are directed toward units that generate electrical power. In contrast, a water pumping system does not specifically require electrical power and all the farmer needs is mechanical power for the pump, whatever the source.

The farm windmill represents the only long term successful utilization of wind energy in the US. A regional count (99-104° W Long, 33-37° N Lat) on USGS maps (3) gives a figure of about 30,000 and with an average power output of around .25 kw. These windmills produce the equivalent of 25 to 50 million kwh of energy annually. Since all the energy produced is used in direct mechanical work, a minimum of three times this equivalent in fossil fuel is saved each year. But the real value of the farm windmill becomes obvious when the cost of providing power to remote locations is considered, which would also have to include the cost of the distribution system required.

The power levels achieved by the farm windmill are too small for use in irrigation and mechanical features of their design preclude a simple scaling up. Power levels of 10 kw and up would be adequate for many rural operations and a 20 kw wind unit could be used for irrigation. The Brace Institute built and tested an irrigation system in Barbados where a 32 ft diameter rotor was coupled through a standard Austin truck transmission to a commercially available centrifugal pump (4). The rotor was not self-starting or orienting, and the blades of fiberglass were fairly expensive.

GENERAL SYSTEM PARAMETERS

The amount of power needed and the wind distribution are the important parameters in determining the size of the rotor. A 20 kw unit was selected on the basis that it would provide enough power to deliver 200 gpm from a 200 ft depth, which represents a small irrigation well in the High Plains. The required wind unit would not be too large. Such a unit will be smaller than the 100 kw (1000 gpm) needed to operate a pivot-point sprinkler for a quarter section, so for this type and size system, smaller wind units could serve only as a supplementary source of energy.

The next problem is how to determine the most effective manner of coupling the wind rotor with the type of pumping system chosen. The wind unit can deliver power which is proportional to the cube of the wind speed, if the rotational speed is proportional to the wind velocity. For optimum power transfer the power characteristics of the load should properly match those of the rotor. The variable speed or constant speed rpm of the wind turbine/load impose different restrictions on the system (5). This problem pervades all mechanical systems wherein efficiency is a consideration in power transfer.

The types of pumps to be considered are the positive displacement pump and the air lift pump (6,7). Currently, irrigation wells have centrifugal pumps which operate essentially at constant rpm and load. Their performance is characterized by a minimum rpm below which no water is delivered and only a small range of rpm above the minimum before damaging speeds are encountered. This behavior generally rules out the direct connection of wind units to centrifugal pumps, but there is a possibility of connecting a wind unit as a supplementary source of power to the existing electric motor or internal combustion engine.

Wheat and sorghum are to be considered as the irrigated crops. Since wheat needs water in the fall and spring, and sorghum needs water in the summer, and since the annual variation in the wind provides more energy in the spring, the greater problem will be to deliver enough power to irrigate sorghum.

ROTOR DESIGN

Since a 20 kw wind unit was selected, the rotor was designed using the average wind speed distribution (2, p. 50) for Amarillo, Texas. Wind speed increases with height. A tower height of 40 ft was selected on the assumption that towers will be constructed of available 20 ft pipe sections, but a detailed analysis of increased energy with height versus increased tower costs has not been considered.

The wind speed distribution at 40 ft was calculated from the Amarillo NWS station data. (Figure 1). From that distribution the average power in kwh/(ft^2-year) was calculated (Figure 2). The peak of the curve occurs around 22.5 mph (10 m/s), therefore the wind unit is designed to give 20 kw at 10 m/s. If a turbine is to be operated at constant rpm, then the initial design point should be at the peak of the power distribution curve. The uncertainties in the computer model and lack of experimental verification preclude a more detailed analysis of where the design point should be in relation to the wind energy distribution and power characteristics of the rotor. An upper speed has to be selected at which the rotor blades feather and at which very little energy is lost.

The radius (16.4 ft = 5 m) of the rotor was then determined from the 20 kw at 10 m/s. The airfoil selected for the calculations was the NACA 0012, because lift and drag coefficient data were available over airfoil attack angles from 0 to 90 degrees. In a later section, different airfoils will be discussed. The solidity was selected from the standpoint of rotor rpm (low solidity means a high rpm), and the number of blades, three, was selected for balance and minimizing yaw problems from gyroscopic effects.

The computer program developed by R. E. Barieau (8), which includes an optimization procedure for the PROP program of Wilson and Lissaman (9), was used to calculate rotor performance. For further information, such as how to operate the program, and for examples of input and output data for the BOB.F4 program, see the companion report by R. E. Barieau. The time-sharing mode allows one to calculate and optimize the parameters of the rotor in a minimum time.

The coefficient of power, Cp, was maximized by changing the tip speed ratio and the twist angles. The difference between the optimized twist angles along the blade and no twist was a five percent increase in Cp. The optimum tip speed to wind speed ratio (X), and twist angles are independent of the wind speed, therefore a rotor gives optimum performance, maximum Cp, at a certain X. The solidity of the rotor can be changed by changing the number of blades or by changing the chord length. A lower solidity, .026, still keeping a high power coefficient,

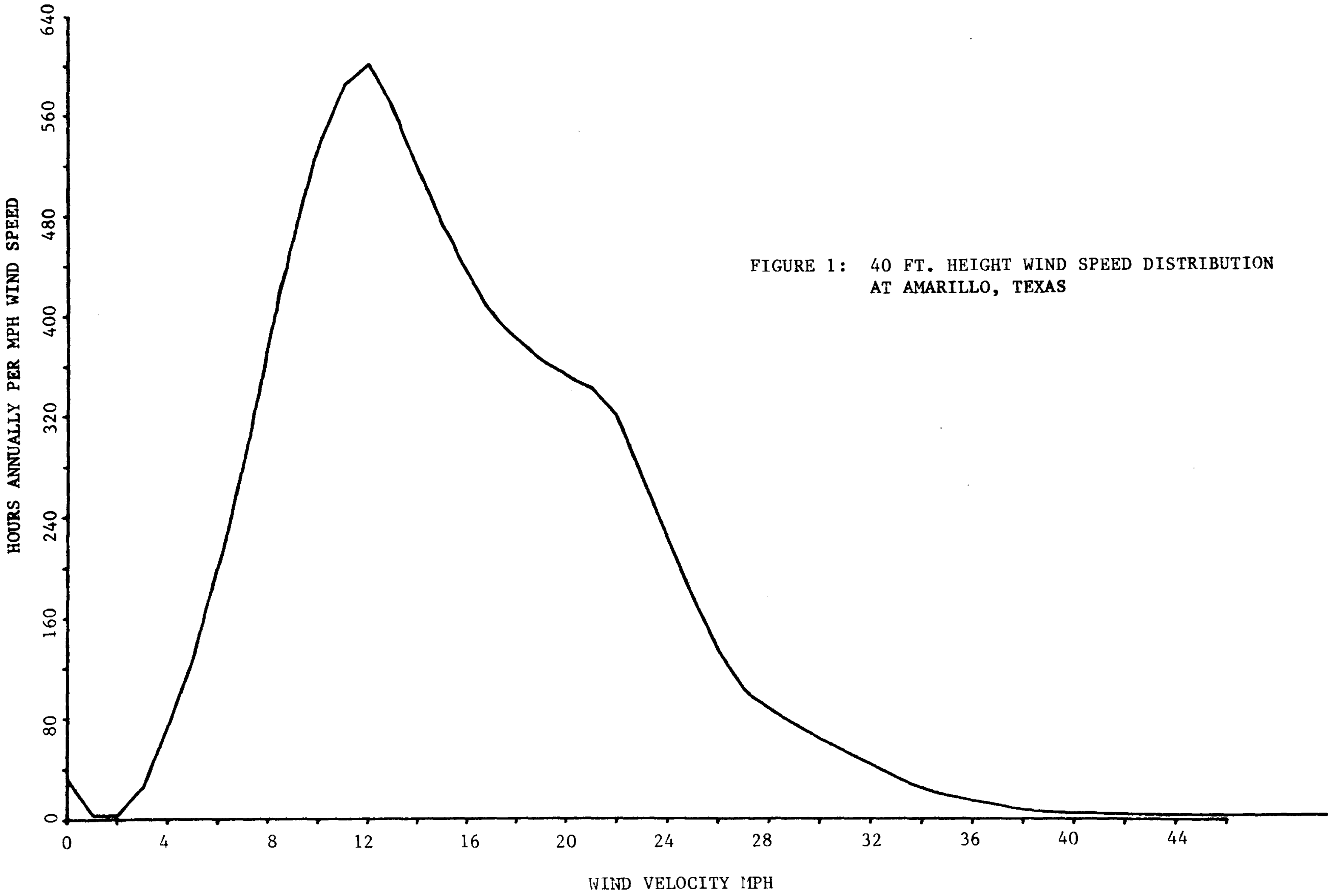


FIGURE 1: 40 FT. HEIGHT WIND SPEED DISTRIBUTION AT AMARILLO, TEXAS

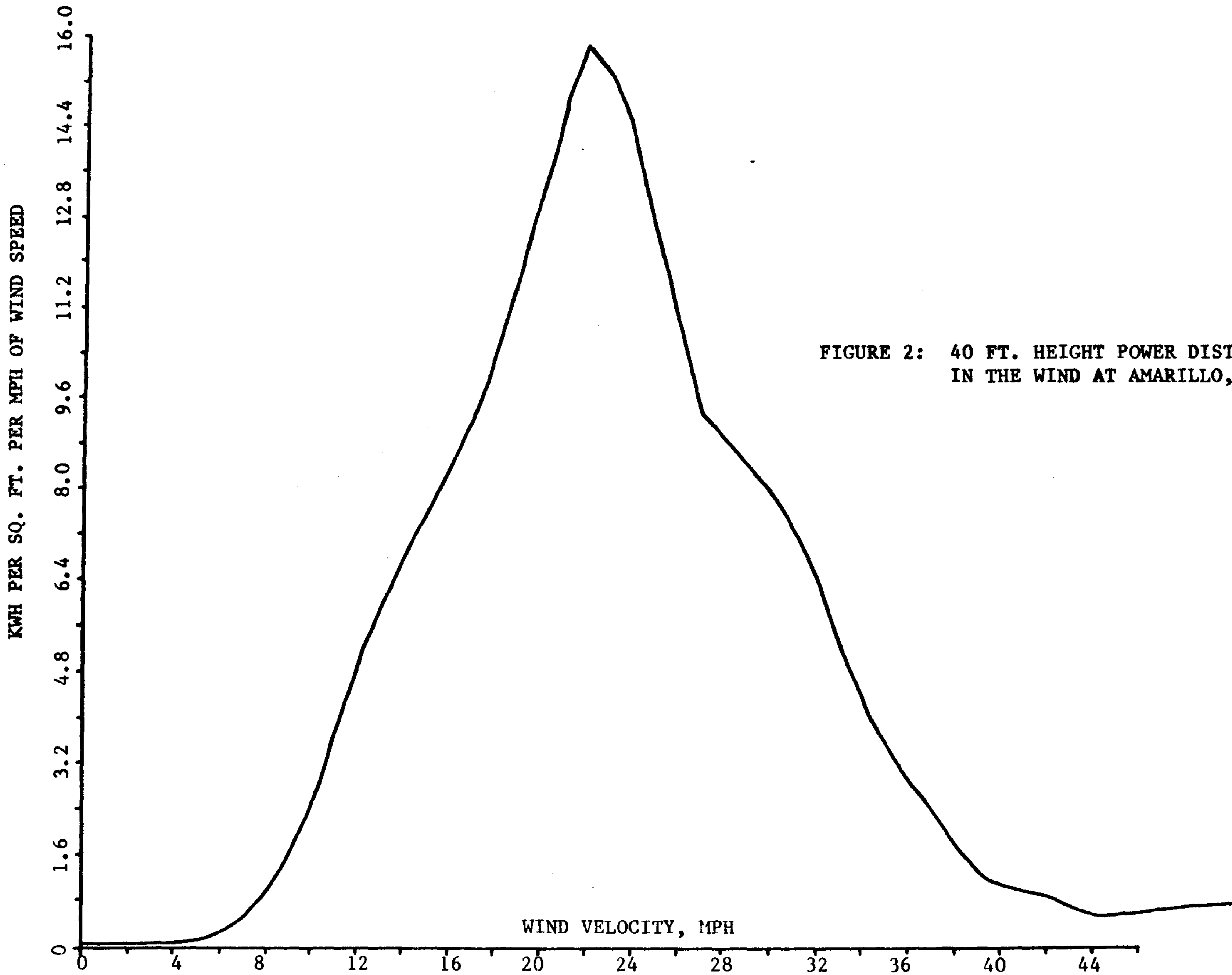


FIGURE 2: 40 FT. HEIGHT POWER DISTRIBUTION IN THE WIND AT AMARILLO, TEXAS

gave an rpm of 186. This was deemed too high and the chord length was increased to give a solidity of 0.076.

The parameters of the optimized rotor (designated as WT5) are the following for a wind speed of 22.4 mph.

Blades; airfoil 0012
Radius, r = 16.4 ft = 5 m
Hub radius, h = 3.28 ft = 1 m
Chord, c = 1.64 ft = .5 m

Area of rotor, A = 845 ft^2 = 78.5 m^2
Solidity = 0.076

Tip speed/wind speed, X = 7.0
Revolutions per minute, ω = 134
Power, P = 23 kw
Power Coefficient, Cp = .52

Table 1: Data Output of BOB.F4 for WT5

Theoretical Performance of a Propeller Type Wind Turbine
Radius - Ft = 16.400
Incremental Fraction for Integration = .1000
Hub Radius - Ft = 3.280
Pitch Angle - Degrees = 0.0000
Number of Blades = 3.0
Altitude of Site Above Sea Level - Ft = 3000.0
Coning Angle - Degrees = 0.000
Number of Data Stations Along Span = 9
NACA Profile = 9999 (NACA 0012 Airfoil)
Standard Axial Interference Method Used
No Tip Loss Model Used
No Hub Loss Model Used
Method of Solution, Newton-Raphson
Solution Number, 1

PERCENT RADIUS	CHORD-FT	TWIST ANGLE-DEGREE
100.0000	1.64000	1.81875
90.0000	1.64000	2.04688
80.0000	1.64000	2.31563
70.0000	1.64000	2.73438
60.0000	1.64000	3.27500
50.0000	1.64000	3.96875
40.0000	1.64000	5.50469
30.0000	1.64000	6.55000
20.0000	1.64000	11.65937

INPUT DATA AS ABOVE EXCEPT FOR TIP SPEED RATIO

TIP SPEED RATIO	RPM	KILOWATTS POWER	TORQUE	POWER COEFF	THRUST LBS	VELOCITY MPH
6.9922	133.76	22.97792	1209.99	0.5203	807.5123	22.4

If a coning angle of 10^{o} is put in to make the rotor self-orienting, then the power output is reduced by the factor $\cos^3(10)$, which gives a power of 22 kw. Once the twist and pitch angles are fixed then the power coefficient versus wind speed (Figure 3) for a constant rpm can be calculated. The curve is not smooth at higher wind velocities because the drag coefficient data have not been smoothed. Figure 4 gives the power coefficient versus tip speed ratio. As stated earlier, for the maximum power output, (constant power coefficient) the rotor needs to operate at a tip speed ratio of 7 over the total range of wind speeds.

ROTOR OPERATION

The computer program simulates a steady-state model and system dynamics can be inferred only from the family of steady-state curves. The power available from the wind unit is proportional to the cube of the wind velocity. The theoretical limit to the amount of power obtainable from a rotor of given area is equal to 59.3% of the power in the wind. Wilson and Lissaman (9) and Glauert (10) are good sources of information on the aerodynamics of wind machines.

The power output of the rotor is

(1) $P = \omega\tau = KCpV^3$

where ω = rotational speed
τ = torque
V = wind speed
Cp = power coefficient
K = a constant depending on density of the air.

As in every type of power transfer system, a careful matching of the load driven by the rotor with the torque-rpm characteristics of the rotor is necessary if optimal energy transfer is to occur.

In the following graphs describing the operation of WT5, the values for the graphs were taken from tables generated by the computer program. Again the curves are not smooth because the input data for drag coefficients are not smooth.

The three modes of rotor operation are the following: (1) constant rpm, (2) constant power coefficient, and (3) constant torque. For WT5 these three modes are indicated in Figure 5 by the curves A, B, and C. Each one of these modes has its advantages and disadvantages and at present it seems that a universal wind machine, i.e. one which could operate satisfactorily in any one of these modes, would be at a disadvantage when compared to a machine designed for a specific mode of operation.

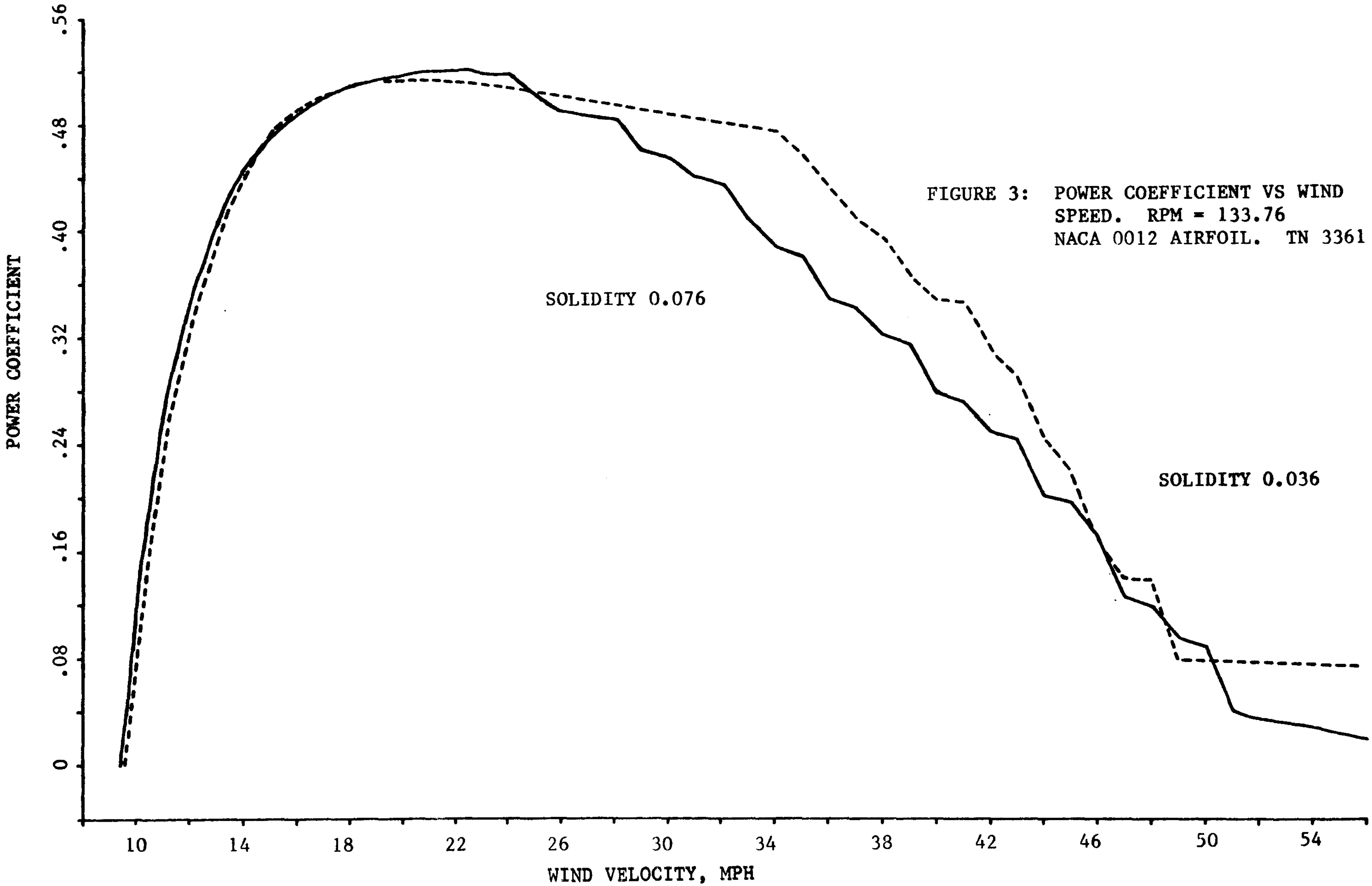


FIGURE 3: POWER COEFFICIENT VS WIND SPEED. RPM = 133.76 NACA 0012 AIRFOIL. TN 3361

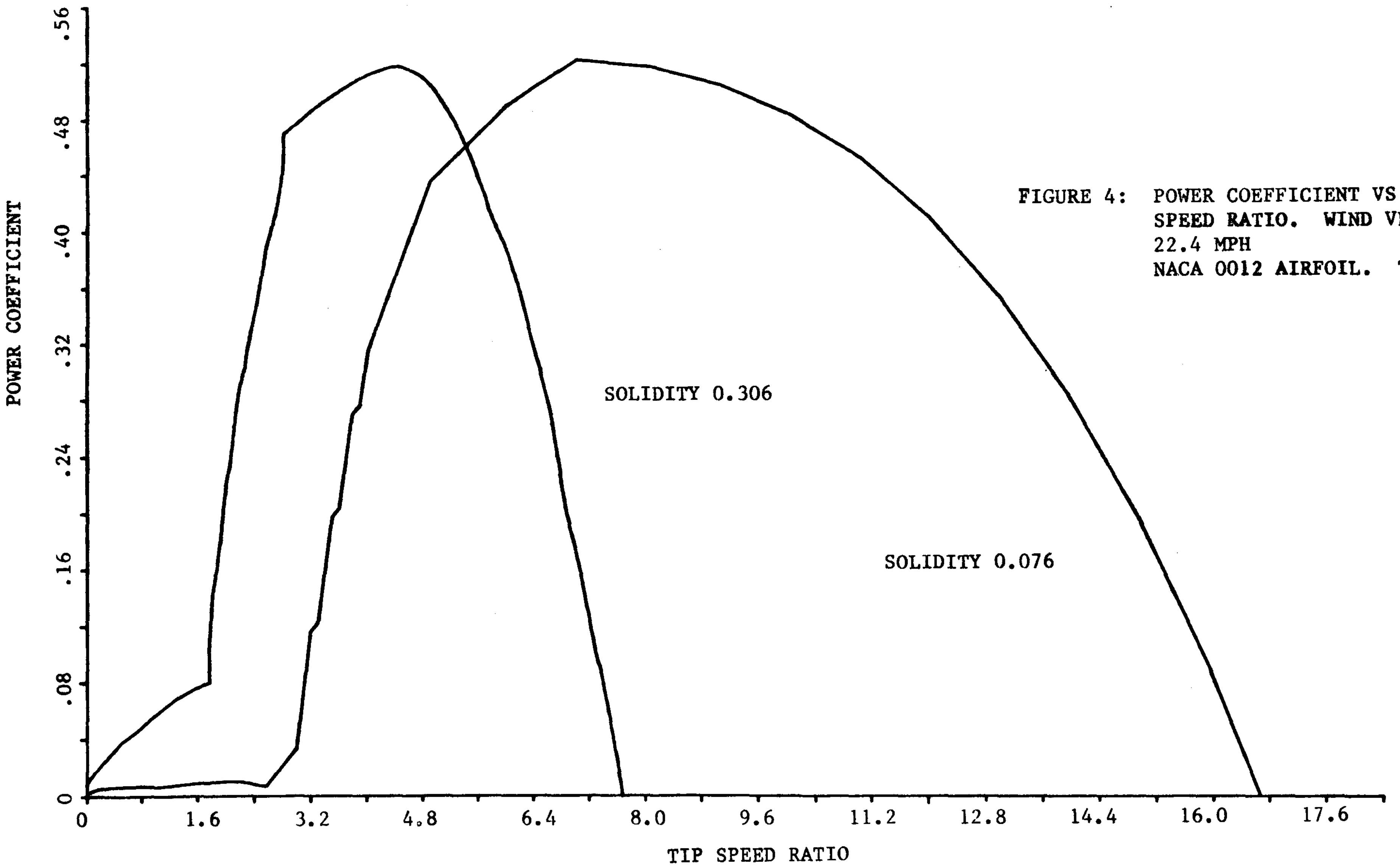


FIGURE 4: POWER COEFFICIENT VS TIP SPEED RATIO. WIND VELOCITY = 22.4 MPH
NACA 0012 AIRFOIL. TN 3361

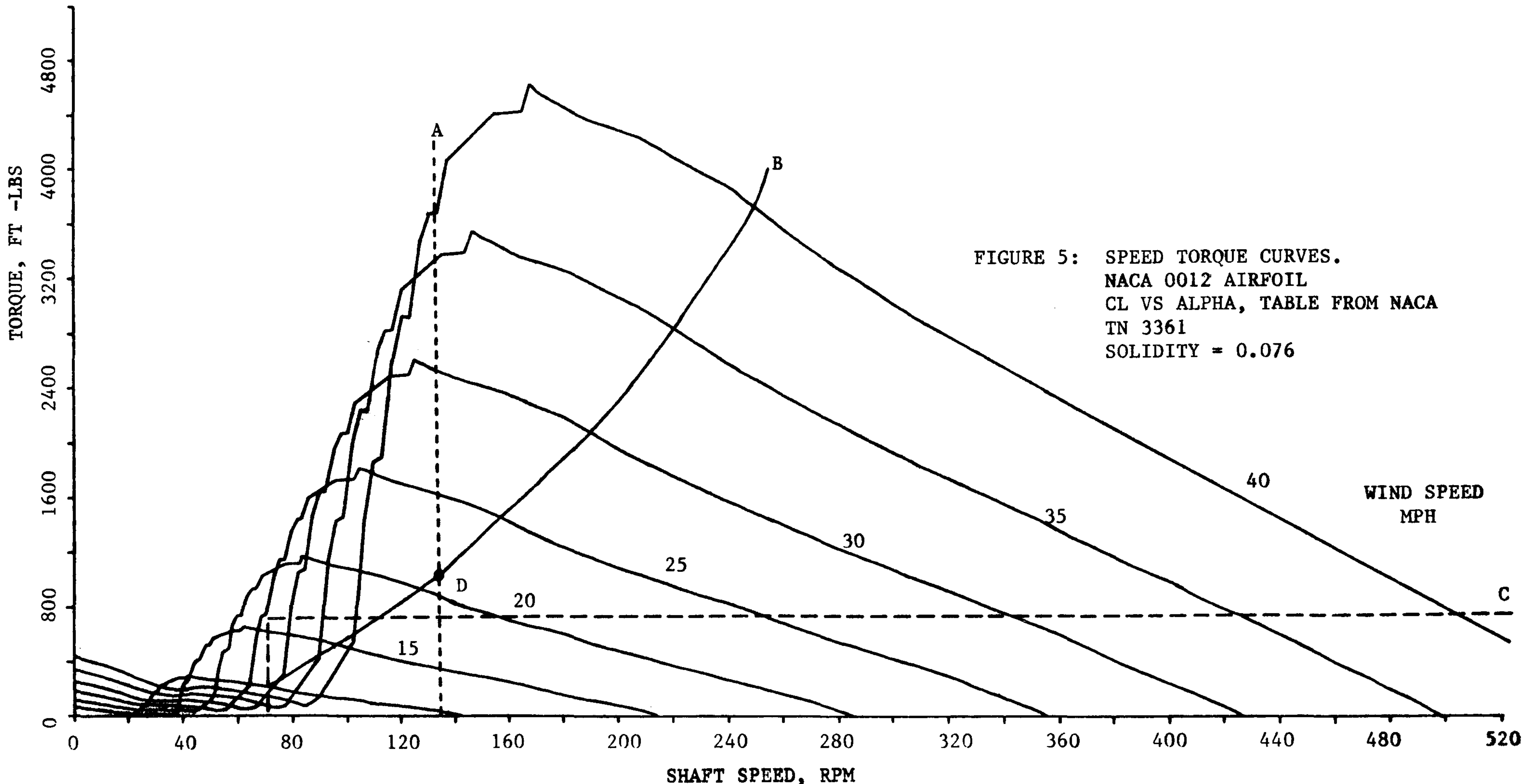


FIGURE 5: SPEED TORQUE CURVES.
NACA 0012 AIRFOIL
CL VS ALPHA, TABLE FROM NACA
TN 3361
SOLIDITY = 0.076

1. Constant RPM. Cp is a maximum at point D (the design point) in Figure 5 and Cp decreases (Figure 3) along the dotted line A on either side of point D. The load can vary only by means of torque changes and the problem is to determine the maximum size of load to provide for the most cost-efficient system. For example, if the load was held constant at D (23 kw) then the pitch of the blades would have to be changed to hold that constant load for all wind speeds above that point.

The amount of energy generated over one year at constant rpm was calculated from the rotor parameters and from the wind distribution curve (Figure 1). This gives a value of 46% compared to the theoretical value of 59% for a perfect rotor and a value of 52% for WT5 at peak efficiency. A problem arises because a large load, 75 kw, is required at 43 mph wind speed. Because Cp decreases at wind speeds above the design point, 42 kw load is needed at 50 mph. The percentage accumulation of actual power (% of 46% possible at constant rpm) is plotted against the wind speed (Figure 6) and a decision can be made on percentage power and size of load. For example, if the blades were feathered at 23 mph (24 kw load), 27% (.59 x .46) of the energy available in the wind is actually captured, or if the rotor is feathered above 32 mph (56 kw load), the rotor efficiency would be 42% (.92 x .46). Of course there will be other efficiencies (frictional, drive train, pump, etc.) which would reduce the actual efficiency but part of the energy above 23 mph can be picked up by changing the pitch to maintain a constant load. If the power is held constant (24 kw) from 23 to 50 mph the efficiency would be 38% and an increase to 31 kw (25 to 50 mph) would give 41% efficiency.

The most common example of this type of operation is the synchronous generator connected to the utility line. For pumping water a possible system would be: wind unit, generator, battery storage, electric water pump. Another possibility is to connect the wind unit to an existing motor, electric or internal combustion engine.

2. Constant Power Coefficient. For curve B (Figure 5) the load torque has to increase as the square of the rpm. This is the mode of operation giving highest efficiency (52% of the available energy), but the problems are: a variable speed transmission is needed, the rpm becomes large with increasing wind speed, and the power rating of the load must be large.

Again the percentage accumulation of power (% of 52%) is plotted against the wind speed (Figure 7) and a decision can be made on percentage power and size of the load. For example, if the rotor is feathered above 23 mph, 30% (.58 x .52) of the energy available would be captured, or if the rotor is feathered above 32 mph (67 kw load, 200 rpm), the rotor would give 46% (.88 x .52) efficiency. The possibility exists for obtaining energy above that wind speed by changing the pitch of the blades. If WT5 was held at constant load (24 kw) from 23 to 50 mph, then the efficiency over one year would be 41%. The economics of a larger load and a variable speed transmission have to be contrasted with the higher rotor efficiency.

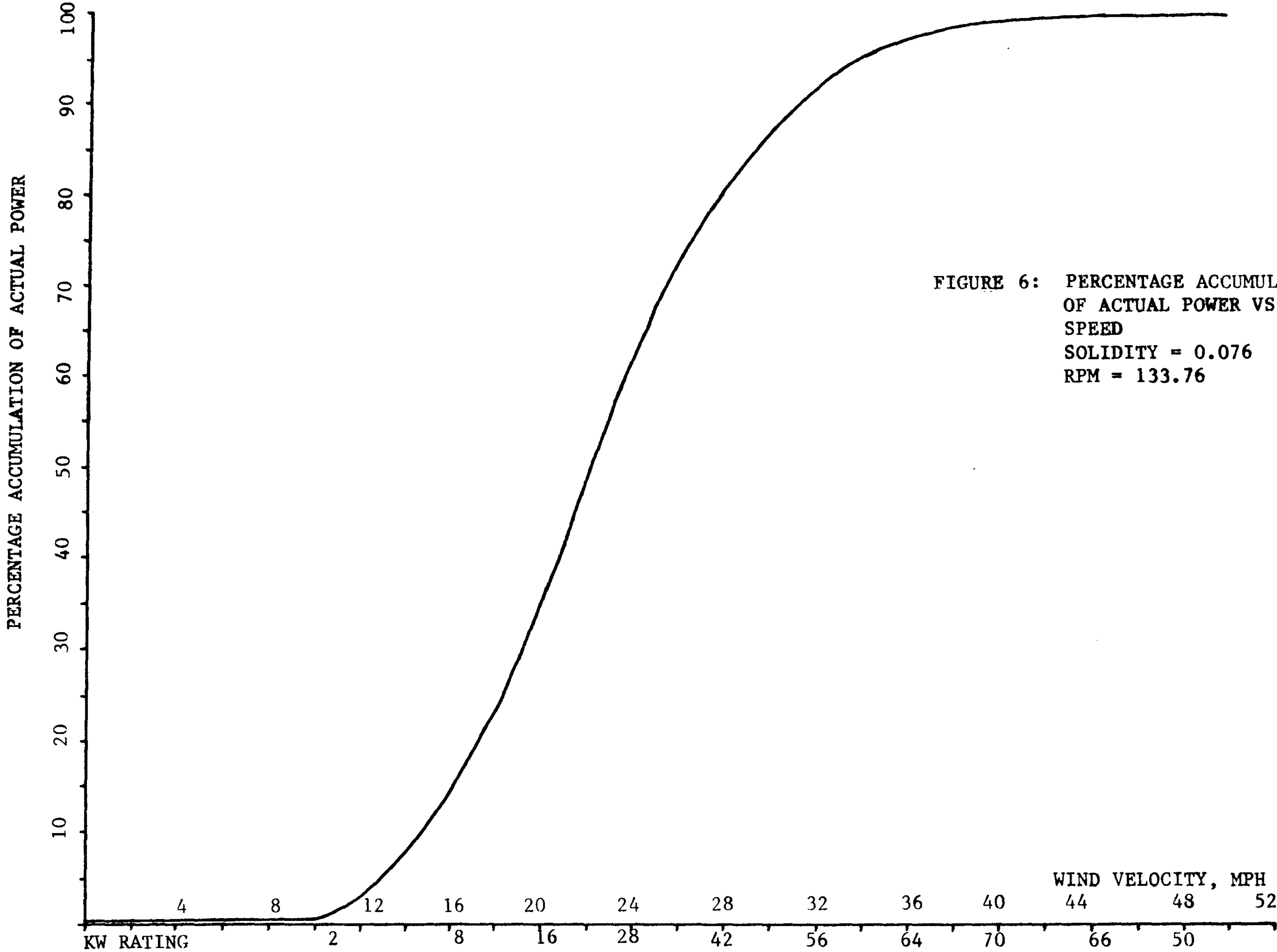


FIGURE 6: PERCENTAGE ACCUMULATION OF ACTUAL POWER VS WIND SPEED
SOLIDITY = 0.076
RPM = 133.76

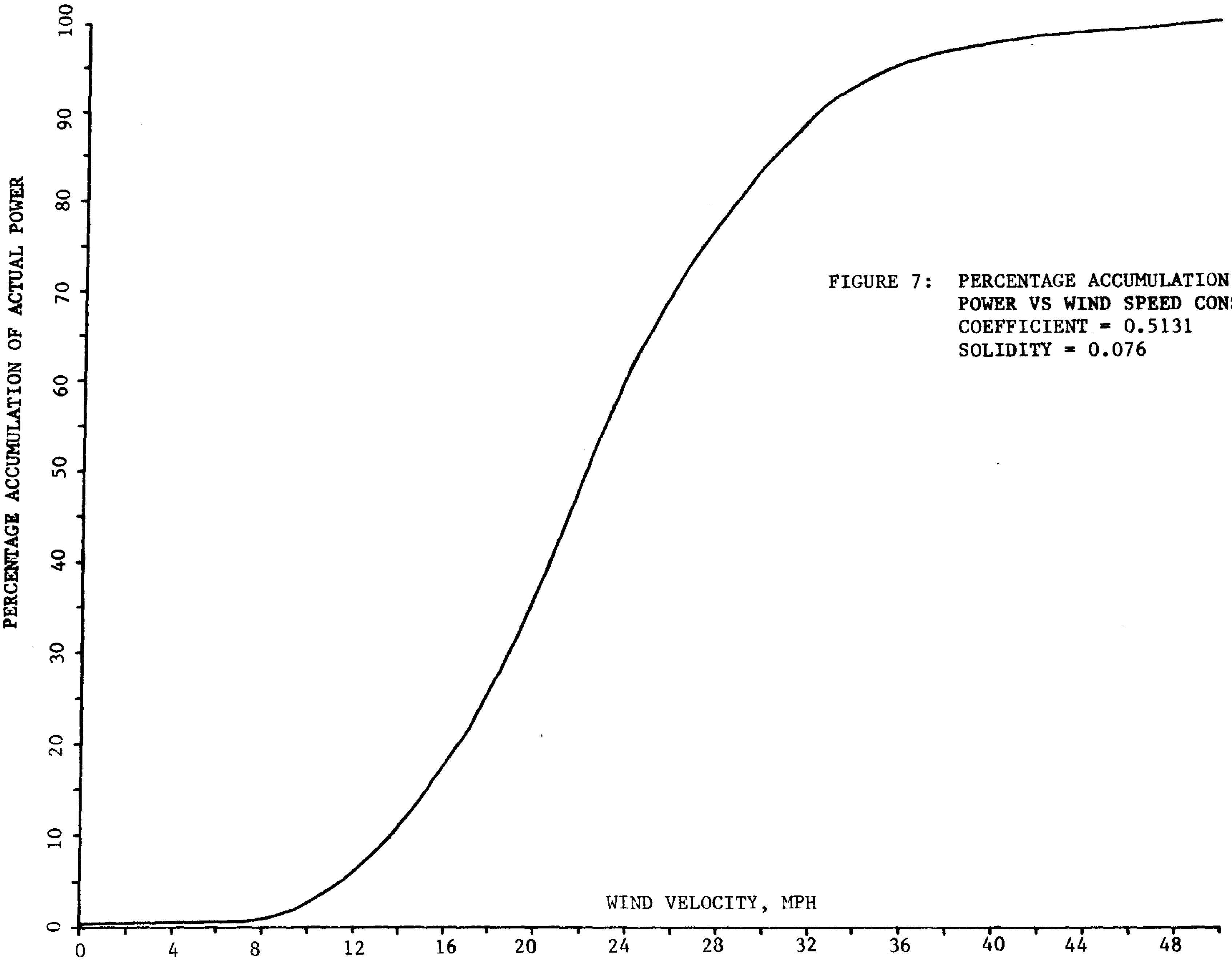


FIGURE 7: **PERCENTAGE ACCUMULATION OF ACTUAL POWER VS WIND SPEED CONSTANT POWER**
COEFFICIENT = 0.5131
SOLIDITY = 0.076

3. Constant Torque. For those operations where the load increases as rpm, the torque is essentially a constant (line C in Figure 5) . This operation is similar to constant rpm operations in that the Cp (Figure 8) goes to zero away from the design point. There are two other restrictions that make this design restrictive; no power can be generated below the starting velocity (16 mph) and the rpm increases rapidly for rotors of low solidity.

If no restrictions on rpm exist and the maximum torque is attained at 16 mph then the efficiency over one year would be 33%, however, at 30 mph the rotor would have to spin at 300 rpm (Figure 5), which is too high for safe operation. Therefore, the overall efficiency will decrease, depending on the limiting rpm chosen. For example, if an upper limit of 200 rpm is chosen then that corresponds to a wind velocity of around 24 mph and the percentage of energy captured would be about 20%.

The analysis illustrates that for constant torque operation, which is essentially provided by a positive displacement pump, the rotor should be designed for high torque. With this in mind, additional calculations were made, this time for a rotor having a solidity of 0.306, four times larger than the original model. For this value of solidity the rpm is lower (Figure 5), Cp is broader (Figure 3) and the torque is larger (Figure 9). The percentage accumulation of power (Figure 10) is calculated from Cp versus wind velocity and the average energy distribution. The problem of obtaining no power until the starting velocity (15 mph) is reached still exists, but an upper limit of 200 rpm now corresponds to a velocity of 35 mph, 41 kw (see Figure 9). The percentage of energy captured would be 33% (.95 x .35), a large increase over the 20% with the low solidity.

For the constant torque operation, Figure 11 gives the reduced tip speed ratio (X/X_s) versus the reduced wind speed (V/V_s), where the subscript s refers to the starting values. This curve is the same, no matter what V_s and X_s and the values of X can be calculated as a function V to be put into the computer program.

AIRFOIL SELECTION

The airfoil used for most of the calculations was the NACA 0012, since the lift and drag coefficients were available for attack angles from 0 to 180 degrees (11). Most data on airfoils (12) are for attack angles over a limited range, up to 30 degrees. Aerodynamic design for a large wind unit (190 ft diameter) was modeled by a computer program (13) and airfoils (NACA 44XX, 230XX) having a lift coefficient around 1.0 gave satisfactory performance.

The power coefficient for the WT5 unit can be calculated with respect to pitch, twist, and tip speed ratio, but different sections of the blade will be at attack angles which are away from the maximum lift. Since the radial velocity changes along the blade, sections of the blade may even give negative contributions, therefore the airfoil should be selected for overall performance, lift over a wide range of attack angles, rather than a high lift/drag at one point. The second criterion

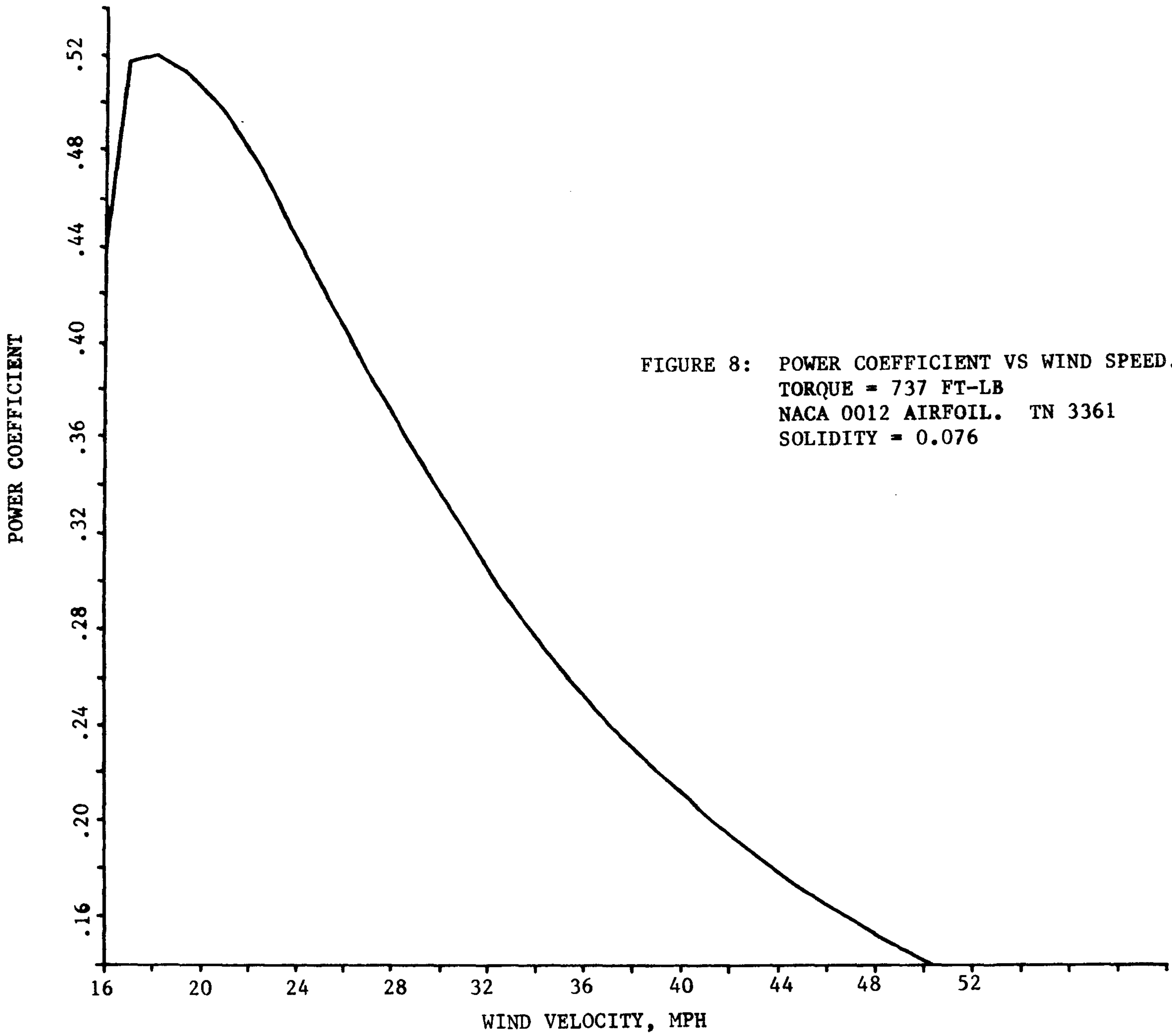


FIGURE 8: POWER COEFFICIENT VS WIND SPEED.
TORQUE = 737 FT-LB
NACA 0012 AIRFOIL. TN 3361
SOLIDITY = 0.076

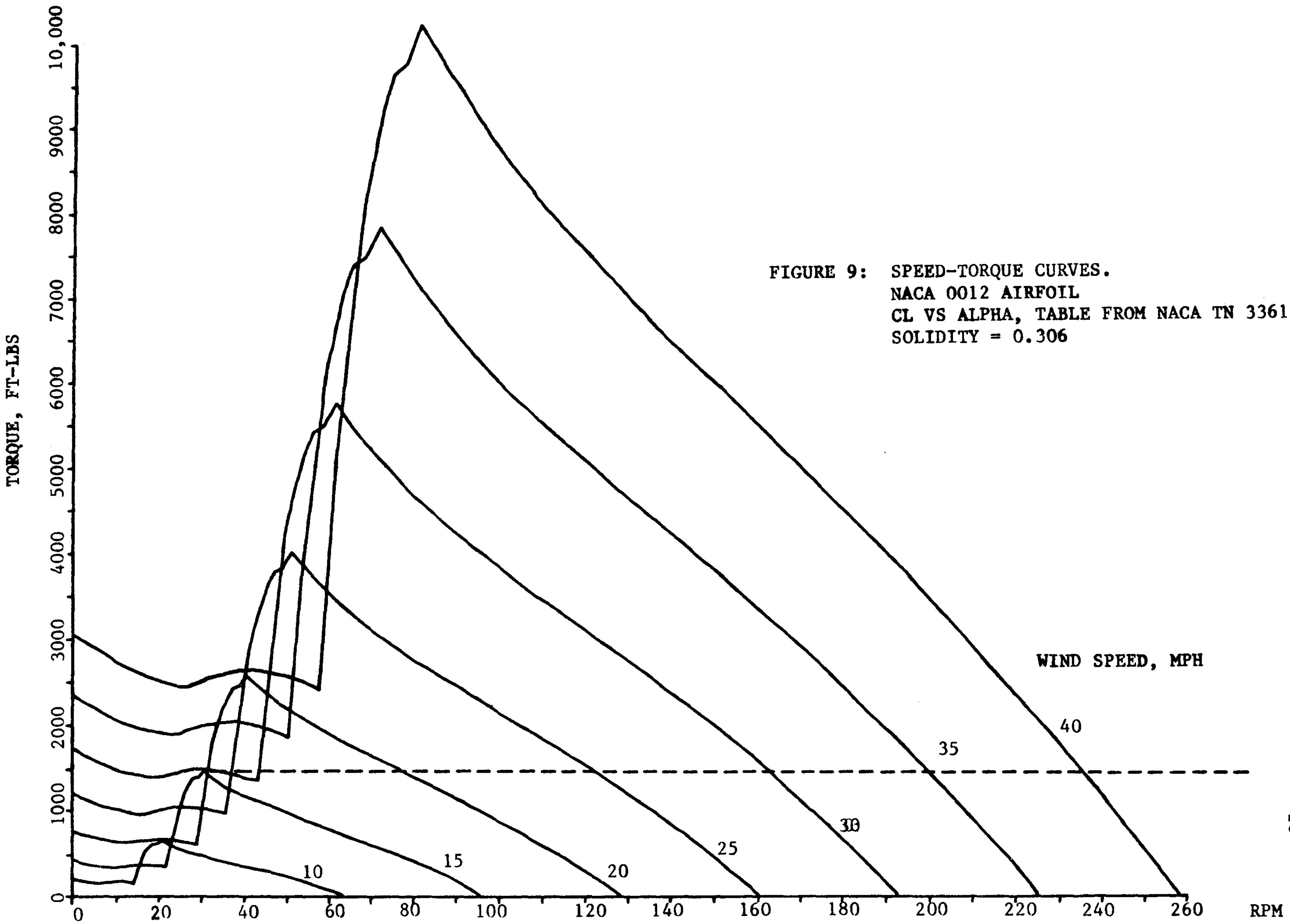


FIGURE 9: SPEED-TORQUE CURVES.
NACA 0012 AIRFOIL
CL VS ALPHA, TABLE FROM NACA TN 3361
SOLIDITY = 0.306

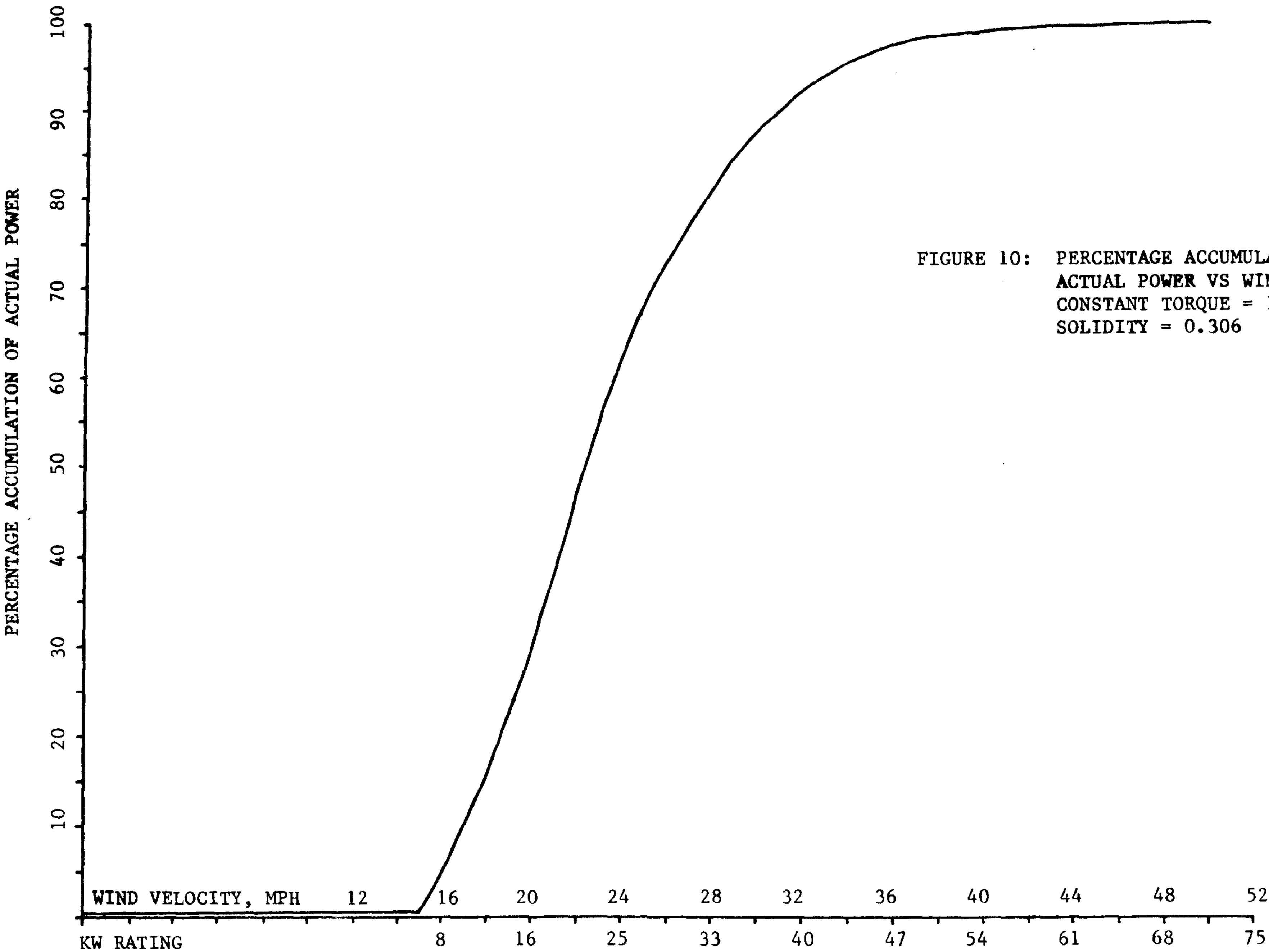


FIGURE 10: PERCENTAGE ACCUMULATION OF ACTUAL POWER VS WIND SPEED
CONSTANT TORQUE = 1645 FT-LB
SOLIDITY = 0.306

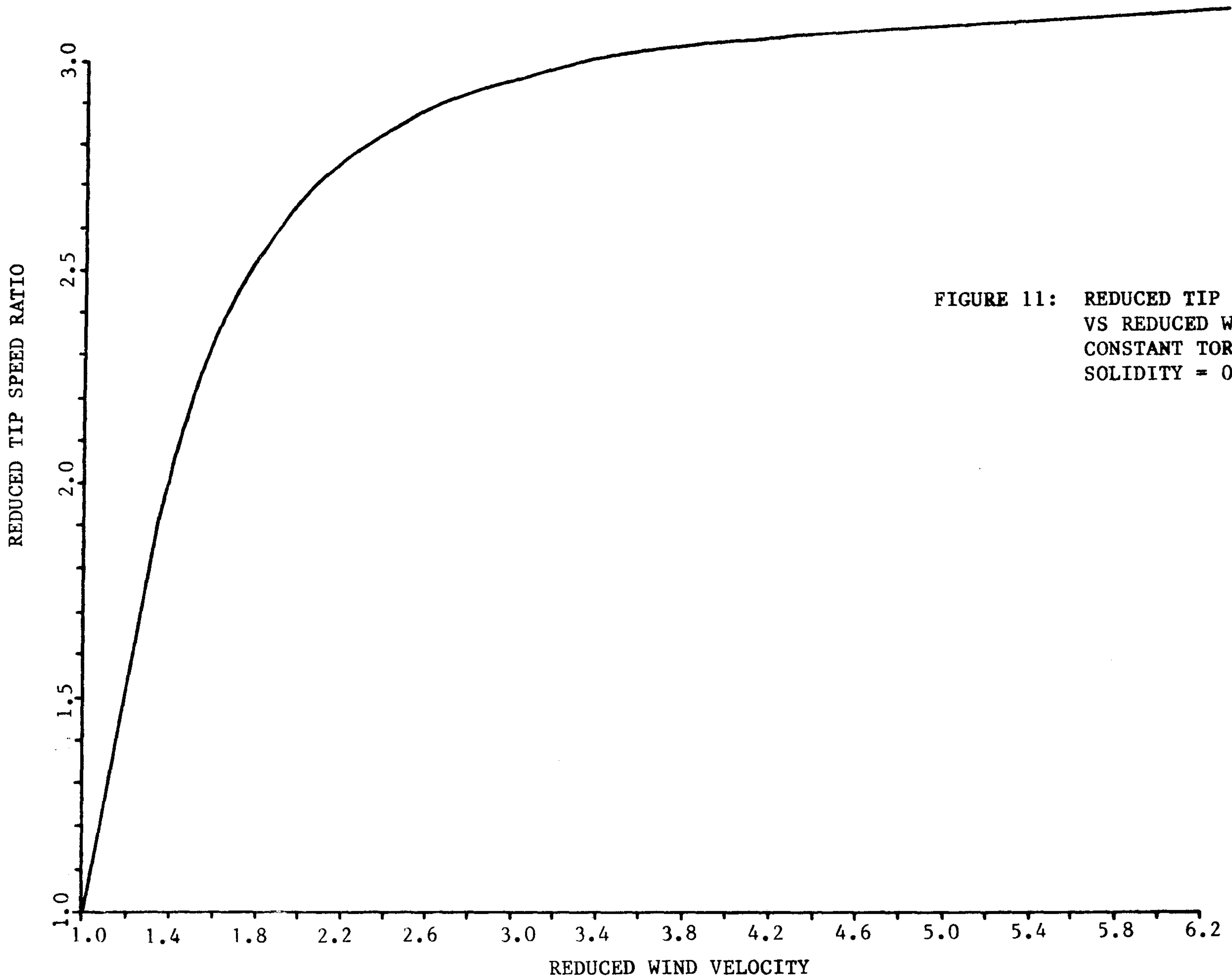


FIGURE 11: REDUCED TIP SPEED RATIO VS REDUCED WIND SPEED FOR CONSTANT TORQUE OPERATION SOLIDITY = 0.306

should be the percentage of annual energy which the unit would deliver and third is the cost of construction. For those rotors where the pitch control is used only at high wind speeds, the large attack angle data are important for calculating starting torques.

The values of lift and drag coefficients were graphed (Figures 12 and 13) for the following airfoils: NACA 0012 Tabular, smoothed data points calculated from tabular values taken from Technical Note (11); NACA 0012 Double Gaussian, which was a non-linear least squares fit for the lift coefficient only; NACA 0012 Wilson, and NACA 4418, from the analytic representations in Wilson's program (9); NACA 23018 refers to data obtained from Professor Wilson that is included in the March 15, 1976 version of their PROP program; and 4418 HA(0012), lift coefficient of the 0012 at attack angles above 32° added to the 4418 data. The last four used an approximation for the high angles of attack as data were available only up to around 22 degrees.

These airfoils were then used in the optimization program to calculate Cp, for a wind speed of 22.4 mph, with respect to tip speed ratio and twist angles, and with respect to tip speed ratio and pitch with zero twist angles (Table 2). Also the efficiency of each airfoil for WT5 was calculated using the yearly wind speed distribution (Fig. 1). In the computer program, a quadratic interpolation was used in estimating the lift and drag coefficients for the 0012 airfoil. The calculation for the 23018 had a provision for the effect of the relative wind speed on the lift coefficient.

TABLE 2. Yearly Efficiencies at Constant RPM, Maximum Power Coefficient, and RPM for Rotor WT5.

	EFF		Cp Max		RPM	
Airfoil	No Twist	Twist	No Twist	Twist	No Twist	Twist
0012 Tabular	43.1	45.6	.4908	.5203	138	134
0012 Double Gaussian	44.2	45.7	.4854	.5212	141	134
0012 Wilson	38.5	37.3	.4727	.4991	132	106
4418	44.3	44.7	.4976	.5046	108	103
4418 HA(0012)	44.3	44.7	.4977	.5046	113	103
23018	32.8	33.5	.4069	.4648	124	108

The data in Table 2 indicate that the Double Gaussian gives a close representation of the 0012, and the use of the Double Gaussian eliminated the multiple solutions in the program. The data for the 0012 Wilson, 4418, and 23018 indicate the importance of having adequate data for lift and drag coefficients over a wide range of angles. Adding the NACA 0012 high angle of attack data to Wilson's data for the 4418 did not lead to an increased efficiency. The 0012 (Wilson) without the high angle data was quite inferior, as was the 23018.

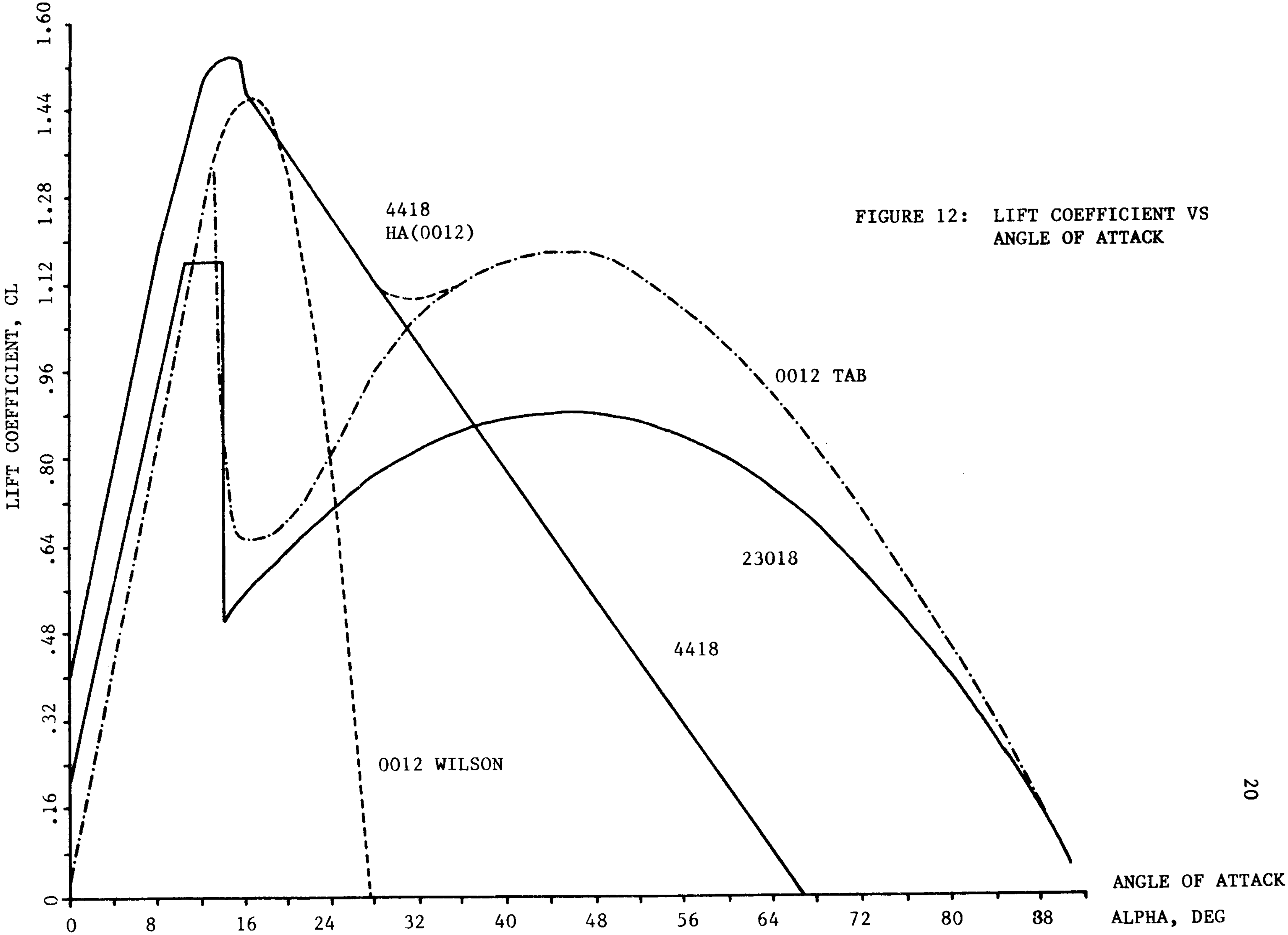


FIGURE 12: LIFT COEFFICIENT VS ANGLE OF ATTACK

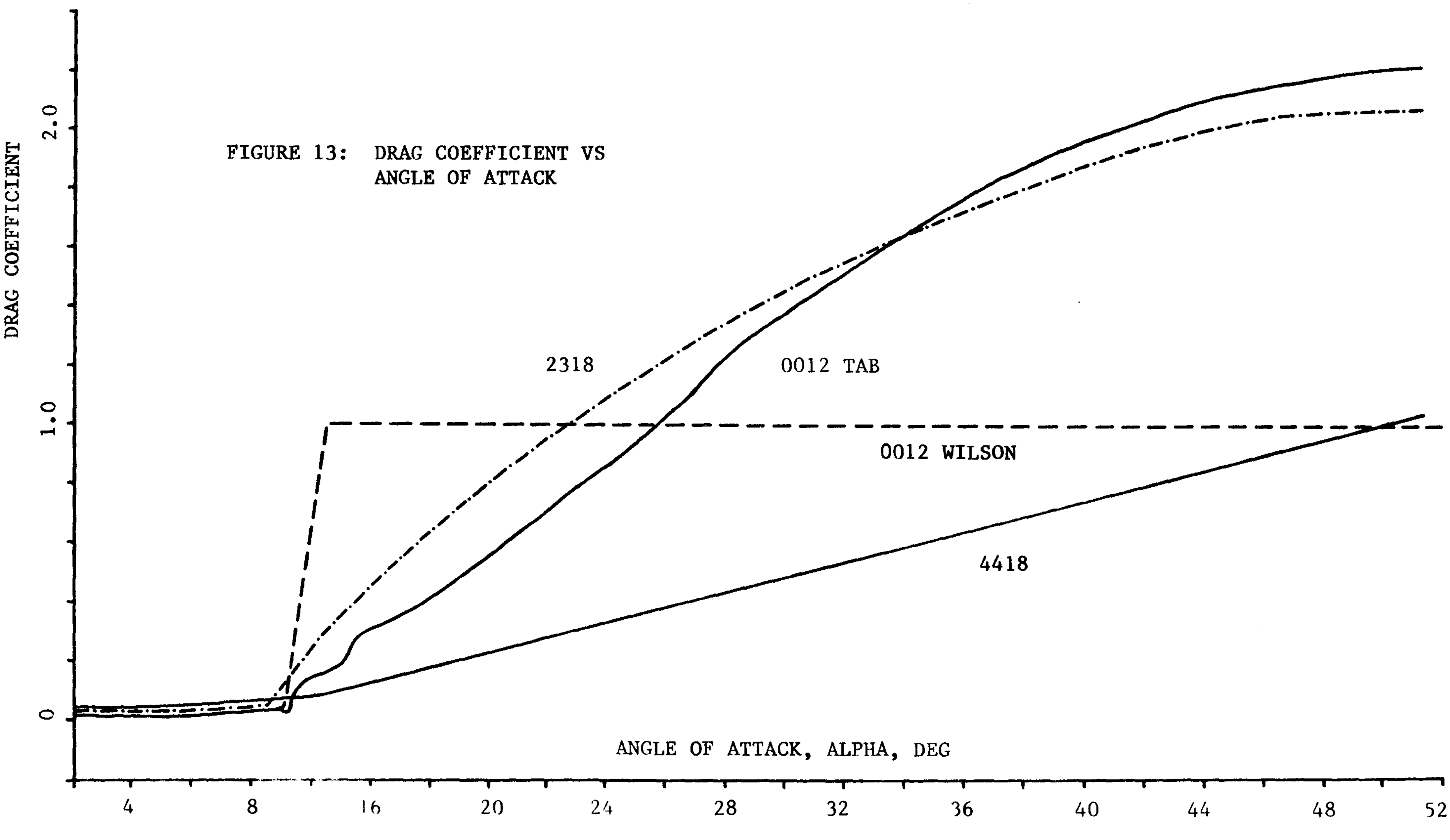


FIGURE 13: DRAG COEFFICIENT VS ANGLE OF ATTACK

The 0012 Tabular airfoil is an adequate airfoil for the computer model of an optimum rotor. Probably a number of airfoils would give satisfactory performance, but until more experimental data are available (13), a decision on which type of airfoil to use involving 2 to 5 percent differences in overall efficiency will be difficult to make from computer models.

PUMPING SYSTEMS

Positive displacement pumps. Rotary pumps are characterized by steady (non-pulsating) rates of discharge, high pressure delivery capabilities, high hydraulic efficiencies (90-95%), capacities proportional to shaft rotational speeds, and brake horsepower proportional to shaft speed and dynamic head. If used downhole, all rotary pumps would obviously require power shafting from ground level down to pump level.

A. Gear pumps. It was found that none of the gear type rotary pumps in production is recommended for pumping water, nevertheless, a few are being used in this manner (14). The primary problem is with proper lubrication if water is the fluid being moved. Downhole configurations of this type of pump could not be found in manufacturer's bulletins, in Thomas' Register, or by means of several telephone conversations with technical personnel in industry. The Keller rotary pump (KROV) incorporating new mechanical features has recently been designed which might be developed into a satisfactory downhole water pump (15), although it is not a conventional gear pump.

B. Moyno or progressive cavity pumps. Applications of this type of pump to water pumping are generally limited to low capacity, high pressure systems. The pump is used primarily to pump high viscosity liquids, slurries, and other liquids presenting special problems. Some designs are subject to unacceptable shaft vibrations when operated at rotational speeds other than two or three recommended speeds. For this reason units with this characteristic would surely offer formidable problems when driven by wind rotors. As far as can be determined, no manufacturer produces this type of pump in a downhole version with the capacity required for the irrigation application under consideration. The maximum rate specified was 52 gpm.

Reciprocating pumps are characterized by a pulsating discharge, the rate of which is approximated by a series of the positive halves of a sine function (16). The farm windmill is of this type. These pulsations occur at a much higher frequency ($\sim$0.1 to 1 Hz) than corresponds to the wind data measurement frequency, therefore, an analysis of the effect of this kind of load on the rotor cannot be made until (1) wind data of comparable frequency or statistically reliable representations are available, and (2) one has a dynamic model for rotor and system performance. About the best that can be done presently is to assume the rotational inertia of the system will smooth out the power demand to an average value. The system thus becomes mathematically equivalent to the positive displacement rotary pump.

The search for commercially available reciprocating pumps meeting the pumping rate specification of 200 gpm was unsuccessful. The units coming closest to providing this capacity are the types used for pumping oil. The largest unit found produces 166 gpm if operated at the extreme limit of its mechanical capabilities. A system incorporating such a pump can scarcely make use of the full 20 kw provided by the rotor even if considerable friction is present. The Harbison-Fischer 4 3/4 inch bore plunger (their largest) would have to execute twelve 15 ft strokes per minute to pump 166 gpm. It is clear that a very large rocker arm would be required, with some provision to drive one end from the wind rotor. The company engineer advises the use of the largest (1 1/8") sucker rod to resist buckling on the down stroke, and a pumping rate of 10 strokes per minute or less. It is clearly impractical, both from technical and economic considerations, to adapt this oilfield equipment to pump irrigation water.

Mathematical Representation of the Positive Displacement Rotary Pump. Despite the absence of a usable off-the-shelf item for the application being considered, the rotary pump will be mathematically modeled for inclusion in the computer systems analysis program. This decision can be justified on three counts: (1) a satisfactory pump may be available, (2) knowledge of how such a pump should perform in a windpowered system would be valuable, and (3) evidence indicating a general need for this kind of pump may stimulate its development by industry.

The power relationship between the pump and the rate of hydraulic work is

$$P_h = k_1 CH \tag{2}$$

where P_h = hydraulic power delivered by pump

C = volumetric flow rate

H = total dynamic head

k_1 = weight/unit volume.

The relationship between the dynamic and hydraulic quantities is found by combining Eqs. (1) and (2),

$$P_h = \varepsilon_p P$$

where ε_p = mechanical efficiency of pump.

Since

$$C = k_2 \varepsilon_v \omega \tag{3}$$

where ε_v = volumetric efficiency of pump and

k_2 = geometric factor characteristic of pump, the torque is

(4) $$\tau = Hk_1k_2(\varepsilon_v/\varepsilon_p), \text{ and}$$

(5) $$P = k_1k_2(\varepsilon_v/\varepsilon_p)H\omega.$$

H would be constant, provided well drawdown and fluid friction are constant over the practical range of pump shaft speeds. Under such circumstances, the torque would be independent of the rate at which water is pumped; the power consumed would be directly proportional to shaft rpm. In any practical pumping system water will be delivered to the surface with different velocities depending on the pumping rate. This velocity head, together with increasing fluid friction will cause a rotary positive displacement pump to require increasing torque for increased water delivery rates.

Illustrated in Figure 14 are the head quantities h_a, h_d, and h_{dd}; which are defined in terms of heights in a system containing an operating pump. In terms of these quantities, the static head is given by

$$H_s = h_d - h_a + C/s$$

where s is the specific well capacity, or volumetric delivery rate per unit length of drawdown. The total head is then found by adding the velocity and friction heads.

If the water is discharged through a circular pipe of diameter, d, the velocity head, h_v, is given by

$$h_v = (4C/\pi d^2)^2/2g.$$

If h_f represents the friction, the total head is

(6) $$H = h_d - h_a + h_f + C/s + (4C/\pi d^2)^2/2g,$$

where h_f is a function of C, but the nature of that relationship is not considered at the moment because it can be estimated only when the details of the total system are known.

Now τ can be written as $\tau(\omega)$ by using Eq. (3), in Eq. (6), and the results in Eq. (4), thus

(7) $$\tau(\omega) = k_1k_2(\varepsilon_v/\varepsilon_p)[h_d - h_a + h_f + k_2\varepsilon_v\omega/s + (4k_2\varepsilon_v\omega/\pi d^2)^2/2g].$$

The last term inside the bracket of Eq. (7) is the velocity head. Because it is important to know in detail the relationship between τ and ω, it is expressed here in terms of ω.

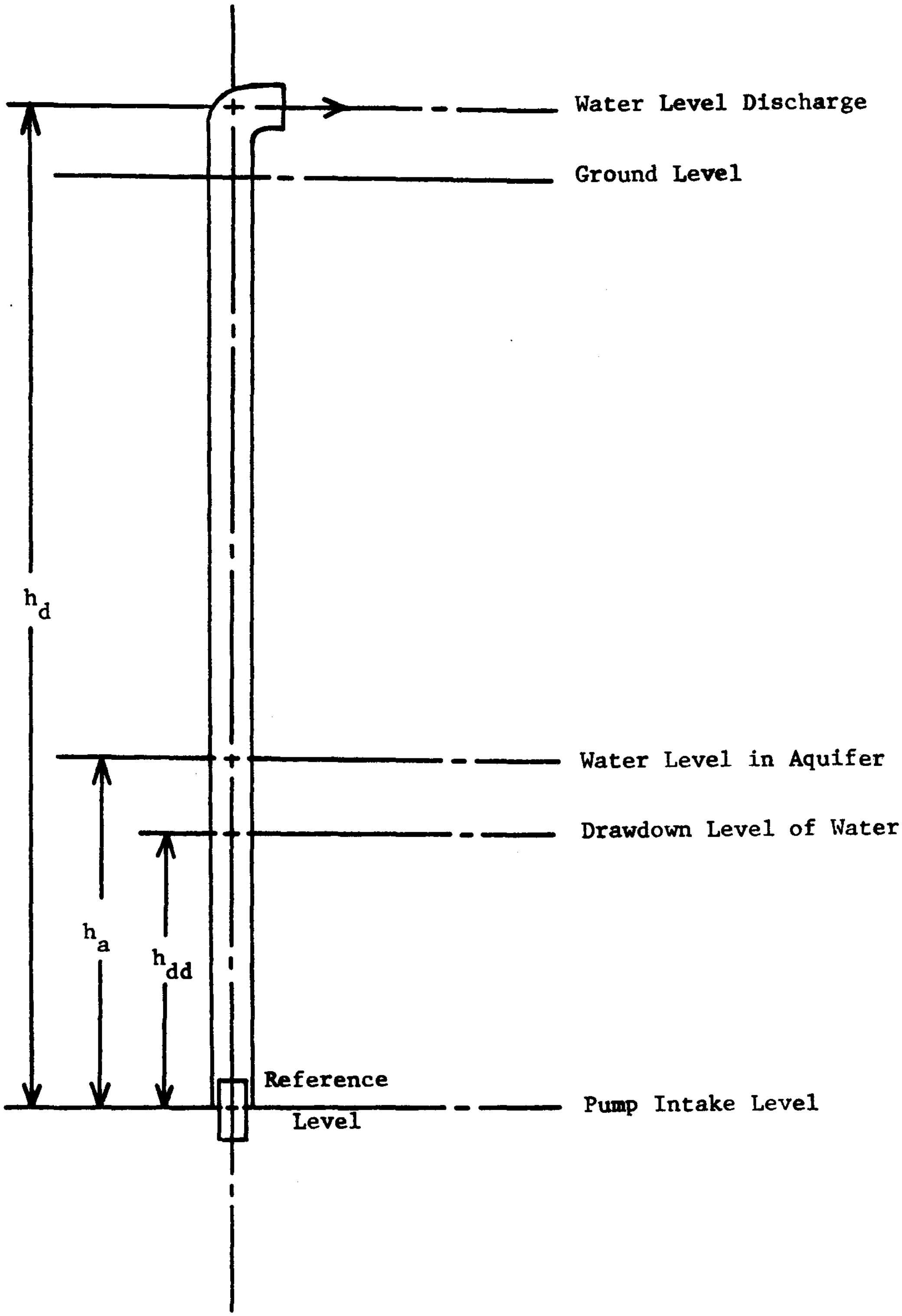


FIGURE 14: REFERENCE FRAME FOR ANALYSIS OF PUMPING SYSTEMS

The positive rotary displacement pump presents a load to the rotor characterized by an angular speed and by a corresponding torque (Eq. 7). Stable operation is achieved when the rotor provides a matching torque and angular speed, whether directly or through a torque (and shaft speed) converter. Corresponding to this combination of τ and ω, the volumetric capacity is given by Eq. (3) and the power used by the system is given by Eq. (1).

In Eq. (7), h_d and h_a are constant, therefore the behavior of h_f and the magnitudes of the coefficients of the ω and ω^2 terms will determine the extend to which τ differs from a constant. By making reasonable assumptions for the piping configuration, estimates can be made for h_f and the last term in the brackets of Eq. (7). The term $k_2\varepsilon_v\omega/s$ represents the well drawdown and can be known only by test pumping the well. We assume here that well drawdown varies linearly with ω and is 20 ft for 200 gpm. From tables in Hicks and Edwards (16, p. 192), a 4-inch pipe is seen to be the economic choice for size. In the simple pumping system to be considered, 250 ft of pipe with two elbows would make up the delivery system. A straightforward evaluation of H in Eq. (7) gives a difference of only 10 to 15% from zero to 200 gpm delivery. In the vast majority of cases, the positive displacement rotary pumping system would provide an essentially constant torque load, and power consumed is nearly proportional to shaft rpm.

We can now examine the nature of the matching of this pumping system with wind rotors. Torque-rpm curves are presented for a low solidity (0.076) rotor in Figure 5. Constant torque lines are horizontal lines in this figure for which line C provides an example. Two fundamental difficulties are immediately apparent: (1) if a low starting torque is chosen, excessive rotor speeds must be attained to deliver adequate power at the anticipated level, and (2) the total system operates far from the curve for optimum efficiency. Conversely, the choice of a higher rotor torque raises the minimum wind speed required to operate the system so that the energy at lower wind speeds cannot contribute to the energy captured.

An examination of Figure 9 reveals that the rotor overspeed problem can be reduced by use of a rotor with high solidity, although the operating lines, as before, differ greatly from the curve giving optimum efficiency. A large rotor of high solidity approximates a scaled-up farm windmill.

Systems made up of smaller delivery pipes will obviously dissipate more energy as friction. A system with 2-inch pipe not only presents considerably more friction to be overcome at identical flow rates, but an h_f term more strongly dependent upon ω^2, because

$$h_f \propto L\omega^2/D^5,$$

in which L is the total length of pipe in the system, and D is the diameter of the pipe. In an irrigation system that must make use of an extensive piping system, such as the drip irrigation system, it appears possible to choose the pipe size and configuration as a means of properly loading the wind rotor. At this time, such systems are in extremely limited use, so it appears unjustified to pursue this idea presently. In the principal method under present consideration, where the water is to be immediately discharged into a pond, then transported by ditch to its point of use, it is clearly impractical to attempt to design a piping system that would cause the pump to present a matched load to the wind rotor. An obvious technical solution is to make use of a torque converter whose shaft turn ratio is appropriately variable.

In a positive displacement pump of the reciprocating type, a mechanism that would vary the length of stroke (displacement) could be used to gain better loading characteristics.

Airlift Systems. The airlift pumping technique has been in use for over one hundred years. In terms of components required, it is a remarkably simple system. Compressed air and a footpiece for delivering air at the bottom of the discharge pipe are the principal requirements. The footpiece is often no more than a short piece of pipe directed upward inside the discharge pipe, or it can be a tapered section with a Venturi throat through which large numbers of small holes have been drilled for the purpose of breaking the injected air into large numbers of small bubbles.

Airlift systems were in extensive use at least until the early 1930's. The answer to the question of why their use has steadily declined appears to lie partly in the difficulty of maintaining acceptable efficiencies over a commonly encountered range of operating conditions, partly in the problem of evolving good designs due to the lack of a good operating theory, and partly in the development of the efficient turbine pumps, which are compatible with the available prime movers.

Performance predictions of airlift systems require that wind rotor-air compressor combinations, and compressed air-footpiece combinations be evaluated. In this section, air compressor and airlift pump characteristics are considered. The question of how airlift systems should perform with a wind rotor will be discussed in a later section after the problem of choosing gear box ratios has been treated.

A. Air compressors. For systems designed to operate with wind as their only source of energy, the compressor used must be capable of continuous operation over a wide range of shaft speeds if any acceptable level of efficiency is to be reached. The power level ($\sim$25 hp) and the air pressure required ($\sim$100 psi), impose additional restrictions on the type of compressor that can be selected. In short, when off-the-shelf compressors are considered, the positive-displacement reciprocating piston type air compressor appears to be the only reasonable choice.

The torque of the reciprocating air compressor is a periodic function of the angular displacement of its crankshaft, but for multiple stage units, these torque pulsations are considerably smoothed by the manner in which the compression and expansion strokes are phased. Moreover, manufacturers present their data as brake horsepower versus rpm. The considerable rotational inertia to be expected in a typical windpowered compressed air system should also help smooth the torque. Finally, the rotor model cannot accomodate such short period torque variations, so it will be assumed that the torque required for a given horsepower and shaft speed is constant. The mathematical representation of the positive-displacement reciprocating air compressor thus becomes identical to that for the rotary positive-displacement water pump. Theoretically, Eq. (7) could be adapted and used, but it is more appropriate to use the manufacturer's data directly because it is available. The fundamental technical difficulties of the positive-displacement water pump-wind rotor combination are also present in the reciprocating air compressor-wind rotor combination.

B. Airlift Pumps. The airlift pump is a device within which many physical processes occur simultaneously, most of which involve the mechanics of the two fluids, water and air. Furthermore, the nature of the flow depends upon water and air flow rates, footpiece design and submergence, pipe diameter, etc. In attempts to treat the system theoretically, so many complications arise that several simplifying assumptions are usually made with the result that the theories evolved to date have been only moderately useful for predicting or confirming experimental results. The model selected often limits the application of the theory to limited ranges for the operating parameters. For example, a relatively recent analysis incorporates the more up-to-date techniques for treating two-phase flow, but in the form presented it is applicable only to lifts much less than those we have need to consider (17). Suggestions were given for modifying the equations to permit treatments of systems with higher lifts, but several uncertainties would be involved, and it is doubtful that the time required would be available in the present study. The flow parameters in the accompanying experimental data did not match those to be treated in our problem so we decided to make use of the semiempirical treatment appearing in the report of Ward and Kessler (7). Their work is also backed by an extensive collection of carefully presented data for systems much more like the ones we must consider. The treatment involves a model with slug or plug flow, and it may suffice as well for bubble or areated flow. It is unlikely to be adequate for the more turbulent forms of two-phase flow, however.

The method in Ward and Kessler's report lumps the effects of all factors affecting the dynamic performance into a single term, H'_{1-2}, with the lift and the velocity head excepted. These are cleanly separable even in the theoretical sense. The term H'_{1-2} is appropriately called the head loss and includes fluid friction at the pipe wall, fluid shear effects, air bubble and plug slip with respect to the water, etc. The volume of air under footpiece conditions required to lift a unit weight of water to its discharge point, A_v is given by

$$A_v = (h_L + H'_{1-2} + W_s^2/2g)K_1 \tag{8}$$

where h_L = static lift of pump
= $h_d - h_{dd}$ (on Fig. 14),
$W_s^2/2g$ = velocity head of discharged water,
K_1 = work obtainable from a unit volume of air at footpiece conditions, when allowed to expand reversibly and isothermally to atmospheric pressure.

The British engineering system of units (ft, lb, sec) is used throughout the report, but any consistent system of units can be used.

Ward and Kessler prepared several graphs for the loss of head per foot, H'_f, of eductor pipe as a function of (1) the average value for the ratio of air to water volumes, R_v, (2) the average linear speed of the air-water mixture, W_a, (ft/sec), and (3) diameter, D (inches, converted to feet in velocity calculations).

The procedure for predicting pump performance consists of:

1. Make an initial assumption for H'_{1-2} to insert in Eq. 8. (0.5 to 1.0 ft water/ft eductor pipe length is generally satisfactory.)

2. Calculate A_v (ft^3 air/lb water), ignoring the velocity head contribution.

3. Calculate R_v and W_a (ft/sec) corresponding to the A_v calculated and pipe size chosen, etc.

4. Read a value for H'_f (ft water head/ft eductor pipe length) from the proper graph in the report of Ward and Kessler.

5. Recalculate H'_{1-2} using the value read for H'_f and the eductor pipe length.

6. Repeat steps 2 through 5 until successive values for H'_{1-2} check satisfactorily.

7. Repeat calculations by inserting values for the velocity head into Eq. 8.

We modified this procedure by developing a functional representation for H'_f of the form

$$H'_f = H'_f(R_v, W_a)$$

in a Taylor series up to and including all 4th degree terms (15 parameters). The graph chosen for reading the values of H'_f, R_v and W_a correspond to the experimental series 6, 7 and 10 in Ward and Kessler's report. This graph was selected because the pipe diameter was their largest (3 inches) and the eductor tubes were the longest for which a graph was provided. At step 4 in the procedure described, the computer program evaluated H'_f in a subroutine and in the first iteration included a value for the velocity head. Iterations were terminated when H'_{1-2} values differed by 0.1 percent.

Our computational procedure was checked by applying it to the experimental data in series 6, 7 and 10. The standard deviations of the relative errors of the calculated flow rates were in all cases less than 15 percent. Since some of the experimental data were obviously inconsistent, we judged our modification of the analytic procedure and our computer program to be satisfactory. When applied to the experimental data of series 9, the calculated values were almost all too high (standard deviation of relative error = 31 percent). This series was obtained for the shortest eductor pipe length, 37.85 ft. Ward and Kessler noted that the pipe elbow used had a greater effect on these results than on the longer eductor lengths. When applied to the data of series 8, the calculated values were almost all too low (standard deviation of relative error = 26 percent). Series 8 experiments were carried out using the eductor pipe of greatest length, 100.64 ft. Systematic errors are clearly present, but considering that our program gives low estimates for the delivery rates of airlift pumps with long eductors, we felt its use would definitely not lead to over-optimistic predictions of pump performance.

Performances of airlift pumps having a diameter of 3 inches, a lift of 200 ft, and several depths of submergence for the footpiece were calculated next. The model predicts that pump efficiency increases with increasing percentage of submergence. Percentage submergence, S, is defined by the relation

$$S = 100\ H_s/(h_s + h_L)$$

where h_s = depth pump is submerged below dynamid level of water,

$h_s = h_{dd}$ (Fig. 14).

The results of this series of calculations for the 3 inch eductor pipe are presented in Table 3.

TABLE 3. Calculated Characteristics of Airlift Pump, 3 inch Eductor Pipe 200 Feet Long.

Length of submerged pipe, ft	200	300	400	550
Percentage submerged	50	60	66.7	73.3
Maximum pump efficiency, percent	47.2	52.5	65.6	62.5
Free air flow at maximum pump efficiency, cfm	38.0	36.0	34.0	32.0
Water flow at maximum pump efficiency	40.8	50.6	57.2	66.2

It should be observed in the table that maximum airlift efficiencies compare favorably with those of turbine pumps. The price to be paid for these higher efficiencies, i.e., greatly deepened wells with correspondingly more casing and delivery pipe and a restricted range of air flow rates, definitely reduces the attractiveness of this system.

Performance predictions for a 6 inch airlift pump were then attempted. Graphs of R_v, W_a and H_f' for 6 inch eductor pipes are unavailable in the report, so the procedure followed was to reduce all flow rates for the 3 inch system by a factor of four for purposes of estimating H_f' in the 6 inch pump and piping system. For 200 ft lift and 200 ft submergence, the calculations indicated an optimum efficiency of 47 percent for flow rates of 163 gpm and 152 cfm for water and air, respectively.

Few experimental data are available against which the predictions of this model can be checked; however, S. F. Shaw has published data on a 6 inch airlift pump having a lift of 214 ft and a submergence of 216 ft (18). A maximum efficiency of 62.7 percent occurs at a flow rate of 450 gpm when 302 ft^3/min of air (as free air) is delivered at 90 psig pressure. The maximum flow rate was 588 gpm at an efficiency of 33 percent.

The results calculated from the mathematical model are conservative both from the point of view of the efficiency and the water flow rate. The model may be adequate but the use of the representation for H_f' for the 3 inch eductor pipe may lead to the considerable difference in results, or, of course, the model used successfully for the 3 inch pump may itself fail when applied to the 6 inch pump. Nevertheless, one must assume that the experimental results are the more reliable, especially since the experimental results for several 6 inch airlift pumps were similar. It thus appears that 200 gpm can be delivered by a 5 inch and possibly a 4 inch airlift pump of the same lift (200 ft) and submergence (200 ft), and at an average efficiency of about 50 percent in a properly designed system.

The foregoing conclusions were checked with empirical formulas given in standard engineering handbooks. From Mark's (19), an eductor diameter of from 4 to 5 inches is indicated. From the Chemical Engineer's Handbook (20), a flow rate of 158 ft^3/min of free air would pump 200 gpm of water, and the formula can be combined with formulas for isothermal reversible work from compressed air to predict an efficiency of 56 percent. The expression obtainable for efficiency does not contain the submergence or pipe diameter, so one cannot deduce how efficiency should change with submergence (or submergence ratio) despite a statement recommending a submergence as low as 41 percent for a lift of 550 ft.

MATCHING THE TORQUE-RPM CHARACTERISTICS OF LOADS TO THOSE OF WIND ROTORS

The power (torque x rpm) from a wind rotor is generally transferred by a gear box to a second machine. Therefore the power transfer characteristic is important because it directly affects the overall efficiency of power extraction from the wind. The mechanical system utilized to link the torque-rpm combinations of the rotor to those of the load must, therefore, be given careful consideration if best all-round performance is to be had.

It is assumed that the torque of a load can be represented as a function of rpm in the form

$$\tau_2 = a_2(RPM_2)^b \tag{9}$$

where subscript 1 refers to the wind rotor, subscript 2 refers to the load, a and b are constants (RPM is related to angular speed by RPM = $\omega/2\pi$). Thus if b = 0, the load presents a constant torque to the rotor; if b = 1, the torque increases linearly with rpm (a > 0). By allowing nonintegral values for b, a wide range of torque-rpm characteristics can be represented. Positive displacement pumps and air compressors have torques which are almost independent of rotational speeds, so they can be represented by Eq. (9) with relatively small positive values for b. The form of Eq. (9) permits the mathematical problem to be solved easily.

<u>The technique of correctly choosing a fixed-turn-ratio gear box.</u> The power coefficient of WT5 is maximized with respect to tip speed ratio X, and all the twist angles along the blades, with the pitch angle held at zero degrees. With the above determined twist angles held constant, the torque is maximized with respect to variations in the tip speed ratio, and X_m is the tip speed ratio and Cpm is the power coefficient corresponding to the maximum torque, τ_{1m}.

Starting at a tip speed ratio of X_m, the power coefficient is determined as a function of X, at constant wind speed, up to a value just before the power becomes negative. The table of Cp values versus X values is independent of the wind speed. The reduced quantities are calculated by dividing each Cp value by Cpm, the first value in the table, and each X by X_m, also the first value in the table.

The problem now is to determine pairs of V and X, (V is the wind speed) that will give wind rotor operating points matching those determined by the load. In solving this problem, it is necessary to postulate that between the windmill rotor and the load is a gear box, having fixed RPM change factor of R. This gear box is necessary because the permissible rotational speeds of currently available constant torque machinery will seldom if ever correspond to wind rotor rotational speeds. R is defined by

(10) $$R = RPM_2/RPM_1.$$

Conservation of energy, through the gear box, requires

$$\tau_1 RPM_1 = \tau_2 RPM_2,$$

if no frictional losses occur. To allow for frictional losses, one can write

(11) $$\tau_1 RPM_1 \varepsilon = \tau_2 RPM_2,$$

where ε is the efficiency of the gear box, which is assumed to be constant.

From Eq. (11), we have

(12) $$\tau_1 = \tau_2 RPM_2/(RPM_1 \varepsilon).$$

If Eqs. (1) and (10) are substituted in (12),

(13) $$\tau_1 = a_2 (RPM_2)^b R/\varepsilon.$$

Substituting for RPM_2 from equation (10), we have

(14) $$\tau_1 = a_2 R^{(1+b)} (RPM_1)^b/\varepsilon.$$

We see from Eq. (14) that if the load satisfies Eq. (9), then the output of the wind rotor has the same functional form, namely

(15) $$\tau_1 = a_1 (RPM_1)^b$$

where $$a_1 = a_2 R^{(1+b)}/\varepsilon,$$

and it follows

(16) $$R = [\varepsilon a_1/a_2]^{1/(1+b)}.$$

Eq. (16) will be used to calculate the RPM ratio of the gear box.

The power output of the wind rotor is proportional to the product of the torque and the RPM. It is also proportional to the product of the power coefficient and the cube of the wind speed. Therefore

(17) $$\tau_1 RPM_1 = K_1 CpV^3,$$

where K_1 is a constant. Substituting for τ_1 from Eq. (15), and since RPM is proportional to X and V,

$$RPM_1 = K_2XV,$$

we have

$$a_1K_2^{1+b}\ X^{1+b}\ V^{1+b} = K_1CpV^3$$

which may be written

$$V^{2-b} = K_3X^{1+b}/Cp \tag{18}$$

where K_3 is a new constant. Since Cp is known as a function of X, we may use Eq. (18) to solve for V as a function of X. We choose a value of V, that will be an operating condition for which $X = X_m$ and thus Cp = Cpm.

$$V_m^{2-b} = K_3X_m^{1+b}/Cpm \tag{19}$$

Then the ratio is

$$\left(\frac{V}{V_m}\right)^{2-b} = \left(\frac{X}{X_m}\right)^{1+b} \frac{Cpm}{Cp}\ . \tag{20}$$

The parameters b and a_2 are determined from the characteristics of the load and having chosen a value of V_m, Eq. (20) is used to calculate V as a function of X. These pairs of values of V and X can be read into the modified PROP program of Wilson and Lissaman, to give the matching operating parameters of the windmill, RPM_1 and τ_1. Then the parameters a_1 and R are calculated from Eqs. (15) and (16), respectively.

Matching for optimum power transfer from wind to the load. The power delivered from a rotor when the wind speed is V is given in Eq. (1). It was pointed out earlier that the optimized PROP program predicts the power coefficient for a given rotor to be independent of wind speed but dependent on X, the tip to wind speed ratio. If we designate by by X_o the value of X giving the optimum value for Cp, then the maximum power, P_{1m}, available from the rotor at ω_1 can be written

$$P_{1m} = K\ Cp(\omega_1r/X)^3, \tag{21}$$

where r is the radius of the rotor. P_{1m} can also be written in the form

$$P_{1m} = C_1\omega_1^3 \tag{22}$$

because all other terms in Eq. (21) are independent of wind speed which is related directly to ω_1 at constant X_o.

We again assume the load to be representable in the form of Eq. (9), so it follows that the power required to drive the load is

$$P_2 = a_2\omega_2^{b+1}.$$

As noted earlier, the law of conservation of energy requires that

$$C_1\omega_1^3 = a_2\omega_2^{b+1}. \tag{23}$$

If the load is linked to the rotor through a constant turn ratio gear box, Eqs. (10) and (22) can be combined to give

$$C_1\omega_1^3\ \varepsilon = a_2R^{b+1}\omega_1^{b+1}. \tag{24}$$

Since all factors other than the ω_1 values in Eq. (24) are independent of ω, $b = 2$. In order for the load to extract maximum power available from the rotor over a range of wind speeds, the load must have the $\tau - \omega$ characteristic

$$\tau_2 = a_2\omega_2^2. \tag{25}$$

This result can be anticipated readily for the case in which the load is directly driven by the wind rotor, but the foregoing demonstrates that the form of the $\tau - \omega$ relation must remain the same when gear boxes are employed.

Only a few available machines can present the ideal load characteristic of Eq. (25) to a wind rotor. The d.c. generator and/or alternator with properly regulated field currents, and the common turbine pump are among these few, but the latter delivers water at a rate proportional to shaft speed, rather than to the cube of the shaft speed. The remaining energy appears as pump head. It is this behavior that will lead in most applications to serious degradation of efficiency when a turbine pump is powered by a wind rotor. It has already been pointed out that positive displacement pumps and air compressors are essentially constant torque devices. A variable displacement air compressor is currently being developed which should give a close approximation to ideal loading (21).

<u>Variable ratio torque converter</u>. Since available air compressors and positive displacement pumps offer poor load matching characteristics to wind rotors, the question of whether some torque converter is capable of improving the power transfer efficiency has to be considered. It is also necessary to know whether a given machine is capable of operating under the conditions imposed by the torque converter. We now deduce the characteristics of a torque converter which would connect machines requiring constant torque drive with wind rotors in such a way as to achieve maximum transfer of power from the wind to the machine.

Power transfer in any kind of torque converter must satisfy Eq. (11). Eq. (22) shows the relation between power and ω for maximum Cp, thus, if all the power available from the rotor at a given ω (decreased by the mechanical efficiency factor) is transferred to a constant torque device, one can write

$$\varepsilon C_1 \omega_1^3 = \tau_2 \omega_2 \tag{26}$$

in which τ_2 is constant. From Eq. (26), it immediately follows that the required turn ratio is

$$\omega_2/\omega_1 = (C_1\varepsilon/\tau_2)\omega_2^2$$

or

$$\omega_2/\omega_1 = C_2\omega_2^2. \tag{27}$$

The use of an additional fixed-ratio gear box would not affect the form of this relation.

The range of rotor shaft speeds will determine the turn-ratio limits. For maximum power extraction, rotor speed is proportional to wind speed. Most of the energy in the wind at Amarillo at 40 ft height lies in the wind speed range from 12 mph to 36 mph (see Fig. 2); thus the turn-ratio should vary from 1:1 at 12 mph to 1:9 at 36 mph. Since the rotor shaft speed at 36 mph is three times its value at 12 mph, the output shaft speed will vary by a factor of 27, moreover, the power handling capabilities of the torque converter during 36 mph winds would need to be 27 times what it is at 12 mph. Such considerations generally apply at every site, although differences in wind regimes will change the wind speed range involved.

None of the conventional positive displacement air compressors, piston or screw type, can be operated over such a wide range of shaft speeds. The maximum ratio of permissible operating speeds according to several manufacturers is at most a little greater than 3. A positive displacement rotary water pump of the required capacity for use in wells has not yet been built, so nothing is known concerning the permissible range of operating speeds for such pumps, yet it appears that this type of pump offers the only possibility for a device that might be successfully coupled through variable torque converters satisfying the condition of Eq. (27).

Three developmental groups in the US are working on mechanically coupled variable ratio torque converters. Two of these (22, 23) intend to produce automotive transmissions but work is still in the developmental stage. A third group is working on transmissions for garden tractors, etc. (24). At least two European groups are involved with such transmissions but our attempts to contact them have thus far been unsuccessful.

Hydraulic torque connectors with continuously variable ratios are manufactured by Sundstrand (25). These devices can apparently provide the rate of speeds and torques required, especially if some fixed turn-ratio units are also used. However, the overall mechanical efficiency of the unit was stated to be in the range of 60 to 70 percent. It is easy to see that the additional energy captured with a more ideally matched rotor-pump system may be lost as heat in the torque converter train. Systems involving a minimum of energy conversions and transmissions are usually more efficient.

SYSTEM PERFORMANCE

Two two types of systems, (1) wind rotor WT5, drive train, rotary positive displacement pump (Fig. 15) and (2) wind rotor WT5, drive train, air compressor, air lift pump (Fig. 16), were modeled for two years (average wind energy 1966, low wind energy 1969). The model is based on the energy available from WT5 and the water budget as determined from crop demand. The data (for Amarillo and Lubbock) available from the National Weather Service are the air pressure, temperature, and wind speed for every 3 hours and the amount of water pumped to the surface is calculated by choosing a mode of operation for WT5, and by estimating the efficiencies of the drive train and the rotary pump or the compressor and air lift pump.

Rotor-Pump Efficiencies. Even though no rotary pump of adequate performance is now commercially available, system (1) was included because such pumps could be developed. The efficiencies of WT5 are a function of the wind speed and the mode of operation is constant torque (Table 4). The cut-in wind speed determines the operating range and the rated wind speed because WT5 is limited to around 200 rpm. The three cut-in wind speeds, 12, 16, and 20 mph, were used. The corresponding possible annual efficiencies for the rotor were 28, 33, and 31%. These efficiencies are [(energy out/energy in the wind over the area of the rotor) for the wind distribution in Fig. 1] with no limitations on power or rpm of WT5. The cut-in wind speed of 16 mph (example in Table 4) then determines the rated wind speed, 22 mph, and for the computer program on system performance the power output was assumed constant for all wind speeds above the rated wind speed.

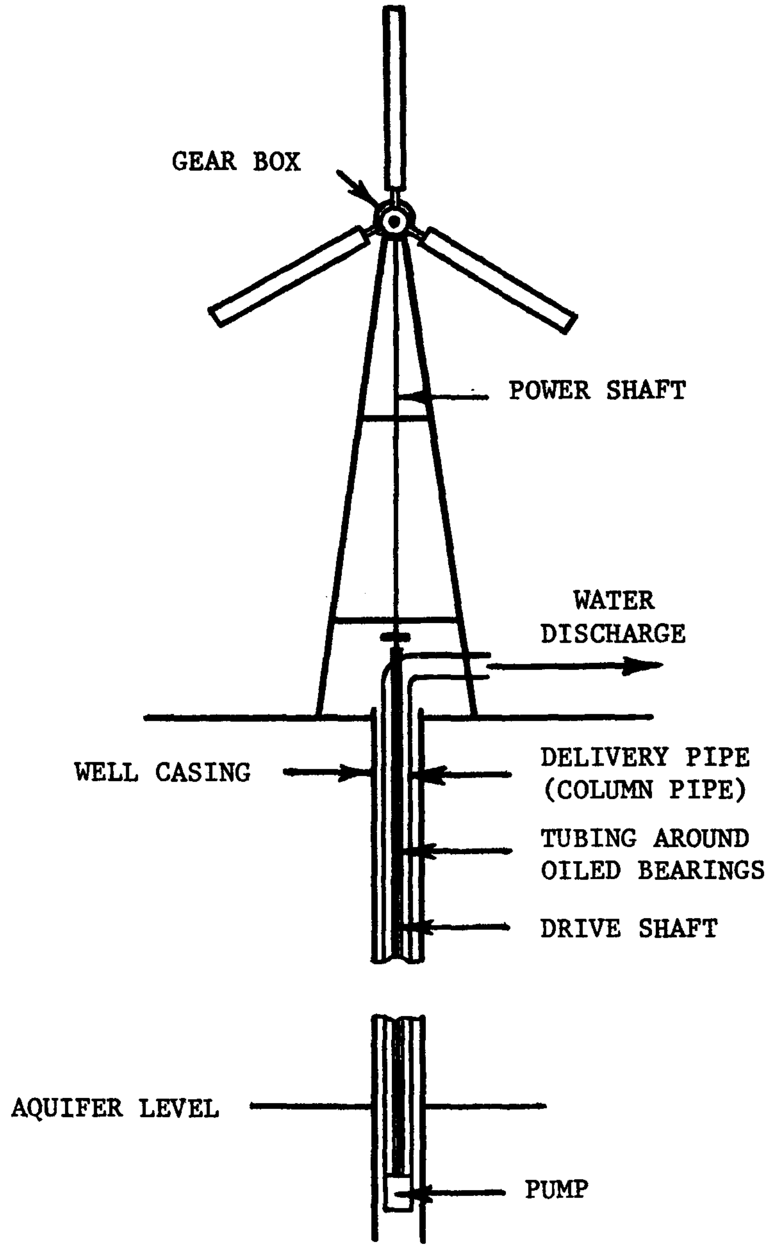


FIGURE 15. ROTARY POSITIVE DISPLACEMENT PUMPING SYSTEM. ROTOR DIAMETER = 33 FT, TOWER HEIGHT = 40 FT.

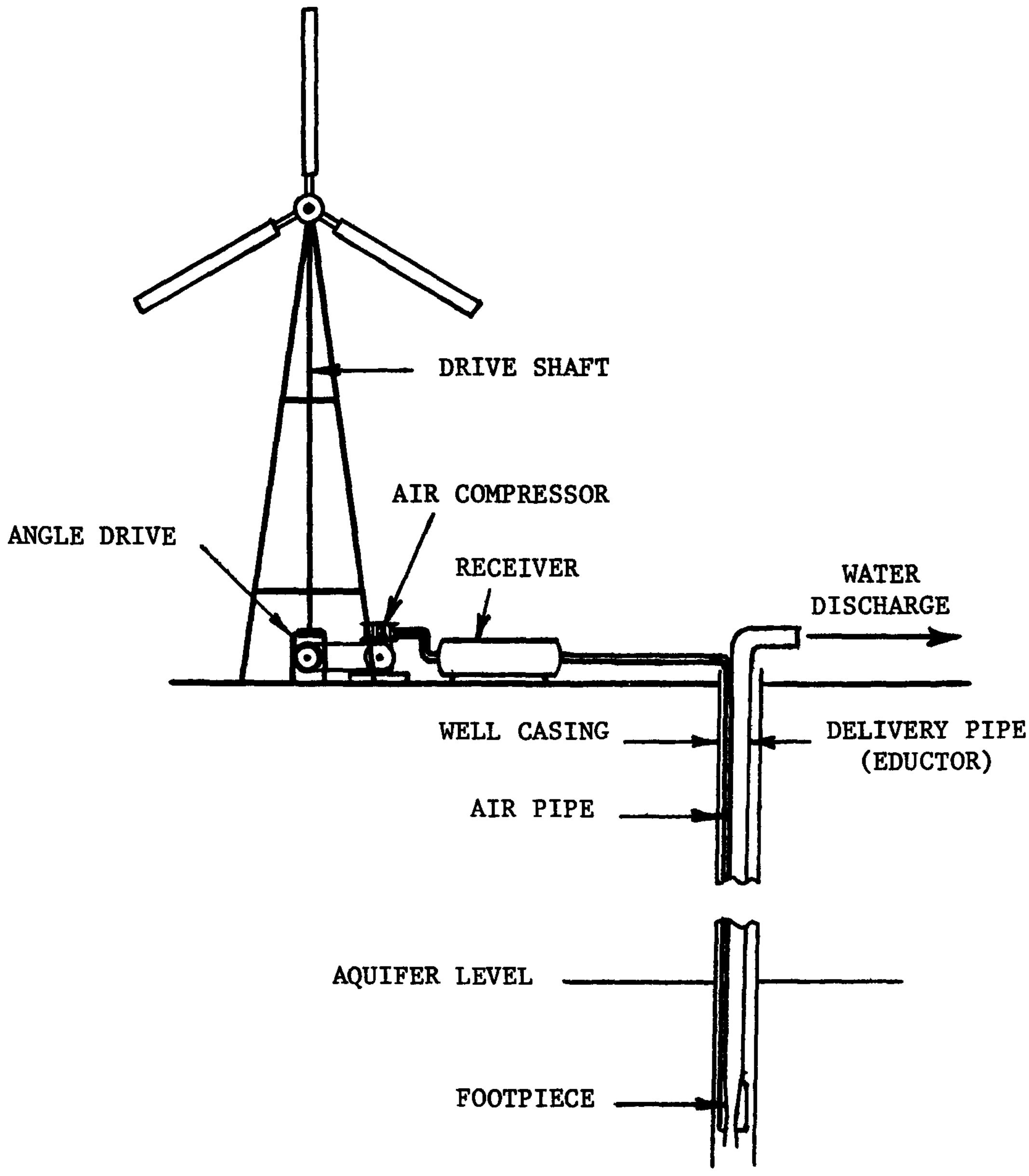


FIGURE 16: AIRLIFT PUMPING SYSTEM. ROTOR DIAMETER = 33 FT,
TOWER HEIGHT = 40 FT.

TABLE 4. Efficiencies of WT5 for constant torque operation, calculated by program BOB.F4.

TIP SPEED RATIO	RPM	KILOWATTS POWER	TORQUE	POWER COEFF	THRUST LBS	VELOCITY MPH
6.44	88.03	7.93	634.78	0.49	385.99	16.0
7.32	106.30	9.58	634.78	0.50	447.61	17.0
8.13	124.91	11.26	634.78	0.49	508.07	18.0
8.88	144.15	12.99	634.78	0.48	569.35	19.0
9.59	163.75	14.76	634.78	0.47	631.68	20.0
10.22	183.41	16.52	634.77	0.45	695.30	21.0
10.80	202.88	18.28	634.78	0.44	760.44	22.0
11.31	222.08	20.01	634.78	0.42	827.34	23.0
11.76	240.96	21.72	634.78	0.40	896.18	24.0
12.16	259.5	23.39	634.78	0.38	967.10	25.0
...						
15.59	665.55	59.98	634.81	0.12	3563.14	50.0

Average power generated over a year = 7.765 KW

Average power in the wind for a 16.4 ft radius windmill = 23.72 KW

Efficiency = .327

The rotor operation for the air lift system had a cut-in wind speed of 20 mph. This was chosen to match the $\tau - \omega$ characteristics of a commercially available air compressor. The torque increased somewhat with rpm (b = .176) as determined from Eqs. (9 - 20) and the efficiency of WT5 as a function of wind speed was calculated for this operation. For these same sets of wind speeds (20-50 mph) the efficiencies of the compressor and the air lift system were also calculated, assuming a submergence of 50% (200 ft lift, 200 ft submergence). The combined efficiency of the compressor and the air lift pump is .236 at 20 mph, .211 at 30 mph, and .209 at 50 mph. When these efficiencies are combined with the rotor efficiency the maximum possible efficiency is .11 at 23 mph, which leaves much room for improvement.

Water Budget. There has been extensive work on evapotranspiration for different areas, crops, and with irrigation (26-28). A number of different formulas can be used to predict evapotranspiration but in general it is closely correlated with the amount of solar radiation. Since the solar radiation was unavailable on a daily basis, the crop demand is based on the following variations: constant demand, variable demand for wheat or sorghum which was determined experimentally for the Southern High Plains (29, 30), and a variable demand which combined wheat and sorghum. When a variable demand was used, a 5 inch preplant irrigation from September 15-30 was assumed for wheat and a 5 inch preplant irrigation from March 1 to 15 was assumed for sorghum (Fig. 17). The total water demand (in/acre) is 32.2 inches for wheat and 28.9 inches for sorghum, and then 5 inch for the preplant irrigation which gives 37.2 and 33.9 inches, respectively. The total combined demand is 71.1 inches which is almost the same as the total constant demand, 73 inches. This is from an average daily demand of .2 inch.

Variations include rainfall and a reservoir for the water, if there is an excess above what is required for soil saturation. The following assumptions were included: at the start of each year the water level of the soil was at one half saturation, and the reservoir level was zero. Evaporative loss and seepage was taken to be .2 in/day from a reservoir assumed to have a surface area of one acre.

Time Correlation for Wind Energy and Water Level. Since the efficiencies of the rotor and pump are known as a function of wind speed, the amount of water pumped can be calculated and then applied to a field. For the different parameters of rotor (height, area, efficiencies, cut-in and rated wind speed), drive train efficiency, pump efficiency, depth to water, size and soil saturation for irrigated plot, rain, reservoir size, and crop demand; the computer program gives a daily summary of energy, water pumped, average gallons/min, and the water level in the field. To find out the possible energy available as water pumped the annual efficiency was calculated even though excess water (above soil saturation) will be pumped part of the year in some cases.

A reservoir is primarily useful for regulation of flow, because all the water in the reservoir (4 acre-ft) would evaporate if an attempt were made to store it from spring to late summer, and a large reservoir would be too expensive.

Even though the soil is saturated at the end of the year, there is the possibility of a deficit during the year because of the variability of supply and demand. The daily values of water level (Fig. 18, 19) are for a rotary pump (eff = 80%), a reservoir of 4 acre-ft, a plot of

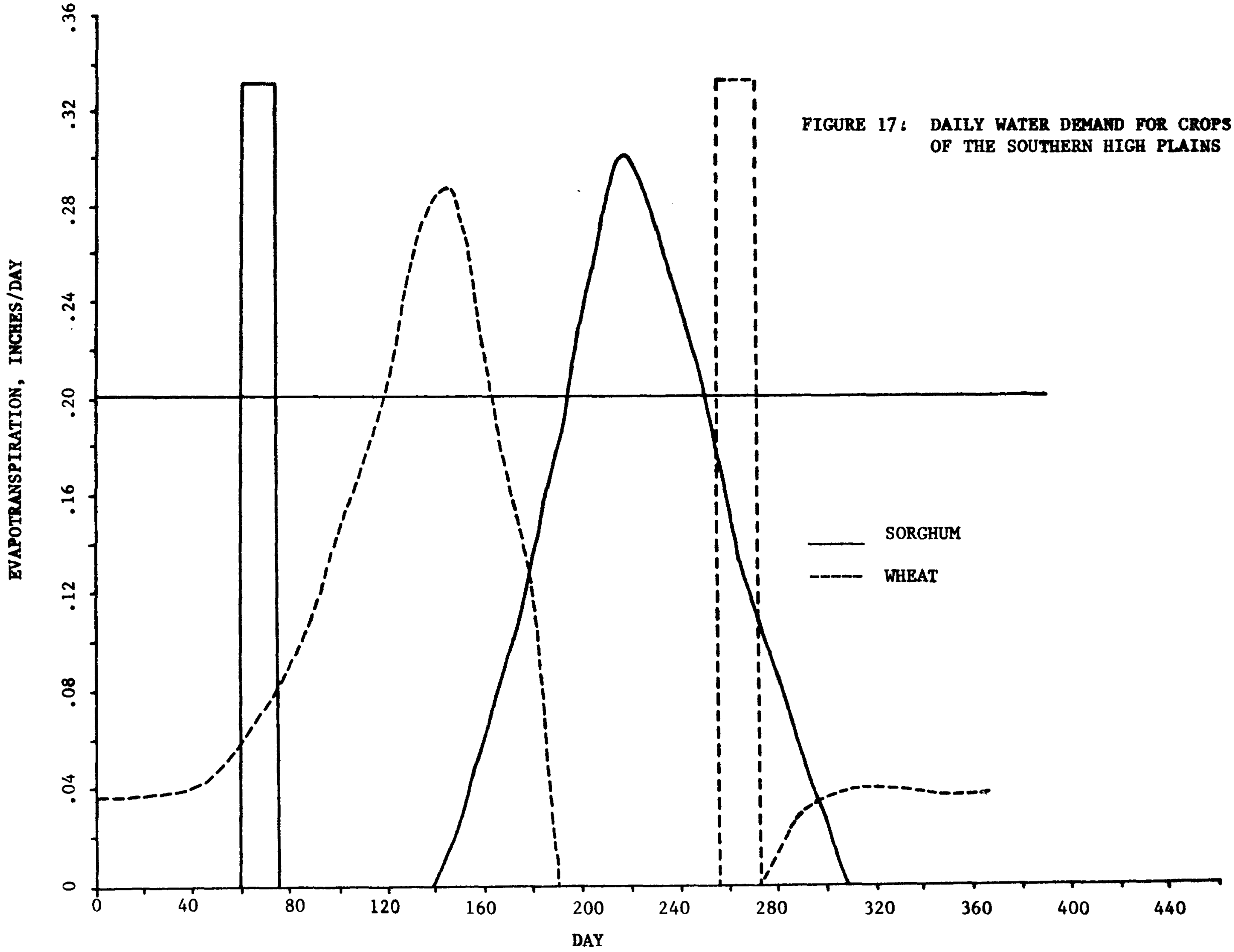


FIGURE 17: DAILY WATER DEMAND FOR CROPS OF THE SOUTHERN HIGH PLAINS

FIGURE 18: DAILY SOIL WATER LEVEL FOR AMARILLO, TEXAS.
SOIL SATURATION = 4 IN.

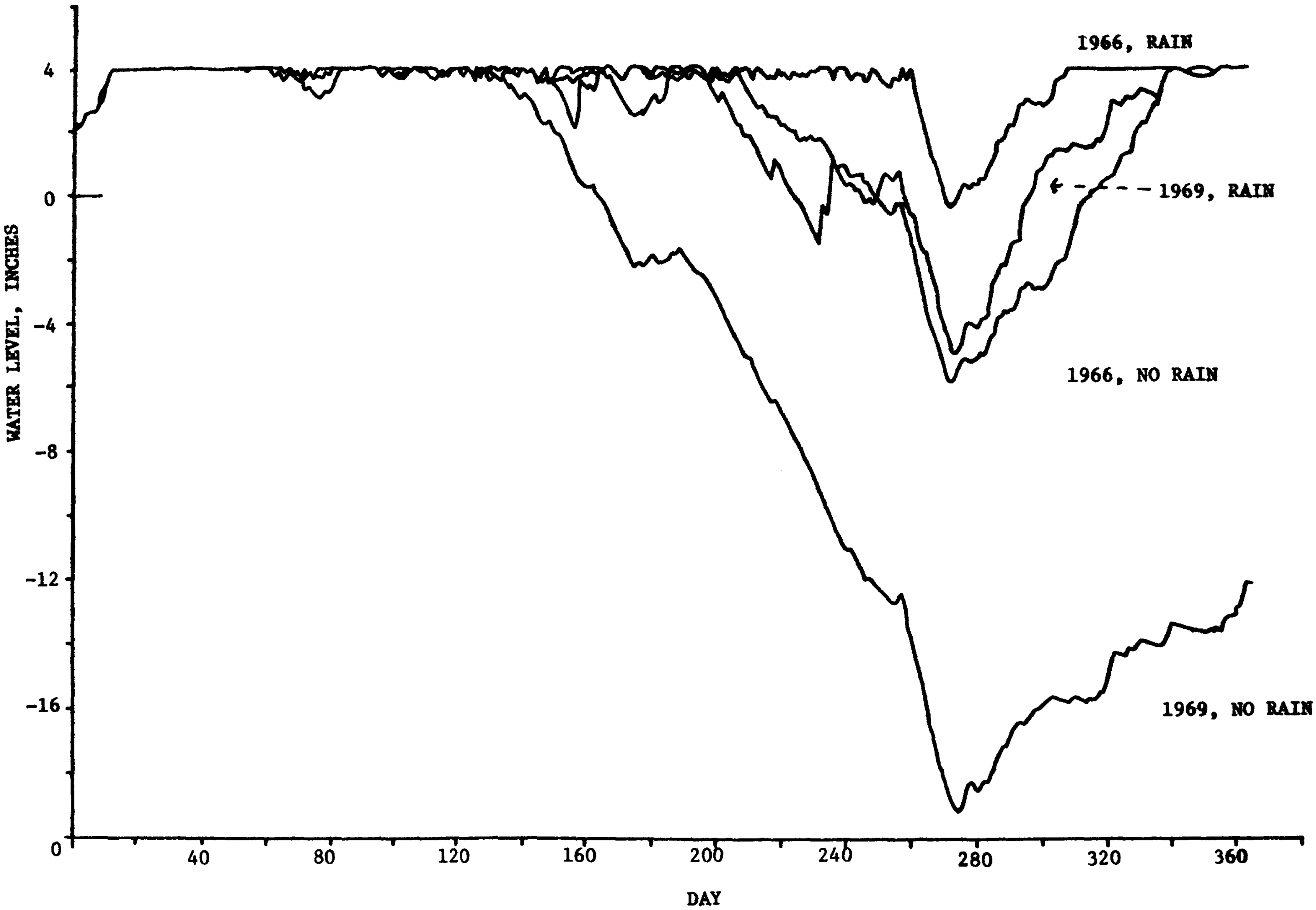
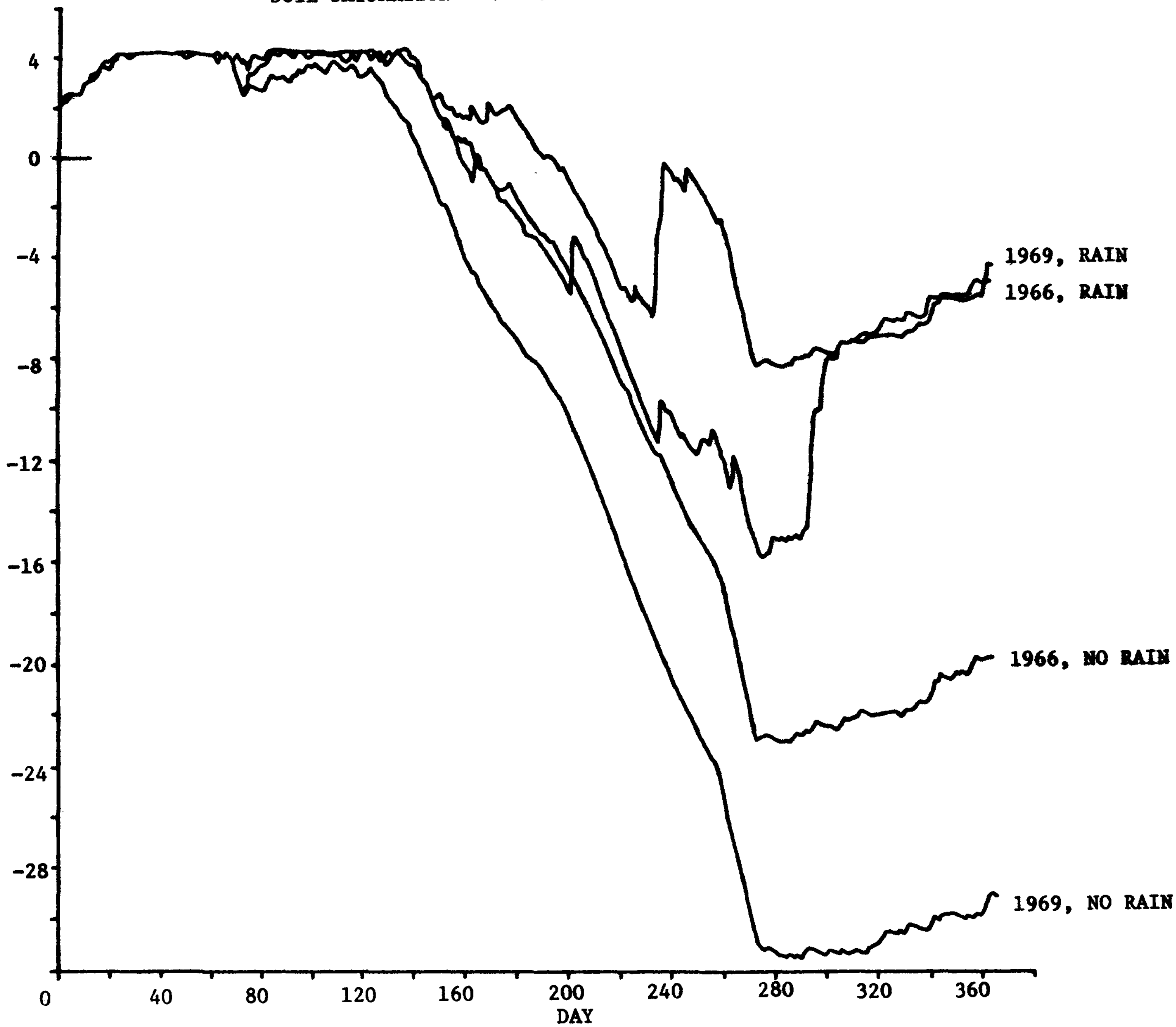


FIGURE 19: DAILY SOIL WATER LEVEL FOR LUBBOCK, TEXAS.
SOIL SATURATION = 4 IN.



40 acres, and the combined crop demand. The worst case is for no rain, low wind energy, and high crop demand (Lubbock 1969). The minimum water levels occurred in October for both Amarillo and Lubbock and is traceable to the preplant irrigation for wheat and the tail end of the demand for sorghum. There is less wind energy available at Lubbock, but for both areas there is sufficient energy through June and if rain is utilized and if the preplant irrigation for wheat could be spread out over a longer time period there would be sufficient water for both sorghum and wheat in the Amarillo area. The possibilities for the Lubbock area are a different demand (cotton, sunflowers, soybeans) and/or a system with a larger rotor.

Other combinations of rain, reservoir, crop demand, and pump efficiencies were tried and some of these are summarized in Table 5. The first three for constant demand indicate that the maximum efficiency 24.5% occurred for the cut-in wind speed of 16 mph. All systems operated up to 50 mph. System 1 had a pump efficiency of 80% and the overall efficiency (water pumped) would be smaller for a less efficiency pump; for example a 60% pump efficiency means the GPM and EFF would be multiplied by 3/4. The airlift system was very inefficient, 5.8%, therefore the efficiency and operating range of the air compressor have to be improved. Another comment about the total system performance, the cut-in wind speed of 16 mph should probably be changed for Lubbock since the wind regime is different, a 20.7% versus 24.5% efficiency for Amarillo.

If only one crop on 40 acres is to be irrigated, system 1 will provide enough water for the Lubbock area. There is also the possibility of limited watering (31-33), which would reduce the demand. In any case, the water demand for wheat coincides with the times of higher wind energy (spring), so an area larger than 40 acres could be irrigated with such a system in the Northern Panhandle region.

<u>Advantages and disadvantages of the two systems</u>. A primary objective of this feasibility study has been to deduce the operating characteristics of windpowered (irrigation scale) water pumping systems. Their performance is dictated by the constant torque restriction of the pumps and as expected, both systems have relative advantages and disadvantages.

A. Rotary positive displacement pump systems. The reciprocating pump type is beset by nearly unsurmountable problems, so the effort was concentrated on the rotary type. Because no rotary positive displacement pump of the required pumping capacity is currently manufactured, we have had to predict performance by use of the best information we could accumulate. In general, this system would have the following advantages:

1. It will pump about four times the quantity of water as an airlift system (same rotor) that can be assembled from available components. Our calculations show that about 25% of the annual energy in the wind would be converted into hydraulic work and appear in the water lifted.

TABLE 5. System Performance: Soil Storage = 4 in, Area = 40 acres, Drive Train Efficiency = 95%

SYSTEM	WINDSPEED cut-in, rated	RAIN in.	CROP DEMAND	DAY 365			
				ANNUAL AVERAGE GPM	ANNUAL EFF %	EXCESS WATER, in.	SOIL WATER LEVEL, in.
AMARILLO 1966							
1 (rotary)	12-19	None	Constant	130	20.4	0	-8.2
1	20-25	None	Constant	138	21.7	2.8	-7.0
1	16-22	None	Constant	156	24.5	5.4	-1.2
1*	16-22	None	Combined	156	24.5	9.9	4.0
1*	16-22	14.9	Combined	156	24.5	24.4	4.0
2 (airlift)	20-50	14.9	Combined	37	5.8	0	-26
AMARILLO 1969							
1*	16-22	None	Combined	117	24.4	8.9	-12
1*	16-22	22.5	Combined	117	24.4	14	3.9
2	20-50	22.5	Combined	23	4.9	0	-25
LUBBOCK 1966							
1†	16-22	None	Combined	88	20.7	0	-20
1†	16-22	18.4	Combined	88	20.7	0	-4.9
2	20-50	18.4	Combined	26	6.1	0	-28
LUBBOCK 1969							
1†	16-22	None	Combined	66	21.3	0	-29
1†	16-22	29.2	Combined	66	21.3	0	-4.5
2	20-50	29.2	Combined	16	5.1	0	-23

* Daily water levels graphed in Fig. 16

† Daily water levels graphed in Fig. 17

2. The system would be no more complicated, at least in principle, than the turbine pump system now in extensive use.

3. With the WT5 horizontal axis rotor considered, the system would be self orienting and self starting. Only infrequently would the unit have to start against the full water lift. The water would leak back through this kind of pump whenever the rotor is not turning. This behavior is in marked contrast to what happens in systems that make use of check valves (reciprocating positive displacement systems).

Its disadvantages are seen to be the following:

1. A refitting of existing wells with the new type of pump would be necessary. Because the evidence indicates such a pump would operate at a considerably slower rotational speed, new and larger drive shafting and new bearings would be required to accomodate the greater torque involved.

2. The windpower system would need to be located in the immediate vicinity of the well, which in many instances could seriously interfere with the agricultural operations.

3. This windpower system would produce nothing of value during the season when irrigation is not needed. Additional investment in a generator could change this, but this additional equipment would not be used in the irrigation season.

4. It would be difficult to couple the outputs from two or more systems should higher power levels be required. (The only practical solution would then be to acquire larger machines, but there may be practical upper limits to rotor sizes, etc.)

B. Airlift pumping systems. An airlift system could be assembled from components now available. The following is a list of favorable characteristics of compressed air-airlift pumping systems:

1. Airlift systems are inherently simple particularly with regard to what must be installed in the well itself: casing, column pipe, air delivery tube, and footpiece. Rotating shaft, bearings, turbine pump, and supporting structures are not required -- no moving mechanical parts are in the well.

2. With the WT5 horizontal axis rotor, the system would be self orienting and self starting. The air compressor cylinders can easily be unloaded for no load rotor starting.

3. Location of the windpower unit is not critical. Compressed air can be conveyed by pipe to considerable distances without serious pressure losses.

4. Power from two or more units can be readily coupled for applications requiring increased power levels. Power systems could be developed on a modular plan.

5. In some locations, underground formations exist in which compressed air might be economically stored.

Recognized disadvantages are:

1. Airlift pumping systems that can be assembled from presently available components are capable of converting only about 6 percent of energy available in the wind to hydraulic work. (It is estimated that this figure could approach 20% with properly designed air compressors.)

2. Wells must be drilled and cased to at least twice the depth to the aquifer level if the best airlift efficiencies are to be had. (This prospect makes conversion to airlift unattractive for those wells already drilled and cased).

The foregoing information points to the rotary positive displacement system as more feasible (if one were available) on the strength of its greater efficiency. On the other hand, the airlift system should have more general applicability where low lifts (< 100 ft) are involved, particularly if a properly designed air compressor could be developed.

COST-BENEFIT ANALYSIS

The preceeding sections indicate that wind power can be used to pump irrigation water in the High Plains. Whether such units will eventually be used depends on their economic feasibility which in turn depends on fossil fuel cost and availability, capital cost of the wind unit, revenue from the sale of food and fibre, the tax rate (subsidizes) for those using wind energy, and the convenience of water on demand with an electric motor or natural gas engine.

The price for a prototype system and the price per unit in a production run of 500 units (Table 6) were estimated for us by Sigma Engineering (34), a mechanical engineering and consulting firm. Even though the airlift system that can be assembled from available components is too inefficient, the prices were included for future reference when a better compressor is developed. Both systems require the same wind unit, WT5 (rotor, gear box, drive train, 40 ft tower), which delivers mechanical power to ground level (Figs. 15, 16).

Both the wind system and the conventional well installation (electric motor or natural gas engine connected to a turbine pump) will have some common costs; a well drilled and cased (12" for 200 gpm, 16" for larger wells) to the red bed (200 to 500 ft) and for the rotary pump system the

TABLE 6. Price Estimates for the Rotary and Airlift Systems. Prices Include Installation Costs.

	Prototype	500 Units
WT5:		
Rotor (32' dia, pitch control)	\$ 9805	\$ 4560
Tower (40' ht)	7145	4450
Design & Tooling	17,175	1150
	\$34,175	\$10,160
Reservoir:	1200	1200
Rotary Pump System:		
Discharge head	\$ 550	\$ 450
1 1/4" oil lubricated drive shaft, 250' @ \$7.06/ft	1765	1560
Installation of pump, shaft, 5" column pipe, \$15/ft	3750	3750
KROV water pump. As this pump has not been built, this is an estimated cost.	23,000	1875
	\$29,065	\$7635
WT5, Reservoir, & Rotary Pump System	\$64,440	\$18,995
Airlift Pump System:		
PVC is suitable for delivery pipe		
6" PVC, 400 ft @ \$2.29/ft	916	780
1" PVC, delivery pipe, 400 ft	62	48
Discharge Head	250	175
Footpiece	75	45
Angle drive	340	250
Curtis Air Compressor C-100 45 hp, stand, air tank	4942	4157
Clutch - control	418	355
Installation	350	350
	\$7353	\$6160
WT5, Reservoir, & Airlift Pump System	\$42,728	\$17,490

drive shaft, column pipe, etc. An estimated cost for a comparable size turbine pump installation (200 gpm, 200 ft lift) is around $8,000, and the difference between that cost and the wind irrigation system is the price of the wind unit, WT5. How long would it take to save $11,000 in fuel costs and as the cost of fuel increases and availability decreases, what would be the value of a wind unit?

The fuel cost for natural gas ($1.00/mcf) is calculated at $.015 per water horsepower hour and for electricity ($.03/kwh) the fuel cost is $.034 per water horsepower hour (35). The water horsepower for the 200 gpm, 200 ft lift is 10.1. Fuel cost is estimated at $1,500/year for irrigating forty acres (combined demand), assuming natural gas is at $2/mcf or electricity at $.03/kwh. From this viewpoint, it would take around seven years to pay for the wind unit from the savings in fuel. If natural gas goes to $4/mcf then the wind unit would be paid off in four years. These figures neglect the fixed charge rate on money. Maintenance costs for the conventional system and the wind unit are assumed to be the same, since no data are available on the reliability of a wind power irrigation system.

Another way to look at the problem is to estimate the price of power to the consumer (36) as generated by the wind unit. If an annual fixed charge rate of .15 and a maintenance cost of 1 mill/kwh are used, then the estimated consumer price of power is around $.013/kwh. This depends on the production price of $10,000/unit. If that price were doubled then the cost of power would be doubled. It is also assumed that the wind unit would have at least a 15 year life. This estimate can be checked against the calculated power output, 67,485 kwh, for WT5 for the Amarillo area in an average year, 1966. Over 15 years at $.03/kwh that would be worth $30,368 minus $1,012 for maintenance.

It must be emphasized that these figures are estimates and are based on the production (500 units) price and will have to be adjusted accordingly when actual prices of manufactured systems are known. Also the wind powered rotary system does not pump water all year round, therefore the above estimates on the value of the power are too large. The wind powered irrigation system can provide an economic alternative when the price of the wind unit is around $1,000/kw. This is comparable to electricity at $.03/kwh. Although the farmer will be sacrificing convenience, he will have the assurance of knowing his fuel costs will not increase and the fuel is non-depletable.

For the large wells, 1000 gpm, which furnish water for pivot point sprinkler systems, comparable size wind units (100 kw) at $1,000/kw would be needed. The main problem will be in acquiring the money for the high initial capital cost. For example, to replace a 150 hp natural gas engine with a stand alone wind unit on 20,000 wells, the cost would be two billion dollars.

SUMMARY

Results. A most important product of this project is the quantification of the inherent limitations on the fraction of energy that can be extracted from the wind depending on the mode of operation. If no restrictions are imposed on rotor rpm and power output, then the maximum annual efficiencies for rotor WT5 are: 52% for constant power coefficient operation, 45% for constant rpm operation, and 35% for constant torque operation. See the section on Rotor Operation for more details.

When the project was initiated we thought a mechanical system would be best and since the project was limited to three months, two pumping systems, positive displacement and airlift were examined in detail. We now realize that both of these systems are constant torque operations, the least efficient way to operate a wind rotor. However, it is possible to use inefficient machines if the product and/or economics are justified. For example, the farm windmill is very inefficient.

The rotary pump system could pump enough water (average 156 gpm) for forty acres in the Amarillo area. The annual efficiency calculated was 24%, assuming a rotary pump of 80% efficiency could be built. The initial cost for this system is $11,000 greater than a comparable conventional system. If natural gas costs $2/mcf or electricity costs $.03/kwh then it would take seven years to pay for the wind system from the savings in fuel. If natural gas becomes unavailable or very expensive then this type of system becomes economically feasible.

The airlift system cannot pump sufficient water since the annual efficiency of this system was calculated to be only 6%. The available air compressors are too inefficient for this system to be economically feasible.

The conclusions from the present study parallel those of many others. That is, the initial costs for utilizing alternative energy sources are considerably greater than for using fossil fuels. However, wind energy is a viable alternative for pumping irrigation water and the research and developmental work needs to be started now, while fossil fuel resources are still readily available.

Discussion. The maximum theoretical efficiency for a wind unit is 59% and for WT5, constant power coefficient operation, the calculated maximum annual efficiency is 52%. However, to operate in this mode, a variable ratio transmission, or a variable volumetric pump or compressor must be available, because the power load must vary as the cube of the rotor rpm. Ninety percent of the energy in the wind usually occurs within a wind speed spectrum where the wind speed varies by a factor of three. For example, ninety-five percent of the energy (Fig. 2) occurs in the range of 12-36 mph. If a variable speed transmission was developed for the wind rotor then the rpm for a fixed volumetric pump or compressor would have to vary by a factor of 27 to match the rotor. Pump manufacturers do not recommend such a large rpm range for their pumps.

If a variable volumetric pump or compressor became available, a constant ratio gear box could be used. In this case the volumetric delivery per stroke or revolution would have to be proportional to the square of the rpm. For a variable speed transmission or a variable volume pump, analyses similar to those in this report will be necessary to determine the economics of total systems which include these devices.

The constant rpm operation has a better annual efficiency than the constant torque operation. It is also possible for this to be a hybrid system where the wind unit is in parallel with an internal combustion engine or electric motor, each delivering power to a gear head or drive shaft. By such an arrangement, a farmer would have irrigation water on demand and with the wind blowing, the energy extracted would decrease the fuel requirements of the internal combustion engine or the electrical power requirements of the electric motor. Such a hybrid unit could make use of irrigation equipment already installed. With an induction motor, it would be possible for the hybrid unit to feed power back into the utility grid when the irrigation equipment is not being used.

This type of operation presents institutional difficulties since the utility company is essentially providing storage through the use of their power lines. A system of this type is to be installed at a school on Prince Edward Island by Dominion Aluminum Fabricating (DAF), and Windworks inverters are installed on some smaller systems.

DAF markets vertical axis wind turbines (3 to 200 kw) and DAF personnel are involved in engineering calculations for a hybrid system, specifically, a diesel-driven electric generator connected to one of their wind turbines. At the present time that data is still unavailable.

At constant rpm operation it is possible to use electricity for power transfer and the important question once again concerns the overall efficiency and the economics of such a system. The possibilities are a hybrid system connected to a conventional turbine pump, airlift with electric motor, and airlift with electric motor and hyraulic air compressor.

The hybrid system, wind unit with induction motor and conventional well, appears to us to have the prospect of being the most economical, and the one requiring the shortest period of time for development. Two drawbacks are recognized with this system: (1) two thirds of the irrigation wells in the High Plains use natural gas engines and (2) the institutional aspects of connecting such a system to the utility grid.

This system has the following appealing features: (1) The conventional turbine pump with shafting and column pipe can be used. (Any of the other options discussed require refitting and in some cases more drilling and casing.) This feature alone will doubtless appeal strongly to the intended user - the irrigation farmer. (2) The rotor can be a (nearly) constant speed type, thus decreasing the problem of troublesome vibrations. (3) The opportunity exists for returning electric power to the line, for the wind rotor can drive an ordinary induction motor as a generator if it remains connected to the line. (Several problems, including power factor correction, relation to peak demand, and phase loading, would need to be worked out.) By this technique, the unit could be producing something of value when irrigation is not required. (4) The rate at which water would be pumped would not be affected by the variability of the wind. (5) The rotor would be the only part of the system that might necessitate developmental work, but this is also true of any other wind powered system. The other items required, gear box, clutch, etc. are already available. For these reasons, this system should provide the smoothest first step in the utilization of wind power for irrigation pumping.

For the natural gas engines, the possibilities are a hybrid system to save fuel, replacement with a hybrid system which uses an electric motor, or by a stand alone rotary pump system or an airlift system (electric motor, hydraulic air compressor). The problem of pumping large quantities of water, 1000 gpm under 60 psi, with a stand alone wind system seems to be very uneconomic at the present time. This is mainly because a large wind unit (200 kw) would be needed and part of the year that capital intensive unit would not be producing any return.

RECOMMENDATIONS

A prototype hybrid system for irrigation pumping should be constructed and tested within the Northern Panhandle Region. This system should include a wind turbine of at least 10 kw rating connected to a conventional irrigation well powered with an induction motor.

A prototype mechanical system should also be constructed and tested. Developmental work is needed to provide a satisfactory pump. This type of system should be tested because it is probable that many situations will exist where electrical or other power will be either unavailable at the well site or will be too expensive to use.

Prototype systems will provide experimental data vital for checking predictions of system performance, and for cost analysis studies. At this writing, nothing is needed more than a comparison or actual rotor performance with predictions made with use of the presently available theoretical model.

Feasibility studies of a nature similar to those in this report are needed for the following wind powered irrigation systems:

1) Hybrid unit consisting of wind rotor (constant rpm, electric motor, hydraulic air compressor, air lift pump.

2) Hybrid system consisting of wind rotor (constant rpm) and natural gas engine.

3) Systems making use of rotors operating at constant (maximum) power coefficient. Developmental work is needed on a variable stroke air compressor.

If large numbers of hybrid wind-electric irrigation systems should be connected to a power distribution system, regardless of whether power is simply saved and/or excess power is fed back into the grid, substantial changes will have to be accomodated within the distribution system, and it will be important to know what is implied technically and economically in this type of operation. For this reason, a study of the institutional aspects of the extensive use of hybrid wind-electric irrigation systems on power networks is needed. It would be rather easy as well as opportune to incorporate the solar energy potential in the model, because the combined wind-solar systems are future possibilities for water pumping and other tasks as well.

Finally, a dynamic model of wind rotor performance should be developed. The model currently in use is basically a steady-state model and dynamic performance can be inferred only as a sequence of steady states.

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