NASA **TECHNICAL**
MEMORANDUM

NASA TM X- 68244

NASA TM X- 68244

CASE FILE
COPY

BASIC PRINCIPLES OF VARIABLE SPEED DRIVES

by Stuart H. Loewenthal
Army Air Mobility Research and Development Laboratory
Lewis Research Center
Cleveland, Ohio

BASIC PRINCIPLES OF VARIABLE SPEED DRIVES

by

Stuart H. Loewenthal
Research Engineer
Army Air Mobility Research and Development Laboratories
NASA Lewis Research Center
Cleveland, Ohio

E-7494

ABSTRACT

An understanding of the principles which govern variable speed drive operation is a prerequisite to successful drive application. The fundamental factors of torque, speed ratio, and power as they relate to drive selection are discussed. A general description of the basic types of variable speed drives, their operating characteristics and their applications are also presented.

BASIC PRINCIPLES OF VARIABLE SPEED DRIVES

by

Stuart H. Loewenthal
Research Engineer
Army Air Mobility Research and Development Laboratories
NASA Lewis Research Center
Cleveland, Ohio

INTRODUCTION

The primary purpose of any drive train is to transport mechanical
energy, usually in rotary form, from the prime mover where it origi-
nates to the load where it is required. Most often the operational
characteristics of the load, principally its torque and speed re-
quirements, are different from those of the prime mover. The inter-
vening device which is called upon to match the prime mover to the
load is referred to as the transmission or drive. Often the load
and prime mover are sufficiently compatible that a simple fixed re-
duction in speed is all that is necessary for satisfactory operation.
The mechanisms which perform single speed reductions, such as gears,
belts, chains, traction devices and the alike, are generally con-
sidered as the simplest, least expensive and most trouble free of all
transmission elements. For these reasons, they are often preferred
to variable speed drives for those systems in which a choice is pos-
sible. However, variable speed drive systems need not be any more
complex or less reliable than fixed ratio drives. In addition, be-
cause they generally minimize starting problems, take over several
functions in the drive train, like clutching and overload protection,
and, at the same time, permit the load to operate at peak perform-
ance at all speeds, variable speed drives often reduce overall system
costs as well.

FUNDAMENTALS OF VARIABLE SPEED DRIVE APPLICATION

It is most important for successful drive application that the drive
system designer have a basic awareness of the parameters that govern
variable speed drive operation in addition to the operating charac-
teristics of the prime mover and the machine to be driven.

Reasons for Variable Speed Drives

Table I summarizes several of the major reasons for utilizing vari-
able speed drives. The tangible benefits of these drives as they re-
late to a process requirement is their ability to get the best per-
formance out of the driven machine, that is, produce the highest

quality product within the constraints of the process and also in-
crease the utilization of the equipment by extending its capabili-
ties. For example, variable speed drives permit you to fine tune the
operating speed on metal working tools, such as lathes, millers,
presses, and forming dies to produce the best part with the least
wear and tear on tool in accordance with the rate of production and
the material being processed.

The operation of a feed-water pump for a power plant boiler is
an example how a variable speed drive can make a system cost effec-
tive. It is the characteristic of a centrifugal pump at constant
speed to produce higher pressures as the flow capacity is decreased.
In contrast, the supply pressure required by a boiler decreases with
reduced flow. In order to match the boiler's diminishing capacity,
the flow from the constant speed pump must be throttled by a regula-
tion valve with a corresponding horsepower penalty. An adjustable
speed pump in this situation need only develop the pressure required
by the system at any flow condition. This results in a sizable sav-
ings of power ordinarily wasted. These are only two examples of many
which could be cited to illustrate how variable speed drives can en-
hance one's process.

Another important area of application is in the start up of high
inertial equipment where a variable speed drive can often permit the
use of a smaller motor than ordinarily would be possible. The rea-
sons for this will become apparent later.

The effectiveness of variable speed drives in the last two cate-
gories listed in Table I, isolation of shock and overload protection,
depends on the type of drive being considered. The best drives in
these categories are the nonpositive engagement type such as the
fluid drives, viz., the variable-fill fluid coupling and torque con-
verter types, although even the more positive mechanical drives, such
as the belt, chain and traction types, will absorb some torsional vi-
bration and either disengage or slip under heavy overloads. A strong
selling point for the variable speed fluid drives is in their ability
to smoothly start up the load and eliminate excess wear and tear on
starters, motors and drive train components and to protect the drive
system if load jamming should occur.

Actually there are a whole host of applications where variable
speed drives are important, if not essential. They range from tex-
tile and printing equipment to extruders, mixers and rock crushers.

Types of Variable Speed Drives

Variable speed drives can be broken down into two main categories;
stepped ratio drives such as those which employ gear shifts to change

2

speed, and stepless speed ratio drives which are capable of continuously changing speed. Although many systems work well with stepped ratio variable speed drives, as those of us who own ten-speed bicycles can attest, there are still other systems that require such fine speed adjustments for best performance that it becomes impractical to string along a multitude of finite gear changes. In this paper we will be concerned with only those of the latter type and principally those drives whose speed ratio can be continuously adjusted independently of load.

Table II summarizes the main types of continuously variable speed drives. The general comments made here regarding variable speed drive selection apply equally well to all the drive types listed. However, only the operational characteristics of the mechanical and fluid drive types will be alluded to since the electrical and hydrostatic types will be covered more thoroughly in other papers contained within this Conference Proceedings.

Variable Speed Drive Selection Factors

Table III lists some of the major design factors to be considered when configuring a drive system. The first item points out the need to have a good working acquaintance with the type of load, and its torque and power requirements throughout its range of operation. Very often the desired speed range of the load in question determines the maximum torque or horsepower which must be supplied and consequently the size of the drive system. Also the speed range or ratio of the driven machine may limit the type of drives available to use. As an example, simple belt, chain and traction drives are traditionally most effective for speed ratios of less than 5 or 6 to 1. They are rarely used by themselves for speed ratios greater than 10 to 1.

The next item, speed regulation denotes the ability of the drive to maintain set speed independently of variations in load torque. As a rule of thumb, the more rigid or positive the drive (for example, a mechanical drive as compared to a fluid drive), the less effect load torque has on drive output speed and consequently the less frequent the control correction.

The control response of a drive is the quickness at which the speed ratio of the drive will change in response to the commands of the speed controller. This may or may not be an important consideration for a particular application. If drive control response and speed regulation are important considerations, then the operational characteristics of the load, motor and control system should be considered as well.

Knowing the efficiency of a drive is not only important from the

standpoint of determining the power that is lost in operating the system but is equally important in sizing the prime mover and determining the amount of cooling that will be required for the drive.

Shock isolation and overload capacity have been mentioned briefly before. It is sufficient to say that these two properties can have a great bearing on drive selection for those systems where component reliability and longevity are a premium.

The remaining four items, reliability, compactness, cost and noise are figures of merit which must be weighed in accordance with the objectives of the given application.

Load Torque Requirements

A common misconception which sometimes causes well-intentioned drive selectors to get gray before their time is that horsepower alone sizes their drive and motor. Many presume that one can simply drive a 100 horsepower load with a 100 horsepower motor, not counting losses of course. This presumption may be true at the rated speed and load but unless the starting torque capabilities of the motor are sufficient, the load may never be driven to its designated operating speed. Of equal significance is that the drive, in accordance with the running torque characteristics of load, must enable the motor to produce sufficient torque at all speeds to accelerate the load to rated speed. What is of cardinal importance is that in order to run a piece of machinery at a given constant speed, the torque being supplied to the machine must be exactly equal to the torque demanded. Secondly, the only way to effect a change in operating speed is to upset this balance. This is essentially the rotary equivalence of Newton's second law. Thus, to do a proper job of drive system selection, the torque requirements of the load must be determined from the point of startup to the point of maximum speed. The motor or prime mover size must be sufficient to meet the most demanding condition.

The three basic load torque categories to be considered are listed in Table IV. They are as follows:

BREAKAWAY TORQUE. Breakaway torque is the torque needed to start the machine from rest. It is almost always greater than the torque required to maintain motion. Breakaway torque is comprised of two parts. The static friction part is the torque necessary to initiate motion of rolling and sliding machine elements. The process demand part is the torque required to overcome a "hard" spot in a machine operation cycle. An example would be the additional torque required to raise a punch press after it has made an impression and has come to rest. Very often flywheels are used to smooth out process demand torques. For most applications it is not necessary to re-

size the motor to meet breakway torques, because the starting torque of an electrical motor is generally appreciably higher than rated running torque. However, for certain types of machinery particularly those having a great number of sliding parts, breakaway torque may be the controlling factor (Table V).

ACCELERATING TORQUE. The torque needed to bring the machine up to operating speed is known as accelerating torque. Generally for most applications the time for the driven machine to reach running speed is not a specification. Usually the motor has sufficient torque to accelerate the load in a reasonably short time period. However, high-inertial machinery (that is, machines with large rotating masses such as flywheels, large fans and bull gears) can cause overheating problems for the drive motor if the motor is forced to operate for prolonged periods of time in a semi-stalled or locked condition. Therefore, in determining the accelerating torque of the motor that is required under these circumstances, it is important to estimate the total inertial load being driven and the maximum permissible starting time. The time a motor can stay on line without overheating during start-up also depends upon the number of start-ups it makes per day. Generally the heat built up by a motor which starts up once every 30 minutes would be dissipated between starts as long as the start-up time is not excessive. For calculation purposes it can be generally assumed that most electrical motors can tolerate a maximum start-up time of 45 seconds. For marginal cases it is best to consult with the manufacturer on this.

The following formula can be used to calculate accelerating torque T_I:

$$T_I = I_p \left(\frac{\omega_f - \omega_i}{\Delta t} \right) \qquad (1)$$

where I_p is polar mass moment of inertia in units of in.lb sec^2, ω_f and ω_i are the final and initial angular velocities, respectively, in units of rad/sec and Δt is the time of acceleration in seconds.

Frequently the inertial load to be driven can be represented by either a cylindrical-rim or flywheel shaped body or a solid disk. The I_p for these bodies can be calculated from the formulas shown in figure 1. In the case of a cylindrical-rim-shaped body whose rim thickness is relatively small, I_p is simply the total mass of the body M times the square of the radius of gyration K where K is approximately equal to the mean radius of the rim R.

Normally when utilizing certain types of variable speed drives

during start-up, the speed ratio across the drive is as large as possible to amplify the starting torque of the motor. In computing the required starting torque of the motor to accelerate the motor from rest to desired operating speed under load, you cannot simply insert into equation (1) the inertial load you have just calculated. This is because the inertial load as viewed from the output side of a drive is never the same as that same inertial load as viewed from the input side. The reason for this is that the kinetic energy, $1/2 \ I_p \omega^2$, on both sides of the drive must be equal and since the shaft on the motor side is turning at a higher speed the equivalent inertial load acting on the motor shaft is proportionately reduced. Thus the dynamically equivalent inertial load I_e, as referred to the motor shaft, is simply the polar mass moment of inertia of the actual load I divided by the square of the reduction ratio R as shown in figure 2.

Motor Characteristics. Having made an estimate of the required breakaway and accelerating torques, the question remains, "what size motor do we need?" The final answer to this question must be deferred until we have examined the load's running torque characteristics but right now we can look at the motor's starting requirements.

Although there are numerable types of prime movers available to choose from for a given application, electrical motors are by far the most commonly used, particularly for stationary applications. The most popular of the a.c. motors in use today are the induction type. The typical operating characteristics of the split phase induction motor sold most frequently is shown in figure 3. This type of motor is designed to operate at essentially constant speed, that is 5 percent slip or less. This means under full load torque a motor with an 1800 rpm synchronous speed will be turning approximately 1710 rpm. If the exact running speed of your load is important, then you must take this into account.

Insofar as starting torque is concerned, the locked rotor torque of this type of motor is nominally 150 percent of full rated torque. It is quite apparent from the current curve in figure 3(b) why it is not advisable to operate for prolonged periods of time in the locked or semi-locked rotor condition. The amount of current drawn by the motor in the locked condition is some six times the nominal rated value. Under these circumstances it would not take long to burn out the motor's windings. The breakdown torque is the maximum torque that can be developed by the motor at rated voltage and frequency. If for some reason this torque is exceeded, there will be a sudden abrupt drop in motor speed. The breakdown torque is typically 200 to 225 percent of full load torque. It occurs at about 70 to 80 percent

of the motor's synchronous speed.

A good example of how a variable slip, variable speed drive such as a fluid coupling or oil shear clutch can help reduce start-up and acceleration torque problems without the ability to multiply motor torque is illustrated in figure 4. The dotted line in figure 4(a) is the torque curve from the typical a.c. induction motor described previously. The superimposed lines represent the output torque-speed characteristics of a fixed-fill fluid coupling. The 100 percent slip line indicates the maximum torque the coupling can accept while the output shaft is stalled. Where this line intersects the motor torque curve indicates the maximum torque available to start the load. It is quite apparent that if the coupling is properly matched to the motor there is up to 50 percent more starting torque available, that is, 225 percent of full load torque with coupling as compared to 150 percent without. Thus, in some instances, variable speed drive elements of this type will permit motor selection to be based on running torque rather than starting load considerations. This results in a reduction in the size of the motor ordinarily required.

A second benefit of variable slip drives is that during extended start-ups, they reduce the current drawn by the motor, some by a factor as great as 1/3. This appreciably reduces the chance of overheating the motor. This is illustrated in figure 4(b) which shows that the current curve of a typical induction motor with a fluid coupling is substantially lower than the current curve without.

The typical running characteristics of fluid couplings is reflected in figure 4(a). At the stall point where the coupling is operating at 100 percent slip all of the motor's output power ends up being dissipated in the drive. For many stop and start systems, the drive's oil cooler must be sized at this point to handle the motor's full power. It can be seen that as the load's speed increases, the amount of slip in both the motor and the coupling decreases since the motor is tending toward rated speed. It is also apparent, that at full rated torque the coupling is at approximately 4 percent slip and thus is transmitting about 96 percent of the input power. Although it is not possible to operate the coupling at 0 percent slip while transmitting torque, a slipping coupling can be used at full load torque and speed without imposing the drastic performance penalties that many suppose.

A variable-fill fluid coupling would have the same characteristics as shown in figure 4(a) when completely filled. As fluid is withdrawn, the angular moment of the fluid which drives the runner or output side of the coupling is reduced. This causes the slip lines on the figure to translate to the right and thereby diminish the amount of torque which the drive can transmit at a given speed ratio.

In this way, torque and speed ratio can be controlled independently making a variable-fill fluid coupling a true adjustable speed drive.

RUNNING TORQUE. Of the various load torque requirements listed in Table IV, the most important is the running torque characteristics of the machine to be driven. A rudimentary understanding of how the machine is expected to behave over its operating range can often spell the difference between a reliable, efficient drive train and one that is substandard. Unfortunately specific published information on the load's operating characteristics over the range of interest is not always available. However, with a little deductive reasoning, a rather good picture of the driven machine's torque and power requirements over its operating spectrum can be formed. Fundamentally, load torque can be categorized as constant torque or variable torque type as will be discussed next.

Constant Torque. Constant torque loads make up the bulk of drive system applications. Constant torque loads are typified by belt, screw or drag-line conveyors; certain types of mixers; jaw and rotary rock crushers; piston and gear pumps and many other types of general machinery. Figure 5 displays the operating characteristics of a constant torque load. The simple belt conveyor used here as an example, illustrates that the difference in tension between the upper and lower strands are independent of the conveyor's speed. Since the product of this tension difference with the pulley radius results in the conveyor's torque, the torque required by the load is also independent of operating speed. This is shown by the dotted line in the plot. The torque required is constant. Since horsepower is directly proportional to torque multiplied by speed, it therefore follows that the required horsepower is also directly proportional to belt speed, being a maximum at maximum speed.

With a mechanical or positive engagement drive, the torque and power to be supplied by the motor will be a maximum at the load's highest speed and will diminish proportionately with lower speed. Keep in mind that due to drive losses, the torque and power which the motor must produce will always be higher than that which the load requires at any given speed.

However, for slipping type drives, such as a variable-fill fluid coupling, the torque to be supplied by the motor will be nearly constant at all speeds. This is because variable slip drives, excluding torque converters, are constant torque devices which do not multiply torque. That is to say, the torque into these drives will be essentially equal to the torque leaving although the power can vary greatly. Because the motor's torque and speed are essentially constant in this case, the power into the drive will also be nearly con-

8

stant and approximately equal to the load's maximum required power. Since the load's required power decreases directly with speed, there is a large difference between input and output drive power at low operating speeds. This represents a sizable power loss for the system which ends up being dissipated by the drive. This is why variable slip drives are rarely used to control speed over wide ranges for constant torque applications.

Variable Torque. Variable torque load requirements are generally more difficult to predict accurately. Basically variable torque loads can be broken down into three types;

(1) Constant horsepower, where hp = constant

(2) Square exponential, where $hp \propto rpm^2$

(3) Cube exponential, where $hp \propto rpm^3$

Figure 6 shows an example of a constant horsepower type load. Some metal cutting and working tools, extruders and mixers fall into this category. The example cited here shows a vertical miller. Its cutting speed would in general vary inversely with cutting depth, hence torque, in such a way that horsepower would remain nearly constant. Obviously associated with constant horsepower type loads is some minimum operating speed. Otherwise at near zero speeds, nearly infinite torques would be required to produce finite horsepower. In practice speed ratios are typically 5 to 1 or less.

With a mechanical or positive engagement variable speed drive, the torque and power to be supplied by the motor will be essentially constant over the operating speed range, disregarding the effect of variable output speed and load on the drive's losses. As a practical matter, the motor is normally sized at minimum load speed since it is likely that drive losses are to be maximum at this point. Using variable slip adjustable speed drives for this type of load will greatly increase the required size of the motor. This is because at minimum load speed, the variable speed drive is at maximum slip since drive input speed is nearly constant. Therefore the drive is absorbing a large percentage of the motor's power. Thus to transmit the required load horsepower at minimum output speed, the drive must receive from the motor many times the required horsepower. Again this is a situation where variable slip drives are impractical.

The operating characteristics of the square exponential and cube exponential loads are shown in figures 7 and 8, respectively. These load types are typically found in those applications in which machinery is called upon to handle fluids or gases. Some mixers, pumps and certain extruders which handle highly viscous fluids or materials

generally fall into the square exponential load category. This is be-
cause in these devices the torque required is proportional to the
shearing stress of the fluid. The shearing stress in turn is propor-
tional to the rate of shearing or speed of operation. Thus the torque
required is directly proportional to speed and the horsepower is di-
rectly proportional to the square of speed as illustrated by the
mixer in figure 7.

The cube exponential load types are characterized by centrifugal
pumps, fans, blowers and other types of turbomachinery where the vis-
cosity of the fluid is relatively low. The fan rotor shown in fig-
ure 8 illustrates the governing parameters for this type of load.
The torque force Q acting on the rotor is a function of the blade's
aerodynamic lift and drag and the angle ϕ at which the blade is set
relative to the axis of rotation. The lift and drag on the blade are
proportional to the density of the fluid ρ, the square of the rotor
diameter d and the square of the relative velocity V_R. The rela-
tive velocity in turn is proportional to the product of rotor rpm N
and rotor diameter d. Since the required torque T is proportional
to the product of the torque force Q with the rotor diameter d it
follows that

$$T \propto \rho d^5 N^2 \tag{2}$$

and

$$HP \propto \rho d^5 N^3 \tag{3}$$

It is apparent that with both the square and cube exponential
load types, the torque and power requirements increase greatly with
speed. This must be taken into account when selecting overall drive
ratio. For example, if you decide to operate a draft fan at twice
its current speed, be prepared to supply and transmit eight times the
horsepower. The most prevalent use of variable slip type drives are
for loads of this type. This is because power and torque require-
ments for these load types are relatively low at low speeds where
variable slip drives are operating in a condition of high slip.

CONCLUDING REMARKS

Putting together a good variable speed drive train requires a systems
approach. Seldom can you select the drive from catalog specifica-
tions alone. You must decide on the normal operating conditions that
the driven machinery will experience, its duty cycle and also any un-
usual conditions such as overload expectancy, high vibration, etc.
From this one can generally estimate the starting, accelerating and
running torque requirements of the given machine over its operating

speed range. Now a choice of adjustable speed drive types can usually be made. Next select the control system to be used. Based on this previous information all that remains is to select and size the prime mover. Finally, go back and look at how all the system elements you have just selected interact. Such a procedure will maximize your chances for a successful drive system application.

BIBLIOGRAPHY

Anon. Power Transmission and Bearing Handbook, Power Transmission Magazine, 1973/74 edition.

Anon. 1971 Mechanical Drive Reference Issue, Machine Design Magazine, Vol. 43, June 17, 1971.

Sheldon, John. "Don't Overlook Hydraulic Variable Speed Drives," Power Transmission Design, June 1971, pp. 31-35.

Wallace, J. H. "Adjustable Speed Drives - The Belt Drive," Power Transmission Design, Feb. 1971, pp. 72-79.

Table I. – Reasons for variable speed drives

PROCESS REQUIREMENT

START UP HIGH INERTIA MACHINERY

ISOLATION OF SHOCK (SOFT START)

OVERLOAD PROTECTION (JAMMING)

CS-66690

Table II. – Major types of variable speed drives

ELECTRICAL

    ADJ. VOLT. D.C. DRIVES

    ADJ. FREQ. A.C. DRIVES

    EDDY-CURRENT CLUTCHES

MECHANICAL

    ADJ. SHEAVE BELT & CHAIN DRIVES

    TRACTION DRIVES

    CLUTCHES

HYDRAULIC/HYDROSTATIC

    POSITIVE DISPL. PUMP-MOTOR DRIVES

FLUID/HYDROKINETIC

    HYDRAULIC COUPLING DRIVES

    TORQUE CONVERTERS

    HYDROVISCOUS DRIVES

CS-66703

Table III. – Drive selection factors

LOAD TYPE, POWER & TORQUE REQ'TS

SPEED RANGE

SPEED REGULATION

CONTROL RESPONSE

EFFICIENCY

SHOCK ISOLATION

OVERLOAD CAPACITY

RELIABILITY

COMPACTNESS

COST

NOISE

CS-66699

Table IV. – Load torque requirements

BREAKAWAY

    STATIC FRICTION

    PROCESS DEMAND

ACCELERATING

    MASS MOMENT OF INERTIA

RUNNING

    CONSTANT TORQUE

    CONSTANT HORSEPOWER

    SQUARE EXPONENTIAL

    CUBE EXPONENTIAL

CS-66700

Table V. – Starting torque characteristics of various machines

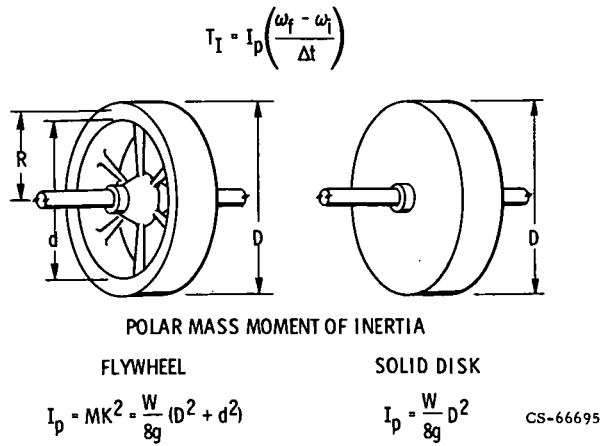| BREAKAWAY TORQUE, % OF RUNNING TORQUE	TYPES OF MACHINE
120 TO 130	GEN. MACH. – ROLLING ELEMENT BRGS
130 TO 160	GEN. MACH. – SLEEVE BRGS
160 TO 250	CONVEYORS & HIGH SLIDING FRICTION MACH.
250 TO 600	MACH. WITH HIGH TORQUE SPOTS IN CYCLE (PRESSES & CAM OR CRANK MECHANISM)
VARIABLE	HIGH INERTIA MACHINERY

CS-66697

$$T_I = I_p\left(\frac{\omega_f - \omega_i}{\Delta t}\right)$$

R

d

D

D

POLAR MASS MOMENT OF INERTIA

FLYWHEEL

$$I_p = MK^2 = \frac{W}{8g}(D^2 + d^2)$$

SOLID DISK

$$I_p = \frac{W}{8g}D^2$$

CS-66695

Figure 1. - Estimating inertial torque.

ACTUAL SYSTEM   DYNAMICALLY EQUIVALENT SYSTEM

N_m

MOTOR   LOAD

N_L

$$R = \frac{N_m}{N_L}$$ I

N_m

MOTOR I_e

$$I_e = \frac{I}{R^2}$$
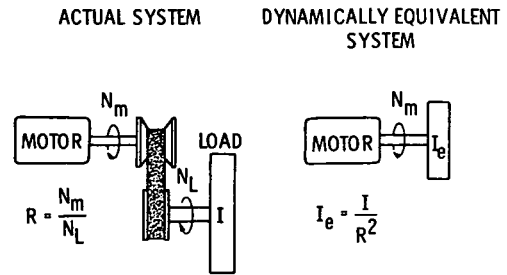
Figure 2. - Dynamic equivalence of inertial load.

CS-66691

LOCKED ROTOR, T

BREAKDOWN, T

PULL UP, T

240
200
160
120
80
40

% OF FULL LOAD TORQUE

FULL-LOAD, T

0 20 40 60 80 100

% OF SYNCHRONOUS SPEED

(a) TYPICAL SPEED-TORQUE CURVE.

600
500
400
300
200
100

% OF FULL LOAD CURRENT, AMP

FULL-LOAD CURRENT

0 20 40 60 80 100

% OF SYNCHRONOUS SPEED

(b) TYPICAL CURRENT CURVE.
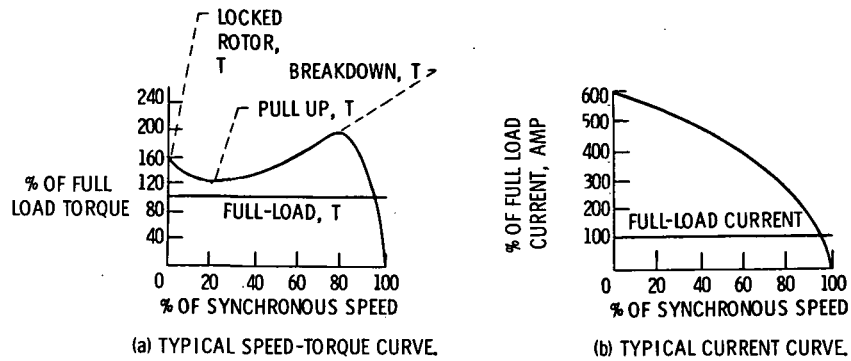
Figure 3. - Characteristics of ac induction motor.

STALL POINT   % SLIP

STARTING TORQUE, % FULL LOAD

200

100   100% SLIP

20
14
8
4
2

0 20 40 60 80 100

COUPLING SPEED, % MOTOR SYNCHRONOUS SPEED

(a) EFFECT ON STARTING TORQUE.

CURRENT DRAW MOTOR ONLY

"LOCK ROTOR"

MOTOR CURRENT, % FULL LOAD

600

400

200

1/3 LOWER CURRENT DRAW

CURRENT DRAW MOTOR WITH COUPLING

0 20 40 60 80 100

MOTOR SPEED, % FULL LOAD

(b) EFFECT ON POWER CONSUMPTION.
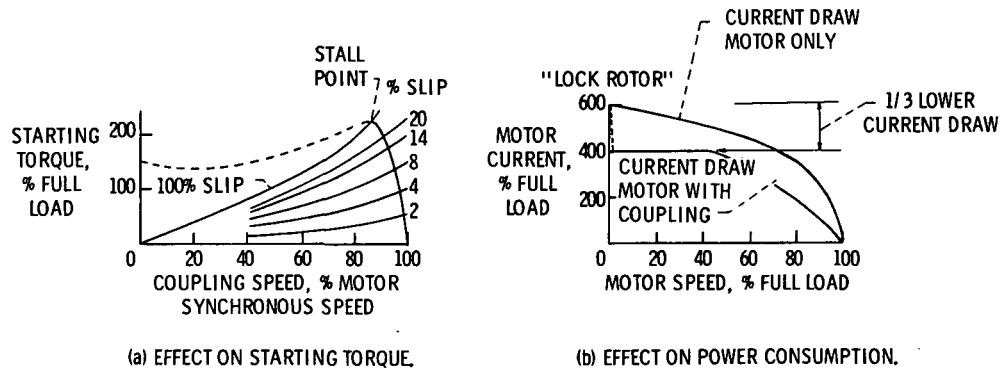
CS-66705

Figure 4. - Effect of fluid coupling on motor start-up.

E-7494

Figure 5

% OF FULL LOAD TORQUE AND HORSEPOWER

% OF FULL SPEED

TORQUE

HP

t_1 = TENSION UPPER
t_2 = TENSION LOWER

CONVEYOR

R   N

$T = (t_1 - t_2) \times R$
$HP \propto TN$

CS-66698
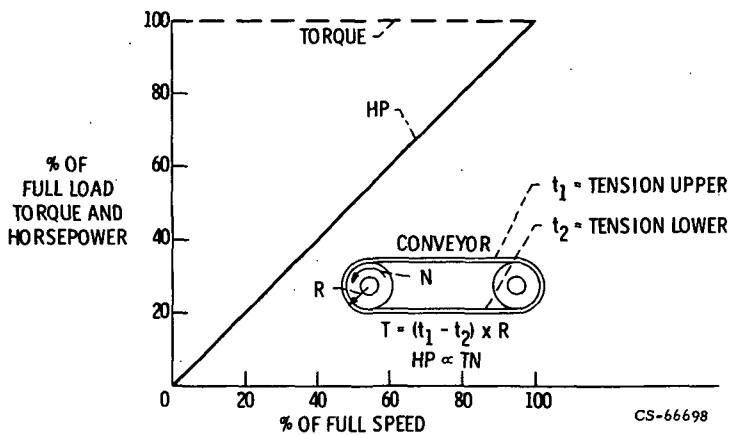
Figure 5. - Running load characteristics.

Figure 6

% OF FULL LOAD TORQUE AND HORSEPOWER

% OF FULL SPEED

HP

TORQUE

MILLING
T, N

HP = CONSTANT

$T \propto \dfrac{1}{N}$

MIN TORQUE →

CS-66693
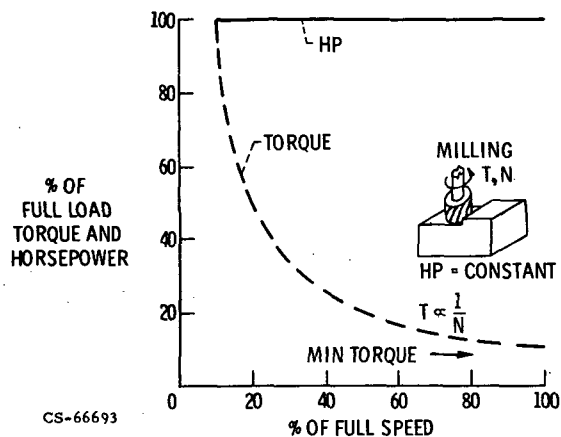
Figure 6. - Running load characteristics.

Figure 7

% OF FULL LOAD TORQUE AND HORSEPOWER

% OF FULL SPEED

MIXER
N, T

$T \propto \tau \propto \mu N$

$HP \propto \mu N^2$

TORQUE

HP

CS-66692
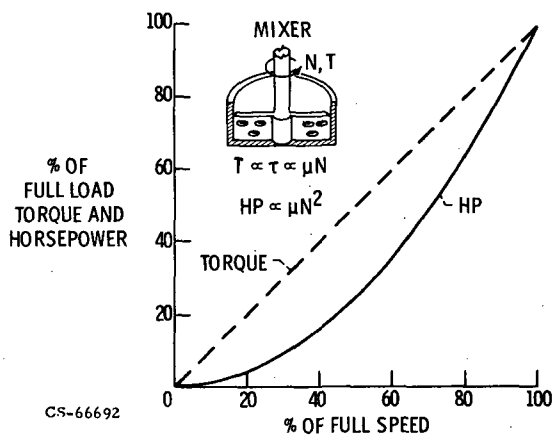
Figure 7. - Running load characteristics.

Figure 8

% OF FULL LOAD TORQUE AND HORSEPOWER

% OF FULL SPEED

FAN
N, T
d

$Q = f(L, D, \varphi) \propto \rho V_R^2 d^2 \propto \rho N^2 d^4$

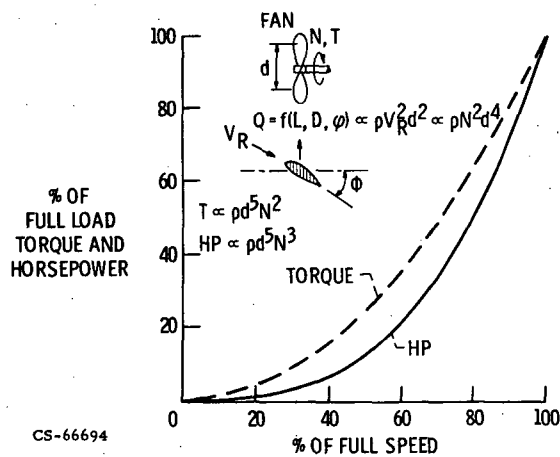V_R ϕ

$T \propto \rho d^5 N^2$
$HP \propto \rho d^5 N^3$

TORQUE

HP

CS-66694

Figure 8. - Running load characteristics.