# VERTICAL AXIS WIND TURBINE DRIVE TRAIN TRANSIENT DYNAMICS

David B. Clauss and Thomas G. Carne
Division 5523
Sandia National Laboratories
Albuquerque, New Mexico 87185

## ABSTRACT

Start-up of a vertical axis wind turbine causes transient torque oscillations in the drive train with peak torques which may be over two and one-half times the rated torque of the turbine. These peak torques are of sufficient magnitude to possibly damage the drive train; safe and reliable operation requires that mechanical components be overdesigned to carry the peak torques caused by transient events. A computer code, based on a lumped parameter model of the drive train, has been developed and tested for the Low Cost 17-Meter turbine; the results show excellent agreement with field data. The code has subsequently been used to predict the effect of a slip clutch on transient torque oscillations. It has been demonstrated that a slip clutch located between the motor and brake can reduce peak torques by thirty eight percent.

## INTRODUCTION

Transient events during operation of Darrieus vertical-axis wind turbines can cause torque levels in the drive line which are unacceptable for many components. Start-up and braking in various ambient conditions are typical events during which peak torques may reach excessive magnitudes. Experience with research vertical axis wind turbines (VAWT) indicates that peak torques of 2 to 3 times rated torque are typical during starting and braking, which implies an undesired overdesign of drive-line components. The objective of the present investigation is to develop an analytical tool which can be used to predict drive-train behavior for several different loading conditions.

Analysis is needed to determine which starting and braking methods are most effective in reducing the peak torques seen in the drive-line. Areas deserving investigation include start-up in high winds, as well as electrical and mechanical methods (clutches) designed to achieve a softer start. The relative effectiveness of low speed vs. high speed braking, and definition of a braking rate which will decrease dynamic amplification to an acceptable level also merit study. Although this paper will deal primarily with turbine start-up in zero ambient wind speed and the effects of a slip clutch on transient response, the model can easily be adapted to study the problems mentioned above.

The model and the numerical results as well are based on experimental data obtained on the Low Cost 17M VAWT installed at Rocky Flats, CO, Figure 1. The method, however, is essentially general to all VAWT's. A Fortran program called DYDTA (DYnamic Drive Train Analysis) numerically evaluates the differential equations of motion and plots results.
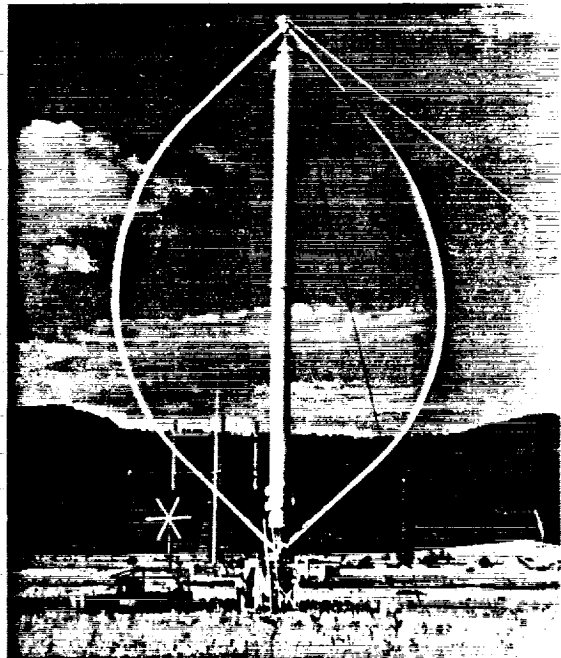


FIGURE 1. DOE/ALCOA Low Cost 17M VERTICAL AXIS WIND TURBINE INSTALLED AT ROCKY FLATS, CO.

## THE DRIVE TRAIN MODEL

Typically, a VAWT drive train consists of the turbine rotor (blades and rotating tower), the transmission, a brake disc and an induction motor/generator which are connected in series by shafts and couplings. Additional mechanical components may be present, and the drive train topography may vary depending upon the specific turbine design. Additional components may include a timing belt (for incremental adjustment of turbine operating speed) and/or a slip clutch. The position of the brake relative to the

transmission and clutch, if present, is the most variable element in drive train topography. For instance on the Low Cost 17M turbine at Rocky Flats, Figure 1, the brake is on the high speed shaft, whereas earlier turbines have had the brake on the low speed shaft.

The transient response depends on the natural characteristics of the system and the functional form of the applied torques. The physical representation of the drive train is shown in Figure 2, along with physical values for the Low Cost turbine. For generality the model includes the slip clutch, as well as applied torques on each inertial element. Several assumptions are made, all of which may not be applicable to a given VAWT design. The low speed shaft is considered to be the only significant stiffness in the system since, in an equivalent system, it appears much softer than both the high speed shaft and the rotating tower. As larger turbines are built and tower height

increases, the tower stiffness may approach the same value of stiffness as the low speed shaft, and at some point tower stiffness may have to be included in the model. On the other hand, the effect of the transmission's gear ratio on the equivalent stiffness of the high speed shaft seems to insure that the high speed stiffness will remain large relative to low speed stiffness, and thus the high speed shaft can be effectively modeled as a rigid element. The motor torque curve as specified by the manufacturer is modified by a constant scale factor less than one, which is related to the voltage drop in the line. For an induction motor/generator, torque is proportional to voltage squared, so that a 20% drop in voltage will cause a 36% reduction in motor torque. The assumption of a constant motor scale factor implies that the voltage drop, and thus the current drawn by the motor are independent of motor speed, which is only approximately true. Also, nonlinear effects such as Coulomb friction, aerodynamic damping, and drive slack are not treated. All losses are represented by viscous damping.

The equations of motions describing the torsional response of the drive train can be easily written by drawing free body diagrams for each inertial element, Figure 3, and equating the torque on each to zero. This method is preferred because it
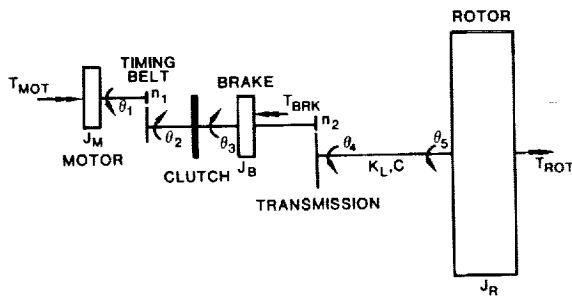


$J_M$ = motor inertia      = 1.291 lb-ft-s$^2$

$J_B$ = brake inertia      = 1.598 lb-ft-s$^2$

$J_R$ = rotor inertia      = 4.042 x 10$^4$ lb-ft-s$^2$

$C$ = Viscous damping      = 1.1 x 10$^3$ lb-ft-s

$K_L$ = low speed shaft stiffness      = 8.626 x 10$^5$ lb-ft

$n_1$ = timing belt ratio      = 1, 40/38, 44/38

$n_2$ = transmission ratio = 35.07

$T_{MOT}$ = motor torque, $T_{MOT}(\omega)$

$T_{BRK}$ = brake torque, $T_{BRK}(t)$

$T_{ROT}$ = aerodynamic torque, $T_{ROT}(\omega, t)$

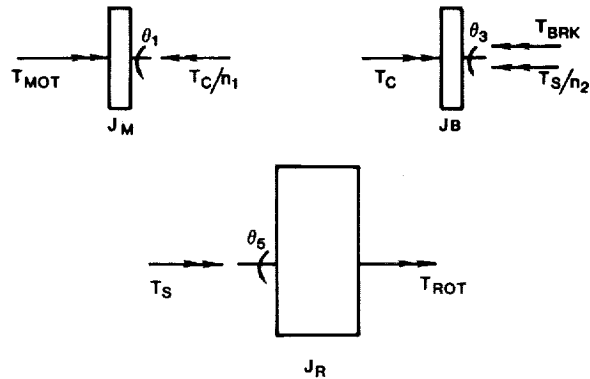FIGURE 2. PHYSICAL REPRESENTATION OF VAWT DRIVE TRAIN.



FIGURE 3. FREE BODY DIAGRAMS FOR WRITING EQUATIONS OF MOTION.

permits treatment of the nonlinear effect of the clutch in a simple way since torque through the clutch can be expressed explicitly. Referring to Figures 2 and 3, $T_C$ and $T_S$ are defined and calculated as follows:

$T_S$ = torque transmitted through low speed shaft,

$$T_S = K_L(\theta_4 - \theta_5) + C(\dot{\theta}_4 - \dot{\theta}_5), \qquad (1)$$

$T_C$ = torque transmitted through slip clutch.

The equations of motion can now be written directly:

$$J_M \ddot{\theta}_1 = T_{MOT} - T_c/n_1, \qquad (2)$$

$$J_B \ddot{\theta}_3 = T_c - T_{BRK} - T_s/n_2, \qquad (3)$$

$$J_R \ddot{\theta}_5 = T_s + T_{ROT}, \qquad (4)$$

where the clutch imposes a constraint of the form

$$|T_c| \leq T_{max}; \qquad (5)$$

$T_{max}$ = maximum torque passed by clutch.

The slip clutch typically consists of two mating frictional surfaces which are connected to the driving and driven shafts, respectively. These frictional surfaces are compressed together by a spring, so that the normal force, surface area, and friction coefficients determine the maximum torque transmitted by the clutch. $T_{max}$ can be adjusted by changing the spring deflection.

The clutch is always operating in one of two conditions; it is either engaged, in which case the velocities on either side of the clutch are equal, or it is slipping, in which case the velocities are unequal and the clutch torque is equal to $\pm T_{max}$. The torque speed characteristic of an ideal clutch is shown in Figure 4. Note that $T_{max}$ is a restoring torque; that is, it is in a direction that will tend to re-engage the clutch.
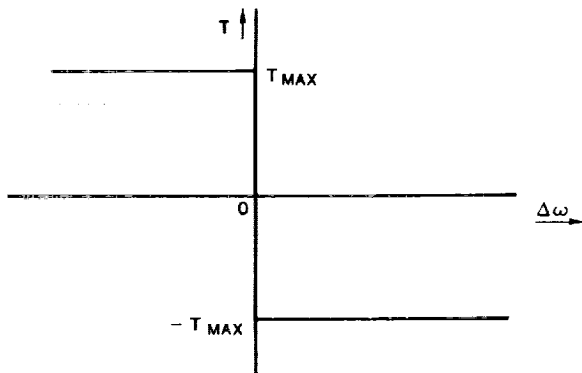


FIGURE 4. TORQUE VS. SPEED DIFFERENCE - IDEAL CLUTCH. ASSUMES STATIC FRICTION COEFFICIENT EQUALS DYNAMIC FRICTION COEFFICIENT.

With this understanding, two distinct sets of unconstrained equations of motion can be written corresponding to the two operating states of the clutch, which will be referred to as the engagement and slip equations, respectively.

Engagement of the slip clutch implies that $\theta_2 = \theta_3$, so that the motor and brake move together. Multiplying equation (1) by $n_1$ and summing the result with equation (2):

$$n_1 J_M \ddot{\theta}_1 + J_B \ddot{\theta}_3 = n_1 T_{MOT} - T_{BRK} - T_s/n_2. \qquad (6)$$

The equations can be simplified by making the following variable transformation and taking advantage of gear relations

$$\psi_1 = \theta_2 = \theta_1/n_1, \qquad (7)$$

$$\psi_2 = \theta_3 = n_2 \theta_4, \qquad (8)$$

$$\psi_3 = n_2 \theta_5. \qquad (9)$$

Thus the complete set of engagement equations can be written

$$(J_{ME} + J_B)\ddot{\psi}_1 = n_1 T_{MOT} - T_{BRK} \qquad (10)$$
$$- K_{LE}(\psi_2 - \psi_3) - C_E(\dot{\psi}_2 - \dot{\psi}_3),$$

$$\ddot{\psi}_2 = \ddot{\psi}_1 \qquad (11)$$

$$J_{RE}\ddot{\psi}_3 = K_{LE}(\psi_2 - \psi_3) + C_E(\dot{\psi}_2 - \dot{\psi}_3) \qquad (12)$$
$$+ T_{ROT}/n_2;$$

where

$$J_{ME} = (n_1)^2 J_M, \qquad (13)$$

$$J_{RE} = J_R/(n_2)^2, \qquad (14)$$

$$K_{LE} = K_L/(n_2)^2, \qquad (15)$$

$$C_E = C/(n_2)^2. \qquad (16)$$

Slipping of the clutch decouples the motor from the brake and rotor, so that the slip equations describe two independent systems as shown in Figure 5. Slip implies a velocity difference across the clutch and that the torque transmitted through the clutch is a constant $\pm T_{max}$. Again taking advantage of equations (7) - (9) and equations (13) - (16), and making the substitution
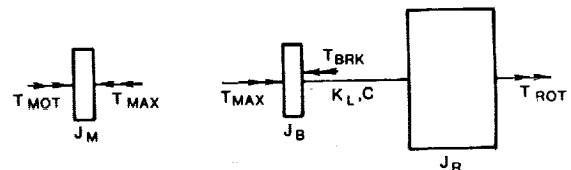
$$T_c = \pm T_{max} \qquad (17)$$



FIGURE 5. REPRESENTATION OF DRIVE TRAIN FOR CLUTCH SLIPPING.

the final form of the slip equations becomes

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$$J_{ME}\ddot{\psi}_1 = n_1 T_{MOT} \mp T_{max} \qquad (18)$$

$$J_B\ddot{\psi}_2 = \pm T_{max} - T_{BRK} - K_{LE}(\psi_2-\psi_3)$$
$$- C_E(\dot{\psi}_2-\dot{\psi}_3) \qquad (19)$$

$$J_{RE}\ddot{\psi}_3 = K_{LE}(\psi_2-\psi_3) + T_{ROT}/n_2$$
$$+ C_E(\dot{\psi}_2-\dot{\psi}_3) \qquad (20)$$

where $T_{max}$ is positive for motor overspeed.

The conditions for initiation of slip and for re-engagement are somewhat more subtle, although intuitively simple. Assume that, at time equal zero, the clutch is engaged. Slip initiates when the clutch torque equals $T_{max}$ and continued engagement would tend to violate the constraint, equation (5). Stated more rigorously the initiation of slip occurs when

$$T_{Ce} = T_{max} \qquad (21)$$

and

$$\frac{dT_{Ce}}{dt} > 0 \qquad (22)$$

where the subscript e indicates the result obtained from the engagement equations. If $T_{Ce}$ is negative, then inequality (22) is directed in the opposite sense. Now assume that the clutch is slipping, so that $\dot{\theta}_2 \neq \dot{\theta}_3$. Re-engagement occurs when the velocities across the clutch become equal again:

$$\dot{\theta}_{2s} = \dot{\theta}_{3s} \qquad (23)$$

and if

$$|T_{Ce}| \leq T_{max}$$

At re-engagement there must be a discontinuity in acceleration, which implies a discontinuity in the clutch torque. This is consistent with the characteristics of the clutch as shown in Figure 4, since when $\Delta\omega = 0$ (as it is at re-engagement), the clutch torque may be any value between positive $T_{max}$ and negative $T_{max}$.

Thus, a basic algorithm would be to use the engagement equations until equations (21) and (22) are satisfied (slip initiates), and then to apply the slip equations until equation (23) is met (re-engagement), at which point the algorithm is applied recursively.

A great deal of insight can be gained from an examination of the natural characteristics of the system, especially in the case of start-up. The remainder of this paper will focus on start-up in a zero ambient wind speed. The motor torque-speed characteristic has a significant impact on the natural characteristics of the system. A typical induction motor curve is shown in Figure 6. It is important to realize that motor
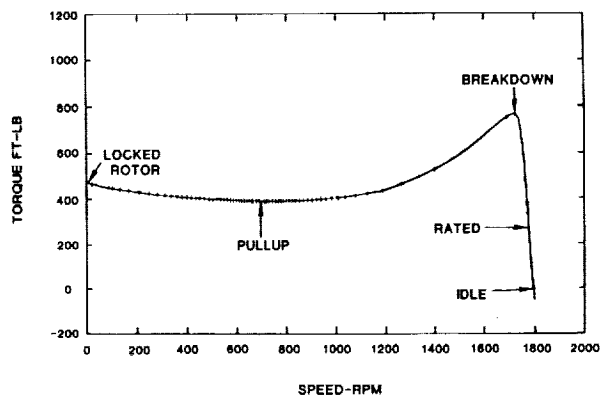


FIGURE 6. TORQUE-SPEED CHARACTERISTIC OF AN INDUCTION MOTOR.

torque is a function of motor speed, not of time. Torque ripple models developed[1] by Reuter have treated the induction motor/generator mechanically as an inertia connected to ground through a linear viscous damper. Referring to Figure 6 the torque-speed relationship is indeed linear for steady state operation ($1740 < \omega < 1860$ rpm). However, during start-up the motor speed goes from 0 to 1800 rpm and the torque-speed relationship cannot be represented as a linear element. From an intuitive standpoint, it is most clear to think of the motor torque characteristic as a non-linear damper, which has a coefficient dependent on rpm, superimposed on a step torque, though in the code and model it is treated as an applied torque. To be clear, the motor torque should not be considered to be an external torque because it is not independent of the state of the system. With this in mind, the free vibration characteristics of the drive train during start-up can be considered.

The natural characteristics of the system also depend on the operating state of the slip clutch. When the clutch is engaged, there is an effectively rigid connection between the motor and the brake so their respective inertias may be lumped together, and the system can be represented as shown in Figure 7. On the other hand, slipping decouples the motor and creates two independent systems, Figure 8. The latter case will not be treated in detail, however, it is apparent that the fundamental frequency of the decoupled system in 8(b) will be higher than that of coupled system in Figure (7), and that the isolated motor, Figure 8(a) moves as a rigid body.

Due to the nonlinearity introduced into the system by the motor and clutch, a linear eigenvalue analysis is not truly applicable. The equivalent motor damping changes continually so that the natural frequencies and mode shapes are a function

of speed. However, an approximate analysis can be done by considering two subcases based on motor speed less than or
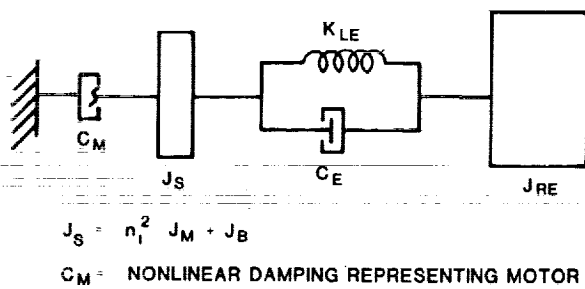


$$J_S = n_1^2 \, J_M + J_B$$

$C_M$ = NONLINEAR DAMPING REPRESENTING MOTOR

FIGURE 7. SYSTEM REPRESENTATION WITH CLUTCH ENGAGED.
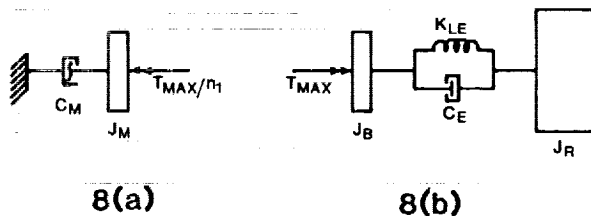


**8(a)**          **8(b)**

FIGURE 8. SYSTEM REPRESENTATION WITH CLUTCH SLIPPING.

greater than the motor breakdown speed, respectively. Up to the breakdown speed the maximum absolute instantaneous motor damping results in a critical damping of approximately minus nine percent, and consequently the damping does not greatly affect the natural frequencies up to breakdown speed. Note that negative damping has the same effect on frequency as positive damping (1 d.o.f., $\omega_d = \sqrt{1 - \xi^2} \; \omega_n$), but the opposite effect on mode shape (in Figure 7, negative damping from the motor tends to increase the oscillations of the motor and brake relative to the rotor). Thus, up to breakdown, damping is negligible and the system can be viewed as shown in Figure 9(a). The natural frequencies and mode shapes of the
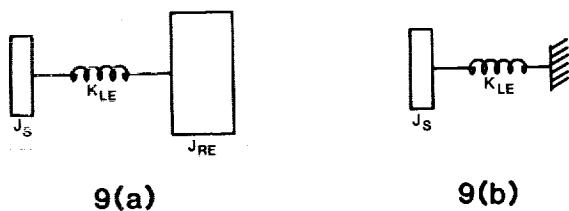


**9(a)**          **9(b)**

FIGURE 9. SYSTEM REPRESENTATION WITH CLUTCH ENGAGED AND MOTOR SPEED < 1720 RPM.

system in Figure 9(a) are easily determined. The results are

$$\omega = 0, \; \sqrt{\frac{K_{LE}(J_S + J_{RE})}{J_S J_{RE}}} \tag{24}$$

$$= 0, \quad 2.586 \quad HZ$$

$$u^{(1)} = \begin{Bmatrix} 1.000 \\ 1.000 \end{Bmatrix}, \quad u^{(2)} = \begin{Bmatrix} 1.000 \\ 1 - \dfrac{J_S + J_{RE}}{J_{RE}} \end{Bmatrix} \tag{25}$$

$$= \begin{Bmatrix} 1.000 \\ -0.083 \end{Bmatrix}$$

The mode shape indicates that the generator/brake is moving with much greater amplitude than the rotor, so that the mode basically represents the motor/brake winding up about the low speed shaft, Figure 9(b). Indeed, if $J_{RE} \gg J_S$ (typically the case for VAWT's), equation (24) can be approximated by

$$\omega \simeq \sqrt{\frac{K_{LE}}{J_S}} \tag{26}$$

which corresponds to the frequency of the system in Figure 9(b).

As previously discussed, when the motor speed is greater than the breakdown speed, the torque is nearly a linear function of speed, with a very large negative slope. This slope, and thus the effective damping, is a function of the rated power, the synchronous speed, and the slip at rating for an induction motor/generator:

$$C_M = \frac{P_{rated}}{\Omega^2 s} \quad , \tag{27}$$

as given in [2].

For the Low Cost turbine this corresponds to a critical damping factor of over 200%, which indicates that the generator/brake oscillation will be very small. As an approximation, the generator can then be considered to be fixed when the motor speed is greater than the breakdown speed, as shown in Figure 10, so that the first mode resembles the rotor winding up on the low speed shaft.
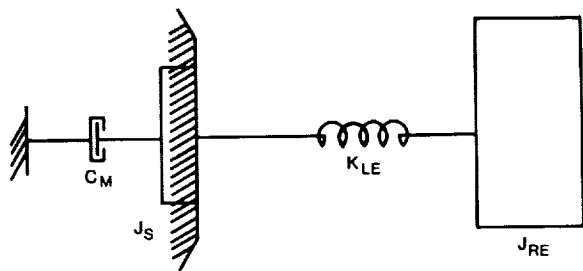
FIGURE 10. SYSTEM REPRESENTATION WITH
CLUTCH ENGAGED AND MOTOR
SPEED > 1730 RPM.

The frequency for the system in Figure 10
is simply

$$\omega \cong \sqrt{\frac{K_{LE}}{J_{RE}}} \ . \qquad (28)$$

## EXPERIMENTAL RECORD - BASE CASE

The Low Cost 17M turbine at Rocky
Flats is instrumented with a torque sensor
located on the low speed shaft. A typical
start-up record taken in zero ambient wind
speed is shown in Figure 11. The
experimental record was used to develop a
base case which could be used to verify
the predictive capabilities of the model,
as well as to fix certain parameters that
could not accurately be calculated
analytically or experimentally. The
parameters which were varied were the
viscous damping coefficient and the motor
scale factor. There are several
characteristics which are typical of a
start-up record: the initial overshoot
and subsequent decay, and then the growth
of torque oscillations to the largest peak
torque, and the frequency shift that
occurs just after the peak is reached.
Note that the maximum torque (40,000
ft-lb) is over 2.5 times the rated turbine
torque (15,000 ft-lb). The time to start
is also an important characteristic.
Several non-linear effects are apparent in
the record as well; the decay envelope is
not smooth, which indicates non-viscous
damping, and gear slack is evident (note
the 'notch' when the sign of the torque
changes near the end of the record). The
model does not attempt to explain or
predict these non-linear effects.

DYDTA results for this base case are
shown in Figures 12-15. Varying the motor
scale factor changes the starting time and
the magnitude of torque, while varying the
viscous damping coefficient primarily
affects the initial rate of decay and to a
lesser extent the magnitude of torque
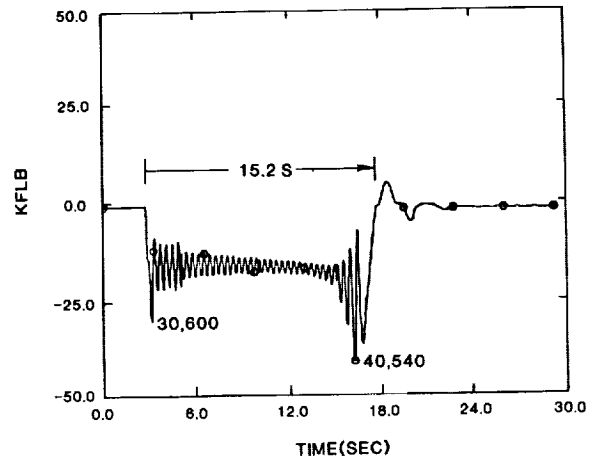oscillations. The final value for the
motor scale factor is 0.665, which



FIGURE 11. EXPERIMENTAL RECORD - LOW
SPEED SHAFT TORQUE VS. TIME
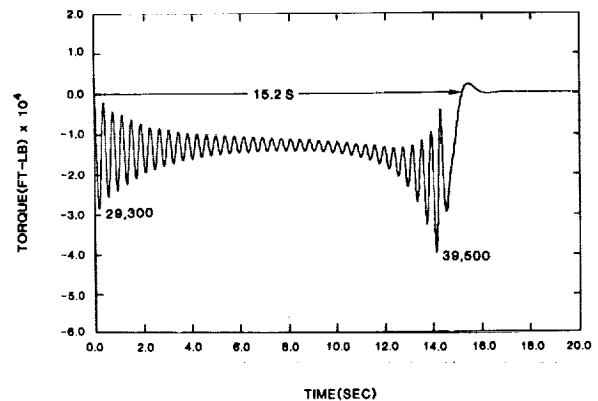FOR THE LOW COST TURBINE
START-UP IN ZERO WIND.



FIGURE 12. DYDTA PREDICTION - LOW SPEED
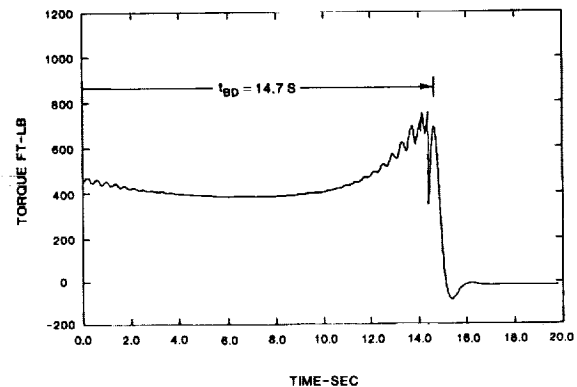SHAFT TORQUE VS. TIME DURING
START-UP IN ZERO WIND - NO
CLUTCH.



FIGURE 13. DYDTA PREDICTION - MOTOR
TORQUE VS. TIME FOR START-UP
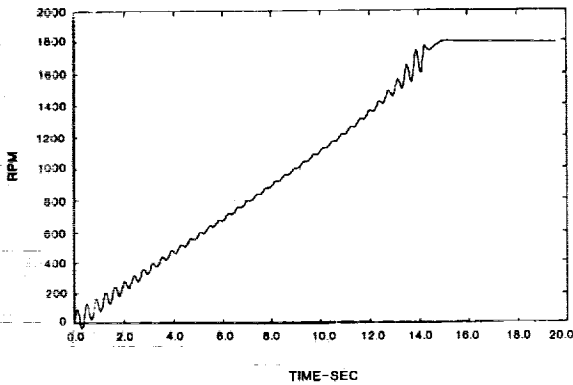IN ZERO WIND - NO CLUTCH.

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FIGURE 14. DYDTA PREDICTION - MOTOR
SPEED VS. TIME FOR START-UP
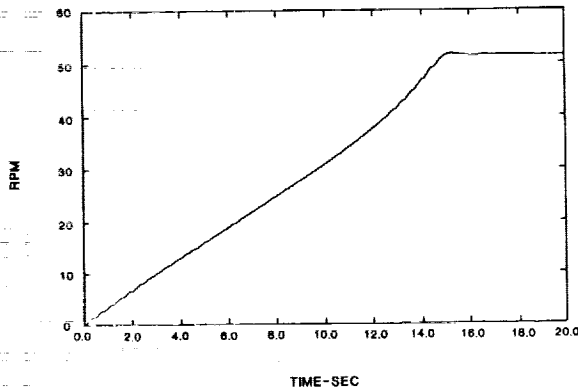IN ZERO WIND - NO CLUTCH.



FIGURE 15. DYDTA PREDICTION - ROTOR
SPEED VS. TIME FOR START-UP
IN ZERO WIND - NO CLUTCH.

corresponds to an 18% voltage drop in the line. This is not unreasonable as the current drawn by the motor is rather high and the line is fairly long at the Rocky Flats site. Obviously the scale factor depends on the particular motor and electrical arrangement for a given VAWT design. The viscous damping coefficient was determined to be $1.1 \times 10^3$ lb-ft-sec, corresponding to a damping factor of approximately one percent. This is insignificant compared to damping provided by the motor and thus the influence of viscous damping due to the structure on the response is negligible.

The results from DYDTA, after adjustment of the scale factor and viscous damping coefficient, show excellent agreement with the field data (Figure 11), and the simple drive train model can be used to explain the characteristics typical of a VAWT start-up in zero wind. DYDTA correctly predicts the magnitude of the initial overshoot and the subsequent decay. The initial overshoot is due to the step in torque which occurs when the

motor is turned on; the amplitude should be approximately twice the initial motor torque (locked rotor torque). When the motor speed is less than the pull-up speed (725 rpm) the motor contributes a small amount of positive damping, and together with the viscous damping due to the structure it causes the oscillations to decay in the typical envelope fashion. However, the rate of decay constantly decreases and eventually becomes zero as the damping from the motor goes from positive at locked rotor to zero at pull-up and becomes negative beyond that.

The growth of torque oscillations occurs when the damping produced by the motor becomes negative, an effect also predicted by DYDTA, Figure 12. Negative damping results in energy addition to the system, which causes an unstable growth in the amplitude of the torque oscillations. The maximum torque seen by the low speed shaft occurs at approximately the time the motor breakdown speed is reached, when the motor damping changes rapidly from negative to positive. As explained earlier, there is a frequency shift at this point as well: prior to breakdown, the total damping is relatively small, and thus it does not significantly affect the 1st fundamental frequency. Since the rotor inertia is much larger than the motor/brake inertia, the vibrational mode basically consists of the motor/brake winding up on the low speed shaft. Past breakdown the large positive damping produced by the motor effectively reduces the motor/brake vibration to zero, so that the first mode shape becomes the rotor oscillating about the low speed shaft. There is a significant decrease in the fundamental vibrational frequency past breakdown due to the large size of the rotor inertia relative to that of the motor/brake. Again, DYDTA is in very good agreement with field data with regard to these phenomena. DYDTA also matches experimental records on starting times, which essentially depends on rotor inertia and mean motor torque.

Figure 13 demonstrates the variation of motor torque as a function of time, which is another critical aspect of VAWT start-up. Up until breakdown the motor draws very high levels of current, and the motor can draw high power for only a short period of time. Past breakdown the current is reduced, so it is clearly advantageous to reduce the time to breakdown ($t_{BD}$) as much as possible. For the base case $t_{BD}$ is about 14.7 seconds. Figures 14 and 15 show the speeds of the motor/brake and rotor, respectively as functions of time. They serve to verify that the turbine successfully comes up to speed, as well as to substantiate what has been said earlier in regard to the mode shapes (the rotor motion is almost entirely a rigid body mode whereas the motor contains a

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significant amount of oscillation about the rigid body mode).

## CONTROL OF TRANSIENTS WITH A SLIP CLUTCH

High torques experienced in the low speed shaft of the Low Cost 17M turbine resulted in the investigation of a slip clutch as a possibility for reducing peak torques. DYDTA has been used to analyze the effect of a slip clutch on torque levels during start-up. The results indicate a clutch could significantly lower the peak torques seen by the low speed shaft, with the following secondary benefit. The clutch also reduces the time to breakdown for the motor ($t_{BD}$), which reduces the current drawn by the motor and protects it from overload. It appears that an appropriately chosen clutch would be well within margins of safety with respect to heat dissipation and power absorption capabilities.

The characteristics of a typical slip clutch were discussed earlier; recall that the clutch passes torque uniformly up to some maximum value ($T_{max}$) which can be adjusted by changing the deflection of a compression spring. With this in mind a parametric study was done using DYDTA to determine what value or range of $T_{max}$ would best reduce torque levels. Obviously $T_{max}$ should be above the rated turbine torque, which is 427 ft-lbs (15,000 ft-lbs referred to the low speed shaft), or else power could be absorbed by the clutch during normal operation.

The optimum value for $T_{max}$ for the Low Cost turbine appears to be 475 ft-lbs (16,700 ft-lbs referred to the low speed shaft), for which the results predicted by DYDTA are shown in Figures 16-21. With the clutch present, the maximum torque in the low speed shaft occurs during the initial overshoot, and its magnitude has been reduced 38% from 40,000 ft-lbs to 25,000 ft-lbs. The growth of torque oscillations does not occur because slipping of the clutch decouples the motor from the drive train, thereby isolating the low speed shaft from the negative damping produced by the motor. Note that the torque in the low speed shaft can exceed $T_{max}$, despite the clutch, because of the inertial reaction of the brake. This implies that the slip clutch should be located as close to the component needing protection as is physically possible.

The clipping action of the clutch is demonstrated in Figure 17; the torque transmitted through the clutch cannot exceed $T_{max}$. As discussed previously, a secondary benefit is that the clutch reduces $t_{BD}$ for the motor. Figure 18 shows that the high torque part of the motor curve is sped through very rapidly, and $t_{BD}$ is reduced to 12.6s from 14.7s for no clutch.
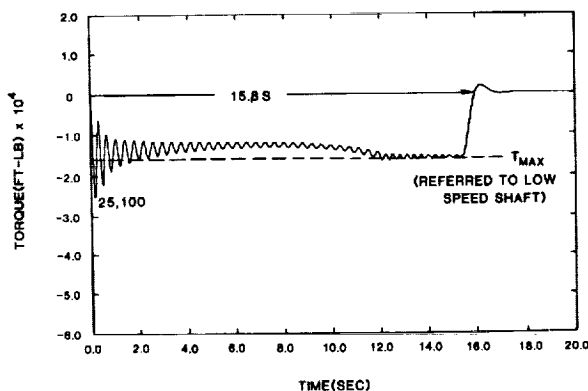


FIGURE 16. DYDTA PREDICTION - LOW SPEED SHAFT TORQUE VS. TIME DURING START-UP IN ZERO WIND WITH CLUTCH.
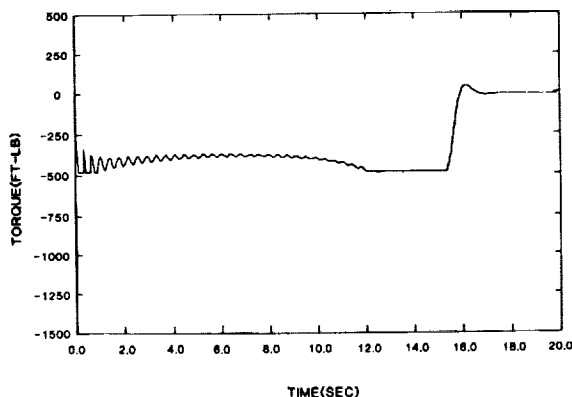


FIGURE 17. DYDTA PREDICTION - TORQUE TRANSMITTED THROUGH CLUTCH VS. TIME DURING START-UP IN ZERO WIND WITH CLUTCH.
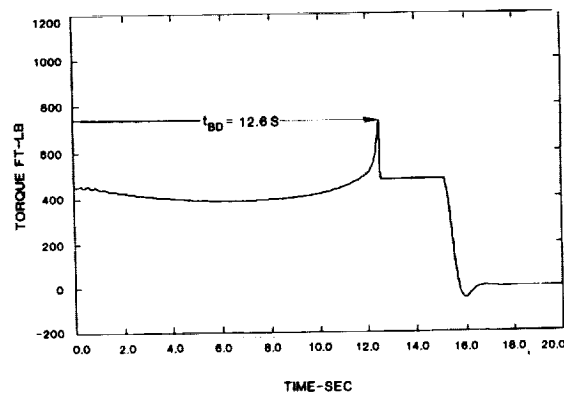


FIGURE 18. DYDTA PREDICTION - MOTOR TORQUE VS. TIME FOR START-UP IN ZERO WIND WITH CLUTCH.
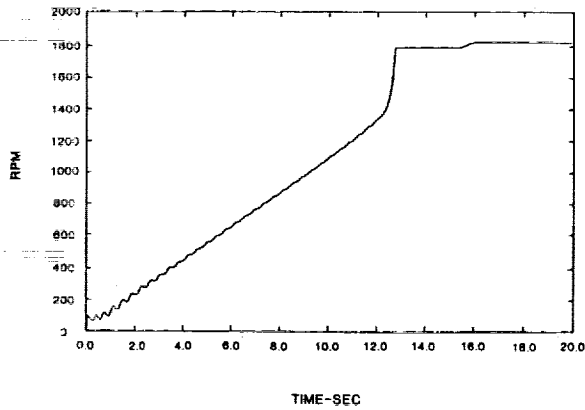
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TIME-SEC

FIGURE 19. DYDTA PREDICTION - MOTOR
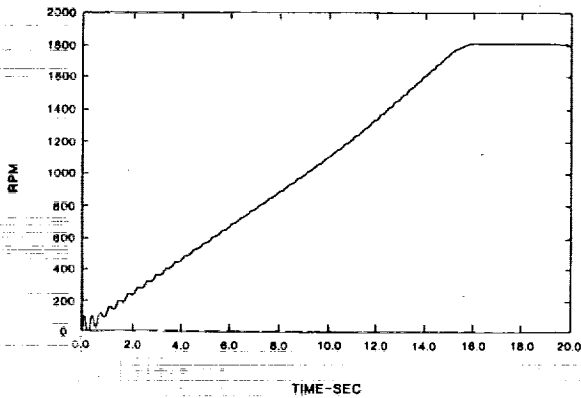SPEED VS. TIME FOR START-UP
IN ZERO WIND WITH CLUTCH.



TIME-SEC

FIGURE 20. DYDTA PREDICTION - CLUTCH
SPEED (ROTOR SIDE) VS. TIME
FOR START-UP IN ZERO WIND.



CLUTCH POWER DISSIPATION VS. TIME
TMAX=475

HEAT DISSIPATED 27 x 10³ LB-FT
(OBTAINED BY INTEGRATING POWER CURVE)

CLUTCH LIMITS FOR 30 SEC SLIP
MAX POWER-56 x 10³ LB-FT/SEC
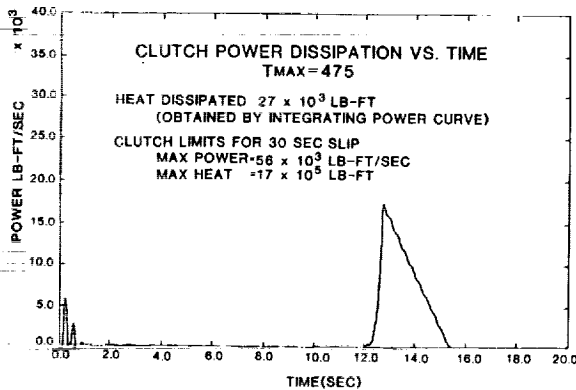MAX HEAT =17 x 10⁵ LB-FT

TIME(SEC)

FIGURE 21. DYDTA PREDICTION - POWER
DISSIPATED IN CLUTCH VS. TIME
FOR START-UP IN ZERO WIND.

Figures 19 and 20 can be compared to
determine at what times the clutch is
slipping. The power dissipated in the
clutch as a function of time, and the
total heat dissipated (obtained by
integrating the power curve) are shown in
Figure 21.

CONCLUSIONS

DYDTA represents an initial step
towards understanding and analyzing
methods of controlling transient behavior
in VAWT drive trains. Results for
start-up in zero wind speed show
exceptional agreement with experimental
records on the Low Cost 17M turbine, thus
providing verification of modeling
accuracy. DYDTA is currently being used
to predict responses for several different
transient operations and possible design
modifications intended to reduce transient
torque levels. It is expected to become a
versatile, easily implemented drive train
design tool.

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Sandia National Laboratories Report No.
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1977.

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QUESTIONS AND ANSWERS

D.B. Clauss

From:  I.P. Ficenec

Q:  What value of damping was used for the transmission system and how was this established?

A:  *The value of damping is 1.1 x 10³ lb-ft-sec, and this was established by varying the damping coefficient in the code until the analytic results agreed well with observed data for start-up.  I might add that this damping coefficient yields a critical damping of ~1%, and consequently does not strongly influence the turbine response.*

From:  G. Beaulieu

Q:  1)  In order to have full automatic control of the turbine, would you also have a maximum power or maximum windspeed cut-off, at which time the turbine would be stopped?

2)  Since you have introduced nonlinearities into your model, what numerical method of solution have you used in your computer code DYDTA?

A:  *1)  Studies involving this aspect of automatic control are in the planning stages.*

*2)  The equations of motion are numerically evaluated using a library routine which employs a variable order Adams method.*

From:  Mike Bergey

Q:  Was the wind data for Oklahoma City from a National Severe Storms Lab report?

A:  *Yes, the data was located for Sandia National Labs by James Connell of PNL.  The National Severe Storm Labs had recorded the data in 1976 and transferred the data to the National Center for Atmospheric Research where Dennis Joseph provided magnetic tapes for dissemination.*

From:  W.C. Walton

Q:  You have treated the rotor as rigid.  Do you think rotor elasticity has any bearing on the dynamic phenomena related to drive train stability and response?

A:  *For VAWT's the size of the Low Cost N-M turbine, we are fairly confident that rotor elasticity does not strongly influence drive train stability or drive train response because the rotor structure appears much stiffer than the low speed shaft.  This is, however, a question we will have to deal with for different turbines.*

From:  P. Anderson

Q:  Does the induction machine model include rotor electrical transients?  What time step size was used for start-up simulations?

A:  *The model does not include electrical rotor transients, nor am I certain of their influence on the mechanical system response.  The step size is shown by the integrator based on the rate of change of the dependent variables, it varies tremendously depending on the smoothness and volatility of these functions.*

From:  Rowley Camedav

Q:  Have you seen reflection of torque spikes back into rotor structure during emergency braking?

A:  *The model which has been used so far treats the rotor structure (turbine tower and blades) as essentially a rigid inertial element, so that analytically we have seen no such reflection .  From a qualitative standpoint, however, there is certainly some reflection of torque spikes onto the rotor structure during emergency braking, although they are probably not as severe as in the low speed shaft.*