NO. UMTA-MA-06-0027-73-1

DEVELOPMENT AND TEST OF AN EDDY-CURRENT CLUTCH-PROPULSION SYSTEM

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FINAL REPORT
OCTOBER 1973

Prepared for
DEPARTMENT OF TRANSPORTATION
URBAN MASS TRANSPORTATION ADMINISTRATION
Office of Research, Development, & Demonstrations
Washington DC 20591

Technical Report Documentation Page

1. Report No. UMTA-MA-06-0027-73-1	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle DEVELOPMENT AND TEST OF AN EDDY-CURRENT CLUTCH-PROPULSION SYSTEM.		5. Report Date October 1973
		6. Performing Organization Code
7. Author(s) *G.J. Adams		8. Performing Organization Report No. DOT-TSC-UMTA-73-8.
9. Performing Organization Name and Address Mobility Systems Equipment Company 6151 West Century Boulevard Los Angeles CA 90045		10. Work Unit No. (TRAIS) UM406/R4703
		11. Contract or Grant No. DOT-TSC-357-1
12. Sponsoring Agency Name and Address Department of Transportation Urban Mass Transportation Administration Office of Research, Development, & Demon. Washington DC 20591		13. Type of Report and Period Covered Final Report, Phase 1
		14. Sponsoring Agency Code

15. Supplementary Notes

*Under contract to: Department of Transportation, Transportation Systems Center, Kendall Square, Cambridge MA 02142

16. Abstract

This report covers the Phase 1 effort which is to develop and to test an AC-propulsion system for personal rapid-transit vehicles. This propulsion system incorporates an AC-induction motor in conjunction with an eddy-current clutch and brake.

Also included are development of the propulsion system, fabrication of the propulsion system, description of the laboratory test program, and analysis of the test results.

17. Key Words Eddy-current clutch AC-propulsion system	18. Distribution Statement DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22151.

19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 202	22. Price

Form DOT F 1700.7 (8-72)   Reproduction of completed page authorized

PREFACE

The research, development, and testing studies documented in this report were carried out as a part of the Component Development Program for Personal Rapid-Transit (PRT) Systems, conducted by Mobility Systems and Equipment Company of Los Angeles under contract DOT-TSC-357 for the Transportation Systems Center, Department of Transportation, Cambridge MA, under the auspices of the Urban Mass Transportation Administration, Department of Transportation, Washington DC.

The principal contracting officers are William J. Rhine, Raymond Ehrenbeck, and Joseph D. Abbas. The studies were conducted under the general supervision of George J. Adams, P.E., and Rhinehart Roberg.

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ABBREVIATIONS

a	acceleration
a_{max}	maximum acceleration
a_t	transient acceleration
AC	alternating current
A_2	function of vehicle frontal area
C_o	Aerodynamic drag coefficient
C_R	Coefficient of rolling resistance
d	distance
d_a	distance required for acceleration
d_c	distance traveled at cruise velocity
d_d	distance required for deceleration
d_t	distance traveled between stations (total trip distance)
DC	direct current
e	mechanical efficiency
E	modulus of elasticity
f_b	natural frequency of beam
f_v	natural frequency of vehicle
F_r	Force required to overcome rolling resistance
f_1	natural frequency of bogie
F_a	Force required for acceleration
F_D	Force required to overcome drag
F_G	Force required to overcome grade
F_p	propulsion system tractive forces required
g	gravitational constant
G	slope of grade
G_R	gear box reduction

HP	Horsepower
I	moment of Inertia
I_R	rotational moment of Inertia
K_b	spring constant for beam
K_s	spring constant for suspension
K_v	spring constant for vehicle
K	overall gear reduction constant
K_1	spring constant for bogie
K_2	spring constant for body
L	length of span
m	mass
MPH	miles per hour
MS&E	Mobility Systems & Equipment Co.
P	applied load
PRT	Personal Rapid Transit
r	radius
RPM	Revolutions per minute
S_A	Surface area (projected)
t	time
t_a	acceleration time
t_c	cruise time
t_d	time required for deceleration
T_i	torque input
T_o	torque output
t_t	trip time
T	torque
T_d	dynamic torque

T_s	static torque
T_{DES}	torque desired
T_{REQ}	torque required
V_c	cruise velocity
V_{inst}	instantaneous velocity
V_{max}	maximum velocity
V_1	initial velocity
V_2	final velocity
V_w	relative wind speed
V	velocity
W	vehicle weight
ρ	density of air
Δ	deflection
Δ_1	deflection of bogie suspension

1.0 INTRODUCTION

Studies on research, development, and testing of components for personal rapid-transit (PRT) systems have been conducted for:

a. establishing basic duty-cycle parameters in the selection of power-drive units,

b. testing AC-drive systems with Eddy-current clutch and brake for PRT applications,

c. developing methods for the testing and evaluation of drive systems, including

   1) Torque requirements, and

   2) Horsepower requirements for various actual and simulated PRT duty cycles.

2.0 PROGRAM OBJECTIVES

The work performed under the subject Contract covered the design, fabrication, test and evaluation of propulsion system components basic to Personal Rapid Transit systems.

These include: The prime mover — an AC Drive with Eddy-current clutch and brake; its suspension system, and the delineation of the operational duty cycles in which PRT systems perform.

The work performed under this program was directed toward advancing the State-of-the-Art of propulsion drives and improving the performance of Personal Rapid Transit systems.

The AC propulsion Drive with Eddy-current Clutch and Brake was selected for study and comparison with other types of drives, proposed for or presently used in electrically powered vehicles.

A 15-Horsepower drive AC motor incorporating an air-cooled Eddy-current clutch and brake was designed and fabricated, applying current State-of-the-Art technology in electric drive units.

Control systems were designed and built applying TransMobile Systems technology to ensure a drive performance which met the requirements of a realistic PRT system duty cycle.

Analysis of PRT system duty cycles supported the selection of the parameters used in the simulated versions of operational systems.

Complete drive and bogie assemblies suitable for a PRT monorail configuration vehicle were fabricated.

The AC Drive with Eddy-current clutch and brake was tested under simulated loads in evaluating its response during two PRT system duty cycles; i.e.:

1. A current operational system ("Jetrail", Love Field, Dallas, Texas)

2. An "optimized" system for an advanced Personal Rapid Transit system.

Data recorded during the test runs was studied to establish the Drive Unit's performance during specific phases of a simulated duty cycle.

An analysis of the data produced a comprehensive picture of the dynamics of the Drive Unit, which were then compared with the predicted performance.

The characteristics of:

a. Static AC Drive Units

b. AC Drive with Eddy-current clutch and brake

c. DC Drives

were compared and evaluated.

The conclusions and recommendations resulting from the study are reported herein.

3.0 PRINCIPLES OF EDDY-CURRENT DRIVES

3.1 OPERATION

A relatively simple method of achieving a variable speed drive for PRT Systems applications is by use of eddy-current coupling. An eddy-current drive relies upon an AC motor for power and a low voltage, DC-controlled eddy-current coupling unit for speed control and torque transmission. An eddy-current clutch and brake function are incorporated within an AC motor frame to form the basic drive system. In appearance the AC drive with eddy-current clutch/brake resembles a normal AC motor, except it is longer due to the rotary elements of the eddy-current clutch/brake assembly. The principal components of the drive are shown in Figure 3-1.

Three main functions are performed by the eddy-current coupling. First, speeds from practically zero up to a speed which approaches the maximum speed of the AC motor, can be achieved. Speed control is available to provide predetermined acceleration rates to the driven load or to develop a constant, regulated, torque-speed. The second function of the eddy-current drive is to transmit torque. The third, is to serve as operational brake to a PRT vehicle.

It is obvious that the eddy-current drive obtains all its energy from the AC motor component of the system. The eddy-current clutch merely transmits energy, or torque, to the driven load at regulated speeds.

It is important to realize that the eddy-current clutch is an effective torque transmitter; the torque required by the load is the same torque that appears at the output shaft, across the air gap of the clutch coil and at the AC motor. Thus, the load dictates what torque is developed by the AC motor

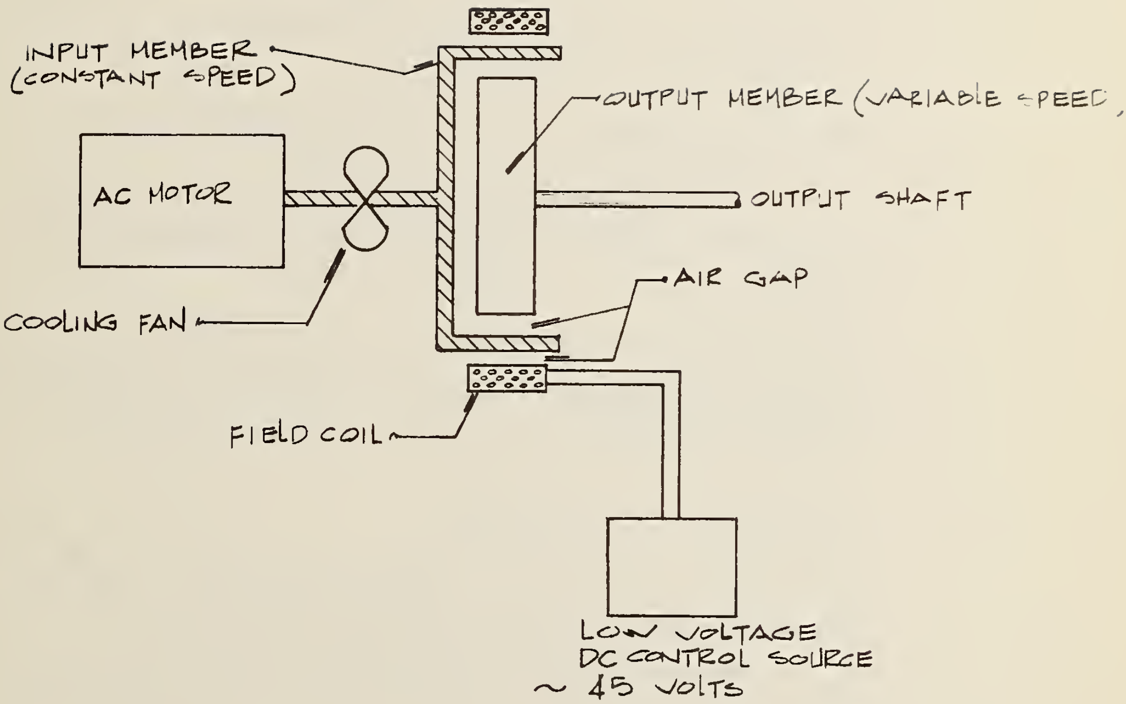


Figure 3-1 EDDY-CURRENT DRIVE COMPONENTS

and transmitted by the clutch. Figure 3-2 illustrates this point. The significance of the illustration is to point out that energy losses occur in the transmission of torque.

Now, since the same torque appears at the motor as required by the load at the output shaft, the difference between the motor input horsepower and the clutch output horsepower is due to the difference in speeds. The general formula for horsepower is given as follows:

$$\text{Horsepower} = \frac{\text{Torque (ft - lbs)} \times \text{Speed (RPM)}}{5252} \quad \text{and}$$

$$\text{Horsepower Motor} = \frac{\text{Torque (load)} \times \text{Speed (motor)}}{5252} \quad \text{and}$$

$$\text{Horsepower Load} = \frac{\text{Torque (load)} \times \text{Speed (output shaft)}}{5252}$$

The difference of the horsepower of the motor and the horsepower of the load is the loss in the clutch and is dissipated as heat in the clutch.

3.1.1 Temperature Characteristics

The eddy-current coupling, being a slip device, must of necessity reject the slip power loss in the form of heat. The magnitude of the slip loss for any speed and load point is calculated by the following formula:

$$\text{Input Horsepower} = \text{Output Horsepower} + \text{Losses (horsepower)}$$

3.1.2 Speed Range

At this point it should be apparent that if there is appreciable slip in a unit, there will be horsepower dissipated

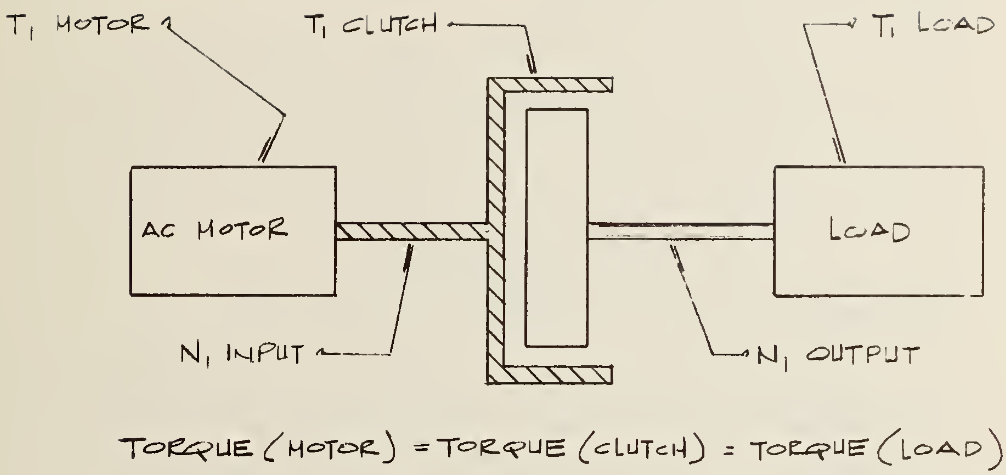


Figure 3-2 EDDY-CURRENT TORQUE TRANSMISSION

as heat in the clutch. If the minimum output shaft speed is rated as something over 100 RPM, then it is because the fan action of the air-cooled unit as designed dissipates only a limited amount of the heat loss that would be developed at lower speeds. This minimum speed due to heat dissipation capacity is based on rated torque and will, of course, decrease the load capability at the lower speeds.

3.1.3 Clutch Torque Transmission

Figure 3-3 shows a typical eddy-current coupling torque curve with rated excitation and additional curves with reduced excitation. Note that as the output speed approaches that of the input member (100%) then the slip decreases and the output torque decreases for a given excitation.

3.1.4 Relationship of Motor and Clutch Characteristics

The characteristic curve of an AC induction motor and an eddy-current clutch have been superimposed on each other in Figure 3-4. Since the AC motor is up to speed before the clutch is energized, it is possible to use the breakdown torque of the AC motor for starting. For a given torque at various output speeds, the AC motor would operate at the same point on its speed/torque curve. For instance, points A and B represent two operating points where the output torque are the same but at different output speeds. In each case, the AC motor would operate at point C to provide the required torque.

3.1.5 Non-Regulated Controls

A non-regulated control would be one where a fixed excitation would be used for the DC clutch field. This excitation would be independent of load and speed. In figure 3-5, the

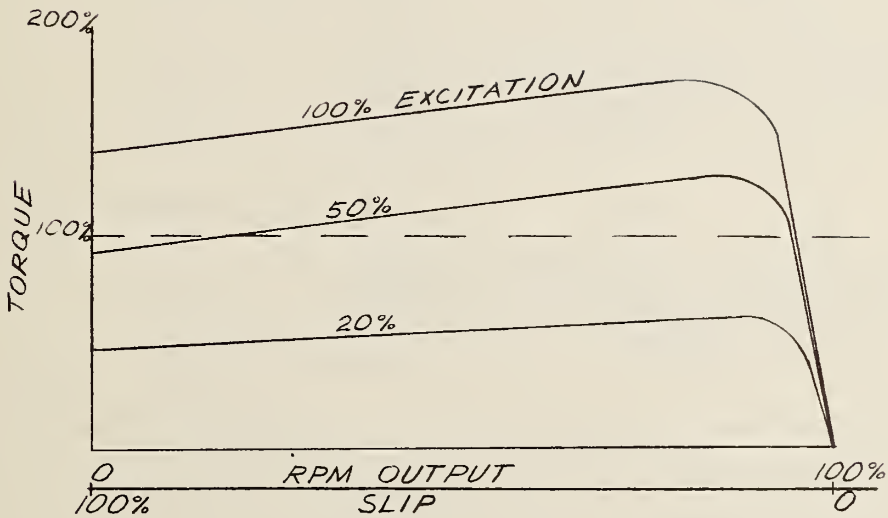


Figure 3-3 CLUTCH EXCITATION TORQUE CURVE

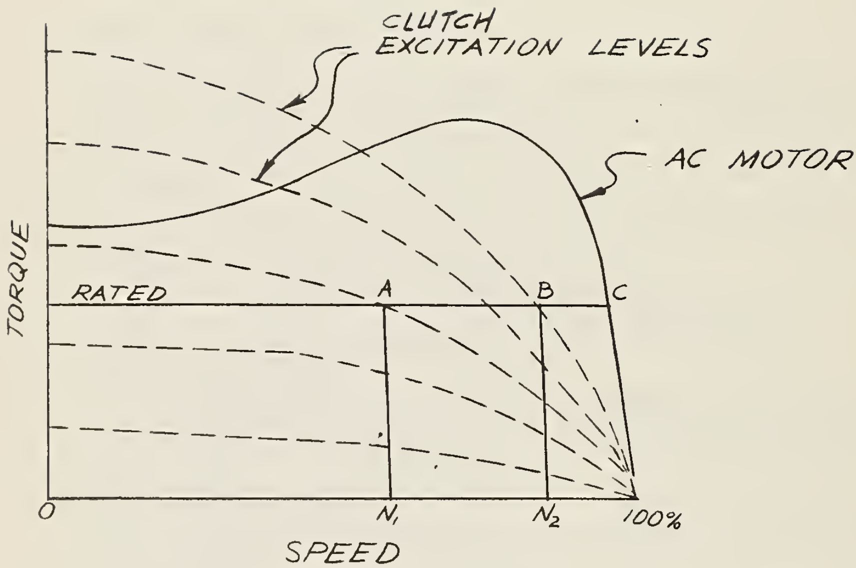


Figure 3-4 . MOTOR AND EDDY-CURRENT CLUTCH CHARACTERISTICS speed/torque characteristics is used to illustrate the clutch operation, with a constant level of excitation and a changing load. For a fixed excitation "A", the drive would operate at a speed of N_1 and a torque of T_1. If the torque were increased to T_2, then the drive would slip back to speed N_2 to maintain this torque.

3.1.6 Regulated Controls

A regulated control would be one where the coupling would be used as a speed regulated drive. It would be necessary to automatically adjust the control excitation to maintain a given output speed. Thus, if the speed were set at N_1 for a torque of T_1, the regulated control would automatically increase its excitation output to maintain the same relative speed for an increase in torque to T_2.

3.1.7 Speed Regulation

Figure 3-6 illustrates a typical speed regulated drive using an eddy-current coupling; the tachometer generator on the output shaft of the drive is used as the speed sensing device.

Because of the extremely small excitation requirements of the eddy-current drive, this type of drive lends itself very readily to electronic control, using solid state converters as the power amplifier for the speed regulating control system.

With the solid state types of controls, precise and smooth speed regulation can be obtained; accuracies of better than 1/10% of top speed can be realized. Features such as controlled rate of acceleration, torque limitation, speed matching and many others can readily be built into the control.

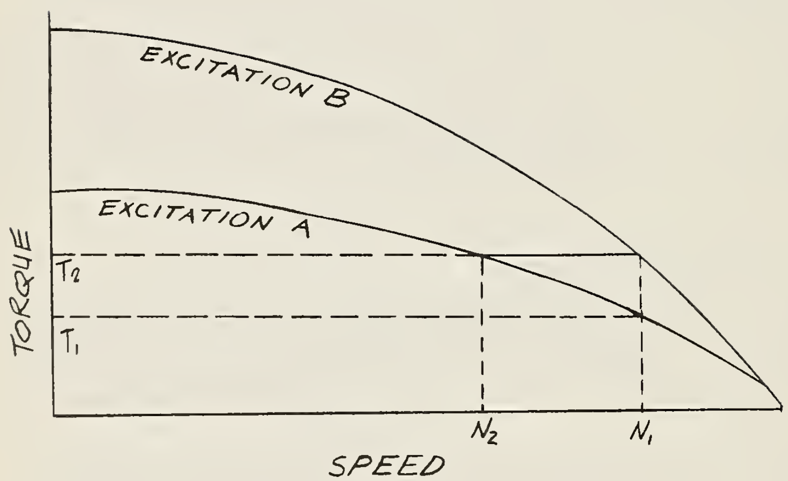


Figure 3-5 CONSTANT EXCITATION LEVEL ON CLUTCH FIELD

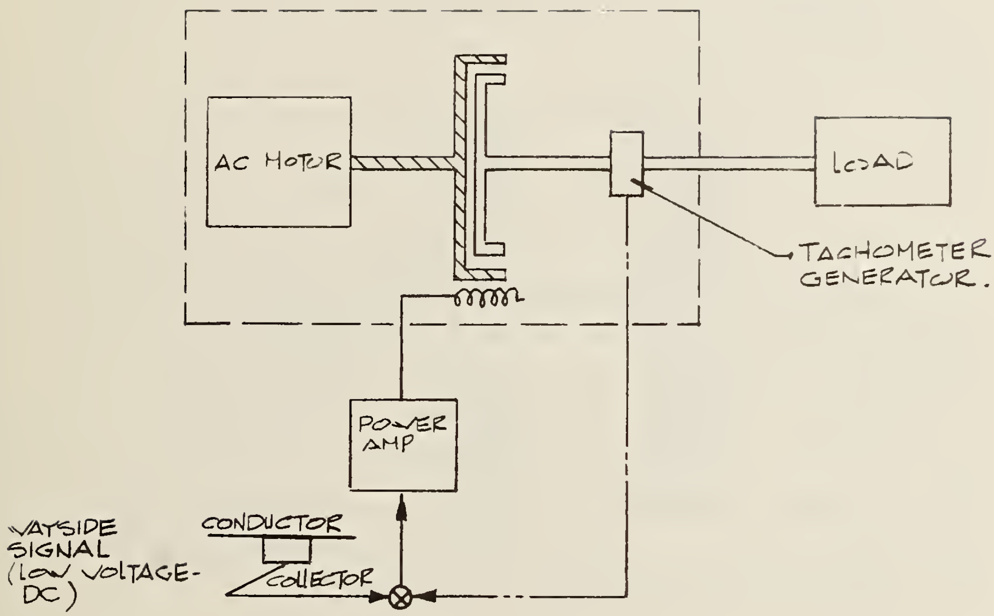


Figure 3-6 SPEED REGULATED EDDY-CURRENT DRIVE

3.1.8 Eddy-Current Brake

Eddy-current braking is an effective means of vehicle braking because it provides smooth and reliable, controlled braking without fading. In an eddy-current brake the input member is stationary becoming part of the magnetic clutch housing. When the output member is stopped, there is no speed difference (slip) between the two members. But, as the speed of the output member increases, the slip between the two members increases proportionately. Thus, the slip is equal to the output speed for an eddy-current brake.

Figure 3-7 shows a combination clutch/brake stationary field magnetic drive. Figure 3-8 illustrates the characteristic curve of an eddy-current brake. It will be observed that the greatest braking torque is obtained where the output speed is the fastest and where the greatest torque would be normally desired. On the other hand, very little torque is developed at low speeds and, therefore, an eddy-current brake cannot be used as a holding brake. A mechanical brake is always required in combination with the eddy-current brake. However, the mechanical brake is only used during the final braking period, when speed decreases from approximately 10% of maximum speed to a full stop.

3.1.9 Speed Regulated - Eddy-Current Clutch/Brake Combination Drive

The eddy-current brake in combination with the eddy-current clutch provides a variable speed drive with superior responses and controlled acceleration-deceleration. A typical control scheme is shown in Figure 3-9. The presence of the error signal inverter in the circuit insures that only one of the two power amplifiers can be turned "ON" at any one time.

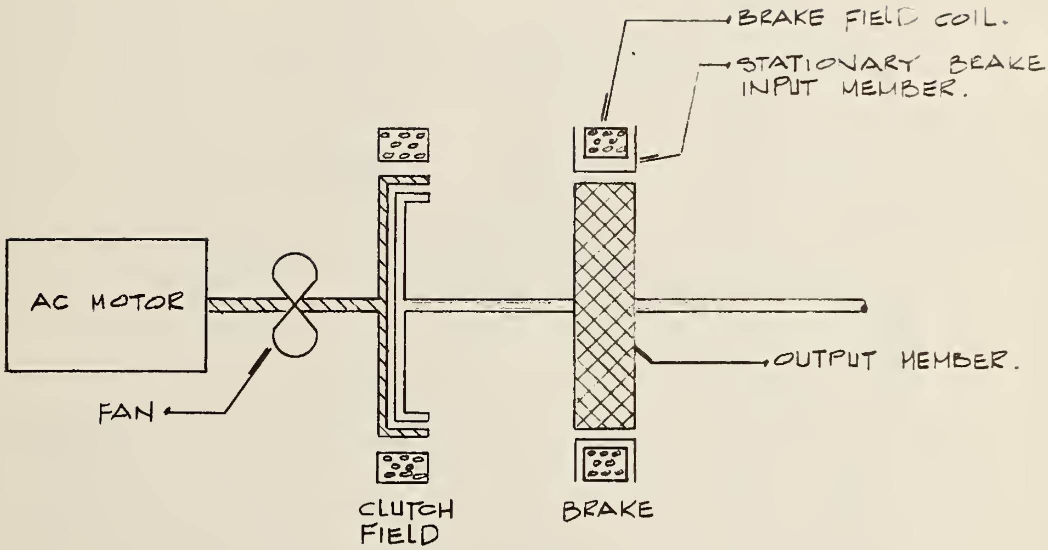


Figure 3-7 STATIONARY FIELD
EDDY-CURRENT CLUTCH AND BRAKE

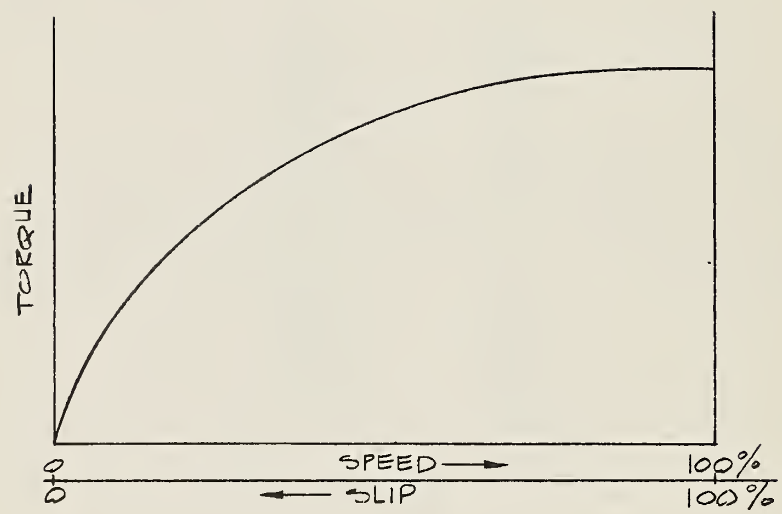


Figure 3-8 EDDY-CURRENT BRAKE PERFORMANCE

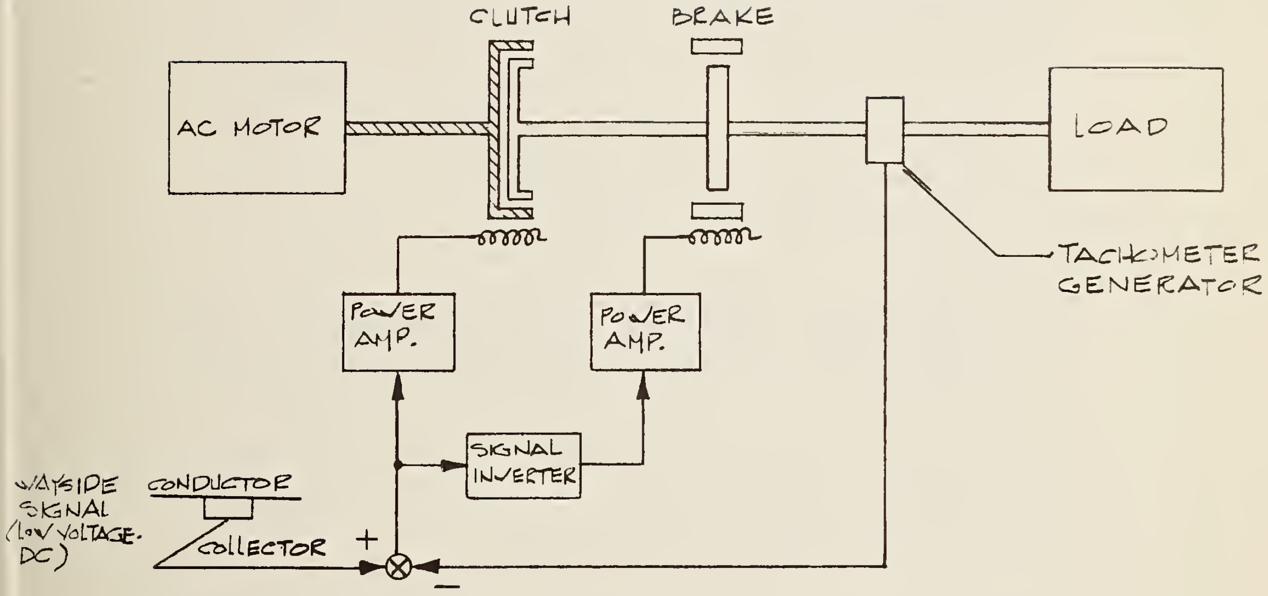


Figure 3-9 TYPICAL SPEED REGULATED EDDY-CURRENT CLUTCH/BRAKE DRIVE

3.1.10 Braking Modes

Since eddy-current braking effort is proportional to the output shaft speed, it becomes ineffective at low vehicle speeds, typically at speeds below 2 miles per hour. Therefore, it cannot be used as a holding or emergency brake. This function is performed by a mechanical brake usually of the disc type operating on the drive wheels. In the service braking mode the disc brake operates when the vehicle speed is such that eddy-current braking is inadequate. In any emergency situation the disc brake operates immediately and brakes the vehicle at a rate consistent with the vehicle design and passenger safety.

3.2 SUMMARY

Mobility Systems & Equipment Company's approach to eddy-current drive for PRT vehicle application transmits torque from an AC motor to an output shaft without mechanical linkage with smooth variable control and rapid response. It is a simple and effective method of providing controlled speed from a constant reference source.

The eddy-current brake in the system provides smooth controlled deceleration and responds to speed control requirements of a PRT vehicle during downhill operation.

Because the drive unit AC motor is normally up to speed before torque is transmitted to the output shaft, the eddy-current clutch permits utilization of maximum drive torque during starting and acceleration.

Unlike a DC drive, the eddy-current AC motor portion of the drive is always at maximum speed, even during dwell periods when no load and very little power is demanded. The continuous operation of the AC motor results in the internal fans providing continuous cooling to the system. This extends the range of operation of this drive over periods of very high torque followed by slow speeds or dwell, without an excessive build-up of internal heating.

If required, a torque limit function can be included in the system such that if torque demands are above a predetermined maximum value the eddy-current drive will increase its slip while maintaining the maximum torque value.

3.3 APPLICATIONS

The relative simplicity, smoothness and ease of operation, accuracy of speed control, high reliability and low maintenance of the AC Eddy-Current Drive in combination with MS&E Company's controls approach resulted in a favorable acceptance of this type of drive for PRT applications.

4.0 DUTY CYCLE

The objectives of the Duty Cycle evaluation are to analyze the basic drive system for a PRT system and establish a representative sequence of operation which can be used to perform laboratory tests of the propulsion drive unit. A duty cycle for a PRT system describes the power or energy requirements necessary to propel the vehicle on its defined route. The duty cycle encompasses all modes of vehicle operation including dwell, acceleration, route grades, all speed levels as well as deceleration and braking.

By examining the duty cycle in combination with passenger loads and vehicle characteristics, it is possible to calculate the power or energy requirement of the drive system. This type of calculation is of great importance for a PRT system because it results in an optimized system with minimum equipment costs and the lowest system operating costs.

4.1 Analogous Operational PRT Systems

In order to give perspective to the system duty cycle being studied, two existing PRT systems were examined. These are the Jetrail system in operation by Braniff Airlines at Love Field in Dallas, Texas and the Los Angeles County Fair system in operation at the Pomona, California fairgrounds. Both of these systems are suspended monorails using AC electric power drive units.

The duty cycle of the Jetrail was reconstructed and tested to show the validity of the duty cycle calculation technique. Then, a higher performance vehicle system was devised for higher speeds, in order to show an optimized matching of a drive unit to a PRT vehicle system.

A brief description and explanation of the two existing PRT systems is given.

Jetrail was constructed in response to a need for transporting airline passengers and baggage from a remote parking area to the airline ticket and boarding area as well as for the return trip from the terminal building to the parking area. Jetrail is a free service provided by Braniff Airlines. The service uses small cars with a capacity of 6 to 8 seated passengers plus baggage.

The system is completely automatic in both dispatching cars and the operation of switches for by-pass spurs. Passengers call for a car by pressing an elevator style button at the loading door. On entering the car there is no operator or complex control consoles. A destination is selected by a simple push button which automatically actuates the entire trip using by-pass spurs as required.

Since utilization of the service is related to airline arrival and departure schedules, the cars are dispatched automatically, on either a continuous flow mode for peak utilization or in a demand mode for lower utilization. Maximum speed is 15 miles per hour.

The Pomona system operates annually during the Los Angeles County Fair season. It is configured for a manual dispatch operation but moves automatically on the route under a "safety block" control system. Its purpose is for amusement. For a small charge passengers can enjoy a leisurely, elevated ride to view the major attractions of the fairgrounds. Each car seats up to 28 passengers and travels a return loop over a distance of one mile.

The vehicle in the optimized system in this study is comparable in weight to the Jetrail system; however, it is designed for speeds of 20 to 30 miles per hour at grades of up to 3%.

4.2 Methodology for Developing a PRT Duty Cycle

Development of a duty cycle for a PRT system entails the application of the basic laws of force to the operational modes of the system. Of first importance then, is a definition of the various modes of operation including system specifications.

Since PRTs usually operate on a separated grade and exclusive right-of-ways, the vehicle varies only within the limits of installation as to the range of service. The established grades and curves encountered and the number of stops on the route are such that actual vehicle operation may be easily programmed. Upon determination of these basic programming modes, a complete duty cycle of operation would be known and the design of the vehicle may begin.

Basic programming may be broken down into the following modes: Travel Mode; Dwell Mode.

4.2.1 Travel Mode

The Travel Mode is the phase in which the vehicle is in actual transit from one station to another. This Travel Mode consists of three independent components: Acceleration Phase; Cruise Phase; Deceleration Phase.

To complete the Travel Mode in terms of drive unit duty cycle, the following parameters must be known: Maximum acceptable rate of acceleration; maximum acceptable rate of deceleration; velocity and distance between stations. From these known quantities, the length of the acceleration, cruise and deceleration phases may be determined.

Passenger safety and comfort limit the design acceleration rates to generally less than .1g laterally for transport vehicle operation and range between .2g and .3g in emergency situations. For computation of selected vehicle systems, a rate of 2.2 ft/sec^2 (.068g) for acceleration and a rate of 3.2 ft/sec^2 (.1g) for deceleration have been designated, although other values are also used. These figures are comparable with other systems studied and within the standards of the industry.

4.2.2 Computation of Travel Mode Performance Parameters

The format for computation of the basic Travel Mode for a system with known parameters may easily be shown by a simple example:

Travel Mode Computation Sample:

Consider a non-stop trip of 2,000 feet length (d_t):

Rate of acceleration:	2.2 ft/sec^2
Rate of deceleration:	3.2 ft/sec^2
Cruise Velocity:	35 MPH = 51.3 ft/sec

(a) Determine acceleration time to cruise velocity:

1. Time to accelerate from rest to 51.3 ft. second = t_a

$$t_a = \frac{v_2 - v_1}{a} = \frac{51.3 - 0}{2.2}$$

$$t_a = 23.3 \text{ sec}$$

t = Time

a = Acceleration Deceleration

v_1 = Initial Velocity

v_2 = Final Velocity

d_t = Total Trip Distance

2. Time to decelerate from 51.3 ft. per second to rest = t_d

$$t_d = \frac{v_2 - v_1}{a} = \frac{0 - 51.3}{-3.2}$$

$$t_d = 16.0 \text{ sec}$$

3. Total distance traveled during acceleration = d_a

$$d_a = \frac{at^2}{2} = \frac{2.2 \times (23.3)^2}{2}$$

$$d_a = 597.1 \text{ ft}$$

4. Total distance traveled during deceleration = d_d

$$d_d = \frac{at^2}{2} = \frac{3.2 \times (16)^2}{2}$$

$$d_d = 409.6 \text{ ft}$$

5. The distance traveled at cruise velocity = d_c

$$d_c = d_t - d_a - d_d = 2000 - 597.1 - 409.6$$

$$d_c = 993.3 \text{ ft}$$

6. The time traveled at cruise velocity = t_c

$$t_c = \frac{d_c}{v_2} = \frac{993.3}{51.3}$$

$$t_c = 19.3 \text{ sec}$$

7. Total trip time = t_t

$$t_t = t_a + t_c + t_d = 23.3 + 19.3 + 16.0$$

$$t_t = 58.6 \text{ sec}$$

4.2.3 Dwell Mode

The second basic component of the duty cycle is the Dwell Mode (loading and unloading) phase which includes times for door operations and passenger entrance and exit. Experience with the Jetrail installation, and other systems, as well as studies of automatic elevators and rapid transit systems, has provided data on which door operation times may be based.

The time for door opening ranges from 1.0 to 2.5 seconds and the time for door closing from 2.5 to 3.5 seconds. Total door operation of 5 seconds would be acceptable as an average operating time.

4.2.4 <u>Duty Cycle Data Preparation Approach</u>

To establish vehicle power requirements basic parameters were identified. The standard form for the organization of these parameters is as follows:

Vehicle System (Name): ______________________

Design Speeds: ______________________

Maximum Number of Passengers: ______________

Drive Unit: ______________

Maximum Acceleration: ______________

Maximum Deceleration: ______________

Gross Vehicle Weight: ______________

Car Weight: ______________

Drive System Weight: ______________

Curb Vehicle Weight: ______________

Max. Passenger Load: ______________
(___@ 150 lb. each)

Baggage: ______________

Car Frontal Surface Area: ______________

Drive Wheel Radius: ______________

Max. Electro-Drive Speed: ______________

Drive Train Efficiency:	________________
System Grades:	________________
Physical Coefficients:	________________
Coefficient of Rolling Friction:	________________
Aerodynamic Drag Coefficient:	________________

4.2.5 Vehicle Forces Factors

For a defined vehicle system, the drive system performance may be determined by examining the various force requirements during the travel mode of the duty cycle. As previously defined, the travel mode includes acceleration, deceleration, a speed profile, distance traveled and grade characteristics.

The vehicle as a point mass moving along a surface may be analyzed by means of Newton's Second Law of Motion:

$$F = m \frac{dv}{dt}$$

or

$$F_p - F_r = m \frac{dv}{dt}$$

where:

F_p = propulsion system tractive forces

F_r = the resistive forces on the vehicle

Application of Newton's law therefore requires a definition and analysis of the propulsion system, the resistive forces and the system acceleration.

Analytically the above expression is correct, when F_p and F_r are simple formulations for numerical solutions.

The difficulty of applying Newton's Second Law of Motion is in establishing the tractive properties of the propulsion system and resistance characteristics of the vehicle.

The calculated acceleration speed time curve and maximum speed are just as accurate as the estimate of tractive effort and resistance. Consequently, the approach taken to determine the performance characteristics of a propelling system -- AC drive unit with eddy-current clutch and brake -- was to establish the best approximation to the actual duty cycle to be simulated.

The propulsion system provides a tractive force, F_p, to overcome resistive forces in the system and for vehicle acceleration. The analytical form for expressing the propulsion system properties is:

$$F_p = F_r + m \frac{dv}{dt}$$

4.2.6 Acceleration Forces

The propulsive force (Fa) for accelerating the vehicle is:

$$Fa = f_a + m \frac{dv}{dt}$$

Since the acceleration forces are a function of vehicle mass and acceleration they cannot be directly dealt with in a laboratory test program. These acceleration forces are therefore treated by the attachment of an inertia flywheel to the drive unit. By proper choice of flywheel characteristics the mass properties of the vehicle are simulated during acceleration-deceleration phases.

4.2.6.1 Inertia Mass Simulation

The method of relating acceleration energy of the vehicle to the acceleration of an inertia mass in a flywheel is developed in the following equations. The inertia mass of the flywheel is calculated through energy considerations. We know that kinetic energy of the vehicle at the end of the acceleration phase is:

$$KE = \frac{W v_2^2}{2g} \tag{4-1}$$

where:

W = vehicle weight (pounds)
v_2 = vehicle velocity at the end of acceleration
g = gravity constant = 32.2 ft/sec^2

For a rotary mass the kinetic energy can be closely approximately by:

$$KE = 0.00017\ I_R(RPM)^2 \tag{4-2}$$

where:

I_R = the total rotational inertia (ft - lb - sec^2)
RPM = speed at the end of acceleration

Equating the kinetic energy of the vehicle and the kinetic energy of the rotating inertia mass enables a calculation of the inertia mass of the flywheel for various weights and speeds of vehicles to be made

$$0.00017\ I(RPM)^2 = \frac{W v^2}{2g}$$

or (4-3)

$$I = \frac{W v^2}{0.00034(RPM)^2 g}$$

Paragraph 7.3.2 provides additional details on the selection of the inertia flywheel.

In studying other transportation systems it is found that vehicle acceleration seldom remains constant and is often represented by the following approximation

$$\frac{dv}{dt} \approx a = a_{max} \left[1 - \left(\frac{v_{inst}}{v_{max}} \right)^{10} \right]$$

where:

a_{max} = maximum constant acceleration rate

v_{max} = maximum velocity

v_{inst} = instantaneous vehicle velocity

This approximation was therefore evaluated for the acceleration phase to aid in determining the compatibility between the drive unit and the duty cycle to be tested.

It is noted that the formulation does result in a relatively high initial "jerk" and a smooth termination of acceleration.

It is further noted that an eddy-current drive has an inherent soft start, the capability for near constant acceleration levels and a soft letup on acceleration as maximum speed is reached.

4.2.6.2 Calculation of Acceleration

Even though acceleration forces are simulated by use of an inertia flywheel it is necessary to calculate the acceleration forces in order to select a drive unit with ample capabilities to meet all the force requirements.

Two methods were used. The obvious method is to assume that acceleration is a constant value and to compute the propulsion requirement for acceleration as

$$F_p = \frac{W}{g} a$$

while this method is useful in determining system requirements it is obvious that a practical system could not tolerate the initial system "jerk" as acceleration commences nor the final system "jerk" as acceleration is removed.

4.2.7 Resistive Forces

The other forces which the propulsion system must overcome are resistive forces associated with the system design which consist of: Rolling resistance, aerodynamic drag, and grade resistance.

The development of these forces are as follows:

(a) Rolling resistance F_r is the force required to keep a vehicle of weight W, rolling; i.e., to overcome mechanical friction. This may be represented by:

$$F_r = W \times C_R$$

where C_R is the coefficient of rolling function.

(b) Aerodynamic drag F_D, is the force required to overcome wind resistance on the vehicle. This may be represented by:

$$F_D = 1/2 \, \rho v_w^2 \, C_D S_A$$

where:

ρ = air density

v_W = wind speed

C_D = drag coefficient of the vehicle

S_A = surface area of the vehicle (projected)

(c) Grade resistance, F_G, is the force required to overcome slope of grades.

$$F_G = W \times \sin\theta$$

if % of grades is less than 10%, then

$$\sin\theta = \frac{\%\text{ of Grade}}{100} = G$$

therefore .

$$F_G = W \times G$$

Total propulsive tractive forces are obtained by the summation of the various resistance and propulsion force components acting on a vehicle.. This is represented as:

$$F_p = F_r + F_D + F_G + F_a$$

4.2.8 <u>Wheel Torque</u>

Force developed in the previous paragraph represents the linear forces acting on the vehicle. Linear forces are converted to torque, T_W, the rotary unit of energy, by considerations of the radius of the drive wheel r, or:

$$T_w = F_p \times r$$

4.2.9 Gear Box Reduction

Knowing torque at the drive wheel it is necessary to consider gear reduction losses and efficiency losses in the transmission of torque between the drive wheels and the output shaft of the propulsion drive system.

4.2.9.1 Drive Wheel RPM

Maximum system velocity dictates the maximum RPM of the drive wheel.

Since the drive wheel travels a distance:

$$d = 2\pi r \times \text{RPM in one minute}$$

and also

$$d = v \text{ (linear speed)} \times 60 \text{ in one minute then:}$$

$$\text{Drive Wheel RPM} = \frac{V \times 60}{2\pi r}$$

If r = ft and v = mph then:

$$\text{Drive Wheel RPM} = \frac{v \text{ (mph)}}{r \text{ (ft)}} \times 14$$

4.2.9.2 Maximum Electro-Drive

The maximum electro-drive of an AC Drive unit is the limiting speed value for efficient motor operation. This value may be obtained upon the examination of the torque-speed curve for that particular unit. Figure 4-1 is the torque-speed curve for Mobility Systems and Equipment Company's 15 HP, AC drive with eddy-current clutch and brake, model PM-OO1MSE. It is seen from this figure that 1,610 rpm may be considered as a relatively efficient

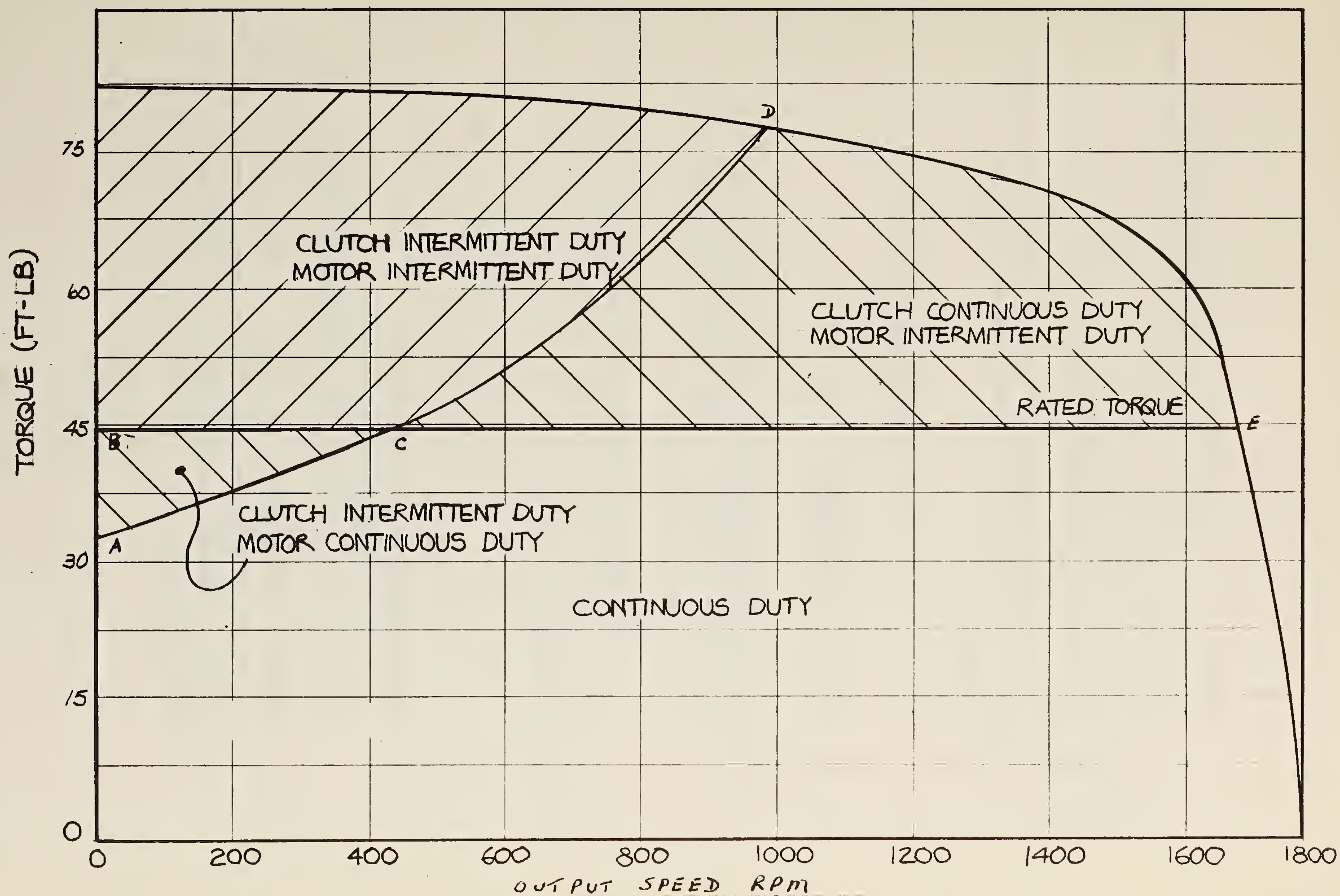


Figure 4-1 PM-001 MSE DRIVE UNIT TORQUE-SPEED CURVE

maximum electro-drive speed. The Jetrail installation uses two 7.5 horsepower drive units, for these units, design rpm speeds are 1,450. (See appendix A.)

4.2.9.3 Mechanical Gear Box Reduction

The actual gear box reduction, G_R, is found by dividing the maximum electro-drive speed by the drive wheel rpm at maximum speed:

$$G_R = \frac{\text{maximum rpm of electro-drive}}{\text{drive wheel rpm}}$$

4.2.10 Mechanical Efficiency

The total drive efficiency of a vehicle incorporates the mechanical efficiency of the entire drive train and spin losses in the gear box. Values of mechanical efficiency range between 75% to 95%.

The actual torque required for vehicle propulsion varies directly with the desired torque, and, inversely, with mechanical efficiency:

$$T_{req} = \frac{T_{des}}{e}$$

4.2.11 Input Torque Required at Gear Box

Input torque at the gear box T_i is supplied by the electro-drive output shaft. Torque required at the drive wheel, T_w, to propel the vehicle was computed in Section 4.2.8. Torque required at the gear box input requires consideration of the gear box reduction and the mechanical efficiency and is computed as follows:

$$T_i = \frac{T_w}{e \times r}$$

4.2.12 Electro-Drive Horsepower Output

Propelling power is a function of the torque and speed (RPM) of the electro-drive output shafts; therefore, for any torque requirement the output speed of the electro-drive output shaft must also be known. For maximum vehicle velocity, steady state operation, this value would be that of the operating maximum electro-drive speed.

Horsepower output may be calculated by:

$$HP = \frac{T_o \times RPM}{5255}$$

Power performance design requirements for the Jetrail and the optimized systems were calculated applying the above format.

4.3 STUDY VEHICLES - SYSTEM DEFINITION

The duty cycle for two systems were calculated and studied. The first system is the existing Jetrail system in its current operational mode at Love Field - Dallas, Texas. Jetrail was selected in order to have a verifiable baseline for comparison. The second system studied is called the Optimized System. It is given this name because it represents maximum vehicle performance for the selected propulsion system with an eddy-current clutch/brake.

In order to calculate the necessary force and energy requirements of the drive unit it is necessary to define both the vehicle and the vehicle's route. For the Jetrail, both the vehicle system and route are well defined (see Paragraph 4.3.1 and Figure 4-2).

4.3.1 Parameters of Jetrail System

Operational and performance parameters of the Jetrail system are as follows:

Vehicle System:	Jetrail
Design Speed	15 mph = 22 ft/sec
Maximum Number of Passengers:	10
Drive Unit:	Two 7.5 HP AC motors with eddy-current clutch/ brake
Max. Acceleration:	1.2 ft/sec^2
Max. Deceleration:	1.8 ft/sec^2
Gross Vehicle Weight:	8,000 lbs
Car Weight:	4,900 lbs
Drive system weight:	1,100 lbs
Curb vehicle weight:	6,000 lbs
Max. Passenger load: (10 @ 150 lbs each)	1,500 lbs
Baggage:	500 lbs
Car Frontal Surface Area:	50 sq ft
Drive Wheel Radius	6 inches = 0.5 ft
Max. Electro-Drive Speed:	1,450 RPM
Drive Train Efficiency:	85%
Gear Reduction Ratio:	3.45:1
System Grades:	<1%
Physical Coefficients:	
Coefficient of Rolling Friction	0.02
Aerodynamic Drag Coefficient	0.52

4.3.1.1 Route of Jetrail System

Figure 4-2 describes the route of the Jetrail system between the Braniff Airlines terminal building and parking area at Love Field - Dallas, Texas.

4.3.1.2 Resistive Forces of Jetrail System

The following data provides calculations pertaining to the Jetrail system.

(a) Rolling resistance:

$$F_r = \text{vehicle weight x coefficient of rolling friction}$$

$$F_r = 8,000 \times 0.02 = 160 \text{ lbs.}$$

(b) Aerodynamic drag:

$$F_D = 1/2\ \rho v_w^2\ C_o\ S_A$$

$$F_D = 1/2 \times 0.024 \times (22)^2 \times 0.052 \times 50$$

$$F_D = 15.1 \text{ lb.}$$

(c) Grade resistance - the system operates on level ground

$$F_G = 0$$

(d) Acceleration (1.5 ft/sec^2 acceleration):

$$F_a = \frac{8,000}{32.2} \times 1.5 \left[1 - \left(\frac{V}{V_{max}}\right)^{10}\right]$$

$$F_a = 372 \left[1 - \left(\frac{V}{V_{max}}\right)^{10}\right]$$

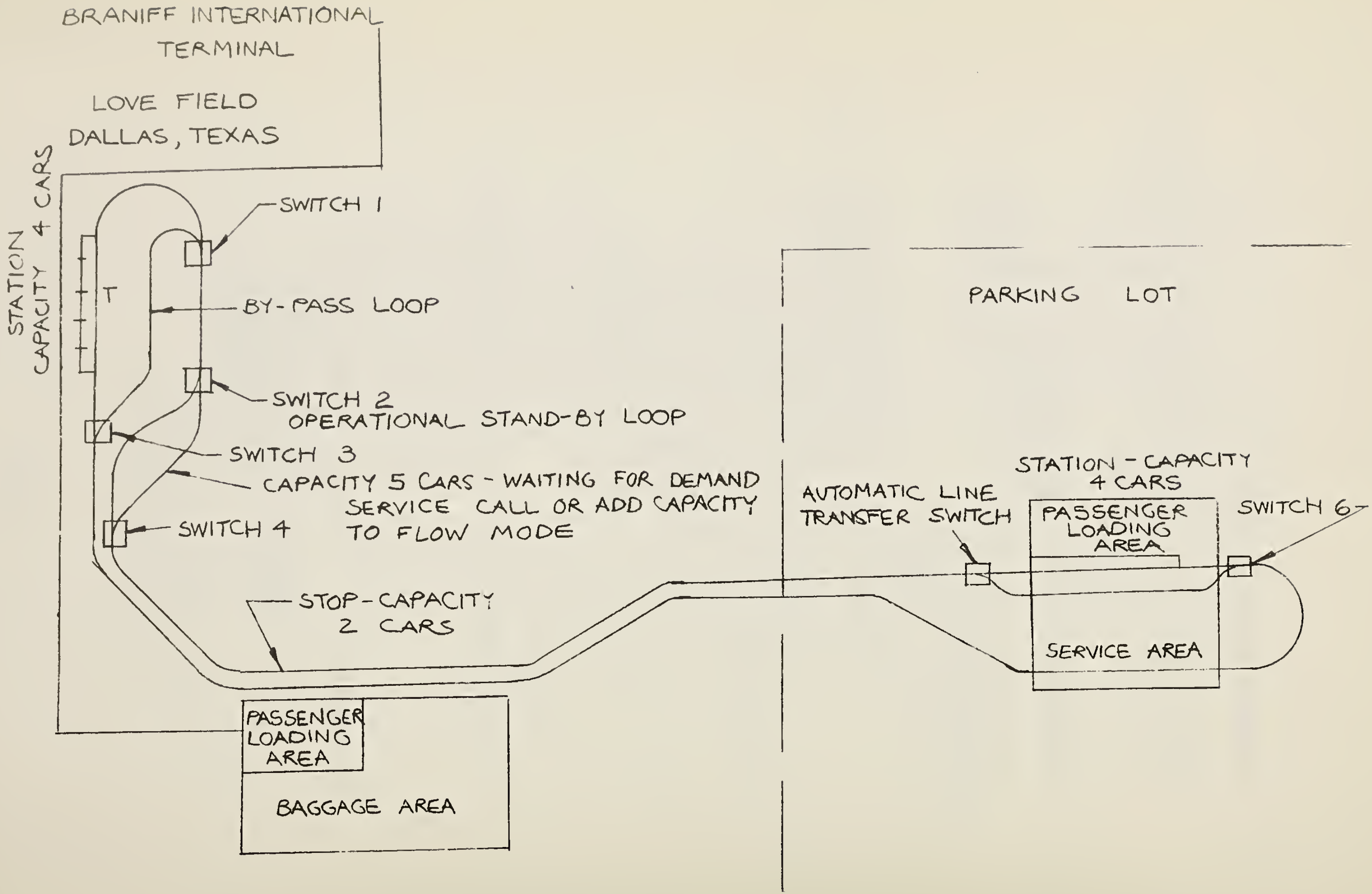


Figure 4-2 "JETRAIL" ROUTE

(e) Drive wheel rpm $= \frac{V\ (mph) \times 14}{r} = \frac{15 \times 14}{.5} = 420$ rpm

(f) Gear box reduction $= G_R = \frac{\text{maximum electro-drive}}{\text{maximum wheel rpm}} =$

$$\frac{1450}{420} = 3.45:1$$

4.3.1.3 Torque Requirements

(a) Torque at gear box input $= T_i = \frac{F_p \times r}{e \times G_R}$

$$T_i = \frac{.5\ F_p}{.85 \times 3.45} = .1705\ F_p$$

The table below lists the torque required by a vehicle accelerating to 15 mph with an acceleration profile described in section 4.3.1.2.d. and zero grade requirements.

Transient Torques

Velocity (mph)	$\frac{V_{inst}}{V_{max}}$	F_G(lb)	F_a(lb)	F_r(lb)	F_D(lb)	F_p (lb)	T_o(ft-lb)
0	0	0	372	160	0	532	90.65
5	.3	0	372	160	2	534	90.99
10	.66	0	364	160	6.7	531	90.5
12	.8	0	331	160	9.5	500	85.2
13	.86	0	297	160	11.5	469	79.9
14	.93	0	167	160	13.5	340	57.9
15	1.0	0	0	160	15	175	29.8

4.3.2 Parameters of Optimized System

In this section a basic duty cycle in terms of trip length, acceleration, deceleration and dwell time is developed. It is also shown how to calculate the forces necessary to make the optimized vehicle meet the duty cycle requirements. To do this, we must first define the vehicle more specifically.

Vehicle System:	Optimized System:
Design Speeds:	20-30 mph
Maximum Number of Passengers:	8
Drive Unit:	Two 15 HP AC drives with eddy current clutch/brake
Maximum Acceleration:	2.2 ft/sec^2
Maximum Deceleration:	3.2 ft/sec^2
Gross Vehicle Weight	8,000 lbs.
Car Weight	4,200 lbs.
Drive System Weight:	2,200 lbs.
Curb Vehicle Weight:	6,400 lbs.
Max. Passenger Load: (8 @ 150 lb. each)	1,200 lbs.
Baggage:	400 lbs.
Car Frontal Surface Area:	50 sq. ft.
Drive Wheel Radius:	10 in. = .833 ft.
Max. Electro-Drive Speed:	1,610 RPM
Drive Train Efficiency:	0.85
System Grades	±3% max.
Physical Coefficients:	
Coefficient of Rolling Friction:	0.02
Aerodynamic Drag Coefficient:	0.52

4.3.2.1 Route for Optimized System

From the knowledge gained by the analyses and study of existing people mover systems it was possible to develop a route system that was representative of the operational conditions required for a full-scale PRT system. This route incorporates typical PRT performance requirements with guideways on 3% to -3% grades, 1/4 to 1/2 mile between stops and deceleration around curves. Figure 4-3 is a diagram of the simulated transit network, which may be considered a large shopping center complex, an air-line terminal in a large city or any other type of urban center.

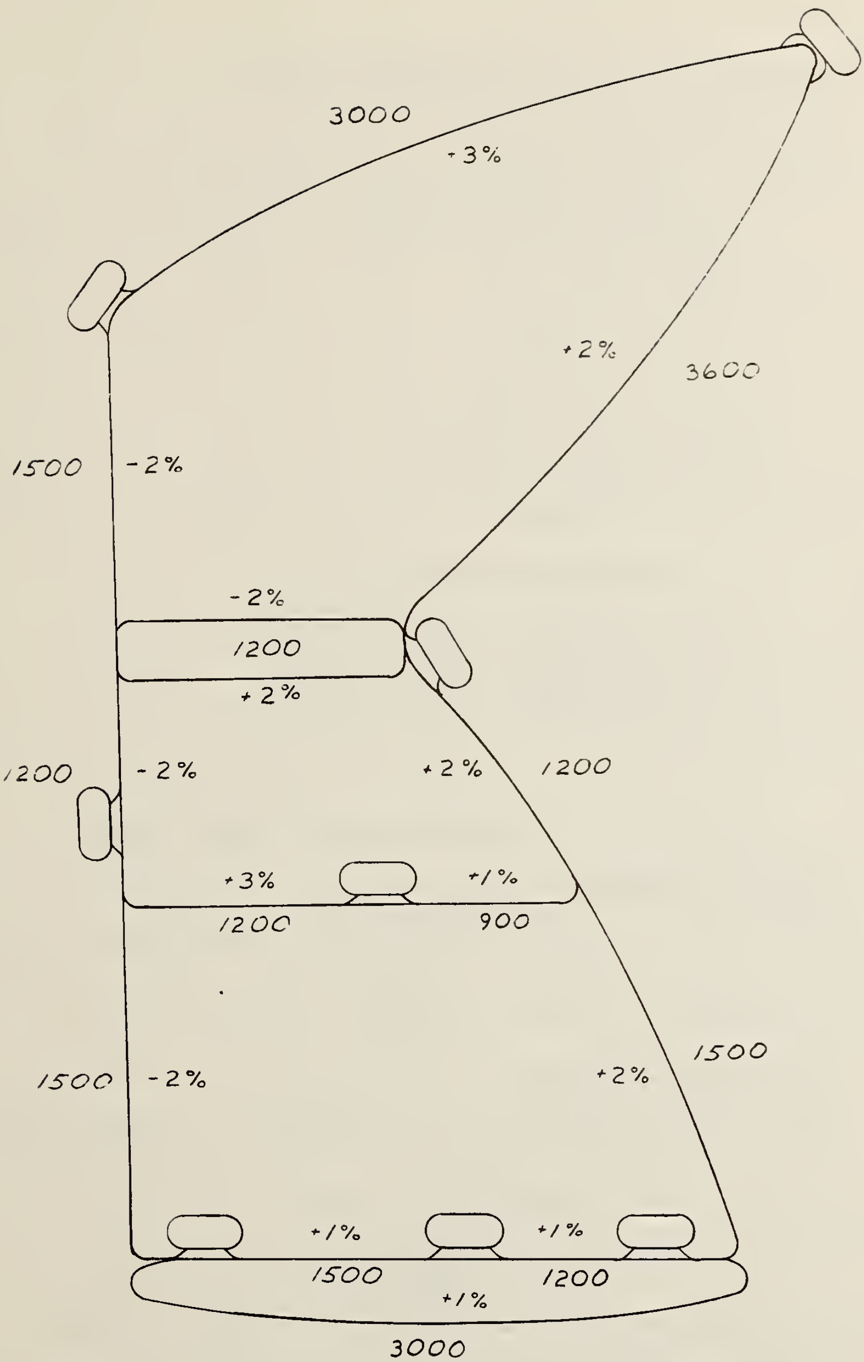


Figure 4-3 OPTIMIZED SYSTEM ROUTE DETAIL

4.3.2.2 Resistance Forces on Optimized Vehicle

(a) Rolling Resistance

F_r = vehicle weight x rolling coefficient

8,000 lbs. x .02 = 160 lbs.

(b) Aerodynamic Drag

$F_D = 1/2 \cdot v_w^2 C_D S_A$

at 30 mph F_D = 60 lbs.

at 20 mph F_D = 25 lbs.

(c) Grade Resistance

F_G = vehicle weight x % of grade

at ±1% 8,000 lbs. x .01 = ± 80 lbs.

at ±2% 8,000 lbs. x .02 = ±160 lbs.

at ±3% 8,000 lbs. x .03 = ±240 lbs.

(d) Acceleration

Calculated at a later time for a 30 mph vehicle.

4.3.2.3 Gear Box Reduction

(a) Drive wheel rpm - $\frac{V(\text{mph}) \times 14}{r} = 30 \times \frac{14}{.833} = 504$ rpm

(b) Gear box reduction $= \frac{1,610}{504} = 3.2:1$

4.3.2.4 Torque Requirements

Torque at the gear box $= T_o = \frac{F_p \times r}{e \times G_R} = \frac{F_p \times 10/12}{\times 3.2}$

1) per $T_{req} = \frac{F_p \times 5/6}{.85} = .98\,F_p$

2) at gear box input $= \frac{.98 F_p}{3.2} = .306$

3) at gear box input per drive unit $= \frac{.306}{2} = .15\,F_p$

For steady state flow, torque requirements are as shown on the following page.

Transient Torques

Grade	Vel (mph)	F_G(lg)	F_r(lb)	F_D(lb)	F_p	T_o Ft-lb/unit
+3%	20	240	160	25	425	43.3
+2%	20	160	160	25	345	35.2
+1%	30	80	160	60	300	44.9
0%	30	0	160	60	220	33.6
-1%	30	-80	160	60	140	21.4
-2%	30	-160	160	60	60	9.2
-3%	30	-240	160	60	-20	-3.0

4.4 DUTY CYCLE FOR SYSTEMS STUDIED

Having derived the route of the two systems and the system parameters including the resistive forces, it is possible to construct a duty cycle of operation for the Jetrail system and the Optimized system.

4.4.1 Duty Cycle — Jetrail System

For the Love Field installation all basic operating parameters are known and the determination of the basic duty cycle is merely computational. A further refinement of the actual duty cycle is the addition of slowing a vehicle around curves. This is accomplished by decelerating the vehicle to a curve speed, before it enters the lead-in section of the curve, cruising the curve at a curve speed and then accelerating back to maximum speed. Computations for such a phase follow the Travel Mode format, with curve velocity and radius as known parameters.

Systems incorporating spiral curves and superelevation of a guideway eliminate the need for this additional analysis and calculation. Cost factors limit extensive use of such design features.

Of the Duty Cycle Tables in Appendix B, Table B.3 indicates the velocity and distances between station stops. This table is a computation of the phases of all possible trip variations of this known system. By arbitrarily changing these trip variations over a one-hour period, Table B.2 is formulated, thereby representing a one-hour schedule of random operation of the basic duty cycle of the Jetrail installation. This duty cycle was used for the dynamometer test program. It encompasses a long time span of operation which ensures that a steady state condition is reached for all drive unit functions during testing.

4.4.2 Duty Cycle - Optimized System

With a typical duty cycle route created as shown in Figure 4-3, it is possible to calculate the Travel Mode and Dwell Mode times of such random configurations. Referring again to Appendix B, Tables B.5 and B.6 are the schedules of two vehicles traveling at different speeds over the same random configuration. Tables 4.1 and 4.2 may be added to Table 4.5 and Table 4.6 to obtain a total duty cycle in excess of 1-3/4 hours. Table 4.5 lacks the sophistication of deceleration around curves; therefore, Table 4.6 was developed as a random routing, including such features as vehicle deceleration and slower curve velocities. Table 4.7 is a finalized, typical basic duty cycle of the Optimized vehicle traveling on the simulated route of Figure 4-3.

5.0 POWER REQUIREMENTS

Upon completing the basic calculations required for a drive unit it is necessary to determine how they can be met using an AC Drive Unit with an eddy-current clutch and brake.

5.1 TORQUE SPEED CHARACTERISTICS

A typical torque-speed characteristic curve of an AC drive unit with eddy-current clutch and brake is shown in Figure 5-1. This is representative of a 15 horsepower model PM001MSE AC drive unit. Specific values for given horsepower drive units vary slightly but the relative curves are comparable.

5.1.1 Operation Cycle of an AC Drive Unit

Figure 5-2 shows curve A-B-C-D, which is a typical power operation cycle of a PRT vehicle. At zero RPM output speed a vehicle is at rest in the station and at "T_0" time, an impulse is sent to the drive unit requesting acceleration to maximum speed. In attempting to accelerate the vehicle the drive unit responds to a torque loading as calculated in Section 4, (say torque A). As the drive unit accelerates the vehicle to maximum speed the torque required travels along the curve, to point B. From point A to point B is a transient plot of the acceleration phase of the drive unit. Once the traveling vehicle arrives at point B it may cruise at maximum speed for as long as desired. To slow for a curve on a guideway, however, the clutch must respond to decreased vehicle speed. In slowing down, the torque curve is traversed from point B to point C. Once the curve is negotiated the vehicle will again accelerate to maximum speed, so that the torque on the drive unit increases from point C to the acceleration curve, at point D. The drive unit then continues to accelerate to point B.

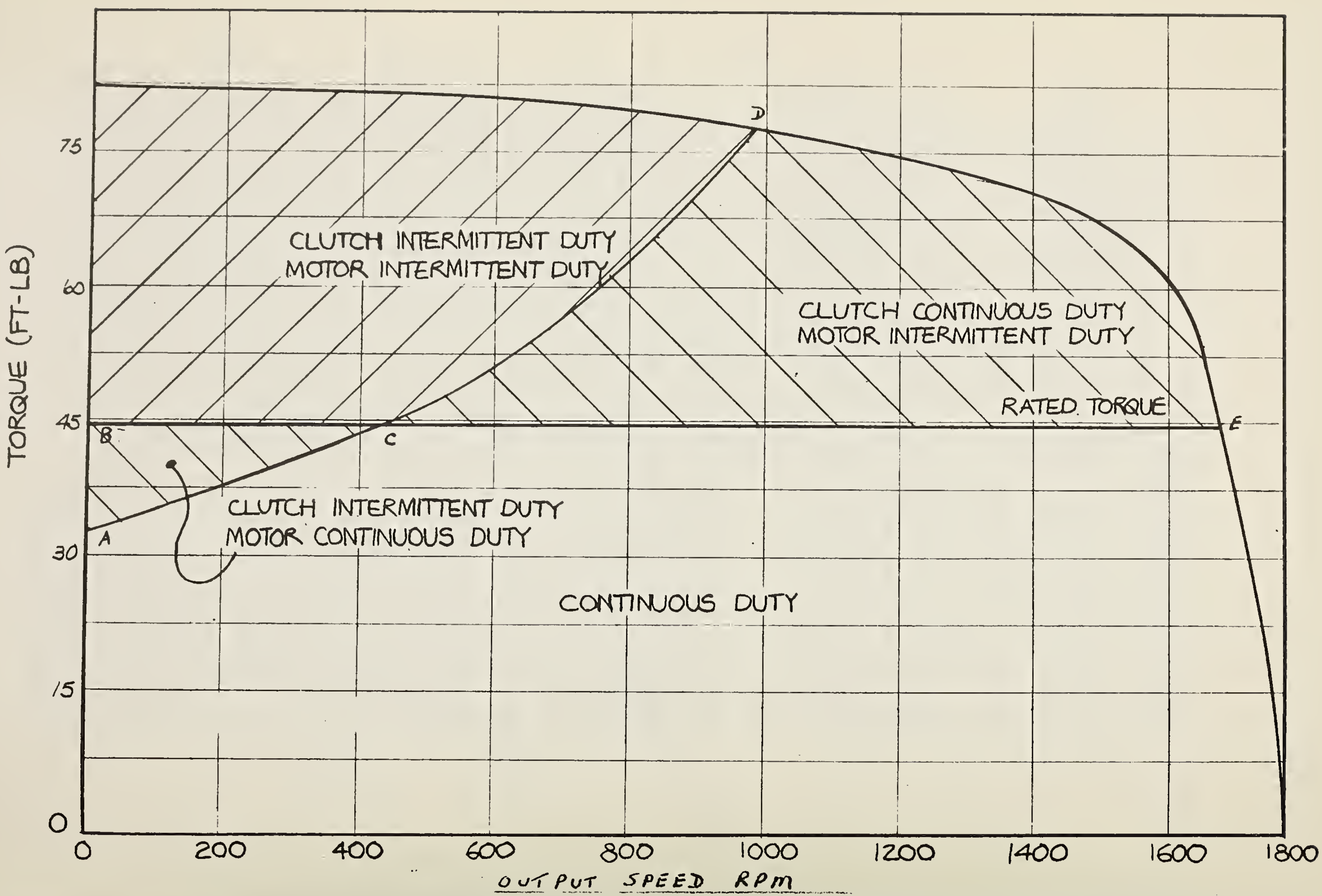


Figure 5-1 PM-001 MSE DRIVE UNIT
TORQUE-SPEED CHARACTERISTIC CURVE

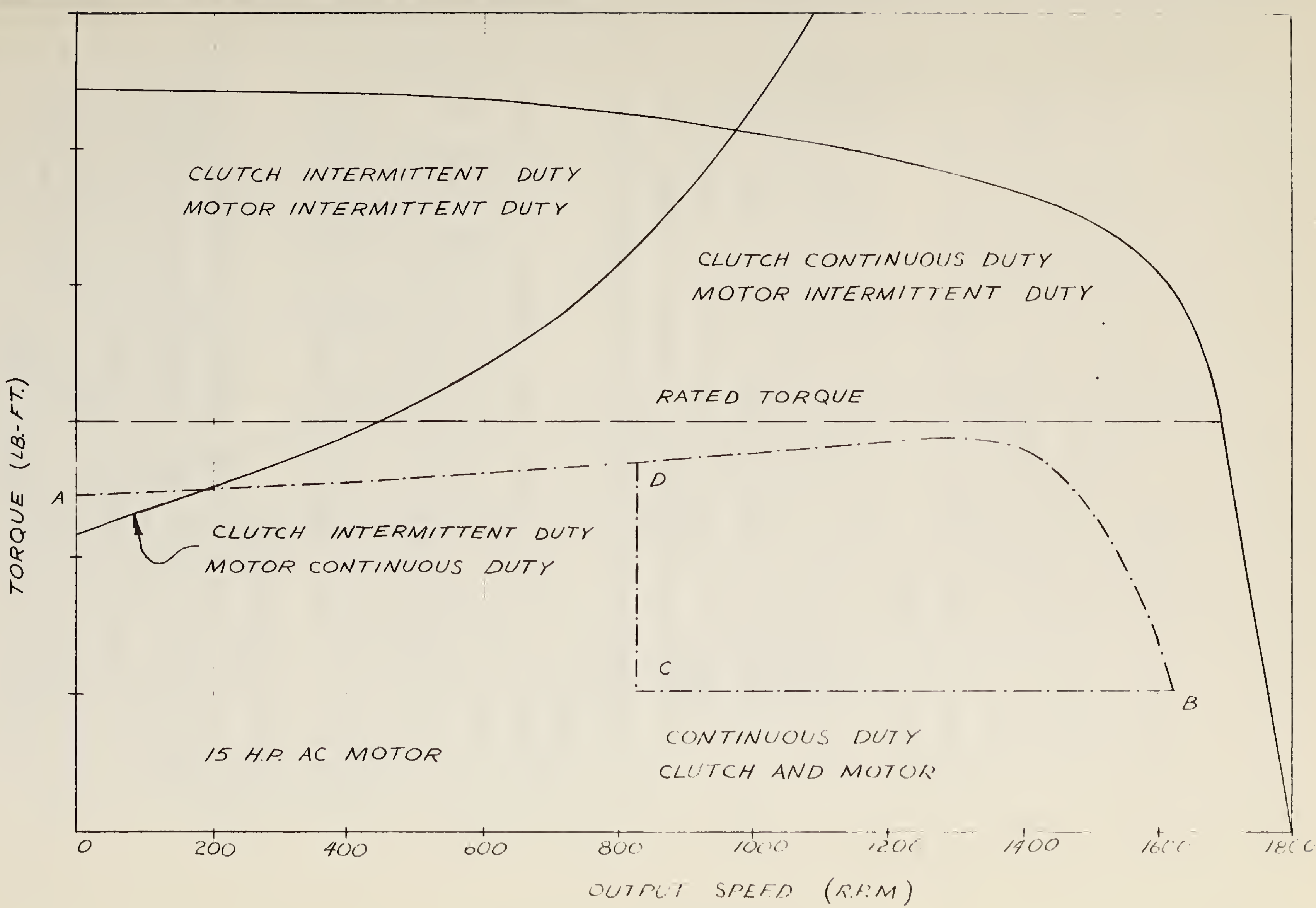


Figure 5-2 DRIVE TORQUE SPEED CURVES

5.1.2 Maximum Vehicle Speed

Figure 5-3 represents several characteristic curves for identical vehicles, in which only top vehicle speed has been varied. The variations in these curves reflect the changes in gear ratios in the drive systems. The effect of this change is greater on the acceleration phase than on the steady state phase.

Gear ratios may also be changed by varying maximum speed of the electro-drive unit. A decrease in the operating speed would require a decrease in the gear reduction ratio and, therefore, would increase the required torque. But, a decrease in the operating speed will also increase clutch slip, which in turn decreases drive unit efficiency; therefore, an electro-drive unit should operate as close to maximum speed as possible.

5.1.3 Grades

Figure 5-4 is a graph of drive unit capability as a function of guideway grades. These curves allow maximum acceleration on a given grade to be calculated, or, conversely, the maximum grade on which a required acceleration may be achieved. Since most systems must reach a compromise in this area, the curves can be used to plan and define an optimum route in terms of distance, grades, acceleration required and power available.

5.1.4 Gross Vehicle Weight

Figure 5-5 shows the power required for a number of passenger load conditions. Again, these calculations reflect the variation of the forces acting on the vehicle. When the power available and curb weight of the vehicle are known, it is possible to design a vehicle to operate over its rated load capacity. Complete knowledge of such design criteria must be known in order to establish an efficient vehicle design.

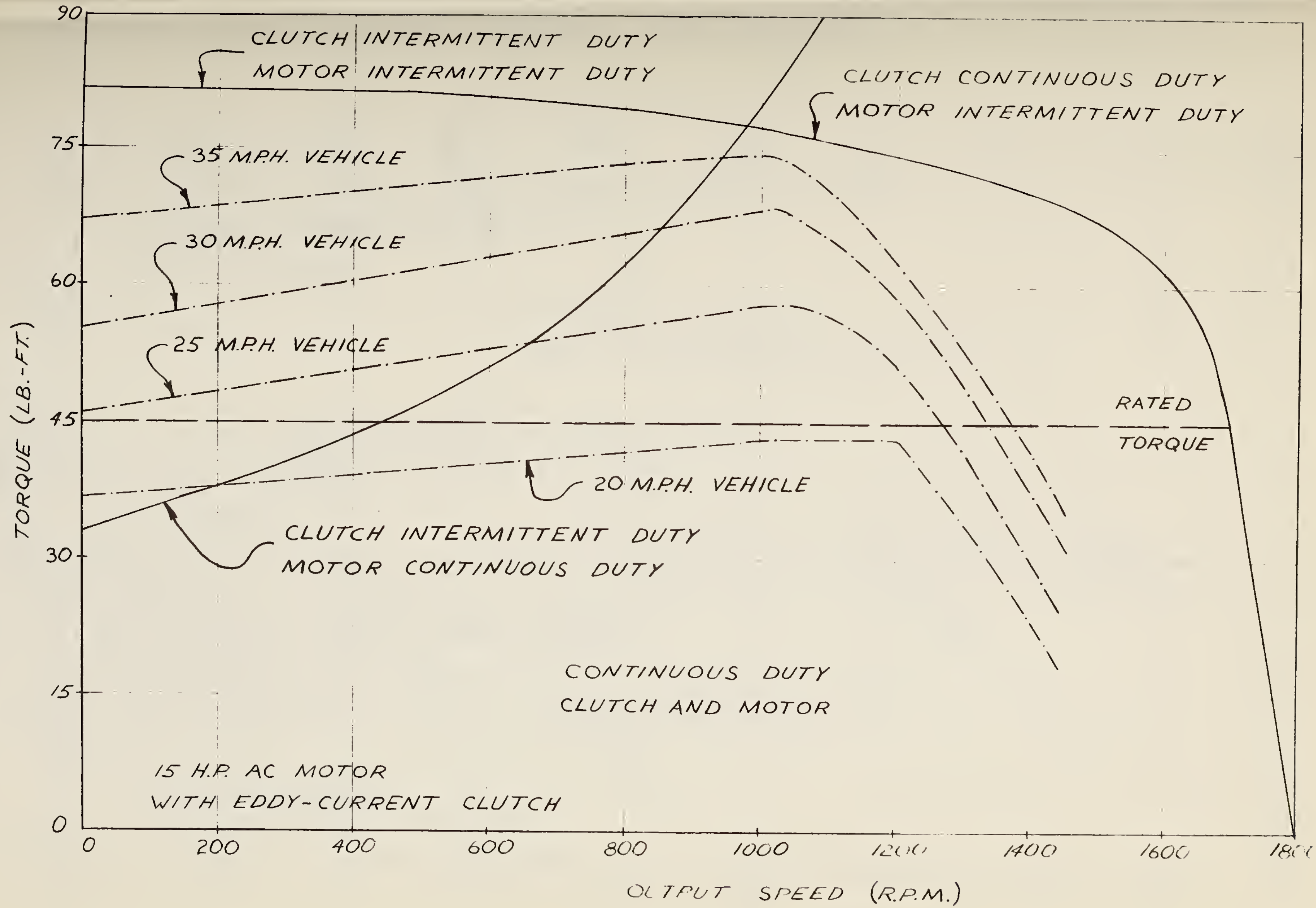


Figure 5-3 SPEED/TORQUE CURVES

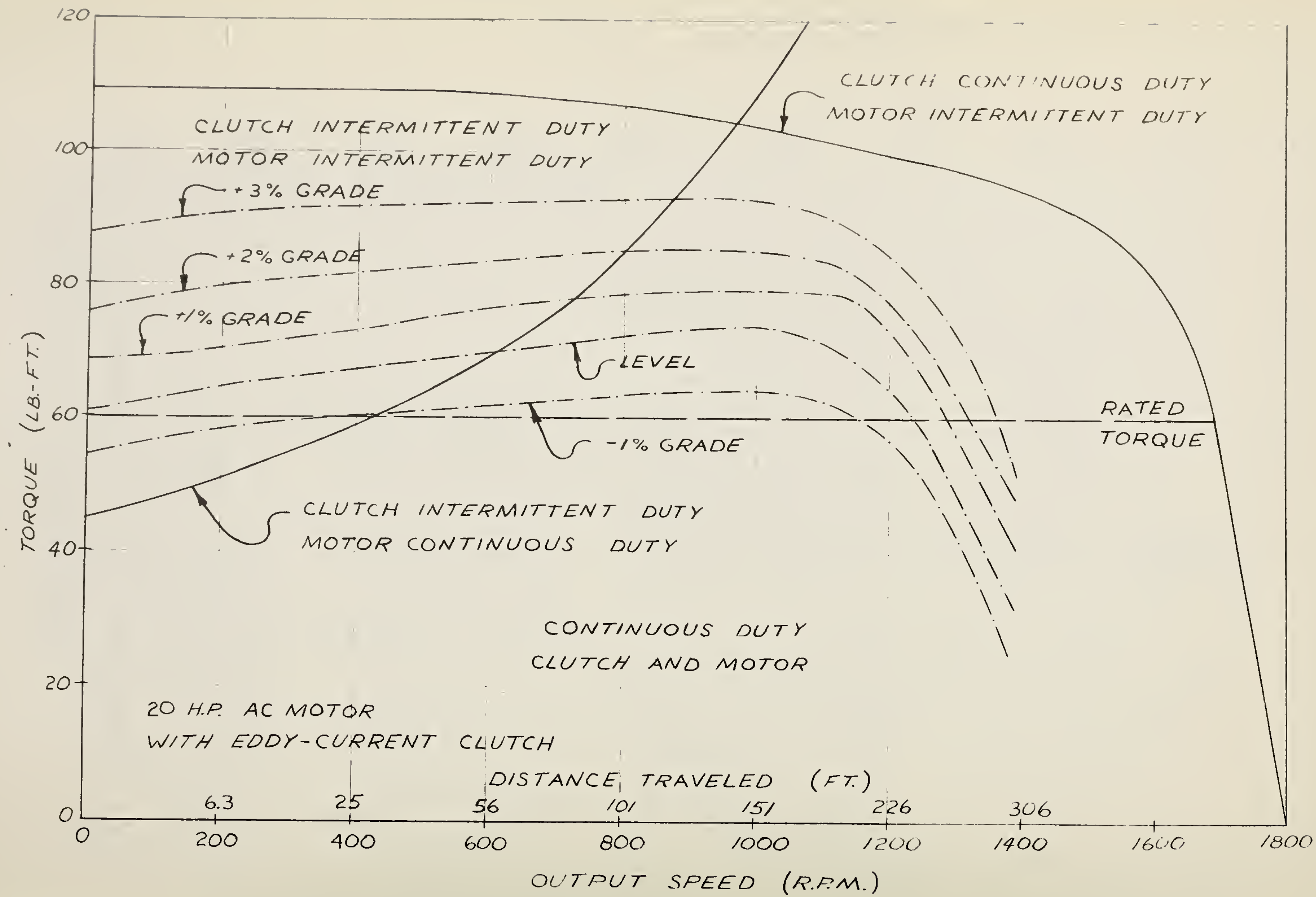


Figure 5-4 GRADE/TORQUE CURVES

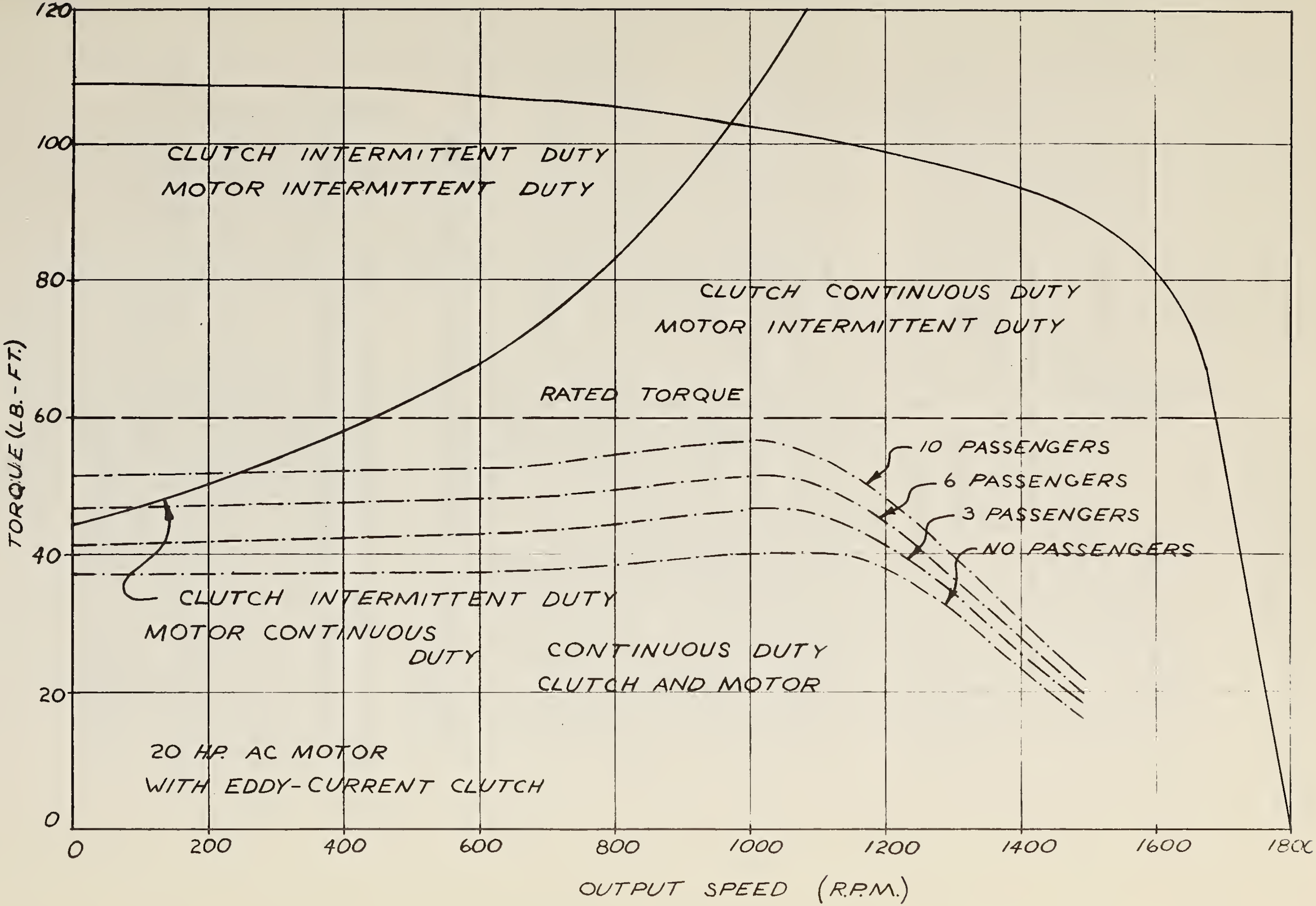


Figure 5-5 PASSENGER LOAD PERFORMANCE

5.1.5 Acceleration Rates

Figure 5-6 demonstrates how acceleration rates vary the power requirements of a given vehicle. Varying the acceleration only affects the drive unit performance during the acceleration phase. In determining the actual power required during the acceleration phase, variation of acceleration rate is the preferred approach. Once actual limits are known, optimal vehicle design can be established.

Preliminary observations of Figure 5-7 can lead to the assumption that if a vehicle were to accelerate on a 3% grade, the AC drive unit would stall. However, the validity of this assumption was questioned as a result of computations conducted on the Dallas (Love Field) Jetrail system.

In this instance, the Dallas vehicle has a reported breakaway surge of 1.8 ft/sec^2. The vehicles were designed to accelerate at 1.2 ft/sec^2, and an averaging mid-phase acceleration may be considered as 1.5 ft/sec^2.

Upon graphing the calculations presented earlier, Figure 5-8 is obtained.

Clearly, Figure 5-8 appears in error, or at least is not representative of calculated performance. The power required was evaluated and, except for the exclusion of a small inertia term, was considered correct. It was therefore concluded that further testing of the motor was required and that until such time as more of the acceleration phase is known, steady state conditions must be considered as the limiting design criterion.

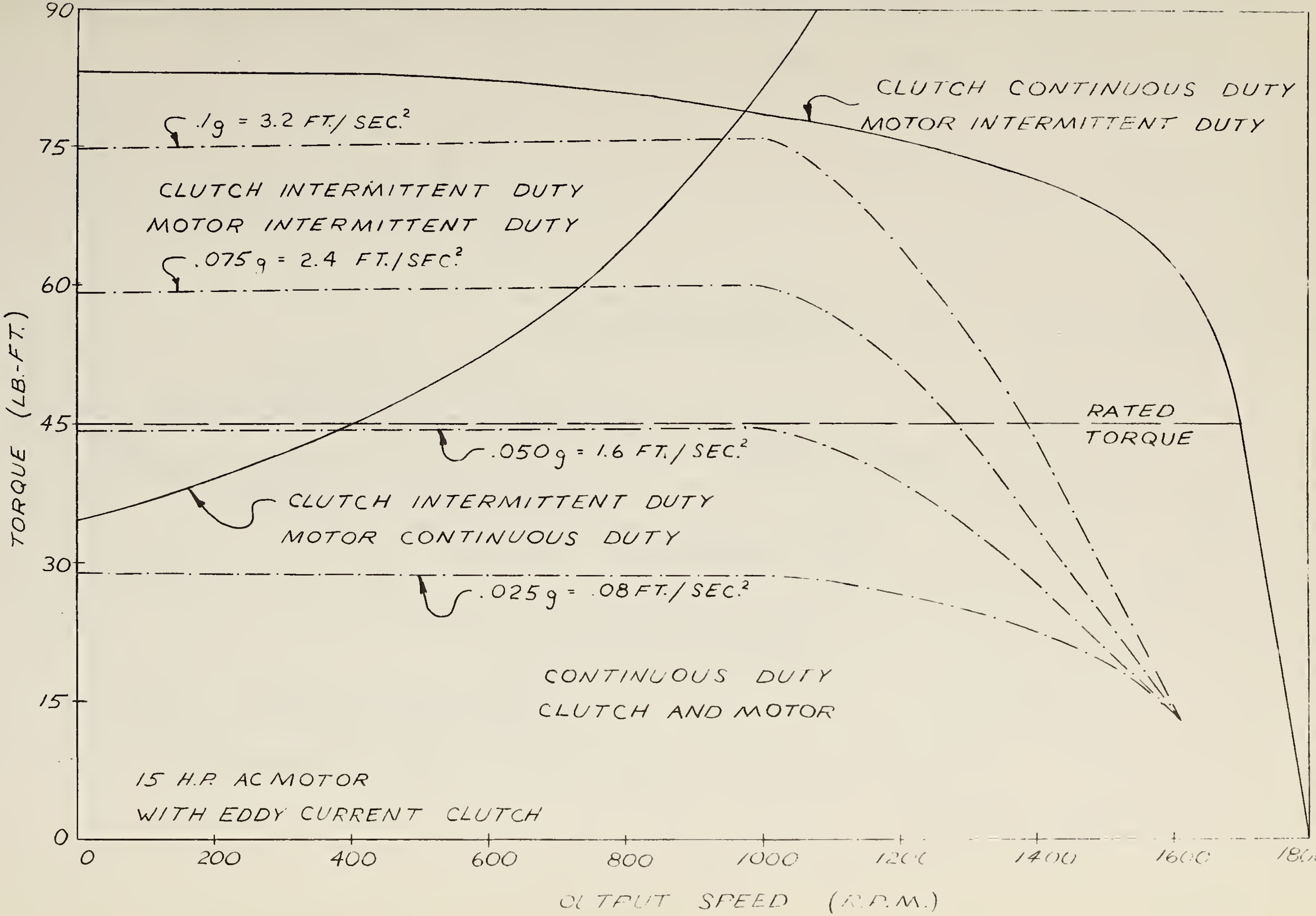
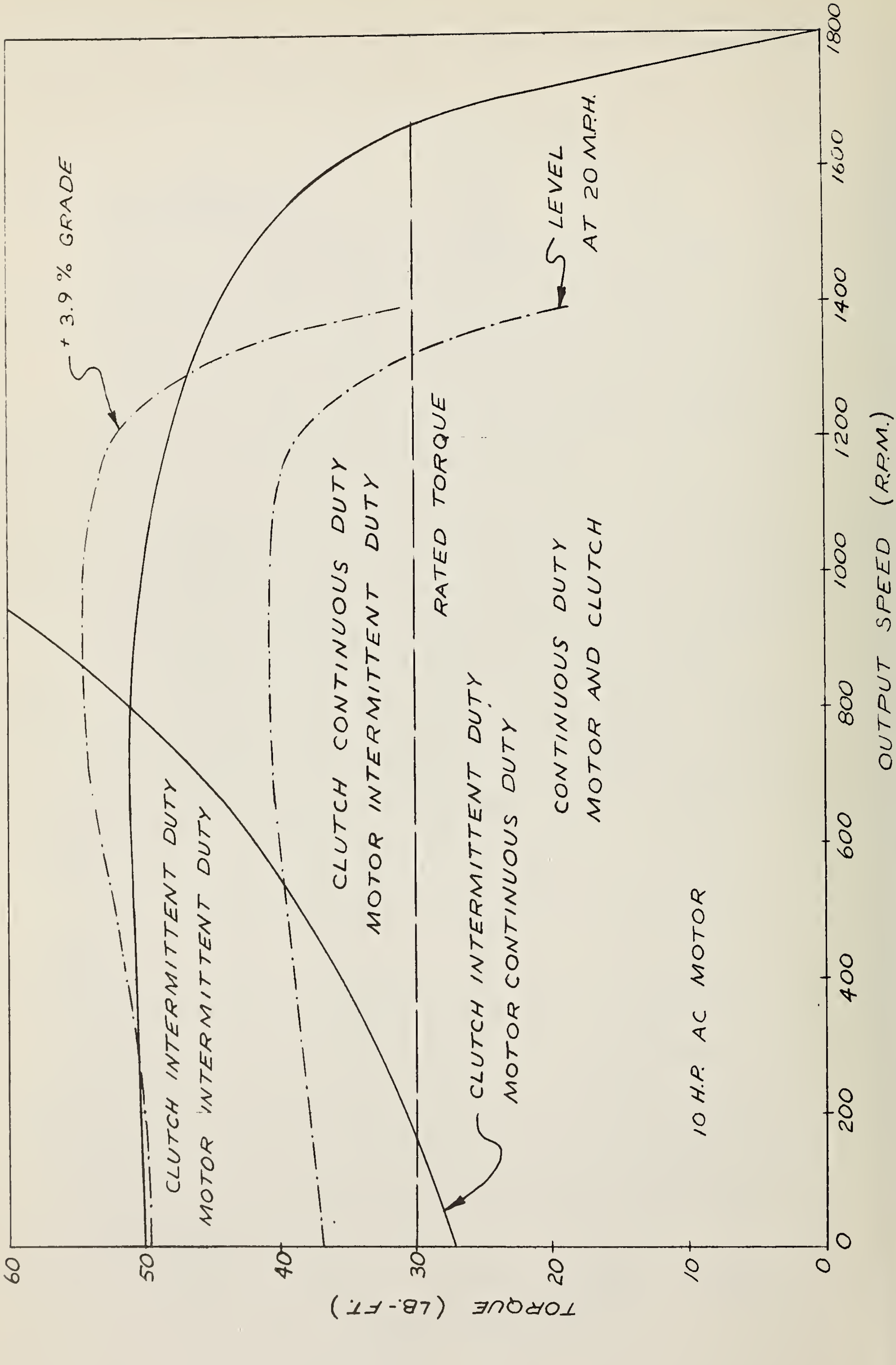


Figure 5-6 TORQUE vs. ACCELERATION


+ 3.9 % GRADE
LEVEL AT 20 M.P.H.
CLUTCH CONTINUOUS DUTY
MOTOR INTERMITTENT DUTY
RATED TORQUE
CLUTCH INTERMITTENT DUTY
MOTOR INTERMITTENT DUTY
CLUTCH INTERMITTENT DUTY
MOTOR CONTINUOUS DUTY
CONTINUOUS DUTY
MOTOR AND CLUTCH
10 H.P. AC MOTOR
TORQUE (LB.-FT.)
0
10
20
30
40
50
60
OUTPUT SPEED (R.P.M.)
0
200
400
600
800
1000
1200
1400
1600
1800

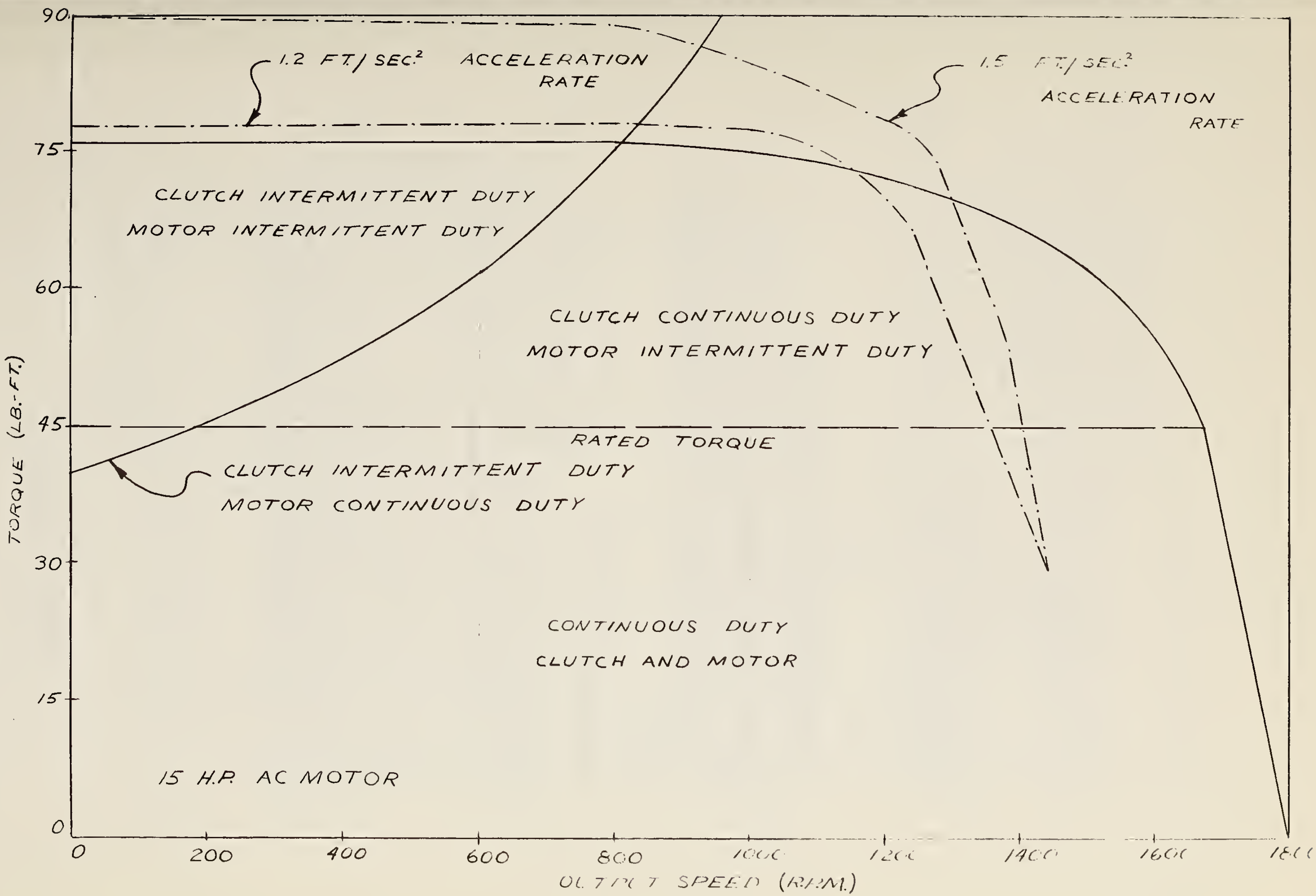


Figure 5-8 DALLAS SYSTEM ACCELERATION CHARACTERISTICS

5.2 SELECTION OF DRIVE PACKAGE

Selection of the drive package entailed the matching of drive system performance capabilities with duty cycle requirements. The duty cycle developed in Section 4 and the propulsion requirements developed in Section 5.1 constitute the basic requirements which were used. In the final selection of the drive package however, it is also necessary to examine the mechanical system to which the drive will be coupled.

5.2.1 Characteristics of the Mechanical System

In examining the applicability of the drive unit for the duty cycle, it is necessary to examine the actual drive being considered along with its specific coupling into the PRT system.

5.2.1.1 Maximum Electro-drive

The maximum electro-drive of an AC drive unit is the limiting speed value for efficient motor operation. This value may be obtained upon the examination of the torque-speed curve for that particular unit. Figure 5-1 is the torque-speed curve for a PM001MSE, 15 horsepower AC drive unit. It is seen from this Figure that 1,610 rpm may be considered as a relatively efficient maximum electro-drive speed. The Jetrail installation used two 7.5 horsepower drive units; for these units, design speeds are 1,450 rpm.

5.2.1.2 Mechanical Components

The actual gear box reduction is found by dividing the maximum electro-drive speed by the drive wheel rpm at maximum speed:

$$\text{Gear box reduction} = \frac{\text{maximum rpm of electro-drive}}{\text{drive wheel rpm}} = R$$

$$R = \frac{S_{ed}}{S_w}$$

Once forces acting on the vehicle system and its mechanical components are known, performance power required may be expressed in terms of torque and rpm, the two components of horsepower.

5.2.2 Power Requirements

5.2.2.1 Mechanical Efficiency

From basic physics, torque = force x radius. For the torque required to power a vehicle this becomes:

$$T = F_t \times r$$

The total drive efficiency of a vehicle incorporates the mechanical efficiency of the entire drive train and spin losses in the gear box. Values of mechanical efficiency range between 75% to 95%.

The actual torque required for vehicle propelling varies directly with the desired torque, and, inversely, with mechanical efficiency:

$$T_{required} = \frac{T_{desired}}{e}$$

For vehicle systems being considered:

$$T_{required} = \frac{F_t \times r}{e}$$

5.2.2.2 <u>Input Torque Required at Gear Box</u>

Input torque at the gear box is supplied by the electro-drive output shaft. Torque required at the gear box input to propel the vehicle is computed as follows:

$$\text{Torque at gear box input} = \frac{F_t \times v\ (\text{mph}) \times 14}{e \times \text{maximum rpm electro-drive}}$$

$$T_r = T_{required} = \frac{\frac{F_t \times r}{e}}{R} = \frac{F_t \times v\ (\text{mph}) \times 14}{e \times S_{ed}\ (\text{rpm})}$$

5.3 CONTROL SYSTEMS

Fundamental considerations of PRT Systems require that the methods used to control the vehicle drive unit be simple and easily applied. The drive unit should respond to one signal which, in its turn, should be easily selected.

5.3.1 Speed Control

Speed control is achieved by applying a DC voltage to the drive unit controller. The signal may be continuously variable or may be applied in discrete steps if specific speeds are to be programmed.

The controller performs two functions: (a) it provides a DC current to drive the eddy-current clutch or brake, and (b) it processes the speed control voltage to insure that its rates of rise and fall do not exceed a value determined by the planned vehicle acceleration and deceleration profiles. These shaping circuits do not influence the speed signal during periods when the vehicle speed is constant.

5.3.2 Controller

Mobility Systems and Equipment Company's controller was used to implement the test program. Speeds could be selected as required to simulate a duty cycle and acceleration and deceleration rates adjusted to meet the system specification requirements.

This permitted a simple, accurate and repeatable control of the programmed duty cycle. It also confirmed that such a control system would be applicable to operational PRT systems when the control signals are derived from wayside signals or safety block subsystems.

5.3.3 Controls

The drive unit consists of a 3 phase, 460v 60 Hz induction motor coupled to an eddy-current clutch and brake. The vehicle speed is adjusted by varying the DC current applied to the clutch coil. This current introduces eddy currents in the rotating part of the clutch which therefore slips with respect to the motor. Similarly if current is applied to the brake coil instead of the clutch coil the eddy currents induced will cause braking of the output drive shaft. In each case the motor runs at its normal slip speed (approximately 1670 rpm).

5.3.4 Speed Control Circuits

The clutch coil is driven by a silicon controlled rectifier, the current output of which is directly proportional to a DC control voltage minus a voltage derived from a tachometer mounted on the output drive shaft. This feedback system ensures that the difference between the demanded speed and the output speed is a designed minimum. When the output speed is higher than the demanded speed, current to the clutch coil is reduced to zero and current to the brake coil is applied until the output shaft again equals the demanded speed. The feedback circuits used ensure that control is essentially linear and that the output speed will follow the demanded speed in either direction.

In a constant speed mode the current will alternate between the clutch and brake at a rate dependent upon the system accuracy.

5.3.5. Drive Unit Controller

The drive unit controller includes the circuitry which:

1. Provides the control current to the eddy-current clutch and brake.

2. Includes the feedback circuitry which regulates the amplitude of the current supplied to the clutch and brake.

3. Accepts the speed control voltage from the way-side subsystem and limits the rate of rise and fall of this signal to provide controllable rates of acceleration and deceleration.

4. Implements a friction brake assembly which is actuated when the AC input voltage falls below a preset value.

5. Controls a trip circuit which limits the maximum speed of the output shaft to a presetable value.

A basic block diagram is shown in Figure 5-9.

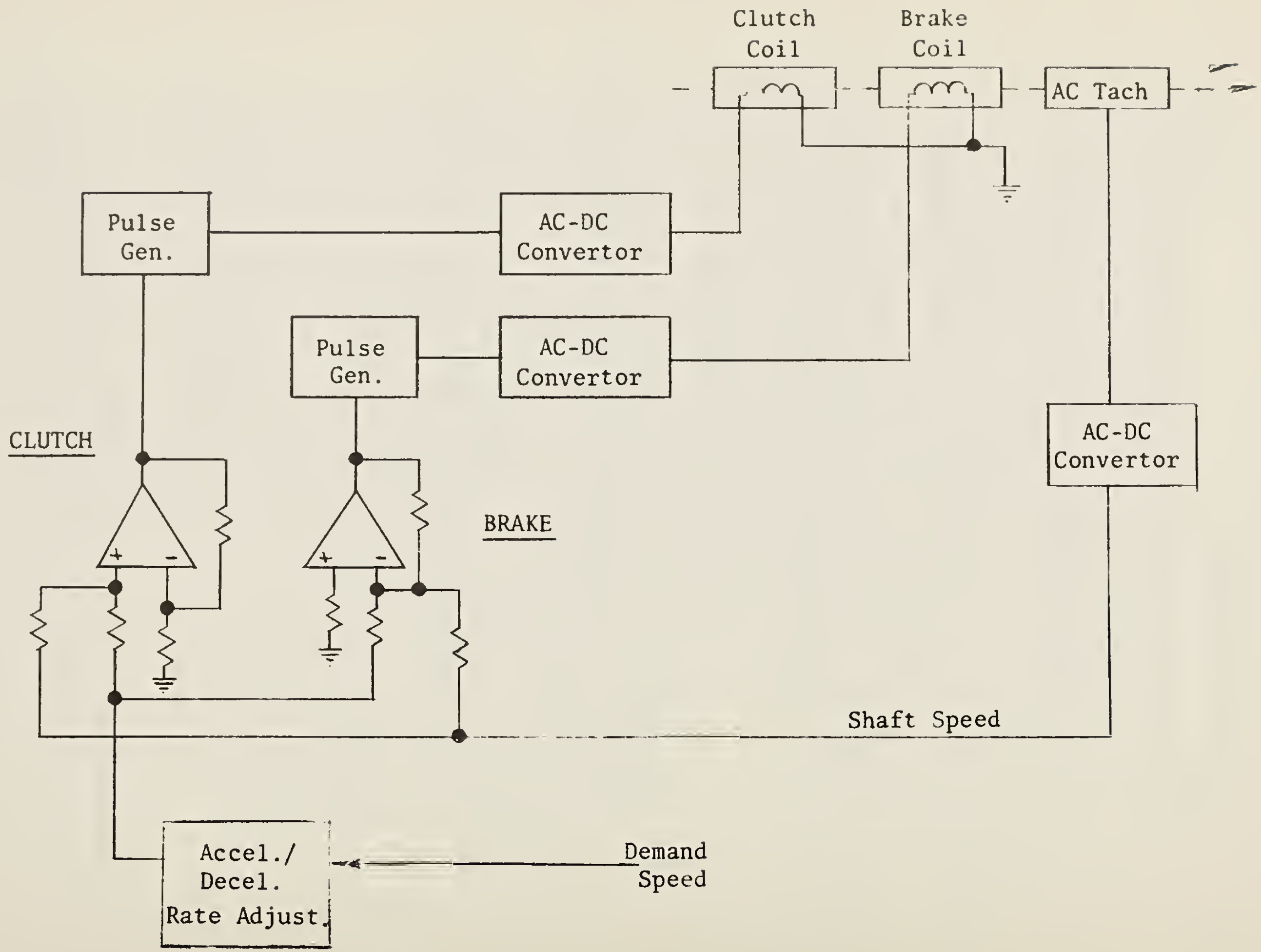


Figure 5-9 CONTROL BLOCK DIAGRAM

5.4 BOGIE AND GEAR DESIGN

MSE has designed and fabricated a bogie drive for overhead monorail Personal Rapid Transit vehicles. The design features, performance requirements and recommendations for further development are described.

5.4.1 General Design Characteristics

The bogie drive system, referred to hereinafter as "the bogie," is an overhead rail mounted system with four drive wheels. Four vertical guide wheels maintain the direction of the bogie on the overhead rail and provision is made for two AC eddy-current drive units. See Figure 5-10 for general layout. The bogie is driven by the AC drive unit with eddy-current clutch for efficient operation at both normal and low velocities. The bogie is adaptable to a variety of monorail rapid transit vehicles and has been designed for a long, low-maintenance operating life.

The bogie has been designed to be used in pairs to support and drive a fully loaded monorail vehicle weighing up to 16,000 pounds. A vehicle of this size could carry 30 adults.

The bogie has been designed to operate with the 15 horsepower AC eddy-current drive system developed by MS&E and described at length in other sections of this report. The bogie may also be used with 20 horsepower drive units. Power is transmitted to each drive wheel by fully enclosed silent chain drives.

The bogie has been designed to meet the following requirements:

Drive System:

A 15 HP Drive unit consisting of motor and eddy-current clutch/brake. The Design shall also be compatible with 20 HP drive unit.

Speeds:

High Gear:	10 MPH - up 6% Grade
	15 MPH - Level
	20 MPH - Down 6% Grade
Low Gear:	3.5 MPH - Up 6% Grade
	5 MPH - Level
	6.5 MPH - Down 6% Grade

Acceleration/Deceleration - 2.5 f/sec^2
Braking Deceleration - 5.5 ft/sec^2
Emergency Braking Deceleration - 9.0 ft/sec^2
Live Load - 5,000 pounds
Total Design Weight - 16,000 pounds
Design Bogie Assembly - for double motor drive unit

The configuration shall be compatible with Love Field/ Braniff International Airlines installation
Design for following curve speeds:
100 ft. radius curve - 10 MPH
25 ft. radius curve - 3.5 MPH

Wind Loads:

Wind velocity - 60 MPH

Vehicle Dimensions:
Side: 30' x 12'
Front: 8' x 12'

Overall dimensions of the bogie are 36.25 inches high, 37.1 inches wide and 46 inches long. The weight of the bogie is approximately 1950 pounds.

Braking is provided under normal operating conditions by the eddy-current clutch of the drive unit. An electric brake is provided to hold the bogie stationary at stops. The electric brake contains a permanent magnet to apply the brake whenever power system voltage drops below 60 volts. The electric brake operates concurrently with the eddy-current clutch to stop the vehicle after the velocity has dropped below 2 MPH, or at higher speeds in case of an emergency situation, such as power failure. Under normal braking conditions deceleration will be at 5.5 ft/sec^2. Under emergency braking conditions deceleration will be 9.0 ft/sec^2. If power failure occurs, a ratchet wheel mechanism automatically grips the rail on which the bogie rides, bringing the vehicle to a quick, safe stop.

5.4.1.1 Drive

The output shafts of the AC eddy-current drive units are connected to the bogie by double universal jointed shafts. The shafts in turn are connected to the input shafts of the transmissions. The universal jointed shafts are provided with splined sleeves which allow for vertical motion of the vehicle cab relative to the rail and for displacement of transmission input shaft center lines from vehicle center lines when the vehicle negotiates a curve.

The transmission drives a differential gear unit having a 4.11 to 1 ratio and double output shafts. The differential gear unit is an off-the-shelf unit manufactured by Spicer Division of Dana Corporation (Model A12-1).

The differential output shafts are connected to silent chain drives which provide the final drive to the wheels at a 0.8444 to 1 ratio. The silent chain drives, manufactured by Morse Chain Division of Borg-Warner Corp., are protected from the elements by a steel case with removable panels.

The drive wheels are 12 inches in diameter with a 3 inch tread. Treads are rubber on cast steel rims.

Details of the drive are as follows:

The drive shaft is 1-3/8 inches diameter cold rolled steel and will rotate at the same rate as the AC drive output shaft.

The universal joints are a standard model manufactured by Boston Gear and have been on the market for many years. At the expected drive shaft rpm, their rated torque capacity is approximately 54 foot pounds.

The transmission will operate at a 1 to 1 gear ratio. Speed control for both normal and low speed bogie operation will be provided by the AC drive with eddy-current brake and clutch using Mr. George Adams' principles of speed control.

The differential has a capacity of 115 foot pounds input torque. Input torque will not exceed 45 foot pounds in any of our tests. The differential is a well proven unit used on golf carts and small vehicles of all sorts. The 4.11 to 1 gear ratio is rather high for our purposes, but no lower gear ratio differential was available.

In order to provide proper matching between wheel rpm and AC drive unit RPM, a slight speed increase was required and is provided by the silent chain drive. The drive uses 45 tooth driven and 38 tooth driven sprockets. Chain size is 1/2 inch pitch and 1-1/2 inches in width and is capable of accommodating a 20 HP AC drive.

The overall gear ratio for the system (K) equals the product of all gear ratios in the system. Therefore,
K = 1 x 4.11 x .844 = 3.47 to 1.

$$\text{Drive wheel rpm } N_w = \frac{\text{velocity (V) (feet/minute)}}{\text{circumference of wheel (2 r) (feet)}}$$

$$\text{AC Drive unit rpm } N_1 = KN_w = 3.47\ N_w$$

The above formulae were used to compute the following tabulation of vehicle speed, drive wheel speed and AC drive unit speed.

Speed Chart

Speed Mode	Vehicle Speed (MPH)	Drive Wheel RPM	AC Drive Unit RPM
Normal - Up 6% Grade	10	280	972
Normal - Level	15	420	1458
Normal - Down 6% Grade	20	560	1944
Low - Up 6% Grade	3.5	98	340
Low - Level	5	140	486
Low - Down 6% Grade	6.5	182	632

Drive wheel loads are analyzed in Appendix C. The drive wheels currently on the bogie are adequate for the 8000 pound vehicle contemplated for testing at the Los Angeles County Fair Grounds, but would have to be enlarged or made with a stronger tread material if a full 16,000 pound vehicle were to be used. The drive wheels operate at about 50%

above rated load on the normal speed down grade (20 mph) portion of the test course. However, the wheels are capable of a short time 100% overload. Very little testing will be done in the downgrade mode.

Shafts, bearings and drives have been analyzed for strength and operating life and have been found to be fully adequate. See Appendices C and D for analysis.

5.4.1.2 Guide Wheels

The direction of the bogie is maintained by pairs of guide wheels which roll along the vertical web of the monorail beam. A pair of guide wheels is present at each end of the bogie. See Fig. 5-10. Pairs of guide wheels are mounted on opposite sides of the web. As a web curves in a turn the guide wheels follow the web. The pairs of driving wheels maintain their path on the flange of the monorail beam, since the drive wheels and guide wheels are connected to the same yoke assembly. The yoke has a thrust bearing to allow the sets of wheels to swivel relative to the cab of the vehicle as the vehicle rounds the curves.

The guide wheels are 4 inches in diameter with a 2 inch wide rubber tread.

5.4.1.3 Suspension

The bogie provides a Belleville washer spring suspension to damp out oscillations imparted to the vehicle by its motion and the motion of the passengers. One major advantage of Belleville washers is that they provide a compact, simple suspension spring system with built-in damping. As a column of Belleville washers deflects under oscillating loads, Coloumb friction between the washers provides a degree of damping to eliminate the oscillations caused by a given shock. The damping factor can currently be evaluated only through a test of the suspension under design loads.

Another major advantage is that all that is required to adapt the bogie suspension to a given vehicle size is to use a Belleville washer of a different thickness and in a different quantity.

The suspension for the bogie constructed as a part of this program was designed as follows:

The weight of the test vehicle was estimated at 8,000 pounds. Each bogie will thus suspend 4,000 pounds. The beam on which the test vehicle will ride is a 27 inch wide flange beam with a weight of 114 pound/foot, with a span of 50 feet. It is desirable to have a ratio between natural frequency of the beam (f_b) and the natural frequency of the vehicle (f_v) on the order of four to one to prevent resonance in the suspension system.

f_b will be at a maximum when the vehicle is at the center of the span. The pertinent equations are:

$$f_b = \frac{3.13}{\sqrt{\Delta_b}} \tag{5-1}$$

Where f_b = natural frequency of the beam (cycle/sec);
Δ = deflection of the beam under a given load (inches).

The deflection Δ of the span will be:

$$\Delta = \frac{Pl^3}{48EI} \tag{5-2}$$

Where P = load (pounds), l = span of beam (inches), E = modulus of elasticity (pounds/square inch), I = moment of inertia (inches^4).

The spring constant (K) for the span will be:

$$K_b = \frac{P}{\Delta} \text{ lb./in.} \qquad (5\text{-}3)$$

For the Los Angeles County Fairgrounds monorail beam under load by the 8,000 pound test vehicle.

$$\Delta_b = \frac{8000 \times 600^3}{48 \times 29 \times 10^6 \times 4080.5}$$

$$= .304 \text{ inches}$$

$$f_b = \frac{3.13}{\sqrt{\Delta_b}} = \frac{3.13}{\sqrt{.304}}$$

$$f_b = 5.68 \text{ cycles/sec}$$

$$K_b = \frac{P}{\Delta} = \frac{8000}{.304}$$

$$= 26320 \text{ pound/inches}$$

5.4.1.4 Suspension Calculations

As has been stated before,

$$\frac{f_b}{f_v} = 4 \qquad (5\text{-}4)$$

and

$$f_v = \frac{3.13}{\sqrt{\Delta_v}} \text{ for the vehicle}$$

By substituting equation (5-1) into equation (5-4) we obtain:

$$\frac{\frac{3.13}{\sqrt{\Delta b}}}{\frac{3.13}{\sqrt{\Delta_v}}} = 4; \frac{\sqrt{\Delta_v}}{\sqrt{\Delta b}} = 4; \frac{\Delta_v}{\Delta b} = 16 \qquad (5-5)$$

Substituting equation (5-3) into equation (5-5) we obtain:

$$\frac{\frac{P}{k_v}}{\frac{P}{k_b}} = 16; \frac{k_b}{k_v} = 16$$

$$k_v = \frac{k_b}{16} = \frac{26,320}{16} = 1650 \text{ lb./in.}$$

The vehicle is supported by two bogies each with a spring constant k_1. The body of the vehicle also has a spring constant k_2. The overall spring system will be as shown in Figure 5-11.

The body of the vehicle is assumed to have the same spring constant (k_2) as the overall bogie suspension (k_s).

From the rules for resolution of spring systems (Ref. Den Hartog, Mechanical Vibrations, 4th ed., 1956, p. 36), the over-all suspension spring constant $K_s = k_1 = 2k$;

and $$k_v = \left[\frac{1}{k_2} + \frac{1}{k_s}\right]^{-1}$$

$k_2 = k_s$ as stated before.

Substituting, we obtain:

$$k_v = \left[\frac{1}{k_s} + \frac{1}{k_s}\right]^{-1} = \left[\frac{2}{k_s}\right]^{-1} = \frac{k_s}{2};$$

$$k_s = 2k_v;$$

$$K_s = 2k_1$$

$$2k_v = 2k_s = 2k_1$$

Therefore $k_v = k_1$

As has been determined before,

$$k_v = 1650 \text{ lb. /in.}$$

Therefore $k_1 = 1650$ lb. /in.

The bogie was constructed with a suspension having a k_1 of 1700 pounds/inch. The Belleville washer used is stocked by Bearing Engineers of Los Angeles. One washer will deflect 0.077 inch under a 4000 pound load. A stack of 31 washers has a k_1 of 1700 pounds/inch. Δ and f for the suspension of an 8,000 pound vehicle may be computed from equations (5-3) to (5-5):

$$\Delta_1 = \frac{P}{k_1} = \frac{4000}{1700} = 2.353 \text{ inches}$$

$$f_1 = \frac{3.13}{\Delta} = \frac{3.13}{2.353} = \frac{3.13}{1.534} = 2.04 \text{ cycles/sec.}$$

f_b is 5.68/2.04 or 2.78 times as great as f_1. There is no danger of the beam and bogie suspension resonating.

The maximum vibration caused by the roadway will come from deflection between the spans - at 15 mph (22 feet/sec.) the period of the vibration would be $\frac{22 \text{ feet/sec.}}{50 \text{ feet}} = 0.44$ cycles/sec., far below the resonant frequencies of the beam or bogie suspension.

The system was surveyed for the presence of frequencies that could cause resonance in the suspension. The tabulations in previous sections for AC drive unit and drive wheel rpm were compared with the beam and suspension natural frequencies. AC drive unit frequencies are far above the frequency of the suspension, so no problem is anticipated. Some drive wheel frequencies are close to the natural frequency of the suspension, such as for the vehicle at low speed on the level. There should be no force to excite resonance in the suspension of the vehicle unless the drive wheels get out of round.

The system will be tested with the Belleville spring suspension and also with no springs in the bogie. It can thus be determined whether the rubber tread of the wheels alone provide enough springing and damping for a comfortable people mover vehicle at speeds of 20 mph or below.

Degrees of freedom are built into various parts of the suspension to allow proper tracking of the wheels and leveling of the vehicle. (See Figures 5-10 and 5-11.) A ball joint at the Belleville washer installation allows the vehicle to remain level while the bogie is on a grade.

A degree of rotation in the horizontal plane is allowed for swiveling of the sets of drive wheels when the bogie negotiates a curve. The yokes which carry the sets of drive wheels and transmissions also swivel in the horizontal plane so that the drive wheels will track properly on curves.

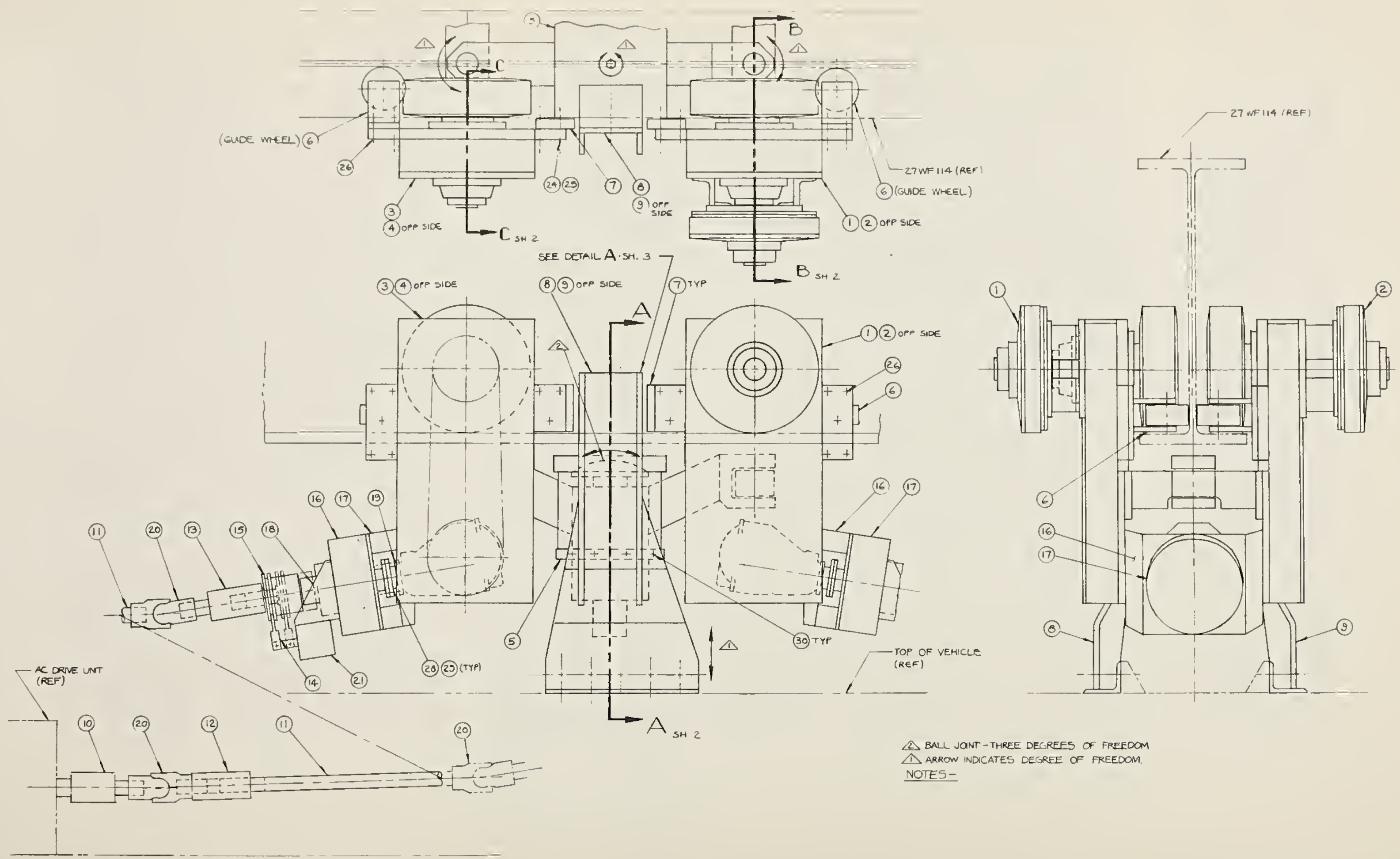


Figure 5-10 BOGIE ASSEMBLY (Reference Pages: 76, 77)

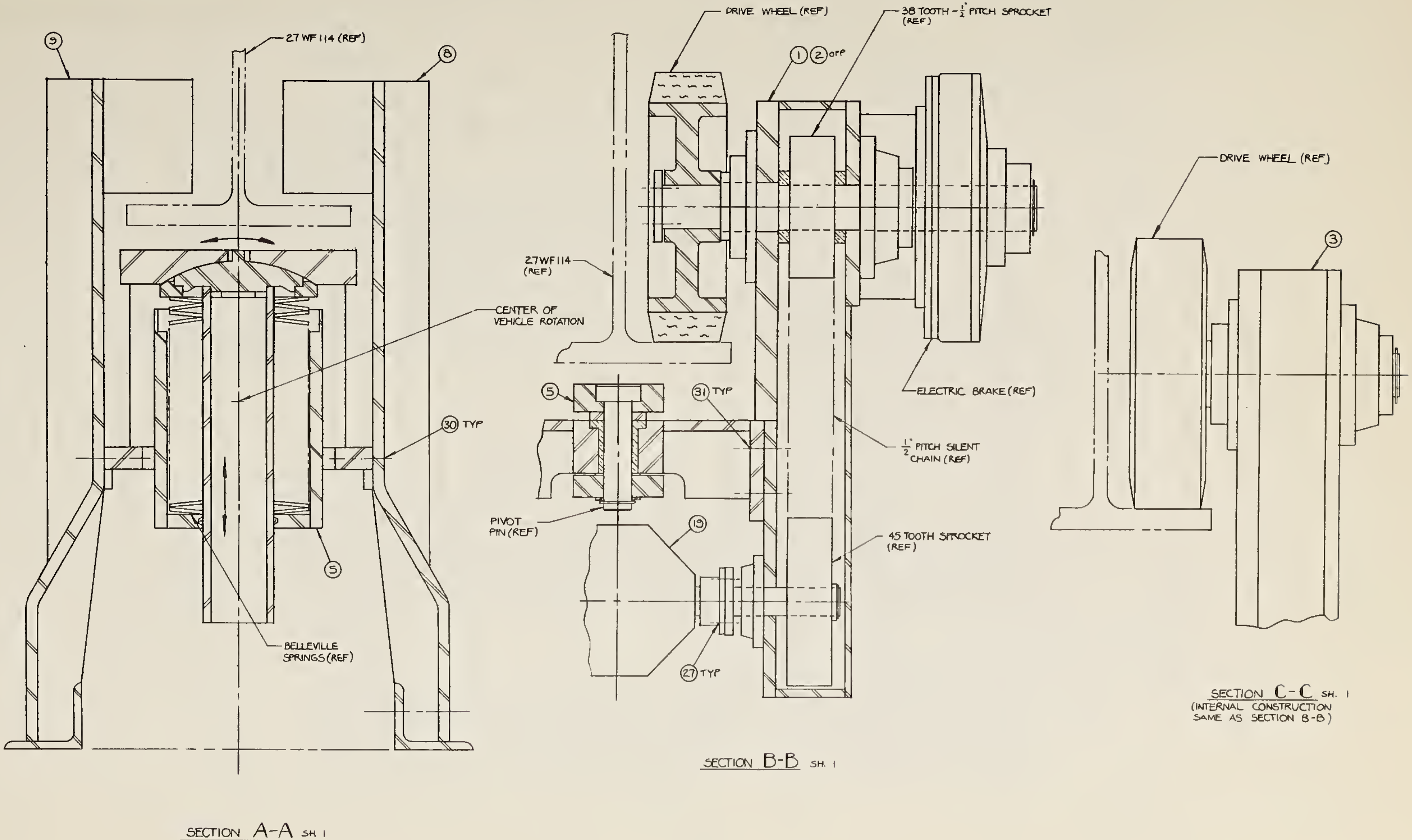


Figure 5-11 BOGIE ASSEMBLY (Reference Pages: 76, 77)

Components of Drive Bogie Assembly

Item	Req'd	Part No.	Description	Overall Dims (in.)	Unit Weight (lbs)
1	1	D-2030-1	Drive Wheel Assy.	28x17.5x12.87	357
2	1	D-2030-2	Drive Wheel Assy.	28x17.5x12.87	357
3	1	D-2031-1	Drive Wheel Assy.	28x12.87x12.18	347
4	1	D-2031-2	Drive Wheel Assy.	28x12.87x12.18	347
5	1	D-2032	Suspension Assy.	31x16.88x13	248
6	4	B-2033	Guide Wheel Assy.	12x7x5.62	.75
7	4	B-2034	Retainer Block	3.62x3.62x1	.75
8	1	D-2035-1	Bracket, R. H.	29.88x14.25x4	75
9	1	D-2035-2	Bracket, L. H.	29.88x14.25x4	75
10	2	C-2036	Adapter	4.5x3.12 DIA	12
11	2	B-2037	Univ. Shaft	23.13x1.375 DIA	10
12	2	B-2038	Coupling	5.5x2.50 DIA	8
13	2	B-2039	Coupling	5.12x2.62 DIA	8
14	2	B-2040	Shifter Fork	3.12x2.50x.75	1
15	2	B-2041	Shifter Gear	5.8x4.18 DIA	5
16	2	C-2042	Transmission	8.0x9.0 DIA	68
17	2	C-2043	Bracket	9.0x13x3.50	2
18	2	C-2044	Bracket	8.75x55x4.25	1
19	2	Model A12-1	Differential	10.81x6.88x9.66	23
20	4	UJNL 22-22	Univ. Joint	5.88x2.25x2.25	3.25
21	2	149-1	Solenoid	3.25x3.25x2.63	2
22	2		Screw, Hex.	#10-32UNF-2Ax2 3/4	2
23	2		Nut, Hex.	#10-32UNF-2B	-
24	12		Bolt, Hex.	1/4-20UNC-2Ax2 1/2	-

Components of Drive Bogie Assembly (continued)

Item	Req'd	Part No.	Description	Overall Dims (in.)	Unit Weight (lbs)
25	12		Nut, Hex.	1/4-20UNC-2B	-
26	20		Bolt, Hex.	5/16-18UNC-2Axl 1/2	-
27	16		Bolt, Hex.	5/16-24UNF-2Ax 3/4	-
28	8		Bolt, Hex.	5/16-24UNF-2Axl	-
29	8		Nut, Hex.	5/16-24UNF-2B	-
30	8		Bolt, Hex.	1/2-13UNC-2Axl 1/4	-
31	16		Bolt, Hex.	1/2-20UNF-2Ax 1/1/4	-
				Total Weight	1952

5.4.1.5 Emergency Braking System

The design of the Bogie incorporates a failsafe mechanical emergency brake to stop the vehicle if electrical power failure occurs. (See Figure 5-12) When electrical power is applied to the bogie drive, the solenoid lowers the racks which in turn causes the ratchet wheels to rotate out of contact with the rail. If electrical power fails when the bogie is in motion, the racks retract, rotating the ratchet wheels into contact with the running surface of the monorail beam. The forward motion of the bogie wedges the ratchet wheels tightly against the beams causing a resistance which quickly brings the bogie (which is coasting without power) to a stop.

The deceleration rate that this system can provide will have to be evaluated by test.

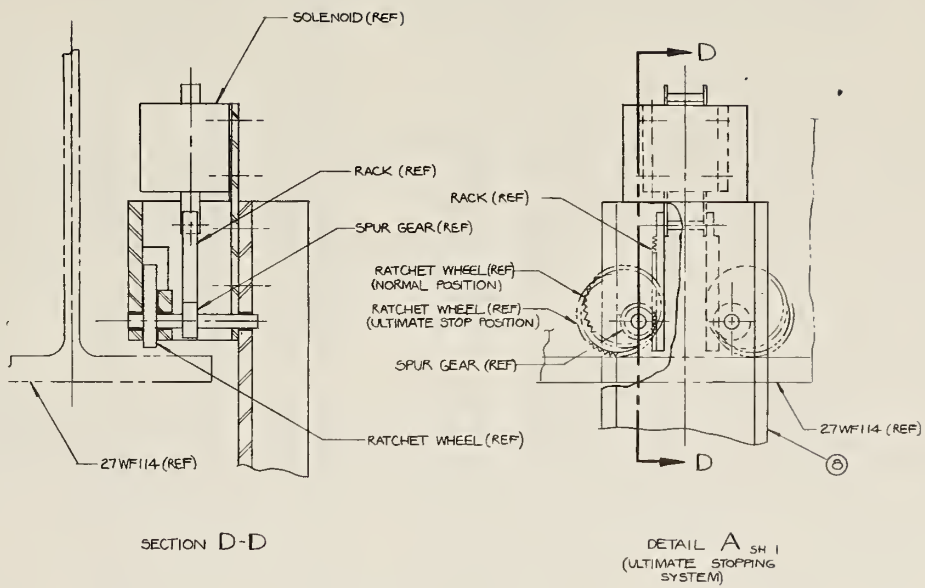


Figure 5-12 BRAKE ASSEMBLY

6.0 FABRICATION

Subsequent to the preparation of a full set of Engineering drawings, the individual components were fabricated and assembled at the MS&E shop, which is well equipped to handle the major portion of work associated with this type of drive equipment. The few elements requiring special tooling such as splines and precision gears were subcontracted to specialty shops acting in support of this effort.

Wherever possible, standard, off-the-shelf items having high reliability and good experience factors were incorporated into the design to ensure system performance and durability. Suppliers having national distribution networks were preferentially selected to preclude excessive down-time in case component replacements were required.

Photographs of the major items are shown in Appendix G; the physical characteristics are as shown in Table 6.1 below, and the drive bogie bill of materials is presented in Table 6.2. The Eddy-Current Drive unit is shown in Figure 6-1 with the major elements identified.

Table 6.1 - Component Sizes and Weights for Drive Package for a 16,000-lb. gross weight PRT Vehicle.

Item	Size (inches)			Weight (lb)
	Length	Height	Width	
Eddy-current drive	40.38	12.50	12.50	480
Bogie drive assembly	47.00	36.25	37.50	1952
Controller	15	52	36.25	70
Operator's console	12	12	16	30

Table 6.2 - Components of the Drive Bogie

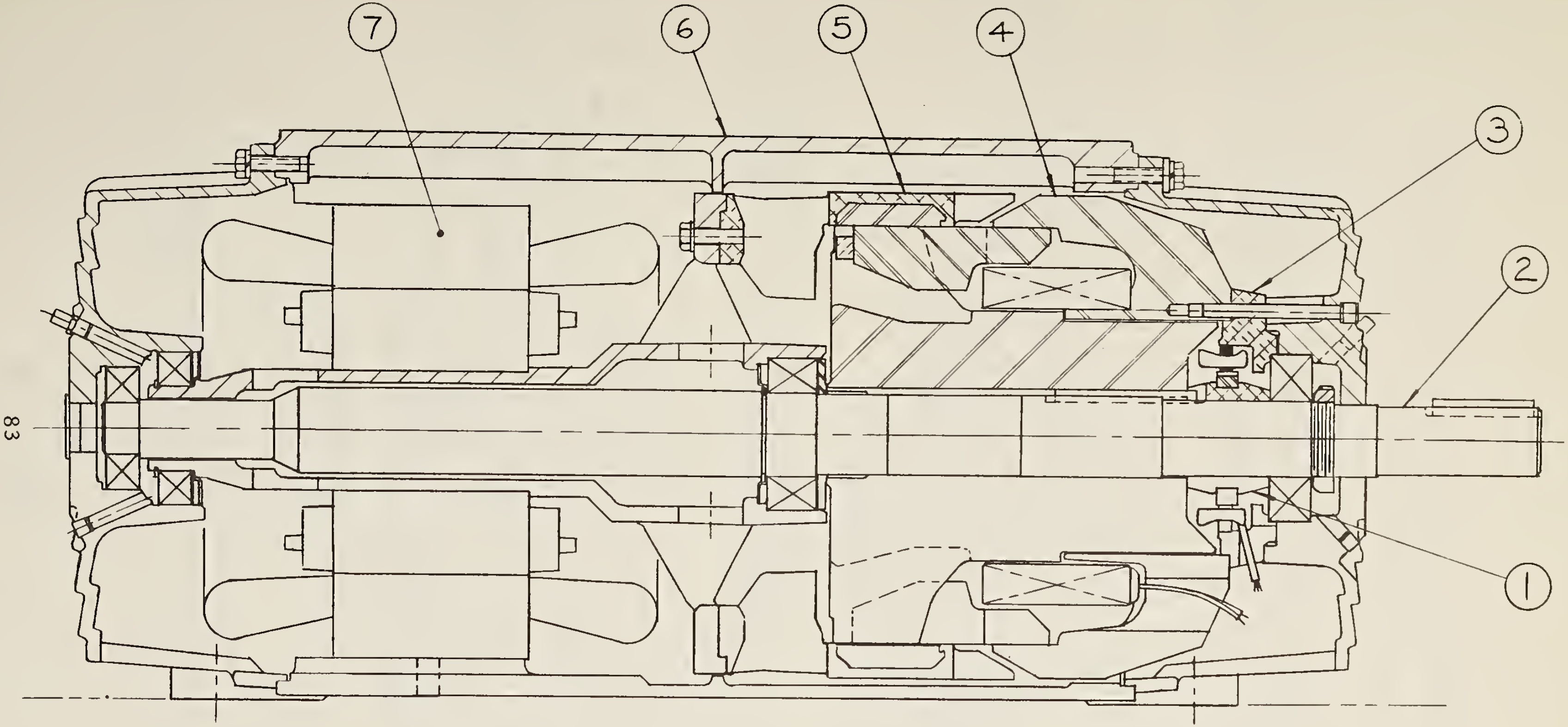
Bill of Materials

Item	Req'd	Part No.	Description	Overall Dims (in.)	Unit Weight (lbs)
1	1	D-2030-1	Drive Wheel Assy.	28x17.5x12.87	357
2	1	D-2030-2	Drive Wheel Assy.	28x17.5x12.87	357
3	1	D-2031-1	Drive Wheel Assy.	28x12.87x12.18	347
4	1	D-2031-2	Drive Wheel Assy.	28x12.87x12.18	347
5	1	D-2032	Suspension Assy.	31x16.88x13	248
6	4	B-2033	Guide Wheel Assy.	12x7x5.62	.75
7	4	B-2034	Retainer Block	3.62x3.62x1	.75
8	1	D-2035-1	Bracket, R. H.	29.88x14.25x4	75
9	1	D-2035-2	Bracket, L. H.	29.88x14.25x4	75
10	2	C-2036	Adapter	4.5x3.12 DIA	12
11	2	B-2037	Univ. Shaft	23.13x1.375 DIA	10
12	2	B-2038	Coupling	5.5x2.50 DIA	8
13	2	B-2039	Coupling	5.12x2.62 DIA	8
14	2	B-2040	Shifter Fork	3.12x2.50x.75	1
15	2	B-2041	Shifter Gear	5.8x4.18 DIA	5
16	2	C-2042	Transmission	8.0x9.0 DIA	68
17	2	C-2043	Bracket	9.0x13x3.50	2
18	2	C-2044	Bracket	8.75x55x4.25	1
19	2	Model A12-1	Differential	10.81x6.88x9.66	23
20	4	UJNL 22-22	Univ. Joint	5.88x2.25x2.25	3.25
21	2	149-1	Solenoid	3.25x3.25x2.63	2
22	2		Screw, Hex.	#10-32UNF-2Ax2 3/4	2
23	2		Nut, Hex.	#10-32UNF-2B	-
24	12		Bolt, Hex.	1/4-20UNC-2Ax2 1/2	-

Table 6.2 - Components of the Drive Bogie (Cont'd.)

Bill of Materials

Item	Req'd	Part No.	Description	Overall Dims (in.)	Unit Weight (lbs)
25	12		Nut, Hex.	1/4-20UNC-2B	-
26	20		Bolt, Hex.	5/16-18UNC-2Axl 1/2	-
27	16		Bolt, Hex.	5/16-24UNF-2Ax 3/4	-
28	8		Bolt, Hex.	5/16-24UNF-2Axl	-
29	8		Nut, Hex.	5/16-24UNF-2B	-
30	8		Bolt, Hex.	1/2-13UNC-2Axl 1/4	-
31	16		Bolt, Hex.	1/2-20UNF-2Ax 1/1/4	-
				Total Weight	1952



1	Generator Rotor	5	Rotor and Spindle Assembly
2	Shaft	6	Housing
3	Generator Field Assembly	7	Stator
4	Clutch Field Assembly		

Figure 6-1 AC EDDY-CURRENT DRIVE (CROSS SECTION)

7.0 TEST PROGRAM

7.1 OBJECTIVES

Objectives of testing were to provide the capability to extrapolate AC Eddy-current drive unit concepts for other design configurations, service requirements, duty cycles and performance of PRT Systems.

The test data provided in this report will assist engineers and designers to apply AC Eddy-current clutch drive systems to PRT Systems, and to achieve predictable performance. It will also aid in the selection of the most satisfactory unit or system in terms of power and acceleration/deceleration controls for a determined duty cycle or application.

7.2 TEST PROGRAM PROCEDURE

The testing of an AC drive unit entailed the demonstration of drive unit performance under the conditions of various operational duty cycles.

The first duty cycle was for an existing PRT system, the "Jetrail", operating at Love Field, Dallas, Texas. The second was a theoretical duty cycle called the "Optimized" system, which was developed to tax the drive unit to near its maximum capacity. Both duty cycles included the detailed characteristics of a particular route which included acceleration, deceleration, aerodynamic forces, friction forces and grades.

7.3 TEST EQUIPMENT AND COMPONENTS

7.3.1 <u>Dynamometer</u>

The dynamometer used in the test was an Eddy-current device using water cooling for heat dissipation. The unit, as pictured in Figures 7-3 and 7-4, Appendix E, is Model No. 1014DG, manufactured by the Eaton Corporation.

Dynamometer loading is produced by applying a variable low voltage to the field coil. Voltage vs. load was not calibrated; therefore, actual dynamometer load was determined through the use of an external static load cell which was connected to a readout device. Prior to the test, the external load cell was calibrated and adjusted using a set of standard weights.

7.3.2 Selection of Inertia Flywheel

Restraining forces (vehicle mass inertia) due to acceleration and deceleration were simulated by a flywheel. Linear force can be converted to rotary force by application of the following formulae as developed in paragraph 4.2.6.1:

$$\text{Inertia IR} = \frac{W\ V^2}{.00034(\text{RPM})^2 g}$$

where W = vehicle gross weight (lbs)
V = vehicle max velocity (mph)
RPM = max drive rotary speed
g = gravitational constant = 3.2.2 ft/sec^2

7.3.2.1 Inertia Mass Data - Jetrail

In Test No. 1, "Jetrail", an 8,000 - pound vehicle system with a top speed of 15 MPH was simulated. The inertia of the flywheel was calculated as follows:

where W = 8,000 lbs.
V = 15 MPH = 22 ft/sec
RPM = 1,610

or

$$I_1 = \frac{(8000)\ (22)^2}{(.00034)(1610)^2(32.2)} = 137 \text{ ft/lb}^2$$

7.3.2.2 Inertia Mass Data - "Optimized" System

Since Test # 2, the "Optimized" system, had a duty cycle with higher grades and greater speeds, two drives were required, or one drive for each "half" of the system. Since it was necessary to test only one drive unit, the weight of the total system was divided in two in order to represent the weight allocation to each individual drive. For the "Optimized" system in Test # 2:

$$\text{where } W = \frac{8000 \text{ lbs.}}{\#\text{ of drive motors}} = \frac{8000}{2} = 4000 \text{ lbs.}$$

$$V = 30 \text{ MPH} = 44 \text{ ft/sec}$$

$$RPM = 1,610$$

or

$$I_2 = \frac{(4000)(44)^2}{(.00034)(1610)^2(32.2)} = 274 \text{ ft/lb}^2$$

7.3.2.3 Flywheel Data

Knowing the inertia mass of the vehicle, the characteristics of the flywheel which will simulate the vehicle's acceleration/deceleration can be calculated. Before calculating the mass of the required flywheel, however, inertia mass must be considered for the 1014DG dynamometer used in the test. Manufacturer's data gives 16 ft-lb^2 for the rotary component used in the test. For Test # 1 therefore, the inertia which must be simulated, I_S, is:

$$I_{S_1} = I_1 - 16 = 137 - 16 = 121 \text{ ft-lb}^2$$

For Test # 2 the inertia to be simulated is:

$$I_{S_2} = I_2 - 16 = 274 - 16 = 258 \text{ ft-lb}^2$$

Simulation of the inertia mass was accomplished by attaching a one inch thick disk of low carbon steel to the shaft of the dynamometer. By selection of the proper disk radius the required inertia mass was available. The following calculations verify the inertia mass of the flywheels used:

For a flywheel, inertia equals:

$$I = \frac{W\ r^2}{2}$$

where W = weight (lbs.)

r = wheel radius (ft.)

For a uniform flywheel of thickness "d" and with a density of "ρ"

$$W = \pi r^2 (d)(\rho)$$

For the flywheel used in Test # 1:

$r = 14.05$ inches

$d = 1$ inch

$\rho = .283 \text{ lb/in}^3$ for low carbon steel

or

$$w = \pi(14.05)^2(1)(.283) = 177 \text{ lbs.}$$

For Test # 2 a second flywheel with a 14.5 inch radius was attached to the first flywheel. The total flywheel weight for Test # 2 was:

$$W_2 = W_1 + \pi(14.5)^2(1)(.283) = 177 + 187 = 364 \text{ lbs.}$$

Therefore, inertia of the flywheel for Test # 1 was:

$$I = \frac{(177)(14.05)^2}{2} = 121 \text{ ft-lb.}$$

Inertia of the flywheel for Test # 2 was:

$$I_2 = I_1 + \frac{(187)(14.5)^2}{2} = 121 + 137 = 258 \text{ ft-lb.}$$

7.3.2.4 Dynamic Torque

Dynamic Torque refers to the rotary force, or torque, required to accelerate vehicles or flywheels up to a given speed level. For a flywheel, torque is:

$$T = .00326\ I\ (\frac{\Delta RPM}{\Delta t}) \text{ ft-lb.}$$

For the single flywheel used in Test # 1:

$$I = 137 \text{ ft-lb}^2$$

or

$$T = .466\ [\frac{\Delta RPM}{\Delta t\ (sec)}] \text{ ft-lb.}$$

For the double flywheel used in Test # 2:

$$I = 274 \text{ ft-lb}^2$$

or

$$T = .4466[\ \frac{\Delta RPM}{\Delta t\ (sec)}\ ] \text{ dt-lb.}$$

Figure 7.1 is a plot of the above equations and is used as a convenient guide to determine dynamic torque when the acceleration rate is specified for either the "Jetrail" or the "Optimized" system.

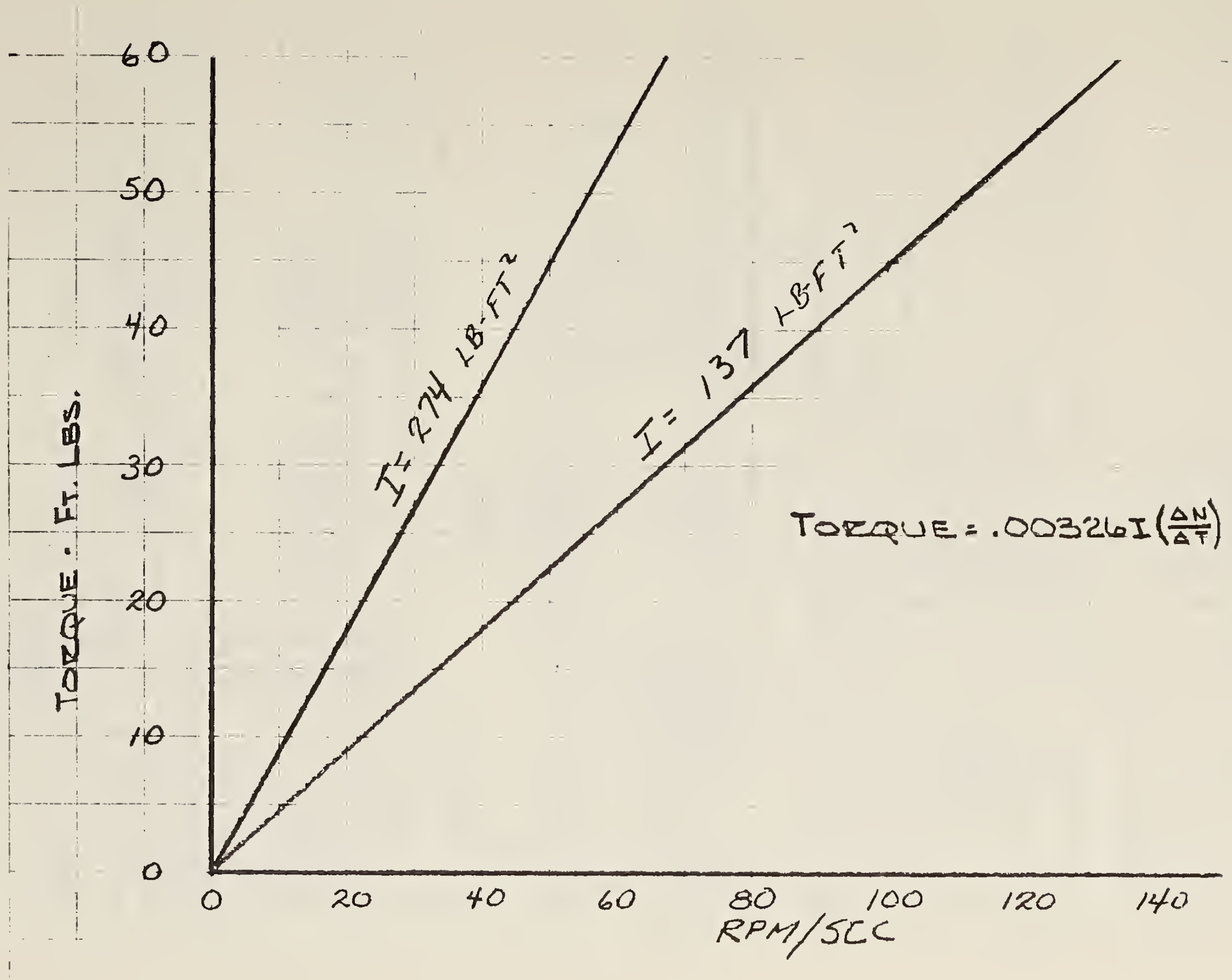


Figure 7-1 TORQUE DURING ACCELERATION

7.3.3 Dynamometer Torque Procedure

The duty cycle being applied during the test entailed various levels of static torque for various time intervals. This required rigid adherence to the torque-time schedule as previously defined in Tables 4.5 and 4.6 (Appendix B).

The procedure required two technicians. One technician observed the torque schedule and a timer. As time approached for a torque change, the second technician was alerted and verbally given the order for a new torque value. The second technician then adjusted the dynamometer field voltage through a small potentiometer setting. The direct readout from the load cell aided him in setting the correct torque value.

After a few initial trial runs, both technicians coordinated their efforts so they were able to make the torque change quickly with the required accuracy.

In Figure 7-3, the torque readout device is visible. The potentiometer for setting torque is not visible.

7.3.4 Test Equipment

All equipment for the test was centered around the AC Eddy-current drive unit and the dynamometer which were firmly attached in test fixtures and solidly anchored to a steel-girder floor.

A diagram of the test setup is shown in Figure 7-2. The photographs in Figures 7-3 and 7-4 (Appendix E) were taken during the test and show the Model PM-001 MSE variable speed drive and the MODEL ASC-110 MSE automatic speed control unit, as well as the overall test setup.

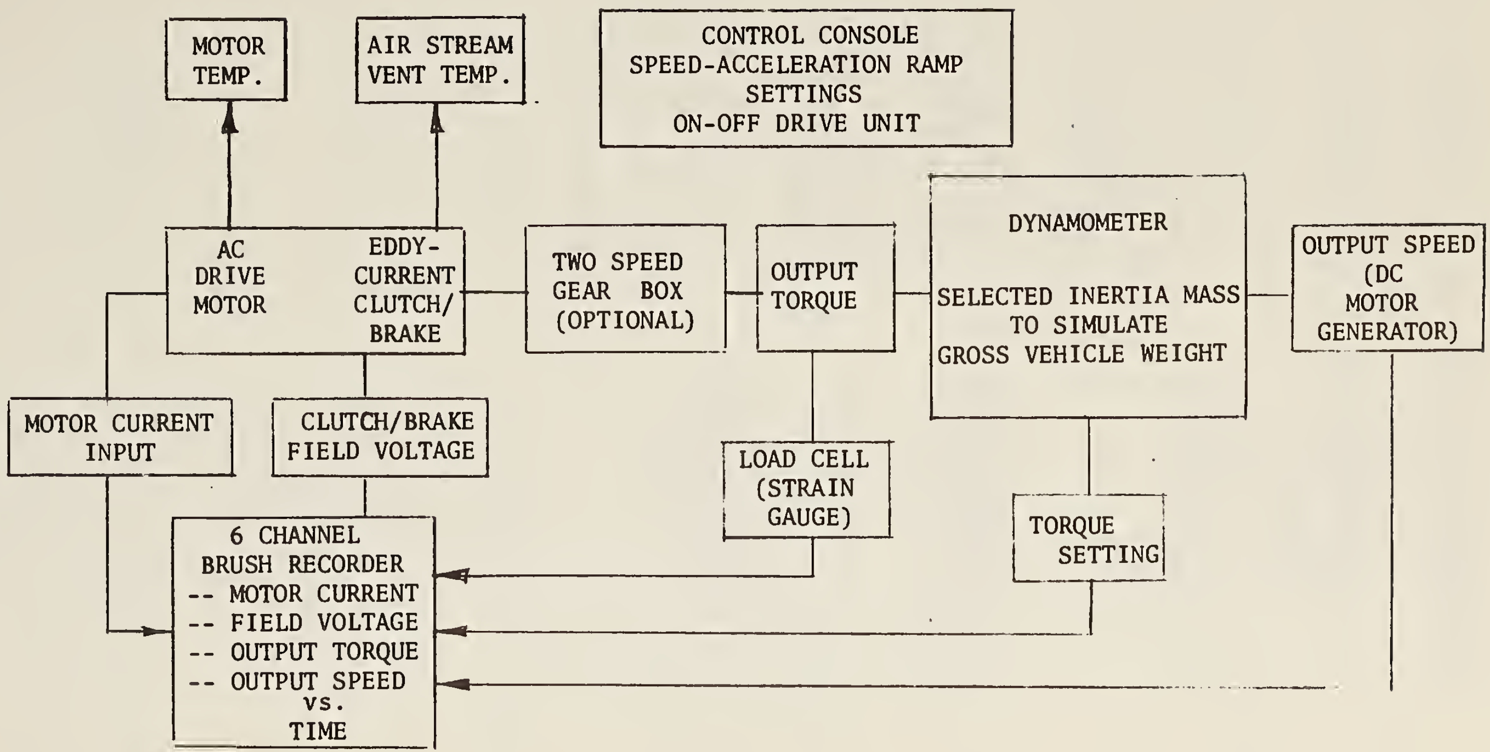


Figure 7-2 DYNAMOMETER TEST ARRANGEMENT

	AC Eddy-current Drive	Description
1)	Variable speed drive, Model PM-001 MSE	15-Horsepower Eddy-current Clutch/Brake drive unit.
2)	Drive Controller, Model ASC-110 MSE	Solid state controller which regulates speed, acceleration and maintains all drive functions and voltage to design levels.
3)	Operator's Panel	A push-button console device which is pre-set for up to four desired speeds. This is the prime input to the Drive Controller.
4)	Mechanical Brake	A friction fail-safe brake which operates when power is removed from the drive unit.

Dynamometer		Description
1)	Eaton — Dynamatic, Model 1014 DG.	Eddy-current dynamometer with water cooling.
2)	Dynamometer Controller	Eddy-current controller device, using a small potentiometer as an input signal.
3)	External tachometer	Was available but not used.
4)	Static torque Load Cell	Measured a calibrated deflection of the dynamometer as a static torque reading.
5)	Dynamometer torque Read-out Device	Connected to the static torque load cell, it gives a direct reading of dynamometer torque absorption. Used to adjust dynamometer torque level.

Data Equipment		Description
1)	Brush Data Recorder	A 6-channel data recorder. (1 channel was inoperative.)
2)	Mercury thermometer	A simple glass thermometer mounted in proximity to the clutch hot air exhaust. Temperatures were monitored periodically.
3)	Meter-thermometer	A metal cased thermometer using a coil-spring dial. Mounted on the drive unit's frame. Temperatures were monitored periodically.

7.4 TEST DUTY CYCLES

The test duty cycle represents the static torque values applied in the dynamometer. It is the amount of torque absorbed by the dynamometer (and dissipated as heat). The test duty cycle, as previously discussed, is representative of the power necessary to move a PRT vehicle along a particular route. Using the technique described in Paragraph 4.2, static torque has been calculated for each increment or phase of the route. Then it was merely a matter of organizing the data into a form that could be tested in the dynamometer.

7.4.1 Jetrail Test Duty Cycle

Figure 4-2 is the guideway route for the Jetrail network. By combining the schedule given in Table 4.4 and the power requirements of Paragraph 4.3.1, the finalized schedule for dynamometer testing, Table 7.1, has been formulated. The test duty cycle for the "Jetrail" system in Table 7.1 (Appendix E) includes performance requirements of the vehicle. Explanations of these terms are as follows:

Under phase conditions, ACC denotes acceleration of the drive unit; CRU denotes cruise of the motor at steady state conditions; DEC denotes deceleration of the drive unit. Velocity, distance, time, output shaft speed of the drive and dynamometer torque are shown for each incremental phase of the duty cycle. Torque requirements during acceleration and deceleration are not listed, as the acceleration of the flywheel inertia mass to the final speed is internally experienced by the drive unit. Inertia mass of the flywheel for test run No. 1 is equivalent to an 8,000 pound vehicle with a single 15 HP drive and a maximum speed of 15 mph.

7.4.2 "Optimized" Vehicle Test Duty Cycle

Motor testing of the optimized vehicle route provided more comprehensive information of the capabilities of an AC drive unit to perform under continuous heavy loading situations and at higher speeds.

Figure 4-3 is the guideway route of the optimized network.

Again using the techniques described in Paragraph 4.2, static torque has been calculated for each increment or phase of the vehicle route. By incorporating these calculations with the finalized schedule of Table 4.7, Table 7 2, Appendi C has been developed.

The test duty cycle for the optimized system in Table 7.2 includes performance requirements of the vehicle. Explanations of performance terms are the same as in Paragraph 7.4.1.

Torque requirements during acceleration and deceleration are not listed since these components of torque are inherently simulated by the inertia mass of the flywheel. Inertia mass of the flywheel for test run No. 2 is equivalent to an 8,000 pound vehicle with two similar 15 HP drive units (30 HP total) with a maximum vehicle speed of 30 mph.

8.0 TEST AND MEASUREMENT PROCEDURE

8.1 PROCEDURE

Tests were conducted over a two day period. The first day of testing, May 24, 1972, was with the "Jetrail" Duty Cycle in which the inertia mass of the flywheel was fixed at 177 pounds. The second day of testing was with the "Optimized" Duty Cycle, which required a larger flywheel mass of 364 pounds to simulate the higher speeds and 30 H.P. of the tested "Optimized" vehicle.

Specific test runs consisted of following the duty cycle schedule of speed and torque shown in Table 7-1 and Table 7-2 Appendix E. The sequence of tests as conducted are described in the log presented in Table 8.1.

During the test run of each duty cycle the speed of the drive was brought up from zero to its cruising speed. Both speed and restraining torque were programmed to give an accurate test of a "real life" PRT duty cycle.

Table 8-1. Test Log Data

Sequence	Description
1.0	First day of testing, start Jetrail duty cycle as described in Test Run #1. Duration approximately 60 minutes.
1.1	Acceleration to maximum speed (1610 RPM . Flywheel inertia only.
1.2	Acceleration to 775 RPM . Flywheel inertia only.
1.3	Acceleration to 385 RPM . Flywheel inertia only.
1.4	Maximum clutch dissipation. Run at low speed with 34 ft-lb of torque, followed by torque at 46 ft-lb. Total run time of 22 minutes. No excess heating observed.
1.5	"Jetrail" system duty cycle described in Test Run #1 was executed. Final run. Duration approximately 60 minutes. (Repeat of Sequence 1.0)
1.6	Acceleration to maximum speed (1610 RPM) then braking.
1.7	Acceleration to 775 RPM , then braking.
1.8	Acceleration to 385 RPM , then deceleration.
1.9	Acceleration to 200 RPM , then deceleration.
2.0	Second day of testing, start "Optimized" system tests, Test Run #2. Duration approximately 60 minutes
2.1	Restart test after 176 seconds.
2.2	9:43 - Overload breaker tripped off.
2.3	9:48 - Overload breaker tripped off.
2.4	10:10 - Overload breaker tripped off. Restart 12 minutes in test. Higher rated current limiter inserted.
2.5	10:35 - Overload Breaker tripped off. Restart at 10:37.
2.6	45 minutes, 7 seconds in Test Run #2 - Overload breaker tripped off.
2.7	Restart cycle at 45 minutes, 35 seconds into Test Run #2.
2.8	Overload circuit breaker tripped off.

Table 8-1. Test Log Data (Continued)

Sequence	Description
2.9	Higher rated current limiter inserted - restart at 10:58.
2.10	Optimized system duty cycle described in Test Run #2. Final run. Time: 11:06.
2.11	Readjust channel 5 (drive current) from 200 to 500 Sensitivity Scale.
2.12	Restart Test Run #2
2.13	Minor schedule readjustment of Test Run #2 due to technician's error.
2.14	End Test #2 - start acceleration - deceleration tests.
2.15	Emergency braking or stop, maximum speed (1610 RPM) to zero RPM. Torque = 21.4 ft-lbs.
2.16	Braking test, maximum speed. Torque = 35.2 ft-lbs.
2.17	Braking test, maximum speed. Torque = 45.0 ft-lbs.
2.18	Acceleration test. Torque = 45.0 ft-lbs.
2.19	10% grade, braking test. Torque = 60.0 ft-lbs.
2.20	Torque = 60.0 ft-lbs, acceleration test. High acceleration rate.
2.21	Torque = 60 ft-lbs. Slow acceleration rate.
2.22	Slow acceleration rate to second speed. Torque = 60.0 ft-lbs.
2.23	Slow acceleration to maximum speed. Torque = 60.0 ft-lbs.
2.24	High acceleration. Torque = 64.0 ft-lbs.
	- End of Laboratory Tests -

8.2 DRIVE CONTROL

During the test all operations related to the function of the drive were performed on a small pushbutton console. Those included Start, Idle, Stop, Speed 1, Speed 2, Speed 3, and Speed 4. Each function was performed by a simulated operator; however, in a working PRT system, the operation would follow a Wayside signal responding to an automatic control system developed by Mobility Systems and Equipment Company.

As can be seen from the replot of the original data in Appendix F, Figures F-1 and F-2, the planned duty cycle was followed as expected. Each phase of dwell, acceleration, cruise and deceleration was executed simply and without complications.

9.0 TEST RESULTS

The testing of Model PM-001 MSE adjustable speed, AC eddy-current clutch/brake was conducted in the Performance Testing Laboratory of the Eaton Corporation, Kenosha, Wisconsin, under the direction of George J. Adams and R. C. Roberg of Mobility Systems and Equipment Company. Monitoring the test was Joseph Abbas of Transporation Systems Center, Department of Transportation.

Tests were conducted over a two day period. The first day of testing, May 24, 1972, was with the "Jetrail" system, in which the inertia mass of the flywheel was fixed at 177 pounds. The second day of testing was with the "Optimized" system, which required a larger flywheel mass of 258 pounds to simulate the additional weight and speed of the "Optimized" system's cars.

Overall test goals of simulating the duty cycle of an operational PRT system, "Jetrail" at Love Field, Dallas, Texas, and an "Optimized" system were successfully met. Data was recorded on a Brush recorder during all phases of the tests, showing specific performance of the drive unit.

9.1 TEST DATA

Original test data is on a continuous recording from a Brush recorder, covering approximately 50 feet of tape for each test. Original data is on file at Mobility Systems and Equipment Company. A replot of the data from the duty cycle tests is included in Appendix F of this Report.

A definition of each data channel is as follows:

9.1.1 Data Measurement Description

Channel	Data Measurement	Description
1	Torque	Torque loading of the dynamometer as measured by a load cell, attached to the dynamometer.
2	RPM	Speed of the drive unit output shaft as measured from the internal tachometer in the drive unit. (Speed verified by an external tachometer on the dynamometer shaft.)
3	Reference Voltage Channel	Reference timing for change of speed signal voltage and all events.
4		Was not used due to an instrument problem.
5	Current	Drive unit current for 230 volts, 3-phase power.
6	Clutch Current	Control current to the eddy-current clutch coil at 50 volts, DC, maximum.
	Frame Temperature	Meter reading of motor housing temperature. Meter attached on top of frame over armature area. Temperature visually observed and recorded.
	Fan Temperature	Thermometer reading of temperature at eddy-current clutch exhaust fan outlet. Temperature visually observed and recorded.

9.1.2 Test Log

The prime objectives in the test entailed operation of the drive system over the previously described duty cycles. However, a number of auxiliary test runs were performed in support of these goals. The sequence of runs performed in support of these tests are identified in the test log in Table 8-1 in the previous section.

9.1.3 Overload Current Limiter - Heater Problem

During the initial run of Test #2 a problem was encountered in the overload circuitry of the controller. Standard procedures called for an overload current limiter heater matched to the full-load rated current of the drive. The drive unit for the test has a full-load rated current of 20.7 amps and normally used a Cutler Hammer No. 1042 heater in the controller, with a current rating of 18.9 to 21.2 amps. This heater caused the overload circuits to "trip off" under conditions of high torque demand.

The heater was changed to a Cutler Hammer No. 1043A with a current rating of 21.3 to 23.9 amps. Under certain high torque conditions which exceeded the drive rating, the over-load circuits "tripped off", so a larger heater was used. After the Cutler Hammer No. 1044H heater, with a 24.0 to 27.0 amps current range was inserted there was no additional "tripping" of the overload circuit in the control unit. Test Run #2, the "Optimized" duty cycle, was then restarted and continued until completion (approximately 60 minutes of total cycle time).

Although the drive was "forced" to operate above its rated range, the test showed the capability of the drive to perform the required duty cycle operations for this optimum capability test.

10.0 TEST ANALYSES

Testing of the AC drive unit with eddy-current clutch/brake entailed operation over two complete duty cycles. Overall results of this testing show conclusively that the eddy-current clutch/brake is a high-ranking drive system for PRT system applications.

Tests entailing duty cycle operations have the general result of showing either satisfactory performance which conforms to expectations or the occurrence of unanticipated difficulties which would curtail the test, or result in substandard performance. Since the tests ran successfully over the complete duty cycle, it is of interest to make a general assessment of performance and analyze the detailed parts of the test which may provide more understanding as to the operation and performance of the AC drive unit with eddy-current clutch/brake. The analysis and presentation of results therefore covers two areas:

1) Overall analysis of the test.

2) Analysis of the performance during distinct phases of each duty cycle.

10.1 OVERALL ANALYSIS OF THE TEST

The defined duty cycle for the Jetrail system and the "Optimized" system were successfully run.

10.1.1 Duty Cycle Test Run #1, Jetrail Simulation

Figure F-1, Appendix F, shows the data which was recorded during the test. The test was run through a number of cycles however, only the last complete cycle is shown. No difficulties were encountered during the test runs.

The drive responded in a normal, satisfactory manner to all commands and loads which were placed on the system.

10.1.2 Duty Cycle Test Run #2, "Optimized" System Simulation

Figure F-2, Appendix F, shows the data which was recorded during the test. Only the last complete cycle of the test run is shown.

Although the drive was operated above it rated range, the test showed the capability of the drive to perform the required duty cycle operations for this optimum capability test without serious performance degradation or over heating effects.

Operations above the normal motor rating, for continuous operation, for intermittent periods of time are within the standard expectations of a PRT.

It is noted that once a duty cycle for a particular application is established, the drive unit will be particularly engineered and fabricated for long term operation such that the drive unit will have a superior performance for that duty cycle.

During the final run of Test #2 the drive responded in a normal, satisfactory manner to all commands and loads which were placed on the system.

10.1.3 Analysis of Performance

Performance criteria for an AC drive unit with eddy-current clutch/brake falls into the following major categories:

1) Predictable or repeatable acceleration/deceleration.

2) Performance level - torque output.

3) Temperature.

10.1.4 Acceleration Phase

The torque speed history of the drive unit during acceleration, under high static torque conditions are shown in Figure 10-1 and 10-2. These test results are superimposed on the torque speed curve representing the rated capability of the drive.

Figure 10-1 is based on data from test Run 1 of the Jetrail system. Static torque was set at the maximum test value of 30 ft-lbs, and the drive was allowed to accelerate from zero to a full speed of 1610 RPM . The resulting total torque developed by the drive, a combination of the static and dynamic torque was then traced at different time points in the acceleration cycle. From this curve it can be seen how the drive traverses the speed-torque curve during acceleration. It is noted that the torque demands shown are, in general, near the maximum rated capability of the drive.

Figure 10-2 is the same as Figure 10-1 except the data is based on test run 2, the Optimized system. Static torque was set at 45 ft-lbs and the drive was again allowed to accelerate from zero to its full speed of 1610 RPM .

The time trace of torque vs. speed is near the maximum rated capability of the drive. It is noted that the acceleration time is substantially increased over the Jetrail simulation since we are dealing with a faster vehicle with a more difficult duty cycle.

In Figure 10-3 through 10-7 the drive speed and total torque output are shown during the acceleration phase with levels of static torque ranging from 4.0 ft-lb to 4.5 ft-lbs. It is noted that the duration of the acceleration phase increases substantially as the level of static torque increases. In Figure 10-8 the duration of the acceleration phase is shown for both the Jetrail system and the Optimized system, for various values of static torque.

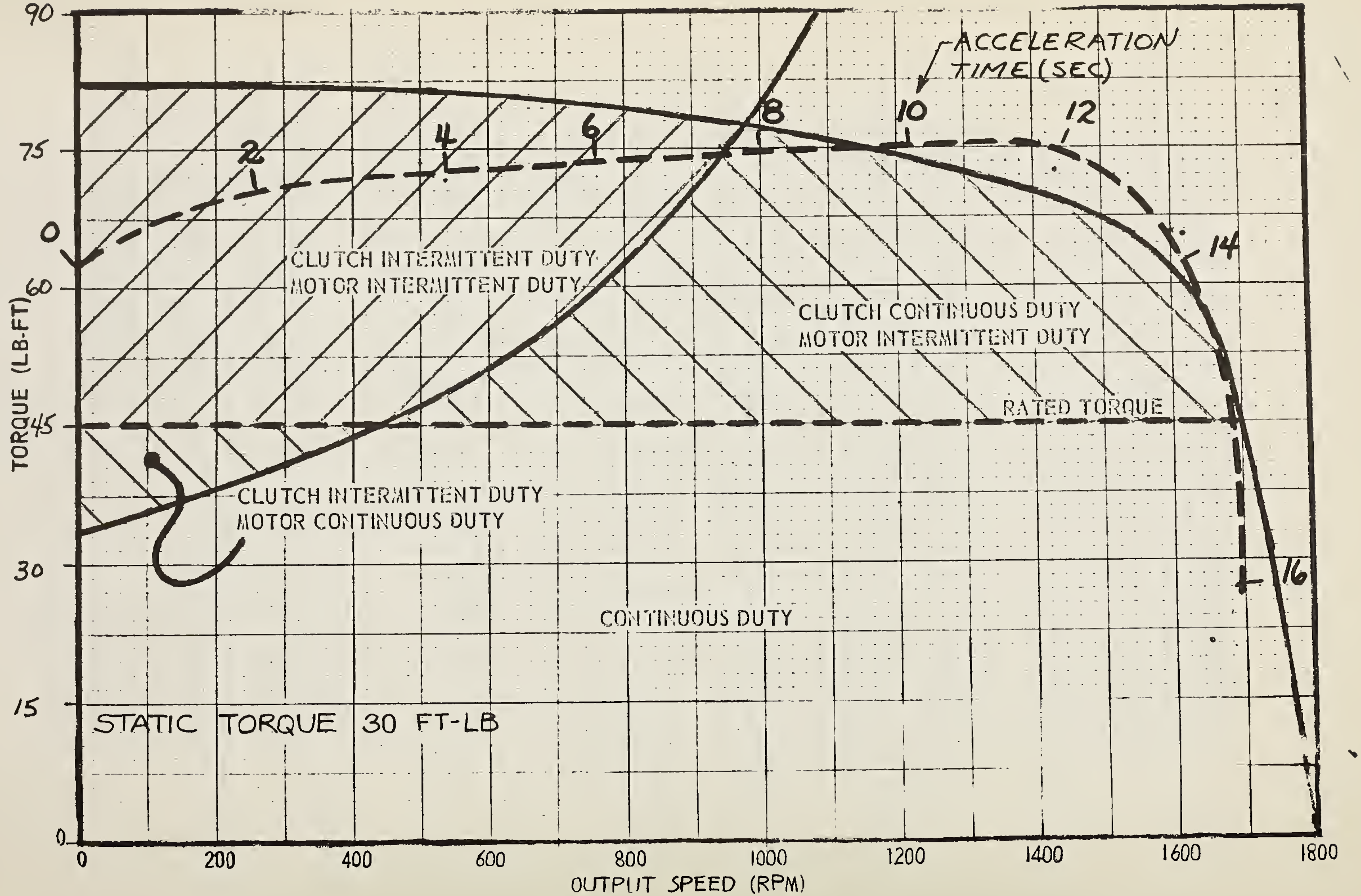


Figure 10-1 TORQUE DEMAND DURING ACCELERATION (0 - 15 m p h)
DALLAS SYSTEM

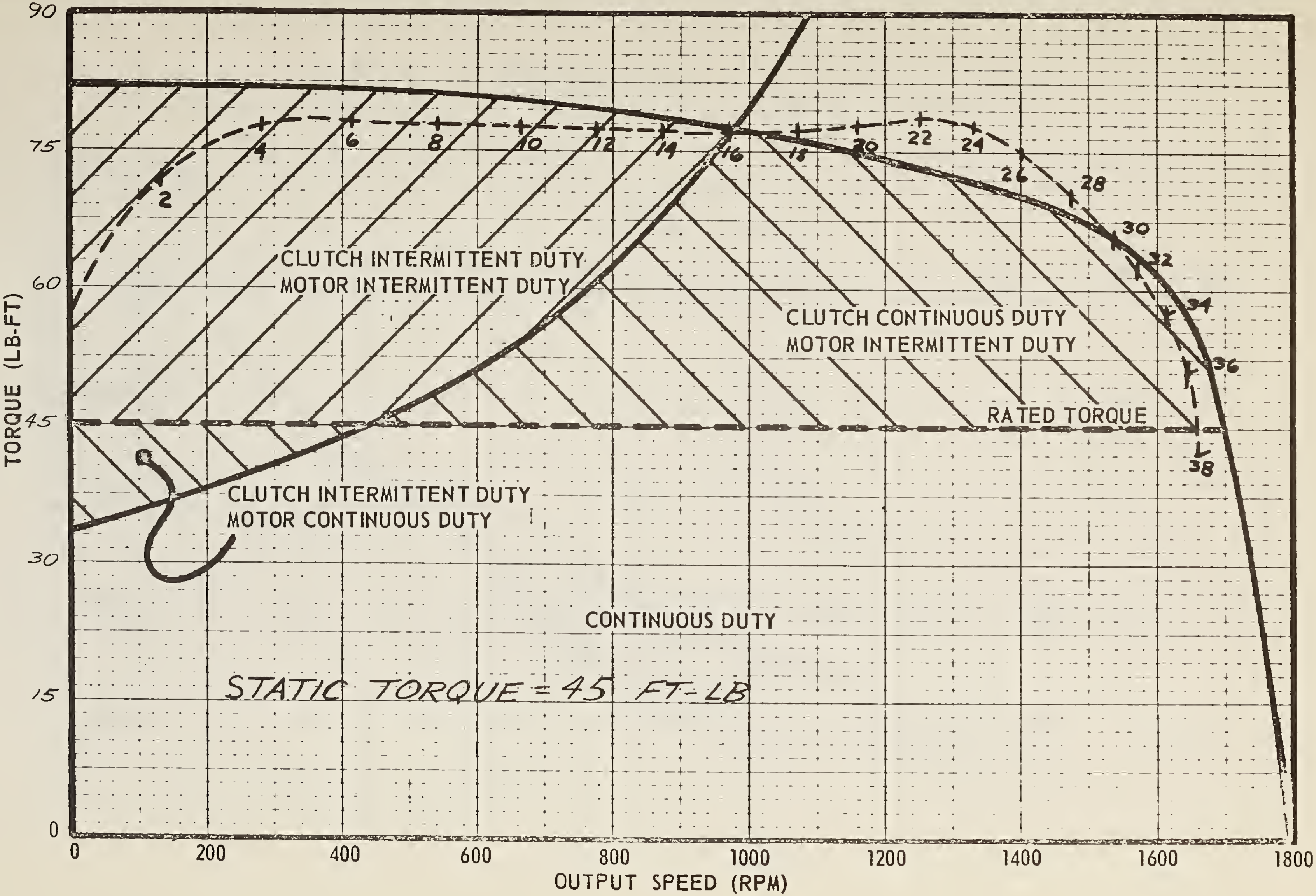


Figure 10-2 TORQUE DEMAND DURING ACCELERATION (0 - 30 m p h) OPTIMIZED SYSTEM

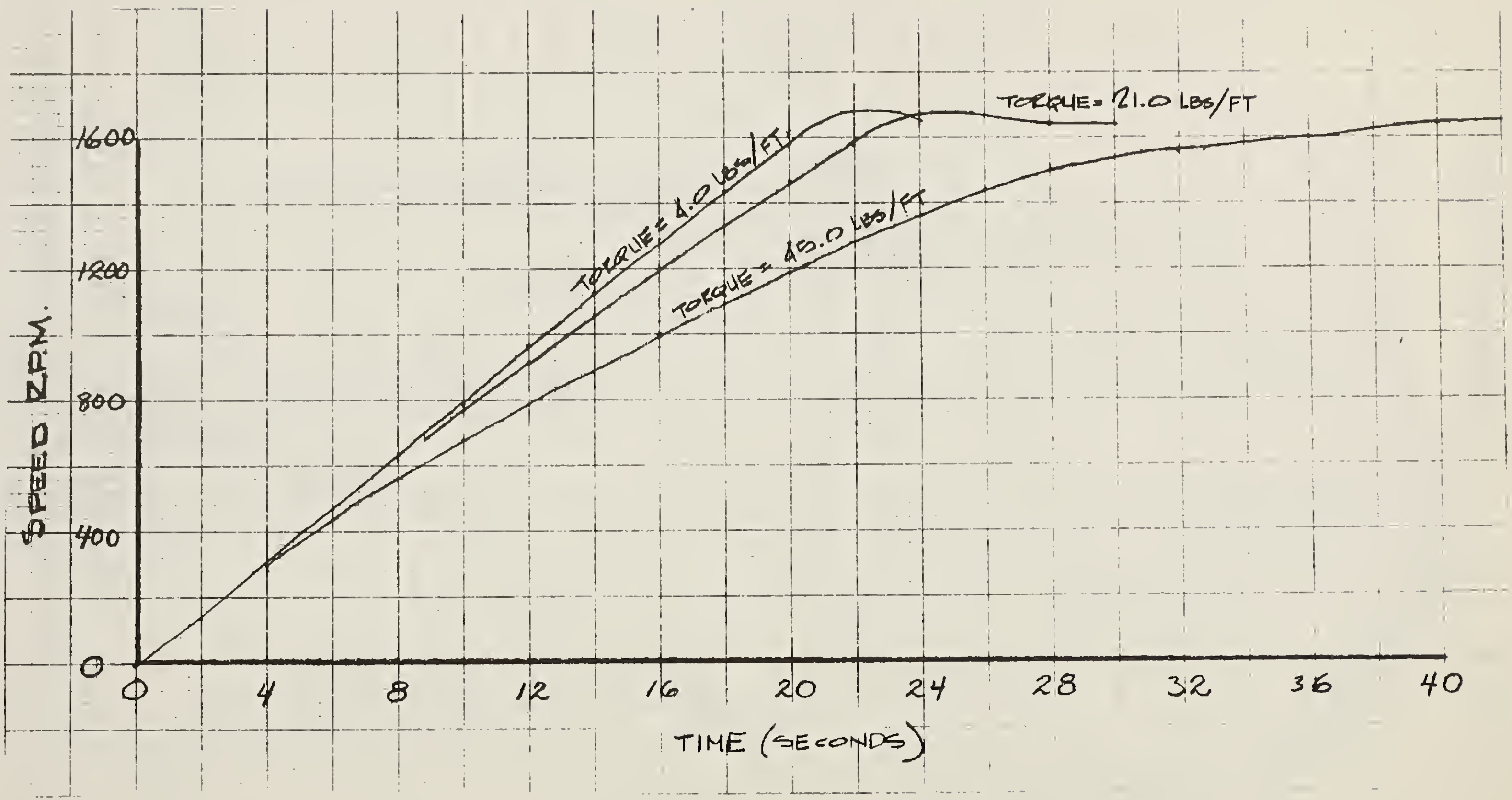


Figure 10-3 ACCELERATION RATES AS A FUNCTION OF INERTIA MASS

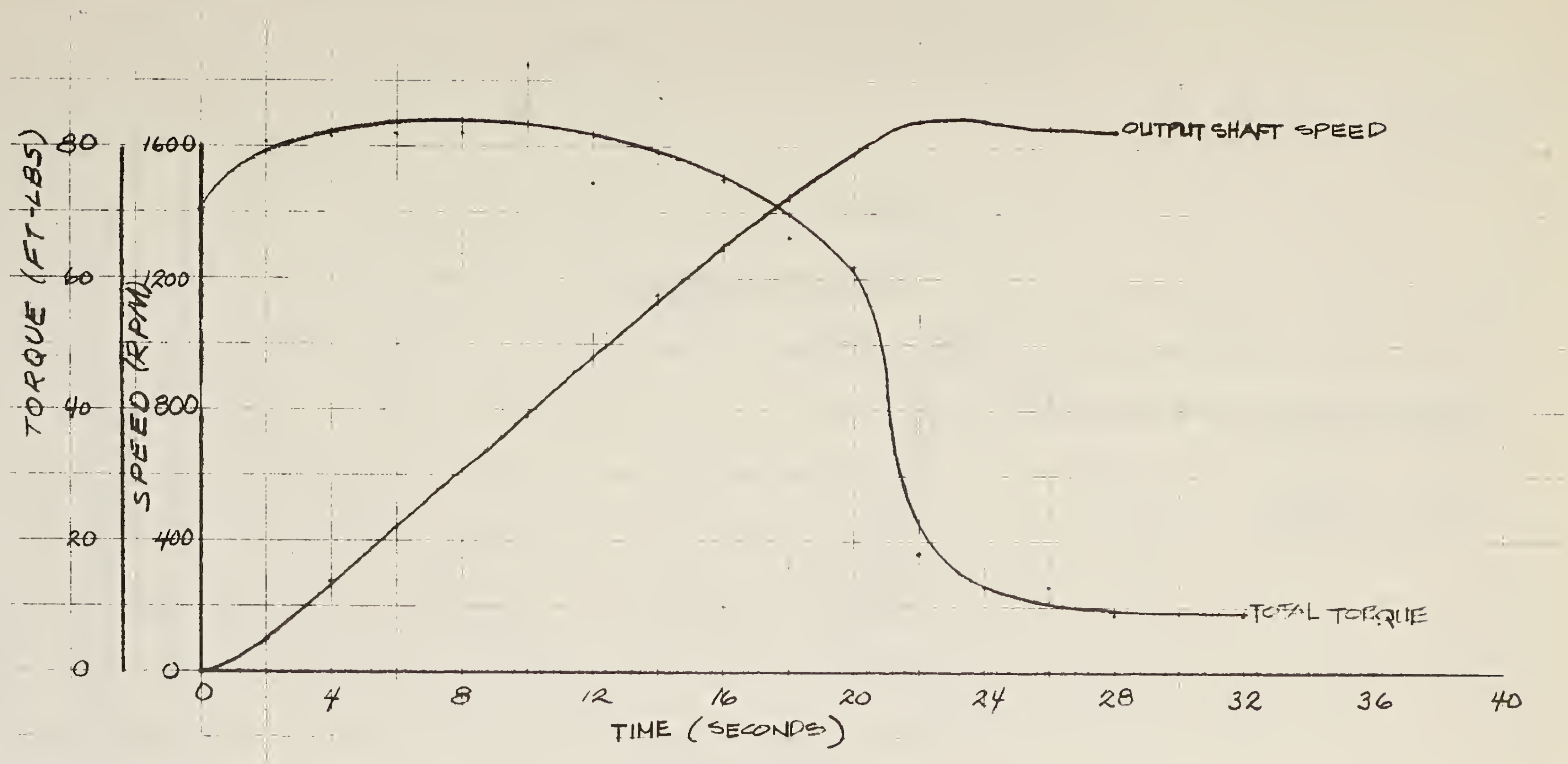


Figure 10-4 ACCELERATION PHASE

4.0 LB./FT. STATIC PRESSURE

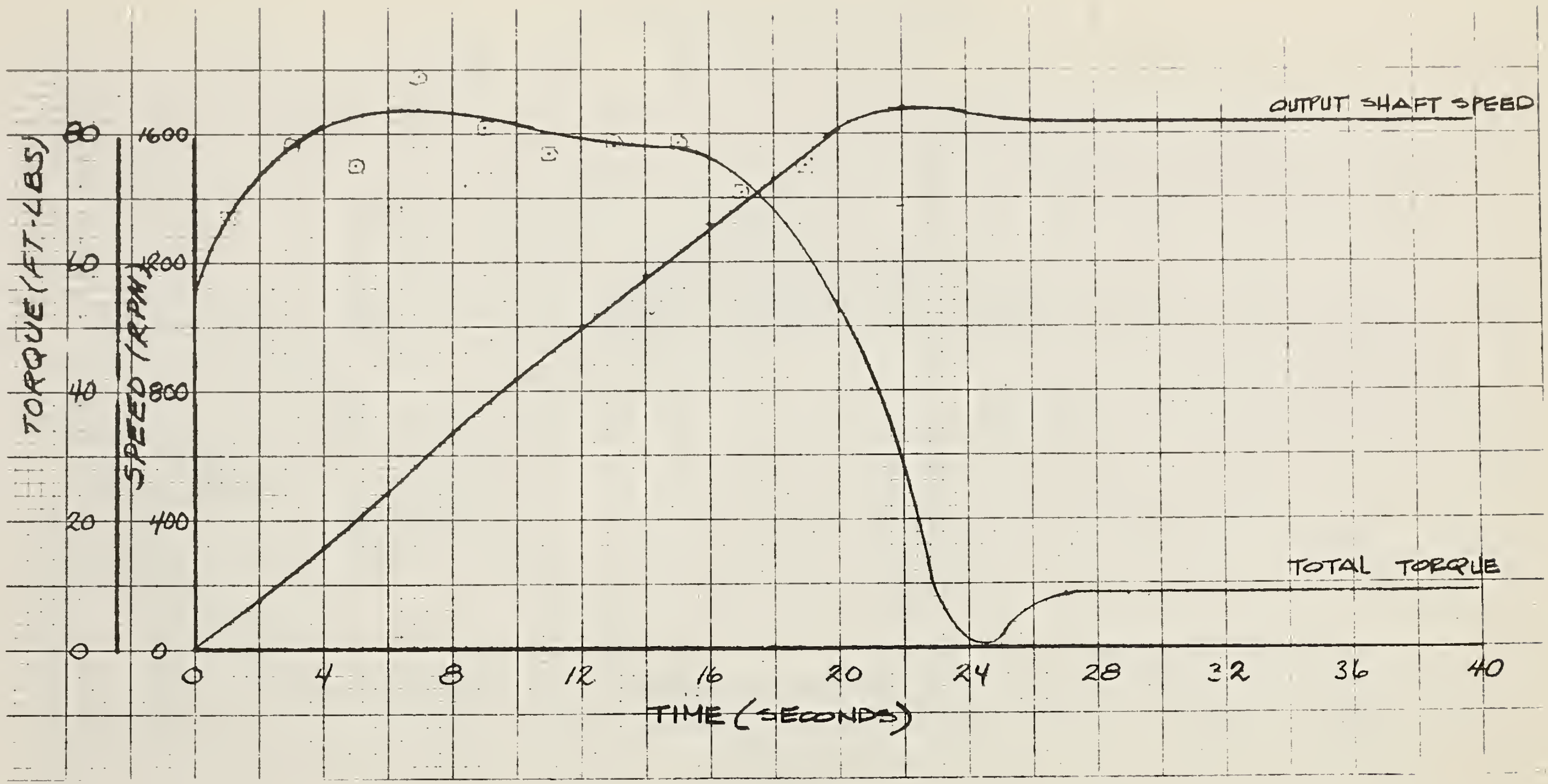


Figure 10-5 ACCELERATION PHASE

9.0 LB./FT. STATIC PRESSURE

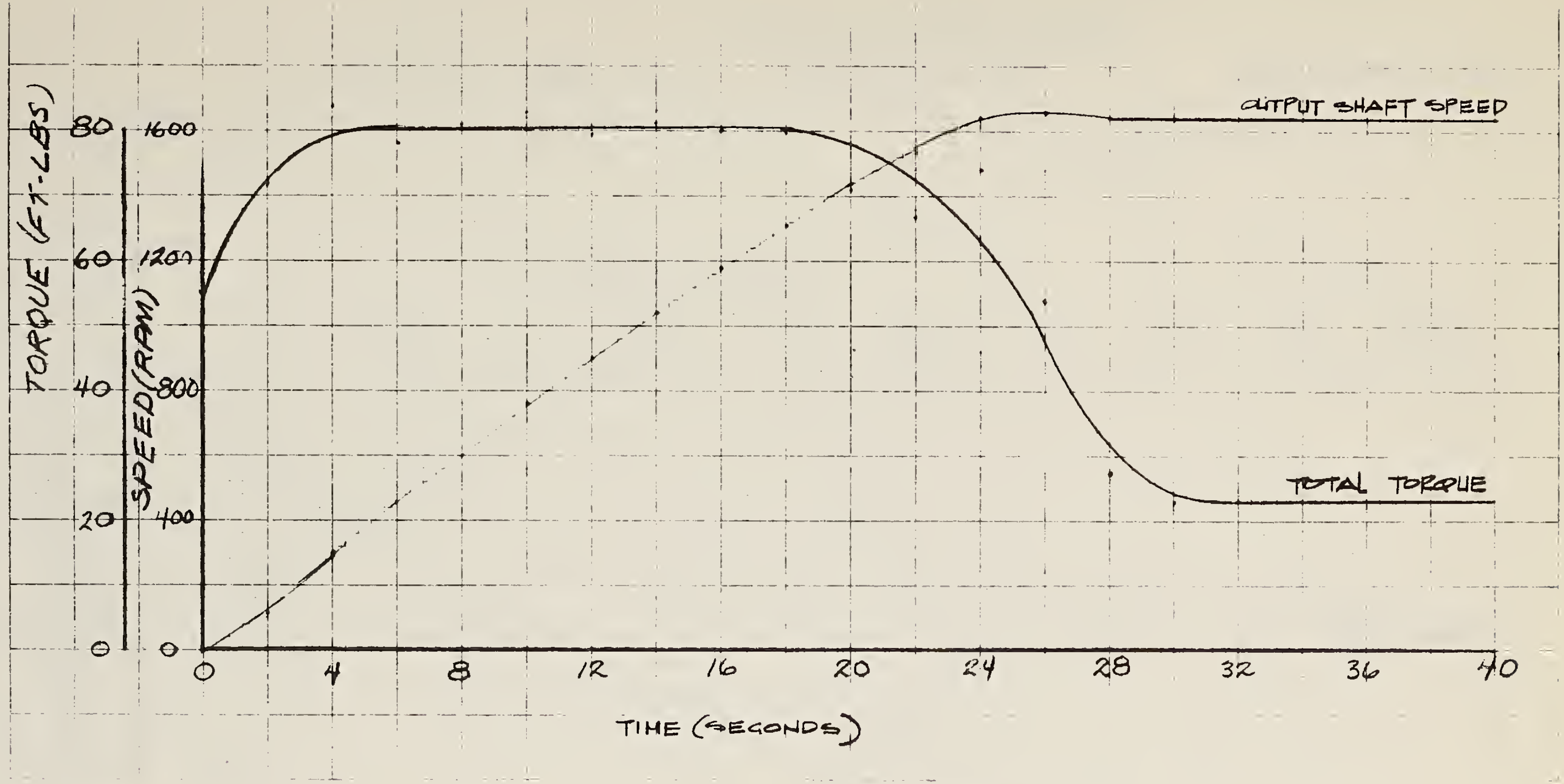


Figure 10-6 ACCELERATION PHASE

21.0 LB./FT. STATIC PRESSURE

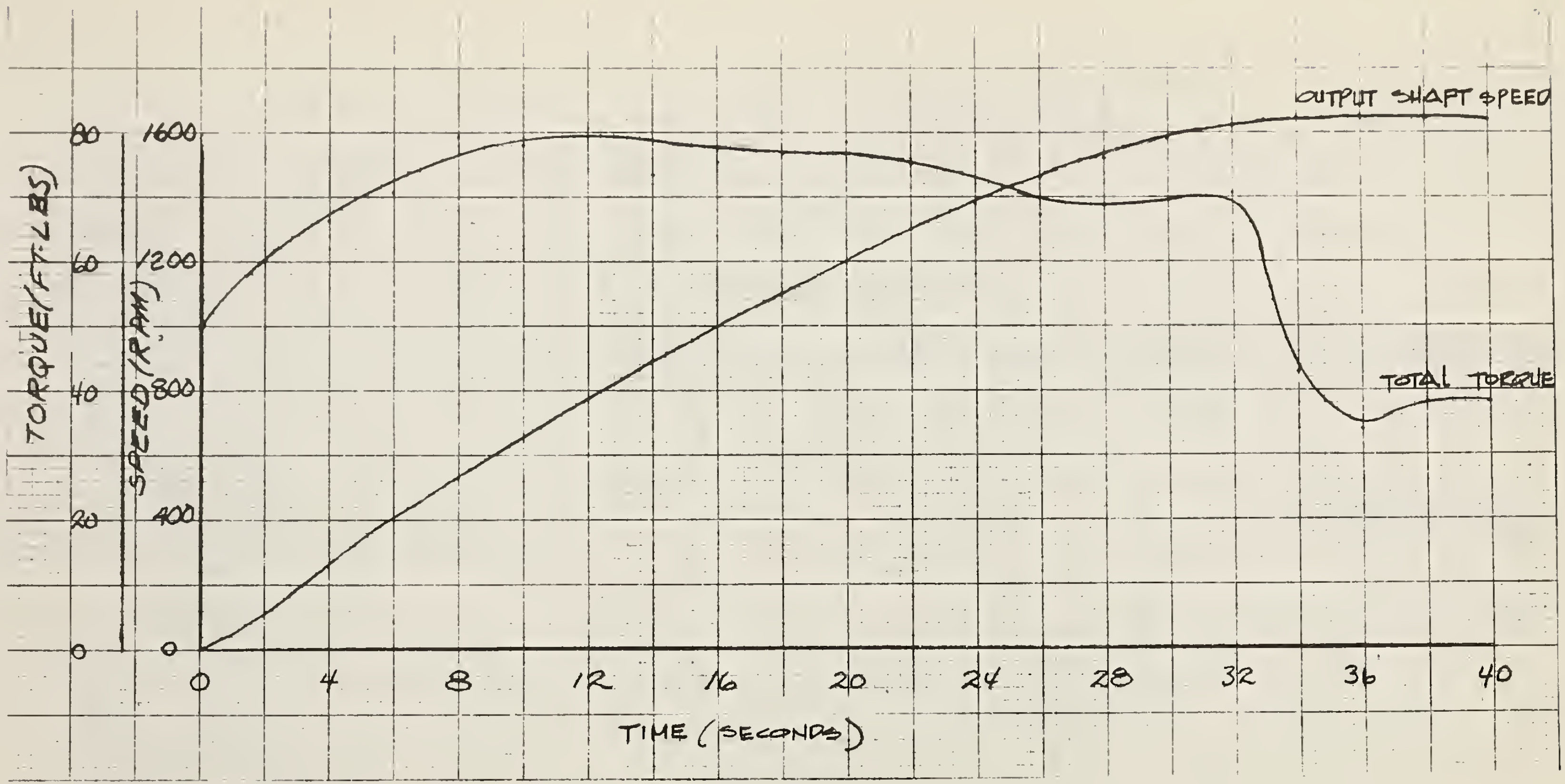


Figure 10-7 ACCELERATION PHASE
35.0 LB./FT. STATIC PRESSURE

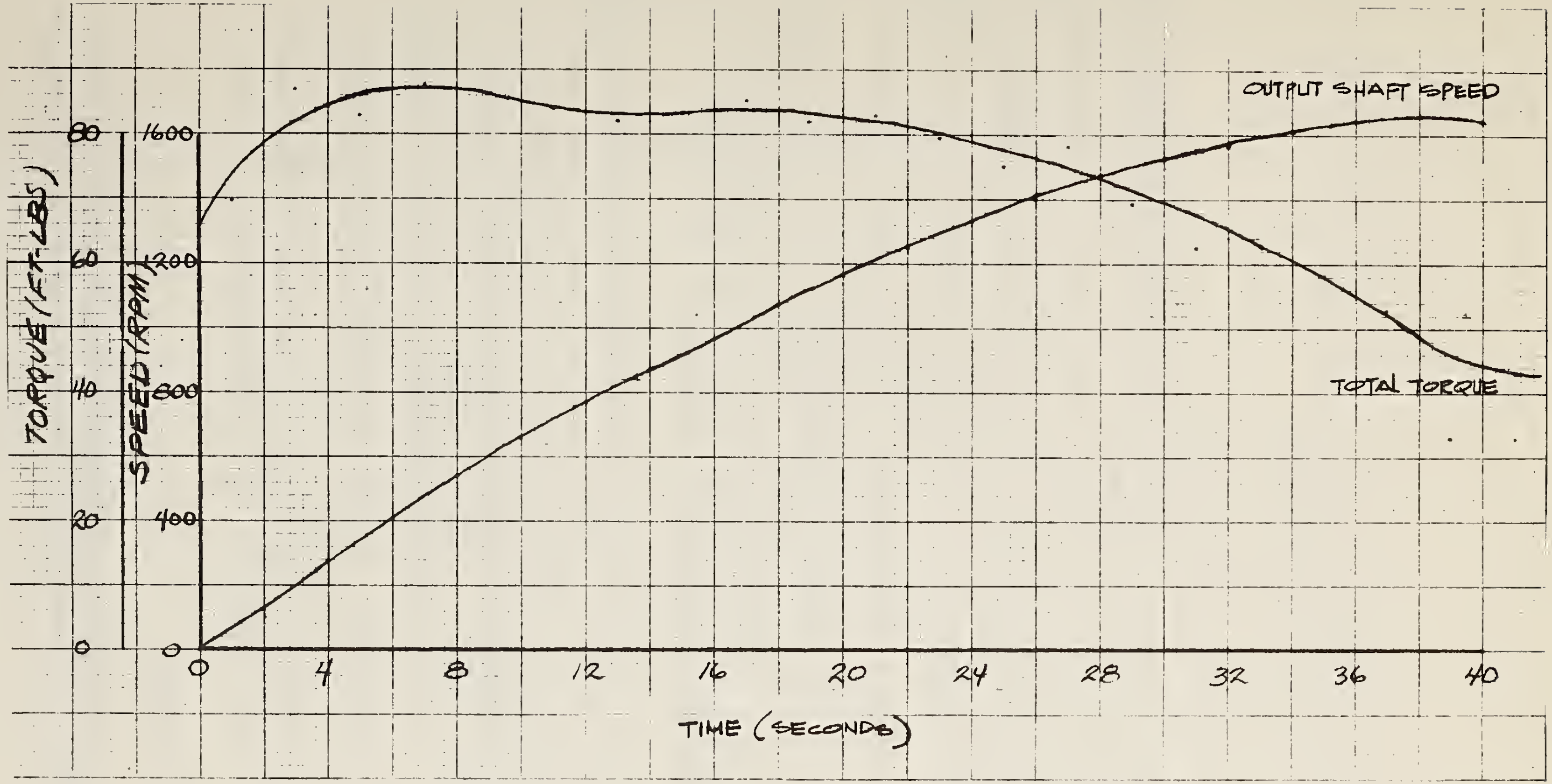


Figure 10-8 ACCELERATION PHASE
45.0 LB./FT. STATIC PRESSURE

Finally in Figure 10-9 the drive system speed during the acceleration phase is compared for various levels of static torque on the optimized system.

10.1.5 Electrical Power Requirements

Power consumed by the drive unit is proportional to the torque demand on the unit. In Figure 10-10, current demand has been plotted for various levels of torque output, up to the maximum torque level of the drive. The current level has been converted to power to show proper power consumption for various levels of torque output, up to the maximum capacity of the drive units.

It should be noted that the AC motor is running continuously even while the vehicle is in the dwell phase at a station. Under conditions where movement of the vehicle is not anticipated the AC motor may be completely shut down, thus, not drawing any power. Motor restart to full speed is accomplished in approximately two seconds.

Since the motor is running during all phases of the duty cycle, including the dwell phases, it might be expected that excessive power is being consumed using the AC eddy-current drive. To identify power consumption during the duty cycle Table 10-1 was prepared which shows the percentage of time the vehicle is in each phase of operation and the power which is consumed during that phase. It is noted that dwell encompasses 7.6% of the time for a complete loop of the Jetrail system. From a total power viewpoint dwell consumes only 3.4% of the total power required for the complete cycle.

10.1.6 Temperature

Visual readings of temperature were made from two thermometers. One thermometer gauge responded to heat dissipated

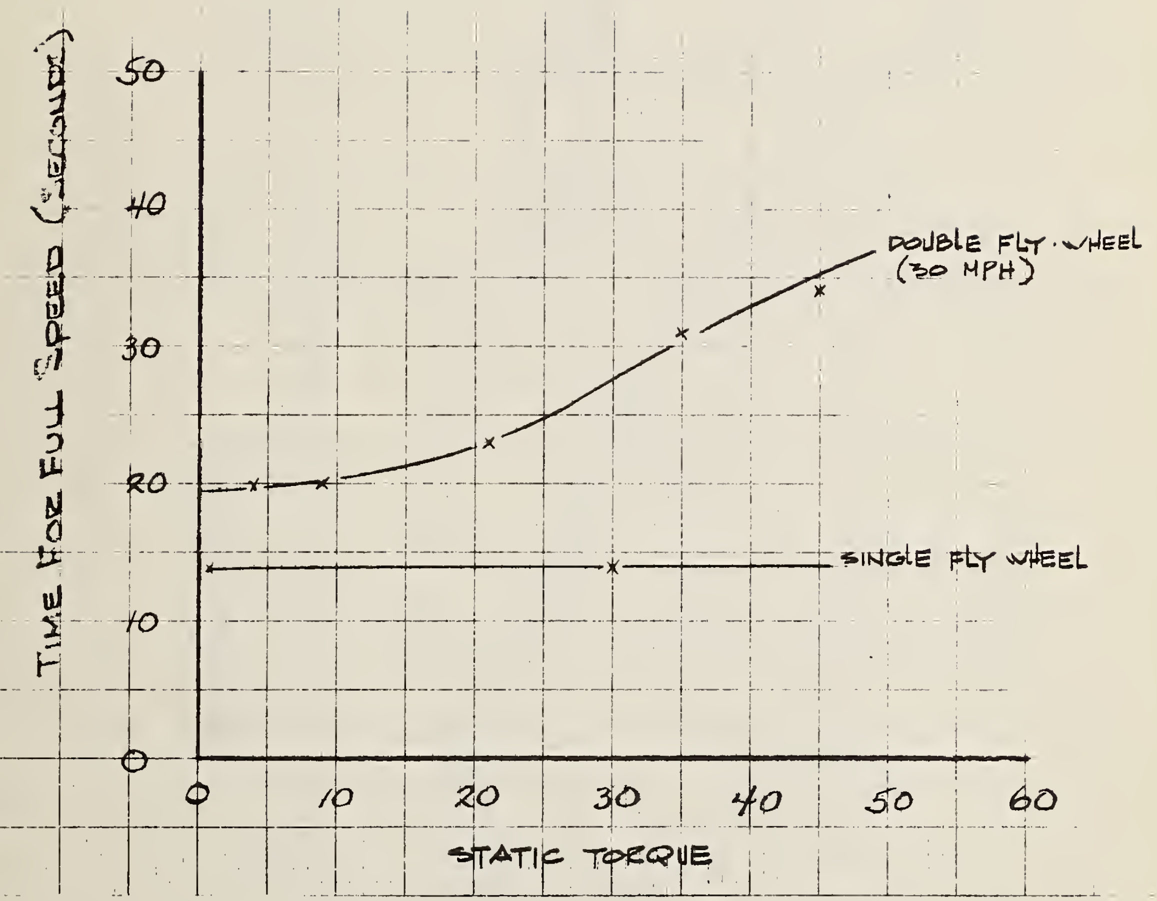


Figure 10-9 DURATION OF ACCELERATION PHASE

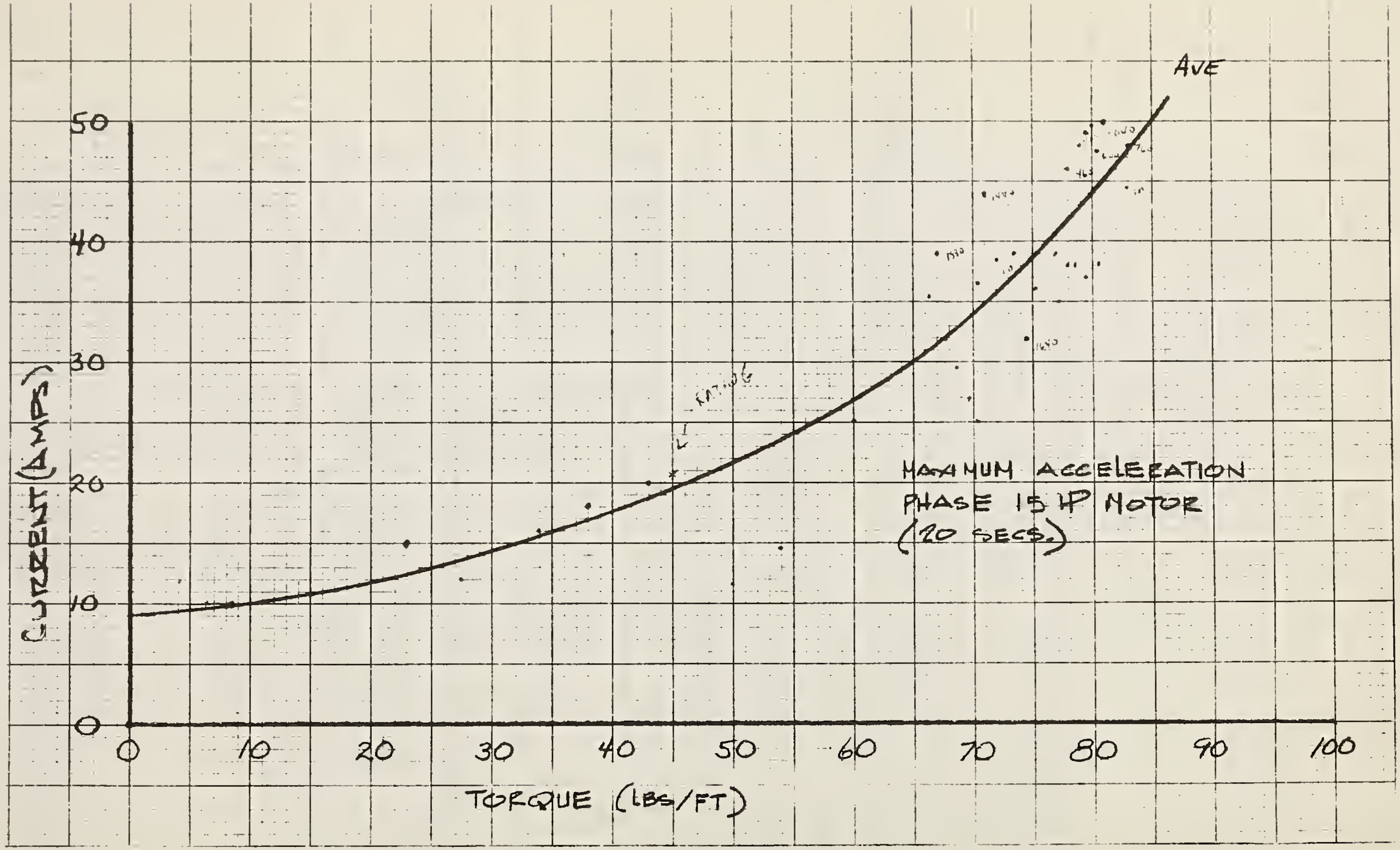


Figure 10-10 DRIVE CURRENT - TORQUE RELATIONSHIP

in the motor field coil and the armature. The other thermometer was inserted into the opening of the eddy-current clutch exhaust fan. Temperature readings were taken periodically during the test and recorded manually on the Brush recorder. Temperature data is shown in Figure F-1 and Figure F-2, Appendix F. No excessive temperatures were observed. The motor maximum temperatures reached 120° for Test #1 and 152° for Test #2.

Table 10.1. Analysis of Duty Cycle - Jetrail System

Phase	No. of Times	Velocity (MPH)	Time (sec)	% of Total Cycle
ACC	3	0-15	44	4.2
ACC	1	0-8	8	.8
ACC	1	2-8	6	.6
CRU	3	15	242	23.1
CRU	5	8	242	23.1
CRU	2	4	93	8.9
CRU	3	2	280	26.8
DEC	3	15-8	17	1.6
DEC	2	8-4	8	.8
DEC	2	4-2	3	.3
DEC	2	2-0	3	.3
DEC	2	8-0	13	1.2
DEC	1	8-2	5	.5
DWL	4	0	80	7.6
		Total	1044	100%

11.0 DRIVE UNIT COMPARISONS

Three types of drives are considered in this study. They are:

- AC Drive with eddy-current clutch and brake
- Static DC Drive
- Static AC Drive

The comparison study examines the general characteristics of the three types of drive systems. General characteristics of the systems include performance and operational features which cannot be precisely defined or quantified.

This comparison does not and cannot recommend any particular drive unit for use in PRT systems but it does show some of the trade offs and factors to be considered when a drive system is to be chosen.

Comparison and evaluation of the drive systems is first done in a general, or qualitative form. In this type of comparison qualities or features which show strong or weak tendencies are pointed out. Evaluation of this type of information is predominantly of a subjective nature, thus the three drive systems studied are merely compared to each other.

11.1 DRIVE SYSTEM STUDY

Three types of drive systems were studied:

AC drive with eddy-current clutch/brake

Static DC drive

Static AC drive (variable frequency)

System data and performance characteristics were obtained from manufacturers of drives. Primary data sources are as follows:

AC Drive with Eddy-Current Clutch/Brake

1. Mobility Systems and Equipment Company, Los Angeles, California

2. The Louis Allis Company, Division of Litton Industries, Sante Fe Springs, California

3. Eaton Corporation, Industrial Drives Division, City of Commerce, California

Static DC Drive

1. General Electric Company, Transportation Industries Sales Operation, San Francisco, California

2. The Louis Allis Company, Division of Litton Industries, Santa Fe Springs, California

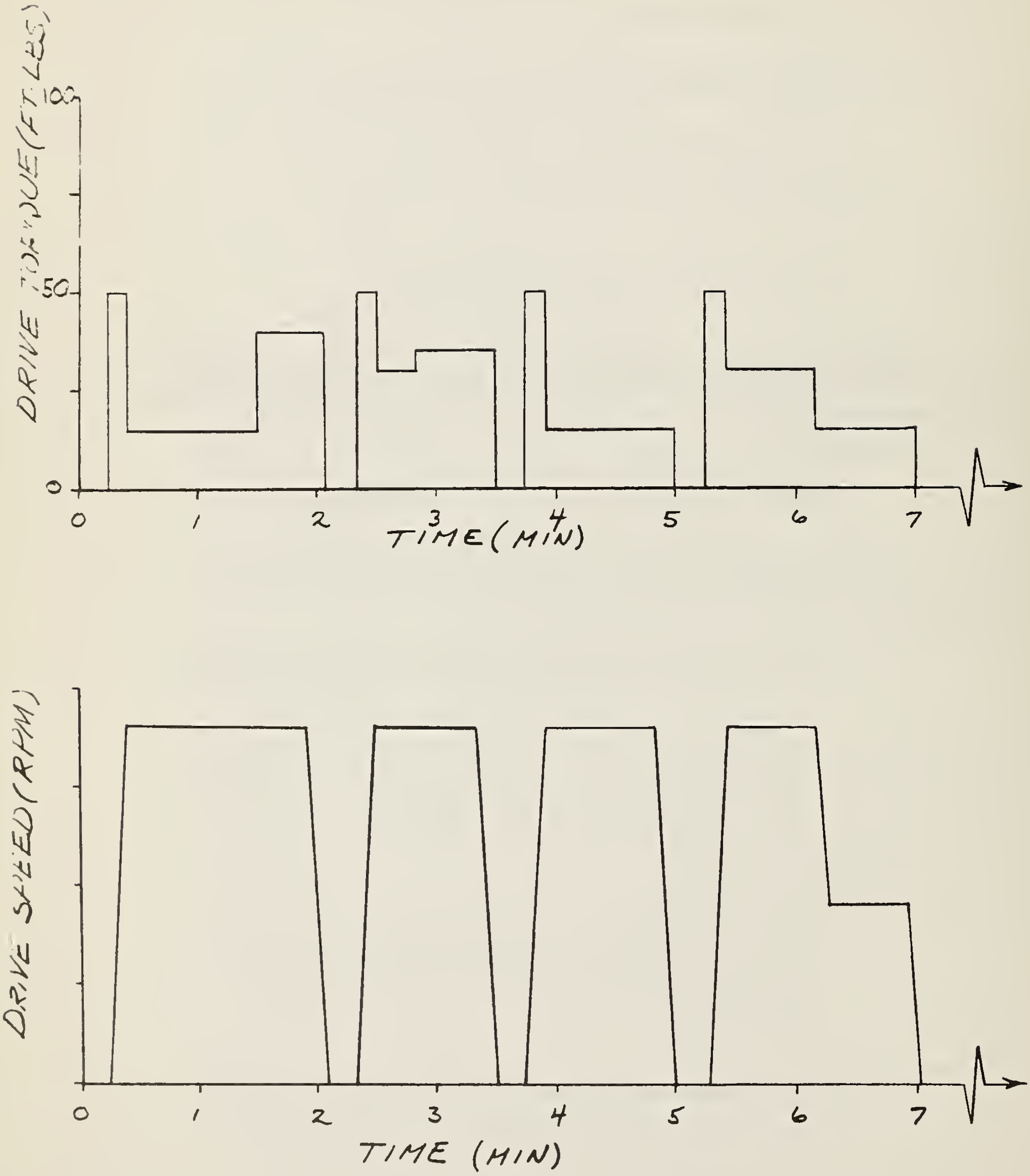


Figure 11-1 TYPICAL DUTY CYCLE

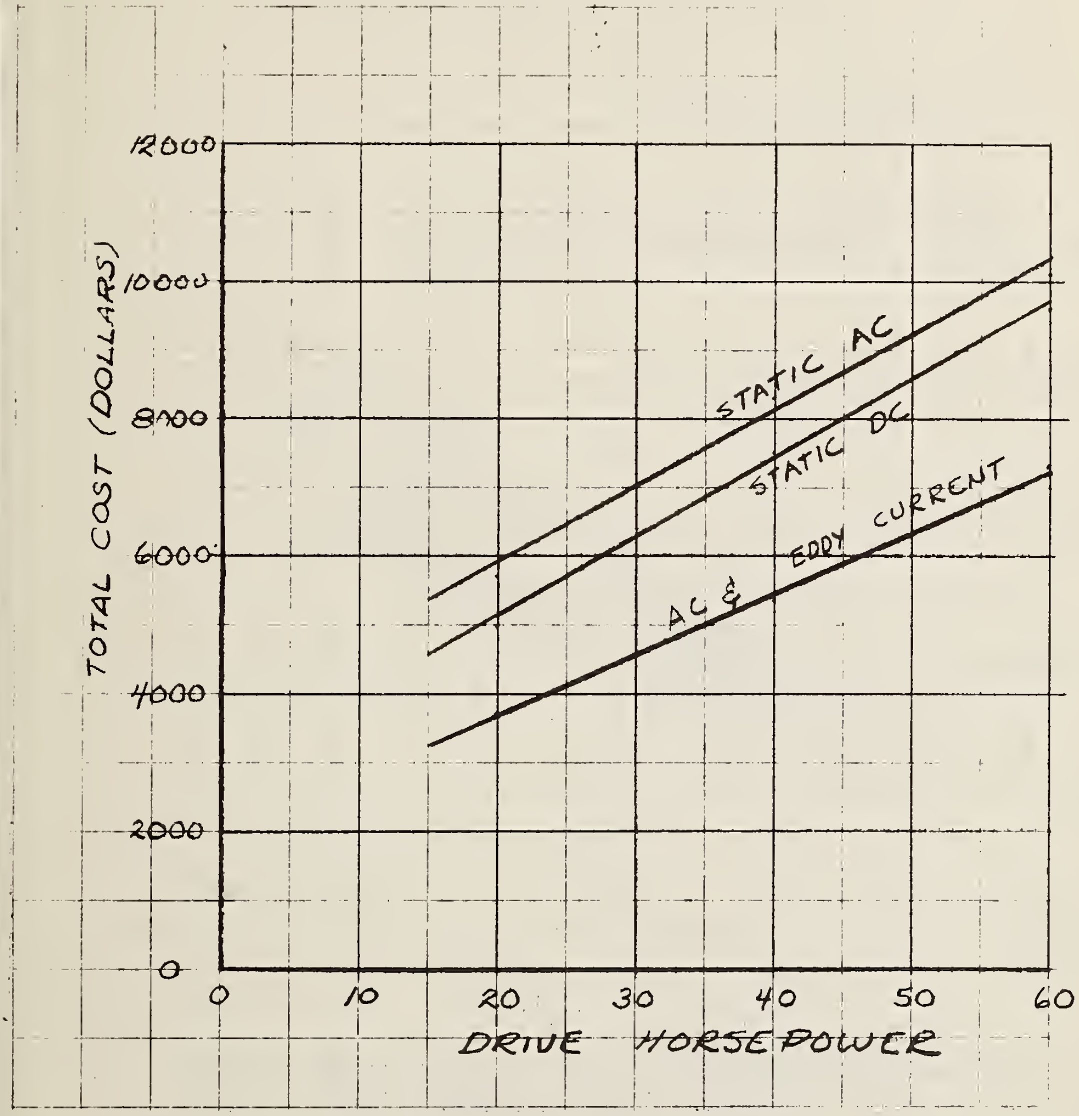


Figure 11-2 DRIVE SYSTEM COMPARISON
COST - SINGLE DRIVE

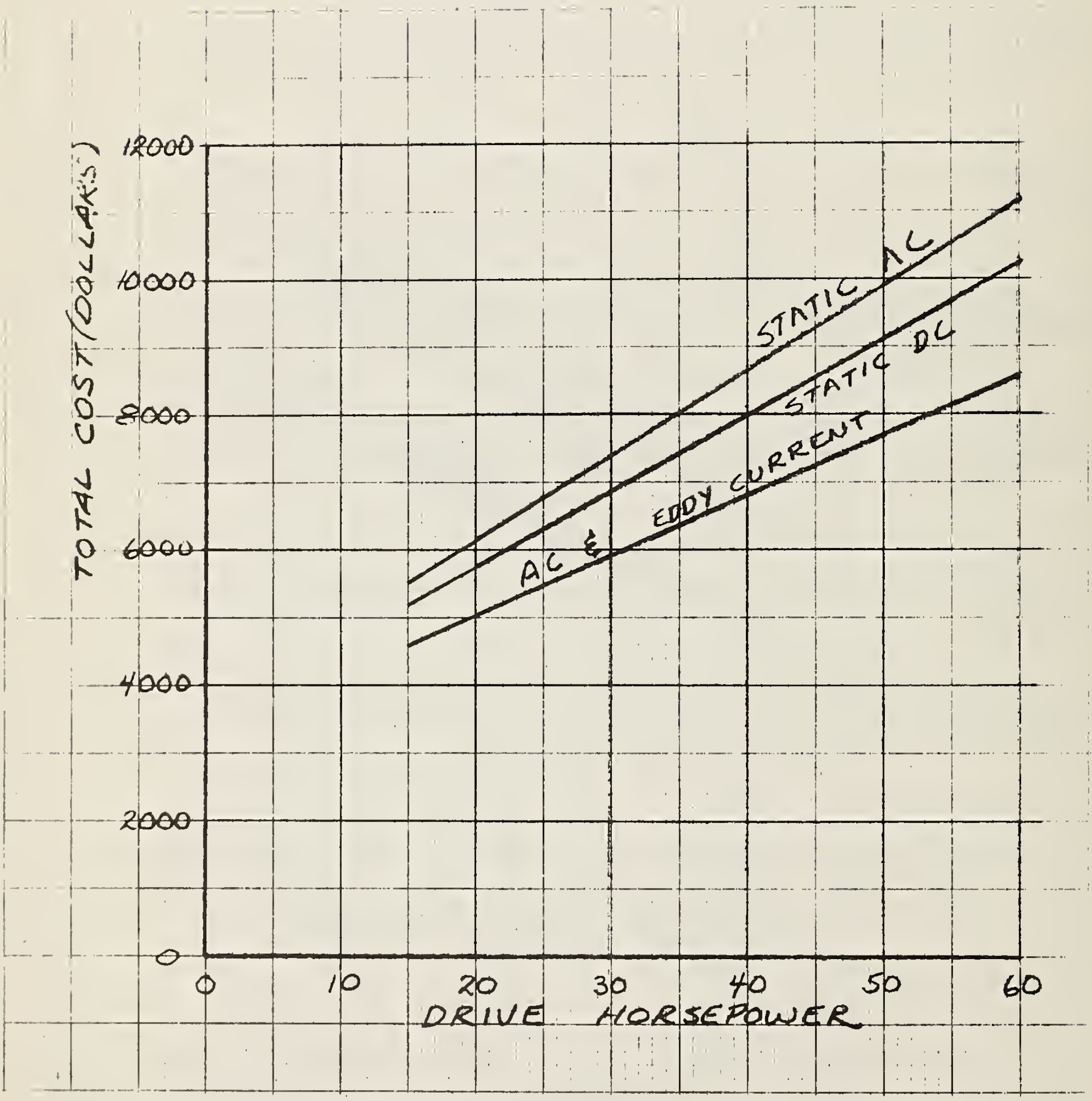


Figure 11-3 DRIVE SYSTEM COMPARISON
COST - DUAL DRIVE

Static AC Drive (variable frequency)

1. The Louis Allis Company, Division of Litton Industries, Santa Fe Springs, California

2. General Electric Company, Transportation Industries Sales Operation, San Francisco, California

11.2 ANALYSIS AND EVALUATION

Drive units with application for PRT systems have been studied, analyzed and evaluated. Although all drives have capabilities for PRT type of vehicles, the drives have been evaluated to show advantages and disadvantages for transportation system application. Results of the study are presented in a series of tables and figures with explanatory annotations.

11.2.1 Selection of a Drive System

A number of important characteristics should be considered in the selection of drive systems for PRT applications. Some of the basic features to be considered are as follows:

Starting - Automatic or manual starting as preferred starting mechanism. Frequency of starting and starting loads. Internal resets. Adjustable break-away.

Stopping - Automatic or manual stopping. Consider emergency stopping as well as normal braking. Stopping accuracy should be defined and holding requirements specified as well as adjustable braking requirements.

Speed Control and Running - Duty cycle and specifications of operation described, including fixed speeds of operation, programmable acceleration/deceleration - logarithmic or linear, etc., speed regulations for load changes or fluctuating line voltage.

Safety Features - Overload protection limits due to high loads or disturbance of line power characteristics.

Multiple Drive Cascading - Requirements for using two or more drive motors with a common control unit.

Careful consideration should be given to PRT system requirements as they will affect the above drive system requirements as well as other system characteristics.

11.2.2 Comparison of Drive System Features

The first step in this study was to make a general comparison of the three categories of drives which were studied. Table 11.4 lists some of the general characteristics of drive systems and gives a subjective evaluation of the AC drive with eddy-current clutch/brake, static DC drive and static AC drive for each of the features or general characteristics listed. Information in Table 11.4 can only show areas where a particular drive has good or poor characteristics.

	AC Drive with Eddy-Current Clutch/Brake	Static DC Drive	Static AC Drive (Variable Frequency)
Size of Drive Mechanism (volume and weight)	Large -- due to mechanical nature of speed control clutch.	Medium -- requires brush rigging and wire-wound rotor assembly.	Small
Size of Controls (electronics) (volume and weight)	Small -- requires small control current to regulate clutch mechanism.	Medium -- handles full propulsion energy of the drive; requires AC to DC conversion controls.	Large -- logic requires electronics for conversion to DC then back to variable frequency AC.
Environment Capability (moisture; dust; temperature range from 40°F to 120° F; corrosion; salt air)	Good -- cannot be totally enclosed, but clutch coil can be encapsulated.	Fair to Good -- has brushes susceptible to moisture; enclosed DC available with veins for cooling, but the frame size must be increased because of heating.	Excellent -- uses standard enclosed AC motor.
Physical Characteristics (frame and shaft)	Good to Excellent -- C & D flange.	Good to Excellent -- C & D flange.	Excellent -- small packaging; available with hollow shaft, C & D flange.
Acoustical Noise	Level -- due to constant speed.	Variable -- due to changing speed. (Noise increase with forced air cooling, even when drive is in dwell or stopped.	Variable -- due to changing speed.
Power Consumption Efficiency	Good -- when operating near maximum speeds; Low -- when operating at low speeds.	Excellent.	Excellent.
First Cost	Low -- simple torque transmission by mechanical unit; simple electronic controls.	Medium -- mechanical unit about the same as Eddy-current but controls required to handle full work load (power) of drive.	High -- drive motor is standard AC but controls are more complex than DC due to AC-DC-AC conversion.

Table 11.4 Comparison of Drive Systems

	AC Drive with Eddy-Current Clutch/Brake	Static DC Drive	Static AC Drive (Variable Frequency)
Maintenance	Low -- very simple electronics, but drive motor / clutch has additional rotating device with corresponding increase in maintenance.	High -- Brushes and commutator must be replaced regularly; wire-wound rotor is affected by environmental conditions; electronics are low-maintenance once "burn in" is accomplished.	Unknown -- standard AC motor is only mechanical equipment; electronics are solid state with high reliability once "burn in" is accomplished.
Torque Overload	200% for 15 seconds; 150% for 5 minutes (from actual testing).	200% for 5 seconds; 150% for 1 minute.	200% for 5 seconds; 150% for 1 minute.
Factory Delivery	12 – 15 weeks (approximate); a wide variety of pre-engineered options are available; no special design engineering required.	20 - 25 weeks (approximately, as manufacturer recommends; specially engineered packages required for customer applications.	20 - 25 weeks (approximately, as manufacturer recommends; specially engineered packages required for customer applications.
State of System Development	Excellent -- uses mainly mechanical components; drives in all horsepower ranges have been in continuous use for years; simple electronics are used with solid state components.	Good -- DC systems have a long operational history; DC motors and solid state rectification and controls are commonly used in industrial drives.	Unknown -- Static AC or variable frequency systems have existed many years; electronic control systems use solid state devices and are very complex; each manufacturer has special control system features; system capability good, but static AC drives lack experience in long time usage.
Life (Time between major overhauls)	10 - 15 years.	6 - 10 years.	Unknown -- estimated to be 10 - 15 years.

Table 11.4 Comparison of Drive Systems (continued)

12.0 CONCLUSIONS AND RECOMMENDATIONS

12.1 CONCLUSIONS

An extensive work package has been completed covering the analysis of PRT duty cycles, the evaluation and selection of a drive system for a test program, performance testing of a 15 HP drive system with eddy current clutch and brake under typical PRT duty cycles, design and fabrication of a bogie assembly and the evaluation and comparison of other types of propulsion systems.

An eddy current drive unit was selected and designed for a duty cycle test program. The drive was tested in the laboratory under a number of complete cycles of operation of the Jetrail, PRT system (currently in operation by Braniff Airlines at Love Field at Dallas, Texas). As expected there were no performance difficulties encountered. Motor temperatures and performance reached nominal steady state values early during the one hour test run.

A second test was also completed with the drive unit. The second run represented an optimized system with a duty cycle designed to show the maximum performance output of the drive unit. During this test the drive was forced to run for short durations above its rated capacity. Steady state temperatures and performance were reached during the one hour test run showing that the drive could tolerate the short durations of high demand without deleterious effects.

The test also showed that the drive unit could be accurately controlled, and is able to execute all the various power and speed demands as well as the acceleration-deceleration requirements necessary for a PRT system.

A bogie system was designed and fabricated which is compatible with the AC drive unit with eddy-current clutch-brake. The bogie is adaptable to PRT vehicles of up to 16,000 pounds gross vehicle weight. The detail design of the bogie has been verified analytically for performance characteristics. The bogie is available for installation and test on a PRT.

Additionally, other types of AC and DC drive systems have been studied, compared and evaluated. The general characteristics of various drive systems, the specific characteristics of drives for particular vehicles were examined subject to the constraints of the simulated duty cycles. Evaluation has shown that no single drive system can be exclusively recommended for all transportation applications. However, as a result of the study as well as the tests and evaluations performed on an AC drive unit it can be shown that the AC drive with eddy current clutch-brake has very favorable characteristics when used in PRT systems having duty cycles similar to those simulated.

Finally, procedures and test data are provided which will allow engineers and designers to apply eddy current drive systems to other design configurations and duty cycles.

12.2 RECOMMENDATIONS

Since acceleration-deceleration phases in the test were simulated by the use of an inertia flywheel it is recommended that an experimental vehicle system be fabricated and tested on an approximate track using the existing eddy current drive unit.

In addition to substantiating the validity of the inertia fly wheel method, the validity of the laboratory test program can be confirmed. Also other tests can be performed including the interaction of the drive unit with the guideway bogie, and car in addition to specific tests on the bogie under operational conditions. Life tests may also be performed.

Since the major elements for a vehicle test are available and since a test track such as the Pomona Fair Grounds System can be utilized, it is expected that such an extension of this program can be successfully carried out. This is therefore strongly recommended.

APPENDIX A

MOTOR TORQUE CHARACTERISTICS

APPENDIX A

Figure A-1 characterizes the performance of a PM-001 MSE drive unit. It defines the torque speed relationships which should be observed for reliable operation of the drive unit. Line B, C, E. specifies the rated torque and the drive may operate continuously (motor and clutch) within the area bounded by points A, C, E. The curve ACD represents a dissipation of 15 HP in the clutch and BE represent 15 HP in the motor. The motor may operate continuously in the ABC region, but the clutch cannot. The clutch dissipation is proportional to the difference between the motor speed and the output shaft speed, and the curve indicates that at output shaft speeds below 400 RPM the torque demands should not exceed those defined by AC on a continuous basis.

Similarly, the clutch may be operated in the area CDE since this represents less than 15 HP dissipation; however, it represents more than 15 HP in the motor and so the motor may not be operated continuously.

The curve ACD is in effect a clutch dissipation curve derived as follows

$$\text{Clutch Dissipation} = \frac{\text{Torque Output X(Motor Speed - Output Speed) HP}}{5252}$$

and the curve ACD represents a clutch dissipation of 15 HP.

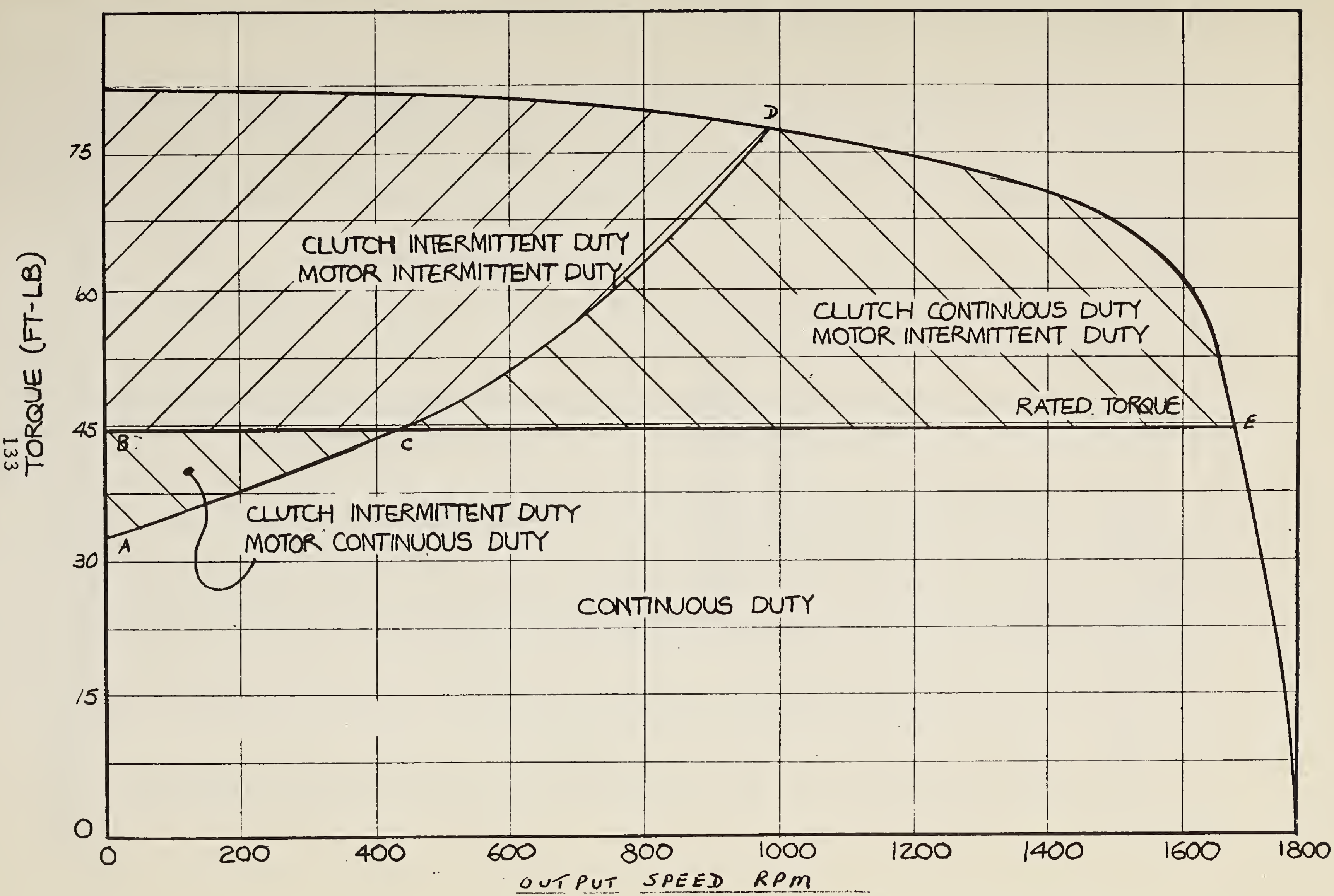


Figure A-1 PM-001 MSE DRIVE UNIT TORQUE-SPEED CURVE

APPENDIX B

DUTY CYCLE TABLES

Table 4.1 - Loading and Unloading Schedule

Station	Trip	Passenger		Load for Trip	Seconds		Dwell Total
		Exit	Enter		Loading	Doors	
1	1-6	0	2	2	3	5	8
6	6-7	2	4	6	7	5	12
7	7-2	3	1	4	5	5	10
2	2-3	2	0	2	2	5	7
3	3-5	3	3	2	7	5	12
5	5-8	0	4	6	5	5	10
8	8-2	3	3	6	7	5	12
2	2-6	2	2	6	5	5	10
6	6-7	4	4	6	9	5	14
7	7-3	3	3	6	7	5	12
3	3-4	1	1	6	3	5	8
4	4-7	4	2	4	7	5	12
7	7-1	0	2	6	3	5	8
1	1-4	6	0	0	6	5	11
4	4-2	0	4	4	5	5	10
2	2-3	2	0	2	2	5	7
3	3-5	1	5	6	7	5	12
5	5-7	3	3	6	7	5	12
7	7-8	3	1	4	5	5	10
8	8-1	2	4	6	7	5	12
1	1-3	3	1	4	5	5	10
3	3-4	3	1	2	5	5	10
4	4-7	1	5	6	7	5	12
7	7-6	3	3	6	7	5	12
6	6-8	4	4	6	9	5	14
8	8-7	4	0	2	4	5	7
7	7-2	2	0	0	2	5	7
2	2-5	0	4	4	5	5	10
5	5-3	4	6	6	11	5	16
3	3-4	3	3	6	7	5	12
4	4-8	2	2	6	5	5	10
8	8-1	3	3	6	7	5	12
1	1-7	4	0	2	4	5	9
7	7-3	1	1	2	3	5	8
3	3-5	1	3	4	5	5	10
5	5-6	0	2	6	3	5	8
6	6-8	2	2	6	5	5	10
8	8-2	3	1	4	7	5	12
2	2-4	2	4	6	7	5	12
4	4-3	6	2	2	9	5	14
3	3-7	2	4	4	7	5	12
7	7-6	3	1	2	5	5	10
6	6-1	1	1	2	3	5	8
1	End	2	0	0	2	5	7
							461 seconds

Table 4.2 - Loading and Unloading Schedule

Station	Trip	Passenger		Load for Trip	Seconds		Dwell Total
		Exit	Enter		Doors	Loading	
1	1-6	0	4	4	5	5	10
6	6-7	2	4	6	5	7	12
7	7-2	3	1	4	5	5	10
2	2-3	2	0	2	5	2	7
3	3-5	3	3	2	5	7	12
5	5-8	0	4	6	5	5	10
8	8-2	3	5	8	5	9	14
2	2-6	3	1	6	5	5	10
6	6-7	2	4	8	5	7	12
7	7-3	2	0	6	5	2	7
3	3-4	4	6	8	5	11	16
4	4-7	6	2	4	5	9	14
7	7-1	0	2	6	5	3	8
1	1-4	6	0	0	5	6	11
4	4-2	0	4	4	5	5	10
2	2-3	2	0	2	5	2	7
3	3-5	1	5	6	5	7	12
5	5-7	1	3	8	5	5	10
7	7-8	5	1	4	5	7	12
8	8-1	2	6	8	5	9	14
1	1-3	3	1	6	5	5	10
3	3-4	5	3	4	5	9	14
4	4-7	1	5	8	5	7	12
7	7-6	4	2	6	5	7	12
6	6-8	5	2	4	5	8	13
8	8-7	2	0	2	5	2	7
7	7-2	2	0	0	5	2	7
2	2-5	0	4	4	5	5	10
5	5-3	4	6	6	5	11	16
3	3-4	3	3	6	5	7	12
4	4-8	2	4	8	5	7	12
8	8-1	5	3	6	5	8	13
1	1-7	4	0	2	5	4	9
7	7-3	1	1	2	5	3	8
3	3-5	1	3	4	5	5	10
5	5-6	0	2	6	5	3	8
6	6-8	2	2	6	5	5	10
8	8-2	3	1	4	5	7	12
2	2-4	2	4	6	5	7	12
4	4-3	6	2	2	5	9	14
3	3-7	2	4	4	5	7	12
7	7-6	3	1	2	5	5	10
6	6-1	1	1	2	5	3	8
1	End	2	0	0	5	2	7
							476 seconds

Table 4.3 - Jetrail Trip Phases

Trip	Phase	Vel.	Dist	Time	Trip Time
PL-T	ACC	0-15	161	14.6	
	CRV	15	1893	86	
	DEC	15-8	96	5.7	
	CRV	8	591	50.3	
	DEC	8-4	34	3.9	
	CRV	4	275	46.1	
	DEC	4-2	7	1.6	
	CRV	2	35	12	
	DEC	2-0	2	1.6	221.8
T-LD	ACC	0-8	46	7.8	
	CRV	8	746	63.5	
	DEC	8-0	38	6.5	77.8
LD-PL	ACC	0-15	161	14.6	
	CRV	15	1539	70	
	DEC	15-8	96	5.7	
	CRV	8	320	27	
	DEC	8-4	34	3.9	
	CRV	4	275	47	
	DEC	4-2	7	1.6	
	CRV	2	316	107.4	
	DEC	2-0	2	1.6	278.8
PL-LD	ACC	0-15	161	14.6	
	CRV	15	1893	86	
	DEC	15-8	96	5.7	
	CRV	8	589	50.1	
	DEC	8-2	36	4.9	
	CRV	2	475	161	
	ACC	2-8	43	5.8	
	CRV	8	600	51	
	DEC	8-0	38	6.5	385.6

Table 4.4 - Jetrail One Hour Duty Cycle

Trip	Phase	Vel.	Dist	Time	Trip Time
PL		0	0	0	0
PL-T	ACC	0-15	161	14.6	14.6
	CRU	15	1893	86	100.6
	DEC	15-8	96	5.7	106.3
	CRU	8	591	50.3	156.6
	DEC	8-4	34	3.9	160.5
	CRU	4	275	46.1	206.6
	DEC	4-2	7	1.6	208.2
	CRU	2	35	12	220.2
	DEC	2-0	2	1.6	221.8
	STOP	0	0	20	241.8
T-LD	ACC	0-8	46	7.8	249.6
	CRU	8	746	63.5	313.1
	DEC	8-0	38	6.5	319.6
	STOP	0		20	339.6
LD-PL	ACC	0-15	161	14.6	354.2
	CRU	15	1539	70	424.2
	DEC	15-8	96	5.7	429.9
	CRU	8	320	27	456.9
	DEC	8-4	34	3.9	460.8
	CRU	4	275	47	407.8
	DEC	4-2	7	1.6	509.4
	CRU	2	316	107.4	616.8
	DEC	2-0	2	1.6	618.4
	STOP	0	0	20	638.4
PL-T	ACC	0-15	161	14.6	653.0
	CRU	15	1893	86	739.0
	DEC	15-8	96	5.7	744.7
	CRU	8	591	50.3	795.0
	DEC	8-4	34	3.9	798.9
	CRU	4	275	46.1	845.0
	DEC	4-2	7	1.6	846.6
	CRU	2	35	12	858.6
	DEC	2-0	2	1.6	860.2
	STOP	0	0	20	880.2
T-LD	ACC	0-8	46	7.8	888.0
	CRU	8	746	63.5	951.5
	DEC	8-0	38	6.5	958.0
	STOP	0	0	20	978.0

Table 4.4 - Jetrail One Hour Duty Cycle (cont)

Trip	Phase	Vel.	Dist.	Time	Trip Time
LD-PL	ACC	0-15	161	14.6	992.6
	CRU	15	1539	7.0	1062.6
	DEC	15-8	96	5.7	1068.3
	CRU	8	320	27	1095.3
	DEC	8-4	34	3.9	1099.2
	CRU	4	275	47	1146.2
	DEC	4-2	7	1.6	1147.8
	CRU	2	316	107.4	1255.8
	DEC	2-0	2	1.6	1257.4
	STOP	0	0	20	1277.4
PL-LD	ACC	0-15	161	14.6	1292.0
	CRU	15	1893	86	1378.0
	DEC	15-8	96	5.7	1383.7
	CRU	8	589	50.1	1433.8
	DEC	8-2	36	4.9	1438.7
	CRU	2	475	161	1599.7
	ACC	2-8	43	5.8	1605.5
	CRU	8	600	51	1656.5
	DEC	8-0	38	6.5	1663.0
	STOP	0	0	20	1683.0
LD-PL	ACC	0-15	161	14.6	1697.6
	CRU	15	1539	70	1767.6
	DEC	15-8	96	5.7	1773.3
	CRU	8	320	27	1800.3
	DEC	8-4	34	3.9	1804.2
	CRU	4	275	47	1851.2
	DEC	4-2	7	1.6	1852.8
	CRU	2	316	107.4	1960.2
	DEC	2-0	2	1.6	1961.8
	STOP	0	0	20	1981.8
PL-T	ACC	0-15	161	14.6	1996.4
	CRU	15	1893	86	2082.4
	DEC	15-8	96	5.7	2088.1
	CRU	8	591	50.3	2138.4
	DEC	8-4	34	3.9	2142.3
	CRU	4	275	46.1	2188.4
	DEC	4-2	7	1.6	2190.0
	CRU	2	35	12	2202.0
	DEC	2-0	2	1.6	2203.6
	STOP	0	0	20	2223.0

Table 4.4 - Jetrail One Hour Duty Cycle (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
T-PL	ACC	0-8	46	7.8	2231.4
	CRU	8	746	63.5	2294.9
	ACC	8-15	115	68	2301.7
	CRU	15	1539	70	2371.7
	DEC	15-8	96	5.7	2377.4
	CRU	8	320	27	2404.4
	DEC	8-4	34	3.9	2408.3
	CRU	4	275	47	2455.3
	DEC	4-2	7	1.6	2456.9
	CRU	2	316	107.4	2564.3
	DEC	2-0	2	1.6	2565.9
	STOP	0	0	20	2585.9
PL-T	ACC	0-15	161	14.6	2600.5
	CRU	15	1893	86	2686.5
	DEC	15-8	96	5.7	2692.2
	CRU	8	591	50.3	2742.5
	DEC	8-4	34	3.9	2746.4
	CRU	4	275	46.1	2792.5
	DEC	4-2	7	1.6	2794.1
	CRU	2	35	12	2806.1
	DEC	2-0	2	1.6	2807.7
	STOP	0	0	20	2827.7
T-LD	ACC	0-8	46	7.8	2835.5
	CRU	8	746	63.5	2899.0
	DEC	8-0	38	6.5	2905.5
	STOP	0	0	20	2925.5
LD-PL	ACC	0-15	161	14.6	2940.1
	CRU	15	1539	70	3010.1
	DEC	15-8	96	5.7	3015.8
	CRU	8	320	27	3042.8
	DEC	8-4	34	3.9	3046.7
	CRU	4	275	47	3093.7
	DEC	4-2	7	1.6	3095.3
	CRU	2	316	107.4	3202.7
	DEC	2-0	2	1.6	3204.3
	STOP	0	0	20	3224.3

Table 4.5 - Optimized System Route Schedule 1 (cont)

	Distance in Feet					Time in Seconds				Power in Ft.-Lbs.		
Trip	% Grade	Total	Acc	Cruise	Dec	Acc	Cruise	Dec	Total	Load	Torque Load	Torque Full Load
6-8	+1	900	306	594	x	16.7	16.2	x	32.9	4	36.3	39
	+2	4,800	x	4,589	211	x	125.1	11.5	136.6	4	42.6	46.2
8-7	+3	3,000	306	2,694	x	16.7	73.5	x	90.2	2	46.8	53.4
	-2	1,500	x	1,500	x	x	40.9	x	40.9	2	17.9	17.9
	+2	1,200	x	989	211	x	27	11.5	38.5	2	40.9	46.2
7-2	-2	2,400	306	1,883	211	16.7	51.4	11.5	79.6	0	17.9	17.9
2-5	-2	1,500	306	1,194	x	16.7	32.6	x	49.3	4	17.9	17.9
	+1	2,700	x	2,489	211	x	67.9	11.5	79.4	4	36.3	39
5-3	-1	3,000	306	2,483	211	16.7	67.7	11.5	95.9	6	24.1	24.5
3-4	+1	1,500	306	983	211	16.7	26.8	11.5	55	6	37.6	35
4-8	+1	1,200	306	894	x	16.7	24.4	x	41.1	8	39	39
	+2	6,300	x	6,089	211	x	166.1	11.5	177.6	8	46.2	46.2
8-1	+3	3,000	306	2,448	211	16.7	67.7	11.5	95.9	6	51.2	53.4
1-7	-2	1,500	306	1,194	x	16.7	32.5	x	49.2	2	17.9	17.9
	+2	1,200	x	989	211	x	27	11.5	38.5	2	40.9	46.2
7-3	-2	3,900	306	3,383	211	16.7	92.3	11.5	120.5	2	17.9	17.9
3-5	+1	2,700	306	2,183	211	16.7	59.5	11.5	87.7	4	36.3	39
5-6	+2	2,700	306	2,394	x	16.7	65.3	x	93.5	6	44.4	46.2
	-2	2,400	x	2,400	x	x	65.4	x	65.4	6	17.9	17.9
	+3	1,200	x	989	211	x	27	11.5	38.5	6	51.2	53.4
6-8	+1	900	306	594	x	16.7	16.2	x	32.9	6	37.6	39
	+2	4,800	x	4,589	211	x	125	11.5	136.5	4	42.6	46.2
8-2	+3	3,000	306	2,694	x	16.7	73.5	x	90.2	4	49	53.4
	-2	2,700	x	2,489	211	x	67.9	11.5	79.4	4	17.9	17.9
2-4	-2	1,500	306	1,194	x	16.7	32.6	x	49.3	6	17.9	17.9
	+1	1,500	x	1,289	211	x	35.1	11.5	46.6	6	37.6	39
4-3	+1	1,200	306	894	x	16.7	24.4	x	41.1	2	35	39
	-1	3,000	x	2,789	211	x	76.1	11.5	87.6	2	23.2	24.5
3-7	+1	2,700	306	2,392	x	16.7	65.3	x	82	4	36.3	39
	+2	2,700	x	2,489	211	x	67.9	11.5	79.4	4	42.6	46.2
7-6	-2	2,400	306	2,094	x	16.7	57.1	x	73.8	2	17.9	17.9
	+3	1,200	x	989	211	x	27	11.5	38.5	2	46.8	53.4
6-1	+1	900	306	594	x	16.7	16.2	x	32.9	2	35	39
	+2	4,800	x	4,800	x	x	130.9	x	130.9	2	40.9	46.2
	+3	3,000	x	2,789	211	x	76.1	11.5	87.6	2	46.8	53.4

Table 4.5 - Optimized System Route Schedule 1

		Distance in Feet				Time in Seconds				Power in Ft.-Lbs.		
Trip	% Grade	Total	Acc	Cruise	Dec	Acc	Cruise	Dec	Total	Pass. Load	Torque Load	Torque Full Load
1-6	-2	2,700	306	2,394	x	16.7	65.3	x	82	4	17.9	17.94
	+3	1,200	x	989	211	x	27	11.5	38.5	4	49	53.4
6-7	+1	900	306	594	x	16.7	16.2	x	32.9	6	37.6	39
	+2	1,200	x	989	211	x	27	11.5	38.5	6	44.4	46.2
7-2	-2	2,400	306	1,883	211	16.7	51.4	11.5	79.6	4	17.9	17.94
2-3	-2	1,500	306	983	211	16.7	26.8	11.5	55	2	17.9	17.9
3-5	+1	2,700	306	2,183	211	16.7	59.5	11.5	87.7	2	35	39
5-8	+2	6,300	306	5,783	211	16.7	157.7	11.5	185.9	6	44.4	46.2
8-2	+3	3,000	306	2,694	x	16.7	73.5	x	90.2	8	53.4	53.4
	-2	2,700	x	2,489	211	x	67.9	11.5	79.4	8	17.9	17.9
2-6	+3	1,200	306	683	211	16.7	18.6	11.5	46.8	6	51.2	53.4
6-7	+1	900	306	594	x	16.7	16.2	x	32.9	8	39	39
	+2	1,200	x	989	211	x	27	11.5	38.5	8	46.2	46.2
7-3	-2	3,900	306	3,383	211	16.7	92.3	11.5	120.5	6	17.9	17.9
3-4	+1	1,500	306	983	211	16.7	26.8	11.5	55	8	39	39
4-7	+1	1,200	306	894	x	16.7	24.4	x	41.1	4	36.3	39
	+2	2,700	x	2,489	211	x	67.9	11.5	79.4	4	42.6	46.2
7-1	+2	3,600	306	3,294	x	16.7	89.8	x	106.5	6	44.4	46.2
	+3	3,000	x	2,789	211	x	76	11.5	87.5	6	51.2	53.4
1-4	-2	4,200	306	3,894	x	16.7	106.2	x	122.9	0	17.9	17.9
	+1	1,500	x	1,289	211	x	35	11.5	46.5	0	33.7	39
4-2	+1	1,200	306	894	x	16.7	24.4	x	41.1	4	36.3	39
	+2	2,700	x	2,700	x	x	73.6	x	73.6	4	42.6	46.2
	-2	2,400	x	2,189	211	x	59.7	11.5	71.2	4	17.9	17.9
2-3	-2	1,500	306	983	211	16.7	26.8	11.5	55	2	17.9	17.9
3-5	+1	2,700	306	2,183	211	16.7	59.5	11.5	87.7	6		39
5-7	+2	2,700	306	2,183	211	16.7	59.5	11.5	87.7	8	46.2	46.2
7-8	+2	3,600	306	3,083	211	16.7	84.1	11.5	112.3	4	42.6	46.2
8-1	+3	3,000	306	2,483	211	16.7	67.7	11.5	95.9	8	53.4	53.4
1-3	-2	4,200	306	3,683	211	16.7	100.5	11.5	129.7	6	17.9	17.9
3-4	+1	1,500	306	983	211	16.7	26.8	11.5	55	4	36.3	39
4-7	+1	1,200	306	894	x	16.7	24.4	x	41.1	8	39	39
	+2	2,700	x	2,489	211	x	67.9	11.5	79.4	8	46.2	46.2
7-6	-2	2,400	306	2,094	x	16.7	57	x	73.7	6	17.9	17.9
	+3	1,200	x	989	211	x	27	11.5	38.5	6	51.2	53.4

Table 4.6 - Route Schedule 2

Distance in Feet						Time in Seconds				Power in Ft.-Lbs.		
Trip	% Grade	Total	Acc	Cruise	Dec	Acc	Cruise	Dec	Total	Load	Torque Load	Torque Full Load
1-6	-2	2,700	197	2,503	x	13.35	85.3	x	98.6	2	9.05	9.05
	+3	1,200	x	1,065	135	x	36.3	9.2	45.5	2	28.5	31.98
6-7	+1	900	197	703	x	13.35	23.9	x	37.2	6	22.8	22.8
	+2	1,200	x	1,065	135	x	36.3	9.2	45.5	6	27.4	27.4
7-2	-2	2,400	197	2,068	135	13.35	70.5	9.2	93	4	9	9.05
2-3	-2	1,500	197	1,168	135	13.35	39.8	9.2	62.3	2	9	9.05
3-5	+1	2,700	197	2,368	135	13.35	80.7	9.2	103.2	2	20.7	22.8
5-8	+2	6,300	197	5,968	135	13.35	203.4	9.2	225.9	6	27.4	27.4
8-2	+3	3,000	197	2,803	x	13.35	95.5	x	108.8	6	32	32
	-2	2,700	x	2,565	135	x	87.5	9.2	96.7	6	9	9.05
2-6	+3	1,200	197	868	135	13.35	29.6	9.2	52.1	6	32	32
6-7	+1	900	197	703	x	13.35	23.9	x	37.2	6	22.8	22.8
	+2	1,200	x	1,065	135	x	36.3	9.2	45.5	6	27.4	27.4
7-3	-2	3,900	197	3,568	135	13.35	121.6	9.2	144.1	6	9	9
3-4	+1	1,500	197	1,168	135	13.35	39.8	9.2	62.3	6	22.8	22.8
4-7	+1	1,200	197	1,003	x	13.35	34.2	x	47.5	4	21.7	22.8
	+2	2,700	x	2,565	135	x	87.5	9.2	96.7	4	26	27.4
7-1	+2	3,600	197	3,403	x	13.35	116	x	129.3	6	27.4	27.4
	+3	3,000	x	2,865	135	x	97.7	9.2	106.9	6	32	32
1-4	-2	4,200	197	4,003	x	13.35	136.5	x	149.8	0	9	9
	+1	1,500	x	1,365	135	x	46.5	9.2	55.7	9	19.6	22.8
4-2	+1	1,200	197	1,003	x	13.35	34.2	x	47.5	4	21.7	22.8
	+2	2,700	x	2,700	x	x	92	x	92	4	26	27.4
	-2	2,400	x	2,265	135	x	77.2	9.2	86.4	4	9	9
2-3	-2	1,500	197	1,168	135	13.35	3918	9.2	62.3	2	9	9
3-5	+1	2,700	197	2,368	135	13.35	80.6	9.2	103.1	6	22.8	22.8
5-7	+2	2,700	197	2,368	135	13.35	80.7	9.2	103.2	6	22.8	22.8
7-8	+2	3,600	197	3,268	135	13.35	111.4	9.2	133.9	4	21.7	22.8
8-1	+3	3,000	197	2,668	135	13.35	90.9	9.2	113.4	6	32	32
1-3	-2	4,200	197	3,868	135	13.35	131.8	9.2	154.3	4	9	9
3-4	+1	1,500	197	1,168	135	13.35	39.8	9.2	62.3	2	20.7	22.8
4-7	+1	1,200	197	1,003	x	13.35	34.2	x	47.5	6	22.8	22.8
	+2	2,700	x	2,568	135	x	87.5	9.2	96.7	6	27.4	27.4
7-6	-2	2,400	197	2,203	x	13.35	75.1	x	88.4	6	9	9
	+3	1,200	x	1,065	135	x	36.3	9.2	45.5	6	32	32

Table 4.6 - Route Schedule 2

Distance in Feet						Time in Seconds				Power in Ft.-Lbs.		
Trip	% Grade	Total	Acc	Cruise	Dec	Acc	Cruise	Dec	Total	Load	Torque Load	Torque Full Load
6-8	+1	900	197	703	x	13.35	23.9	x	37.2	4	21.7	22.8
	+2	4,800	x	4,665	135	x	159	9.2	168.2	4	26	27.4
8-7	+3	3,000	197	2,803	x	13.35	95.5	x	108.8	2	28.5	32
	-2	1,500	x	1,500	x	x	51.1	x	51.1	2	9	9
	+2	1,200	x	1,065	135	13.35	36.3	9.2	58.8	2	24.5	27.4
7-2	-2	2,400	197	2,068	135	13.35	70.5	9.2	93	0	9	9
2-5	-2	1,500	197	1,303	x	13.35	44.4	x	57.7	4	9	9
	+1	2,700	x	2,565	135	x	87.4	9.2	96.6	4	21.7	22.8
5-3	-1	3,000	197	2,668	135	13.35	90.9	9.2	113.4	6	13.6	13.6
3-4	+1	1,500	197	1,168	135	13.35	39.8	9.2	62.3	6	22.8	22.8
4-8	+1	1,200	197	1,003	x	13.35	34.2	x	47.5	6	22.8	22.8
	+2	6,300	x	6,165	135	x	210.2	9.2	219.4	6	27.4	27.4
8-1	+3	3,000	197	2,668	135	13.35	90.9	9.2	113.4	6	32	32
1-7	-2	1,500	197	1,303	x	13.35	44.4	x	57.7	2	9	9
	+2	1,200	x	1,065	135	x	36.3	9.2	45.5	2	24.5	27.4
7-3	-2	3,900	197	3,568	135	13.35	121.6	9.2	144.1	2	9	9
3-5	+1	2,700	197	2,368	135	13.35	80.7	9.2	103.2	4	21.7	22.8
5-6	+2	2,700	197	2,503	x	13.35	85.3	x	98.6	6	27.4	27.4
	-2	2,400	x	2,400	x	x	81.8	x	81.8	6	0	0
	+3	1,200	x	1,065	135	x	36.3	9.2	45.5	6	32	32
6-8	+1	900	197	703	x	13.35	23.9	x	37.2	6	22.8	22.8
	+2	4,800	x	4,665	135	x	159	9.2	168.2	4	26	27.4
8-2	+3	3,000	197	2,803	x	13.35	95.5	x	108.8	4	30.2	32
	-2	2,700	x	2,565	135	x	87.4	9.2	96.6	4	9	9
2-4	-2	1,500	197	1,303	x	13.35	44.4	x	57.7	6	9	9
	+1	1,500	x	1,365	135	x	46.5	9.2	55.7	6	22.8	22.8
4-3	+1	1,200	197	1,003	x	13.35	34.2	x	47.5	2	20.7	22.8
	-1	3,000	x	2,865	135	x	97.7	9.2	106.9	2	12.9	13.6
3-7	+1	2,700	197	2,503	x	13.35	85.3	x	98.6	4	21.7	22.8
	+2	2,700	x	2,565	135	x	87.5	9.2	96.7	4	26	27.4
7-6	-2	2,400	197	2,203	x	13.35	75.1	x	88.4	2	9	9
	+3	1,200	x	1,065	135	x	36.3	9.2	45.5	2	28.5	32
6-1	+1	900	197	703	x	13.35	23.9	x	37.2	2	20.7	22.8
	+2	4,800	x	4,800	x	x	163.6	x	163.6	2	24.5	27.4
	+3	3,000	x	2,865	135	x	97.7	9.2	106.9	2	28.5	32

Table 4.7 - Basic Duty Cycle for Optimized System

Trip	Phase	Vel.	Dist	Time	Trip Time
1-6	ACC	0-30	440	20	20
	CRU	30	2061	46	66
	DEC	30-15	230	7	73
	CRU	15	155	7	80
	ACC	15-30	330	10	90
	CRU	30	570	13	103
	DEC	30-0	300	13	116
	STOP	0	0	20	136
6-8	ACC	0-30	440	20	156
	CRU	30	140	3	159
	DEC	30-8	320	10	169
	CRU	8	78	7	176
	ACC	8-30	400	14.6	190
	CRU	30	570	13	203
	DEC	30-15	230	7	210
	CRU	15	155	7	216
	ACC	15-30	330	10	227
	CRU	30	2970	67.5	295
	DEC	30-0	300	13.7	309
	STOP	0	0	20	328.8
8-2	ACC	0-30	440	20	349
	CRU	30	2830	64	413
	DEC	30-15	230	7	420
	CRU	15	155	7	427
	ACC	15-30	330	10	437
	CRU	30	2070	47	484
	DEC	30-0	300	13.7	498
	STOP	0	0	20	517.5
2-4	ACC	0-30	440	20	537
	CRU	30	1290	29	566
	DEC	30-15	230	7	573
	CRU	15	155	7	580
	ACC	15-30	330	10	590
	CRU	30	870	20	610
	DEC	30-0	300	13.7	624
	STOP	0	0	20	644

Table 4.7 - Basic Duty Cycle for Optimized System (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
4-7	ACC	0-30	440	20	644
	CRU	30	440	10	674
	DEC	30-8	320	10	684
	CRU	8	78	7	691
	ACC	8-30	400	14.6	706
	CRU	30	2000	45.5	751
	DEC	30-0	300	13.7	765
	STOP	0	0	20	785
7-1	ACC	0-30	440	20	805
	CRU	30	2840	64.5	869
	DEC	30-8	370	10	880
	CRU	8	78	7	887
	ACC	8-30	400	14.6	901
	CRU	30	2300	52	953
	DEC	30-0	300	13.7	966
	STOP	0	0	20	986
1-5	ACC	0-30	440	20	1006
	CRU	30	3530	80	1086
	DEC	30-15	230	7	1093
	CRU	15	155	7	1100
	ACC	15-30	330	10	1110
	CRU	30	2070	47	1157
	DEC	30-0	300	13.7	1170
	STOP	0	0	20	1191
5-6	ACC	0-8	31	5	1196
	CRU	8	78	7	1203
	ACC	8-30	400	14.6	1218
	CRU	30	1980	45	1263
	DEC	30-8	320	10	1273
	CRU	8	78	7	1280
	ACC	8-30	400	14.6	1294
	CRU	30	480	11	1305
	DEC	30-8	320	10	1315
	CRU	8	78	7	1322
	ACC	8-30	400	14.6	1337
	CRU	30	570	13	1350
	DEC	30-15	230	7	1357
	CRU	15	155	7	1364
	ACC	15-30	330	10	1374
	CRU	30	570	13	1387
	DEC	30-0	300	13.7	1401
	STOP	0	0	20	1421

Table 4.7 - Basic Duty Cycle for Optimized System (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
6-1	ACC	0-30	440	20	1441
	CRU	30	140	3	1444
	DEC	30-8	320	10	1454
	CRU	8	78	7	1461
	ACC	8-30	400	14.6	1475
	CRU	30	570	13	1488
	DEC	30-15	230	7	1495
	CRU	15	155	7	1502
	ACC	15-30	330	10	1512
	CRU	30	2950	67	1579
	DEC	30-8	320	10	1589
	CRU	8	78	7	1596
	ACC	8-30	400	14.6	1611
	CRU	30	2300	52	1662
	DEC	30-0	300	13.7	1676
	STOP	0	0	20	1696
1-3	ACC	0-30	440	20	1716
	CRU	30	3530	80	1796
	DEC	30-15	230	7	1803
	CRU	15	155	7	1810
	DEC	15-0	75	7	1817
	STOP	0	0	20	1837
3-2	ACC	0-30	440	20	1857
	CRU	30	1940	44	1901
	DEC	30-8	320	10	1911
	CRU	8	78	7	1918
	ACC	8-30	400	14.6	1933
	CRU	30	1980	45	1978
	DEC	30-8	320	10	1988
	CRU	8	78	7	1995
	ACC	8-30	400	14.6	2010
	CRU	30	480	11	2021
	DEC	30-8	320	10	2031
	CRU	8	78	7	2038
	ACC	8-30	400	14.6	2052
	CRU	30	500	11	2063
	DEC	30-0	300	13.7	2077
	STOP	0	0	20	2097

Table 4.7 - Basic Duty Cycle for Optimized System (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
2-7	ACC	0-15	110	10	2107
	CRU	15	155	7	2114
	ACC	15-30	330	10	2124
	CRU	30	1450	33	2154
	DEC	30-8	320	10	2167
	CRU	8	78	7	2174
	ACC	8-30	400	14.6	2189
	CRU	30	500	11	2200
	DEC	30-0	300	13.7	2213
	STOP	0	0	20	2233
7-5	ACC	0-8	31	5	2238
	CRU	8	78	7	2245
	ACC	8-30	400	14.6	2260
	CRU	30	480	11	2271
	DEC	30-8	320	10	2281
	CRU	8	78	7	2288
	ACC	8-30	400	14.6	2303
	CRU	30	2070	47	2350
	DEC	30-15	230	7	2356
	CRU	15	155	7	2364
	ACC	15-30	330	10	2374
	CRU	30	2070	47	2421
	DEC	30-0	300	13.7	2434
	STOP	0	0	20	2454
5-8	ACC	0-8	31	5	2459
	CRU	8	78	7	2466
	ACC	8-30	400	14.6	2481
	CRU	30	2070	47	2528
	DEC	30-15	230	7	2535
	CRU	15	155	7	2542
	ACC	15-30	330	15	2557
	CRU	30	2970	67.5	2624
	DEC	30-0	300	13.7	2638
	STOP	0	0	20	2658
8-7	ACC	0-30	440	20	2678
	CRU	30	2330	53	2731
	DEC	30-15	230	7	2738
	CRU	15	155	7	2745
	ACC	15-30	330	10	2755
	CRU	30	850	19	2774
	DEC	30-8	320	10	2784
	CRU	8	78	7	2791

Table 4.7 - Basic Duty Cycle for Optimized System (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
8-7	ACC	8-30	400	14.6	2806
	CRU	30	480	11	2817
	DEC	30-8	320	10	2827
	CRU	8	78	6.7	2833
	DEC	8-0	21	4	2937
	STOP	0	0	20	2857
7-4	ACC	0-8	31	5	2862
	CRU	8	78	7	2869
	ACC	8-30	400	14.6	2884
	CRU	30	580	13	2897
	DEC	30-8	320	10	2907
	CRU	8	78	7	2914
	ACC	8-30	400	14.6	2928
	CRU	30	2070	47	2975
	DEC	30-15	230	7	2982
	CRU	15	155	7	2989
	ACC	15-30	330	10	2999
	CRU	30	2070	47	3047
	DEC	30-0	300	13.7	3060
	STOP	0	0	20	2080
4-3	ACC	0-30	440	20	3100
	CRU	30	440	10	3110
	DEC	30-8	320	10	3120
	CRU	8	78	6.7	3127
	ACC	8-30	400	14.6	3141
	CRU	30	2280	51	3192
	DEC	30-8	320	10	3202
	CRU	8	78	7	3209
	DEC	8-0	21	4	3213
	STOP	0	0	20	3234
3-8	ACC	0-30	440	20	3254
	CRU	30	1940	44	3298
	DEC	30-8	320	10	3308
	CRU	8	78	7	3315
	ACC	8-30	400	14.6	3329
	CRU	30	2070	47	3376
	DEC	30-15	230	67	3383
	CRU	15	155	7	3390
	ACC	15-30	330	10	3400
	CRU	30	2970	67	3467
	DEC	30-0	300	13.7	3481
	STOP	0	0	20	3501

Table 4.7 - Basic Duty Cycle for Optimized System (cont)

Trip	Phase	Vel.	Dist	Time	Trip Time
8-1	ACC	0-30	440	20	3521
	CRU	30	2260	51	3572
	DEC	30-0	300	13.7	3586
	STOP	0	0	20	3606

APPENDIX C

BOGIE DRIVE SYSTEM WHEEL LOADS

APPENDIX C

BOGIE DRIVE SYSTEM WHEEL LOADS

Assuming that live load is evenly distributed;

$$2F_{3y} = 16,000 \text{ lb}$$

using 60 mph winds;

$$F_6 = C_D q A$$

C_D = Drag coefficient = 1.05 (Ref Marks p. 11-75)

$q = \frac{1}{2} \rho V^2$; $\rho = 0.0024$ slug/ft^3

$v = 88$ ft/sec

$A = 30 \times 12 = 360$ ft^2

$$F_6 = 1.05 \left(\frac{1}{2} \times .0024 \times 88^2\right) 360 = 3624 \text{ lb} \qquad F_6$$

Using 10 mph speed around 100 ft. curve,

$$F_7 = \left(\frac{W}{g}\right) \frac{V^2}{r}$$

where $W = 2 F_{3y} = 16000$ lb

$g = 32.2$ ft/sec^2

$V = 10$ mph $= 14.7$ ft/sec

$r = 100$ ft

$$F_7 = \left(\frac{16000}{32.2}\right) \left(\frac{14.7^2}{100}\right) = 1074 \text{ lb} \qquad F_7$$

From the car free body diagram,

$\Sigma F_x = 0;\quad F_6 + F_7 - 2\,F_{3x} = 0$

$3624 + 1074 - 2F_{3x} = 0$

$2\,F_{3x} = 4698;\ F_{3x} = 2340$ lb $\qquad F_{3x}$

$\Sigma F_y = 0 \quad 2\,F_{3y} = 16000$ lb

$\Sigma M_A = 0 \quad 16000e - 66(F_6 + F_7) = 0$

$e = \dfrac{66x\ 4698}{16000} = 19.38$ inches $\qquad e$

$e = 66\ \sin\varphi = 19.38$ in.

$\sin\phi = \dfrac{19.38}{66} = .2937 \qquad \phi = 17^\circ 6'$

From the bogie drive diagram:

$\Sigma F_y = 0 \quad F_{1y} + F_{2y} - F_{3y} = 0$

$F_{1y} + F_{2y} = 8000$ lb

$\Sigma M_c = 0 \quad 6.4\ F_{1y} - 8.5\ F_{3x} - 3.2\ F_{3y} = 0$

$6.4\ F_{1y} - 8.5 \times 2349 - 3.2 \times 8000 = 0$

$F_{1y} = 7120$ lb $\qquad F_{1y}$

$F_{2y} = 8000 - 7120 = 880$ lb $\qquad F_{2y}$

F_{1y} is divided between 2 wheels;

Wheel load $= \dfrac{7120}{2} = 3560$ lb $\qquad$ Max. wheel load - most severe expected conditions

$$\text{Design load} = \frac{16000}{8} = 2000 \text{ lb/wheel}$$ Design load

$$\text{Expected load} = \frac{8000}{8} = 1000 \text{ lb/wheel}$$ Expected load

Load Ratings for Bogie Drive Wheel Fork Lift Tires Part No. WWM1230		
Velocity (mph)	Rated Load (lb)	Max. Short-Time Overload (lb)
5	1600	2000
10	1400	2000
15	1100	2000
20	700	2000

Conclusions

1. Wheel selected adequate for 8000 lb test vehicle on level.
2. Wheel overloaded on 20 mph downgrade.
3. Wheel not adequate for 16000 lb vehicle specified in MSE layout lb-100.
4. The vehicle should be operated at low speed when strong winds are expected, as during a thunderstorm or hurricane.

Guide Wheel Loads

16000 lb Vehicle

F_{3x} resisted by guide wheels on one side of rail.

$$F_{3x} = \frac{1}{2}\left(F_6 + F_7\right) = \frac{1}{2}\left(3624 + 1074\right) = 2350 \text{ lb}$$ F_{3x}

2 guide wheels share load:

$$\text{Load/wheel} = \frac{1}{2} F_{3x} = \frac{1}{2} \times 2350 = 1175 \text{ lb}$$

<u>8000 lb Vehicle</u>

Using 60 mph winds;

$$F_6 = C_D q A$$

C_D = 1.15 (Ref. marks Pg. 11-75) vehicle more nearly square

$$q = \frac{1}{2}\rho V^2; \ \rho = 0.0024 \text{ slug/ft}^3, \ V = 88 \text{ ft/sec}$$

$$A = 12 \times 12 = 144 \text{ ft}^2$$

$$F_6 = 1.05\left(\frac{1}{2} \times .0024 \times 88^2\right) 144 = 1539 \text{ lb}$$ F_6

Using 10 mph speed around 100 ft. curve,

$$F_7 = \frac{W}{g}\left(\frac{V^2}{r}\right)$$

where W = 8000 lb, g = 32.2 ft/sec^2, V = 10 mph = 14.7 ft/sec

r = 100 ft.

$$F_7 = \frac{8000}{32.2}\left(\frac{14.7^2}{100}\right) = 537 \text{ lb}$$ F_7

$$\text{Load/wheel} = \frac{\frac{1}{2}(F_6 + F_7)}{2} = \frac{\frac{1}{2}(1539 + 537)}{2} = 519 \text{ lb/wheel}$$

Load Ratings for Bogie Guide Wheel (Forklift Tires Part No. WWM 420)	
Velocity (mph)	Rated Load (lb)
5-10	970
15	770
20	550

The present guide wheels will be adequate both for the 8000 lb and the 16000 lb vehicle. There will be an overload on the tires for the 16000 lb vehicle, but this will be short time loading for worst case conditions.

APPENDIX D

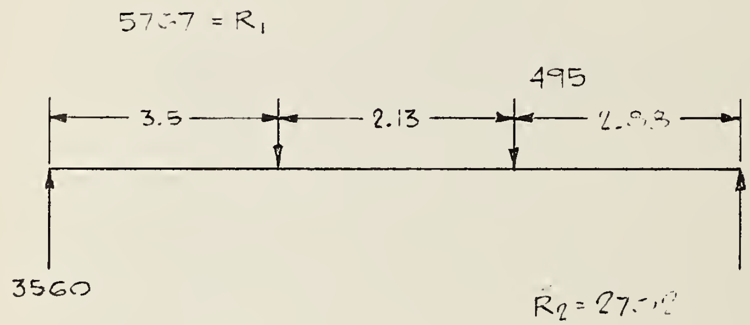
BOGIE DRIVE SYSTEM SHAFT LOADS

APPENDIX D

BOGIE DRIVE SYSTEM CHECK OF SHAFT LOADS

Upper Shaft

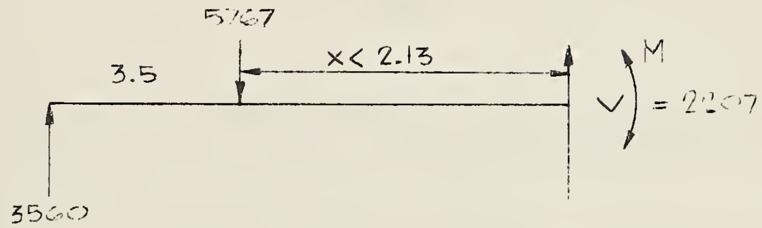
The maximum shaft load is expected during the wind and centrifugal force overload on the 16,000 lb vehicle. Reference Appendix II, "Bogie Drive System - Check of Bearing Capacity" for derivation of shaft loads.



Shaft Loading Diagram

Shaft dia. = 2 inches. Section Modulus $S = .7854R^3 = .7854\ in^3$.
Material: 1045 stl - yield strength F_y = 59,000 PSI.

Moment equation derivation:



Shaft Freebody Diagram

$$\Sigma M_V = 0; M - 5767X + 3560 (X + 3.5) = 0$$

$$M - 2207X + 12460 = 0$$

$$M = 2207X - 12460$$

At $X = 2.13$, $M = 2207 \times 2.13 - 12460 = -7759$ in lb

At $X = 0$, $M = -12460$ in lb $= M_{max}$

$$f_b = \frac{M_{max}}{S} = \frac{12,460}{.785} = 15,873 \text{ PSI}$$

$F_y = 59,000$ PSI; Factor of Safety $= \frac{F_y}{f_b} = \frac{59000}{15,873} = 3.72$

The upper shaft will be adequate.

Lower Shaft

Lower shaft takes torsion, little bending. From Machinery's Handbook, 14th ed., p 506

$$d = \sqrt[3]{\frac{321,000\ HP}{nS}}$$

where d = shaft diameter = 1.25"; HP = power to be transmitted = 10 hp; n = shaft rpm = 355 rpm; and S = shaft stress in psi (to be determined).

Rearranging the above equation to solve for S, we obtain:

$$S = \frac{321,000\ HP}{nd^3} = \frac{321,000 \times 10}{355 \times 1.25^3}$$

$$S = 4612\ PSI$$

This stress level is still well within the 4,000 - 6000 psi range considered allowable for commercial practice.

The lower shaft will be adequate.

BOGIE DRIVE SYSTEM CHECK OF BEARING CAPACITY

The following symbols will be used in this analysis:

L = Wheel load (lb.). May range from 1000 to 3560 lb.

F = Tension in chain (lb.)

R's = Reactions at bearings (lb.)

HP = Horsepower transmitted = 10 hp.

N = RPM of driving sprocket = 355 rpm.

$$HP = \frac{FrN}{5250}$$

$$\therefore \quad F = \frac{5250\ HP}{Nr} = \frac{5250 \times 10}{355 \times \frac{1}{2} \times \frac{7.168}{12}} = 495\ lb$$

From the shaft loading diagram

$$\Sigma M_{R_2} = 0; \ -2.88F - 5R_1 + 8.5L = 0$$

$$-5R_1 = 2.88F - 8.5L - 1426 - 8.5L$$

$$R_1 = \frac{1426 - 8.5L}{-5} = -285.1 + 1.7L$$

$$\Sigma F_y = 0 ; \quad L + R_2 - R_1 - F = 0$$

$$L + R_2 - R_1 - 495 = 0$$

$$R_2 = R_1 - L + 495$$

The above equations may be solved to obtain the value of the reactions:

Reactions for Various Wheel Loads			
L(lb)	R_1(lb)	R_2(lb)	Comment
1000	1415	910	Normal load for 8000 lb. vehicle.
2000	3115	1610	Normal load for 16000 lb. vehicle.
3560	5767	2702	Wind plus centrifugal force over-load on 16,000 lb. vehicle.

R_1 is taken by a Type E-2 $\frac{3}{16}$ Dodge piloted flange bearing. Load capacity exceeds 3430 lb. for 30,000 hour service life at 420 rpm. The bearing at R_1 is adequate. (Ref. p. 10-45 of Dodge Manufacturing Company Catalog D70.)

R_2 is taken by a Boston Gear 06976-11F pillow block. Load capacity exceeds 3000 lb at 300-400 rpm for a service life of 7500 hours. The bearing at R_2 is adequate.

$R_3 = F = 495$ lb. R_3 is taken by a Boston Gear 06954-6F flange bearing. The load rating is in excess of 2000 lb. at 400-500 rpm. The bearing at R_3 is adequate.

The load ratings for the Boston Gear bearings are taken from Boston Gear Catalog 60, pp. 820, 826.

APPENDIX E

SIMULATED DUTY CYCLE TABLES

APPENDIX E

Table 7.1 - Test Run 1
Jetrail Duty Cycle

Trip	Phase	Vel. (mph)	Dist. (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
PL		0	0	0	0	0	0
PL-T	ACC	0-15	161	14.6	14.6	0-1610	
	CRU	15	1893	86	100.6	1610	30
	DEC	15-8	96	5.7	106.3	1610	
	CRU	8	591	50.3	156.6	775	30
	DEC	8-4	34	3.9	160.5	775	
	CRU	4	275	46.1	206.6	385	30
	DEC	4-2	7	1.6	208.2	385	
	CRU	2	35	12	220.2	195	30
	DEC	2-0	2	1.6	221.8	195	
	STOP	0	0	20	241.8	0	0
T-LD	ACC	0-8	46	7.8	249.6	0-775	
	CRU	8	746	63.5	313.1	775	30
	DEC	8-0	38	6.5	319.6	775	
	STOP	0		20	339.6	0	0
LD-PL	ACC	0-15	161	14.6	354.2	0-1610	
	CRU	15	1539	70	424.2	1610	30
	DEC	15-8	96	5.7	429.9	1610	
	CRU	8	320	27	456.9	775	30
	DEC	8-4	34	3.9	460.8	775	
	CRU	4	275	47	407.8	385	30
	DEC	4-2	7	1.6	509.4	385	
	CRU	2	316	107.4	616.8	195	30
	DEC	2-0	2	1.6	618.4	195	
	STOP	0	0	20	638.4	0	0
PL-T	ACC	0-15	161	14.6	653.0	0-1610	
	CRU	15	1893	86	739.0	1610	30
	DEC	15-8	96	5.7	744.7	1610	
	CRU	8	591	50.3	795.0	775	30
	DEC	8-4	34	3.9	798.9	775	
	CRU	4	275	46.1	845.0	385	30
	DEC	4-2	7	1.6	846.6	385	
	CRU	2	35	12	858.6	195	30
	DEC	2-0	2	1.6	860.2	195	
	STOP	0	0	20	880.2	0	0

Table 7.1 - Test Run 1
Jetrail Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
T-LD	ACC	0-8	46	7.8	888.0	0-775	
	CRU	8	746	63.5	951.5	775	30
	DEC	8-0	38	6.5	958.0	775	
	STOP	0	0	20	978.0	0	0
LD-PL	ACC	0-15	161	14.6	992.6	0-1610	
	CRU	15	1539	7.0	1062.6	1610	30
LD-PL	DEC	15-8	96	5.7	1068.3	1610	
	CRU	8	320	27	1095.3	775	30
	DEC	8-4	34	3.9	1099.2	775	
	CRU	4	275	47	1146.2	385	30
	DEC	4-2	7	1.6	1147.8	385	
	CRU	2	316	107.4	1255.8	195	30
	DEC	2-0	2	1.6	1257.4	195	
	STOP	0	0	20	1277.4	0	0
PL-LD	ACC	0-15	161	14.6	1292.0	0-1610	
	CRU	15	1893	86	1378.0	1610	30
	DEC	15-8	96	5.7	1383.7	1610	
	CRU	8	589	50.1	1433.8	775	30
	DEC	8-2	36	4.9	1438.7	775	
	CRU	2	475	161	1599.7	195	30
	ACC	208	43	5.8	1605.5	195	
	CRU	8	600	51	1656.5	775	30
	DEC	8-0	38	6.5	1663.0	755	
	STOP	0	0	20	1683.0	0	0
LD-PL	ACC	0-15	161	14.6	1697.6	0-1610	
	CRU	15	1539	70	1767.6	1610	30
	DEC	15-8	96	5.7	1773.3	1610	
	CRU	8	320	27	1800.3	775	30
	DEC	8-4	34	3.9	1804.2	775	
	CRU	4	275	47	1851.2	385	30
	DEC	4-2	7	1.6	1852.8	385	
	CRU	2	316	107.4	1960.2	195	30
	DEC	2-0	2	1.6	1961.8	195	
	STOP	0	0	20	1981.8	0	0

Table 7.1 - Test Run 1
Jetrail Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
PL-T	ACC	0-15	161	14.6	1996.4	0-1610	
	CRU	15	1893	86	2082.4	1610	30
	DEC	15-8	96	5.7	2088.1	1610	
	CRU	8	591	50.3	2138.4	775	30
	DEC	8-4	34	3.9	2142.3	775	
	CRU	4	275	46.1	2188.4	385	30
	DEC	4-2	7	1.6	2190.0	385	
	CRU	2	35	12	2202.0	195	30
	DEC	2-0	2	1.6	2203.6	195	
	STOP	0	0	20	2223.0	0	0
T-PL	ACC	0-8	46	7.8	2231.4	0-775	
	CRU	8	746	63.5	2294.9	775	30
	ACC	8-15	115	68	2301.7	775	
	CRU	15	1539	70	2371.7	1610	30
	DEC	15-8	96	5.7	2377.4	1610	
	CRU	8	320	27	2404.4	775	30
	DEC	8-4	34	3.9	2408.3	775	
	CRU	4	275	47	2455.3	385	30
	DEC	4-2	7	1.6	2456.9	385	
	CRU	2	316	107.4	2564.3	195	30
	DEC	2-0	2	1.6	2565.9	195	
	STOP	0	0	20	2585.9	0	0
PL-T	ACC	0-15	161	14.6	2600.5	0-1610	
	CRU	15	1893	86	2686.5	1610	30
	DEC	15-8	96	5.7	2692.2	1610	
	CRU	8	591	50.3	2742.5	775	30
	DEC	8-4	34	3.9	2746.4	775	
	CRU	4	275	46.1	2792.5	385	30
	DEC	4-2	7	1.6	2794.1	385	
	CRU	2	35	12	2806.1	195	30
	DEC	2-0	2	1.6	2807.7	195	
	STOP	0	0	20	2827.7	0	0
T-LD	ACC	0-8	46	7.8	2835.5	0-775	
	CRU	8	746	63.5	2899.0	775	30
	DEC	8-0	38	6.5	2905.5	775	
	STOP	0	0	20	2925.5	0	0

Table 7.1 - Test Run 1
Jetrail Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
LD-PL	ACC	0-15	161	14.6	2940.1	0-1610	
	CRU	15	1539	70	3010.1	1610	30
	DEC	15-8	96	5.7	3015.8	1610	
	CRU	8	320	27	3042.8	775	30
	DEC	8-4	34	3.9	3046.7	775	
	CRU	4	275	47	3093.7	385	30
	DEC	4-2	7	1.6	3095.3	385	
	CRU	2	316	107.4	3202.7	195	30
	DEC	2-0	2	1.6	3204.3	195	
	STOP	0	0	20	3224.3	0	0

PL = Parking Lot
T = Terminal
LD = Loading Dock
(Ref. — Figure 4.2)

Table 7. 2 - Test Run 2 Optimized System Duty Cycle

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
1-6	ACC	0-30	440	20	20		
	CRU	30	2061	46	66	1610	21.4
	DEC	30-15	230	7	73		
	CRU	15	155	7	80	805	21.4
	ACC	15-30	330	10	90		
	CRU	30	570	13	103	1610	21.4
	DEC	30-0	300	13	116		
	STOP	0	0	20	136	0	0
6-8	ACC	0-30	440	20	156		
	CRU	30	140	3	159	1610	21.4
	DEC	30-8	320	10	169		
	CRU	8	78	7	176	430	21.4
	ACC	8-30	400	14.6	190		
	CRU	30	570	13	203	1610	21.4
	DEC	30-15	230	7	210		
	CRU	15	155	7	216	805	21.4
	ACC	15-30	330	10	227		
	CRU	30	2970	67.5	295	430	21.4
	DEC	30-0	300	13.7	309		
	STOP	0	0	20	328.8	0	0
8-2	ACC	0-30	440	20	349		
	CRU	30	2830	64	413	1610	21.4
	DEC	30-15	230	7	420		
	CRU	15	155	7	427	805	21.4
	ACC	15-30	330	10	437		
	CRU	30	2070	47	484	1610	21.4
	DEC	30-0	300	13.7	498		
	STOP	0	0	20	517.5	0	0
2-4	ACC	0-30	440	20	537		
	CRU	30	1290	29	566	1610	35.2
	DEC	30-15	230	7	573		
	CRU	15	155	7	580	805	35.2
	ACC	15-30	330	10	590		
	CRU	30	870	20	610	1610	35.2
	DEC	30-0	300	13.7	624		
	STOP	0	0	20	644	0	0

Table 7. 2 - Test Run 2 Optimized System Duty Cycle (cont)

Trip	Phase	Vel.	Dist.	Time	Trip Time	RPM	Torque
4-7	ACC	0-30	440	20	644		
	CRU	30	440	10	674	1610	35. 2
	DEC	30-8	320	10	684		
	CRU	8	78	7	691	430	35. 2
	ACC	8-30	400	14. 6	706		
	CRU	30	2000	45. 5	751	1610	35. 2
	DEC	30-0	300	13. 7	765		
	STOP	0	0	20	785	0	0
7-1	ACC	0-30	440	20	805		
	CRU	30	2840	64. 5	869	1610	35. 2
	DEC	30-8	370	10	880		
	CRU	8	78	7	887	805	35. 2
	ACC	8-30	400	14. 6	901		
	CRU	30	2300	52	953	1610	35. 2
	DEC	30-0	300	13. 7	966		
	STOP	0	0	20	986	0	0
1-5	ACC	0-30	440	20	1006		
	CRU	30	3530	80	1086	1610	35. 2
	DEC	30-15	230	7	1093		
	CRU	15	155	7	1100	805	35. 2
	ACC	15-30	330	10	1110		
	CRU	30	2070	47	1157	1610	35. 2
	DEC	30-0	300	13. 7	1171		
	STOP	0	0	20	1191	0	0
5-6	ACC	0-8	31	5	1196		
	CRU	8	78	7	1203	430	33. 6
	ACC	8-30	400	14. 6	1218		
	CRU	30	1980	45	1263	1610	33. 6
	DEC	30-8	320	10	1273		
	CRU	8	78	7	1280	430	33. 6
	ACC	8-30	400	14. 6	1294		
	CRU	30	480	11	1305	1610	33. 6
	DEC	30-8	320	10	1315		
	CRU	8	78	7	1322	430	33. 6
	ACC	8-30	400	14. 6	1337		
	CRU	30	570	13	1350	1610	33. 6
	DEC	30-15	230	7	1357		
	CRU	15	155	7	1364	805	33. 6
	ACC	15-30	330	10	1374		
	CRU	30	570	13	1387	1610	33. 6
	DEC	30-0	300	13. 7	1401		
	STOP	0	0	20	1421	0	0

Table 7. 2 - Test Run 2 Optimized System Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
6-1	ACC	0-30	440	20	1441		
	CRU	30	140	3	1444	1610	33. 6
	DEC	30-8	320	10	1454	1610	33. 6
	CRU	8	78	7	1461	430	33. 6
	ACC	8-30	400	14. 6	1475		
	CRU	30	570	13	1488	1610	33. 6
	DEC	30-15	230	7	1495		
	CRU	15	155	7	1502	805	33. 6
	ACC	15-30	330	10	1512		
	CRU	30	2950	67	1579	1610	33. 6
	DEC	30-8	320	10	1589		
	CRU	8	78	7	1596	430	33. 6
	ACC	8-30	400	14. 6	1611		
	CRU	30	2300	52	1662	1610	33. 6
	DEC	30-0	300	13. 7	1676		
	STOP	0	0	20	1696	0	0
1-3	ACC	0-30	440	20	1716		
	CRU	30	3530	80	1796	1610	9. 18
	DEC	30-15	230	7	1803		
	CRU	15	155	7	1810	805	9. 18
	DEC	15-0	75	7	1817		
	STOP	0	0	20	1837	0	0
3-2	ACC	0-30	440	20	1857		
	CRU	30	1940	44	1901	1610	9. 18
	DEC	30-8	320	10	1911		
	CRU	8	78	7	1918	430	9. 18
	ACC	8-30	400	14. 6	1933		
	CRU	30	1980	45	1978	1610	9. 18
	DEC	30-8	320	10	1988		
	CRU	8	78	7	1995	430	9. 18
	ACC	8-30	400	14. 6	2010		
	CRU	30	480	11	2021	1610	9. 18
	DEC	30-8	320	10	2031		
	CRU	8	78	7	2038	430	9. 18
	ACC	8-30	400	14. 6	2052		
	CRU	30	500	11	2063	1610	9. 18
	DEC	30-0	300	13. 7	2077		
	STOP	0	0	20	2097	0	0

Table 7. 2 - Test Run 2 Optimized System Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
8-7	ACC	0-30	440	20	2678		
	CRU	30	2330	53	2731	1610	44. 9
	DEC	30-15	230	7	2738		
	CRU	15	155	7	2745	805	44. 9
	ACC	15-30	330	10	2755		
	CRU	30	850	19	2774	1610	44. 9
	DEC	30-8	320	10	2784		
	CRU	8	78	7	2791	430	44. 9
	ACC	8-30	400	14. 6	2806		
	CRU	30	480	11	2817	1610	44. 9
	DEC	30-8	320	10	2827		
	CRU	8	78	6. 7	2833	430	44. 0
	DEC	8-0	21	4	2937		
	STOP	0	0	20	2857	0	0
-4	ACC	0-8	31	5	2862		
	CRU	8	78	7	2869	430	44. 9
	ACC	8-30	400	14. 6	2884		
	CRU	30	580	13	2897	1610	44. 9
	DEC	30-8	320	10	2907		
	CRU	8	78	7	2914	430	44. 9
	ACC	8-30	400	14. 6	2928		
	CRU	30	2070	47	2975	1610	44. 9
	DEC	30-15	230	7	2982		
	CRU	15	155	7	2989	805	44. 9
	ACC	15-30	330	10	2999		
	CRU	30	2070	47	3047	1610	44. 9
	DEC	30-0	300	13. 7	3060		
	STOP	0	0	20	2080	0	0
4-3	ACC	0-30	440	20	3100		
	CRU	30	440	10	3110	1610	-3
	DEC	30-8	320	10	3120		
	CRU	8	78	6. 7	3127	430	-3
	ACC	8-30	400	14. 6	3141		
	CRU	30	2280	51	3192	1610	-3
	DEC	30-8	320	10	3202		
	CRU	8	78	7	3209	430	-3
	DEC	8-0	21	4	3213		
	STOP	0	0	20	3234	0	0

Table 7. 2 - Test Run 2 Optimized System Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
-7	ACC	0-15	110	10	2107		
	CRU	15	155	7	2114	805	9. 18
	ACC	15-30	330	10	2124		
	CRU	30	1450	33	2154	1610	9. 18
	DEC	30-8	320	10	2167		
	CRU	8	78	7	2174	430	9. 18
	ACC	8-30	400	14. 6	2189		
	CRU	30	500	11	2200	1610	9. 18
	DEC	30-0	300	13. 7	2213		
	STOP	0	0	20	2233	0	0
7-5	ACC	0-8	31	5	2238		
	CRU	8	78	7	2245	430	43. 3
	ACC	8-30	400	14. 6	2260		
	CRU	30	480	11	2271	1610	43. 3
	DEC	30-8	320	10	2281		
	CRU	8	78	7	2288	430	43. 3
	ACC	8-30	400	14. 6	2303		
	CRU	30	2070	47	2350	1610	43. 3
	DEC	30-15	230	7	2356		
	CRU	15	155	7	2364	805	43. 3
	ACC	15-30	330	10	2374		
	CRU	30	2070	47	2421	1610	43. 3
	DEC	30-0	300	13. 7	2434		
	STOP	0	0	20	2454	0	0
	ACC	0-8	31	5	2459		
	CRU	8	78	7	2466	430	43. 3
	ACC	8-30	400	14. 6	2481		
	CRU	30	2070	47	2528	1610	43. 3
	DEC	30-15	230	7	2535		
	CRU	15	155	7	2542	805	43. 3
	ACC	15-30	330	15	2557		
	CRU	30	2970	67. 5	2624	1610	43. 3
	DEC	30-0	300	13. 7	2638		
	STOP	0	0	20	2658	0	0

Table 7. 2 - Test Run 2 Optimized System Duty Cycle (cont)

Trip	Phase	Vel (mph)	Dist (ft)	Time (sec)	Trip Time (sec)	Output Shaft Speed (rpm)	Torque (ft-lb)
8	ACC	0-30	440	20	3254		
	CRU	30	1940	44	3298	1610	-3
	DEC	30-8	320	10	3308		
	CRU	8	78	7	3315	430	-3
	ACC	8-30	400	14.6	3329		
	CRU	30	2070	47	3376	1610	-3
	DEC	30-15	230	67	3383		
	CRU	15	155	7	3390	805	-3
	ACC	15-30	330	10	3400		
	CRU	30	2970	67	3467	1610	-3
	DEC	30-0	300	13.7	3481		
	STOP	0	0	20	3501	0	0
8-1	ACC	0-30	440	20	3521		
	CRU	30	2260	51	3572	1610	-3
	DEC	30-0	300	13.7	3586		
	STOP	0	0	20	3606	0	0

APPENDIX F

TEST RUN RECORDINGS

Table F-1. Test Log Data

Sequence	Description
1.0	First day of testing, start Jetrail duty cycle as described in Test Run #1. Duration approximately 60 minutes.
1.1	Acceleration to maximum speed (1610 RPM's. Flywheel inertia only.
1.2	Acceleration to 775 RPM's. Flywheel inertia only.
1.3	Acceleration to 385 RPM's. Flywheel inertia only.
1.4	Maximum clutch dissipation. Run at low speed with 34 ft-lb of torque, followed by torque at 46 ft-lb. Total run time of 22 minutes. No excess heating observed.
1.5	"Jetrail" system duty cycle described in Test Run #1 was executed. Final run. Duration approximately 60 minutes. (Repeat of Sequence 1.0)
1.6	Acceleration to maximum speed (1610 RPM's) then braking.
1.7	Acceleration to 775 RPM's, then braking.
1.8	Acceleration to 385 RPM's, then deceleration.
1.9	Acceleration to 200 RPM's, then deceleration.
2.0	Second day of testing, start "Optimized" system tests, Test Run #2. Duration approximately 60 minutes
2.1	Restart test after 176 seconds.
2.2	9:43 - Overload breaker tripped off.
2.3	9:48 - Overload breaker tripped off.
2.4	10:10 - Overload breaker tripped off. Restart 12 minutes in test. Higher rated current limiter inserted.
2.5	10:35 - Overload Breaker tripped off. Restart at 10:37.
2.6	45 minutes, 7 seconds in Test Run #2 - Overload breaker tripped off.
2.7	Restart cycle at 45 minutes, 35 seconds into Test Run #2.
2.8	Overload circuit breaker tripped off.

Table F-1. Test Log Data (Continued)

Sequence	Description
2.9	Higher rated current limiter inserted - restart at 10:58.
2.10	Optimized system duty cycle described in Test Run #2. Final run. Time: 11:06.
2.11	Readjust channel 5 (drive current) from 200 to 500 Sensitivity Scale.
2.12	Restart Test Run #2
2.13	Minor schedule readjustment of Test Run #2 due to technician's error.
2.14	End Test #2 - start acceleration - deceleration tests.
2.15	Emergency braking or stop, maximum speed (1610 RPM's) to zero RPM's. Torque = 21.4 ft-lbs.
2.16	Braking test, maximum speed. Torque = 35.2 ft-lbs.
2.17	Braking test, maximum speed. Torque = 45.0 ft-lbs.
2.18	Acceleration test. Torque = 45.0 ft-lbs.
2.19	10% grade, braking test. Torque = 60.0 ft-lbs.
2.20	Torque = 60.0 ft-lbs, acceleration test. High acceleration rate.
2.21	Torque = 60 ft-lbs. Slow acceleration rate.
2.22	Slow acceleration rate to second speed. Torque = 60.0 ft-lbs.
2.23	Slow acceleration to maximum speed. Torque = 60.0 ft-lbs.
2.24	High acceleration. Torque = 64.0 ft-lbs.
	- End of Laboratory Tests -

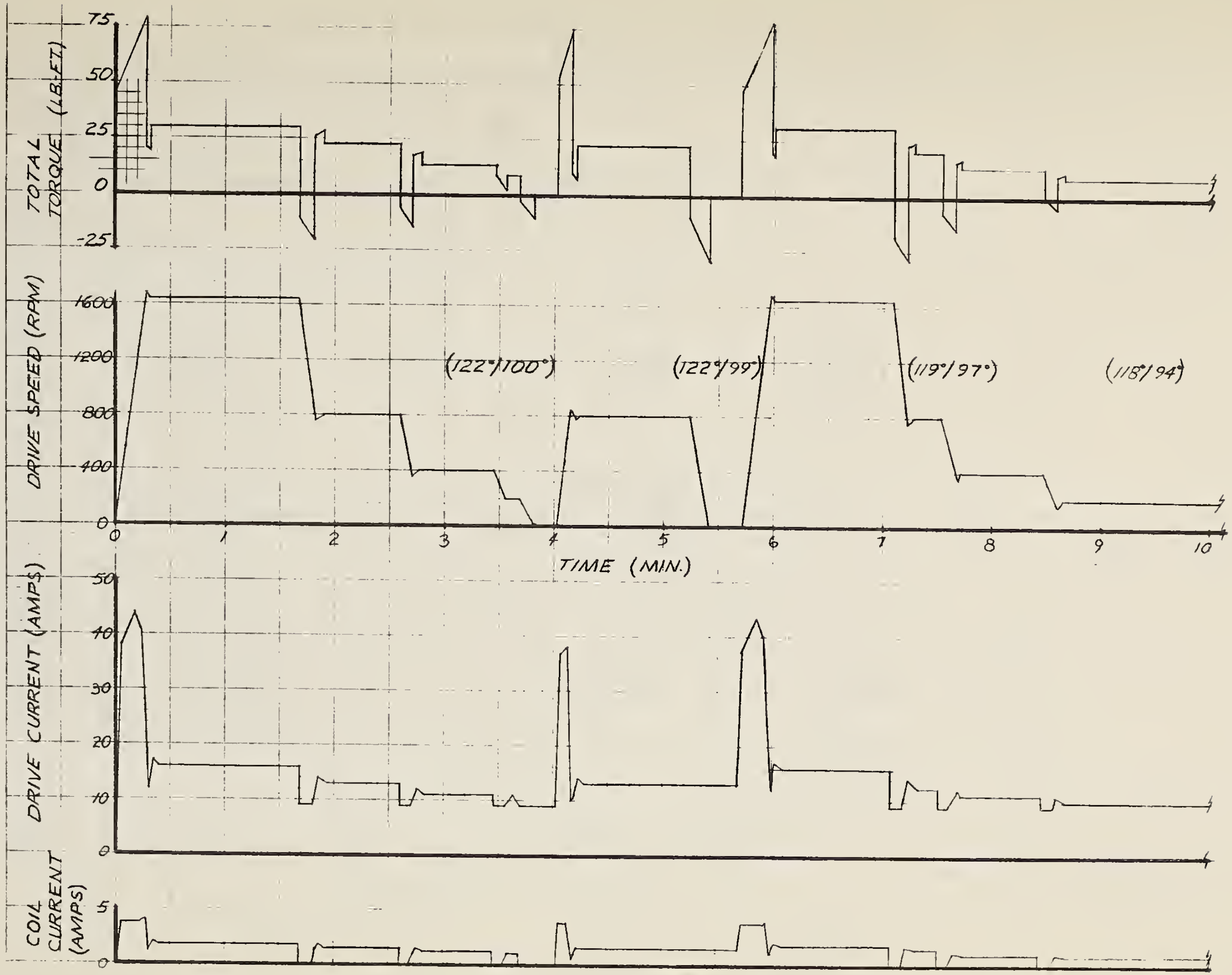


Figure F-1(1) TEST RUN 1 RECORDINGS

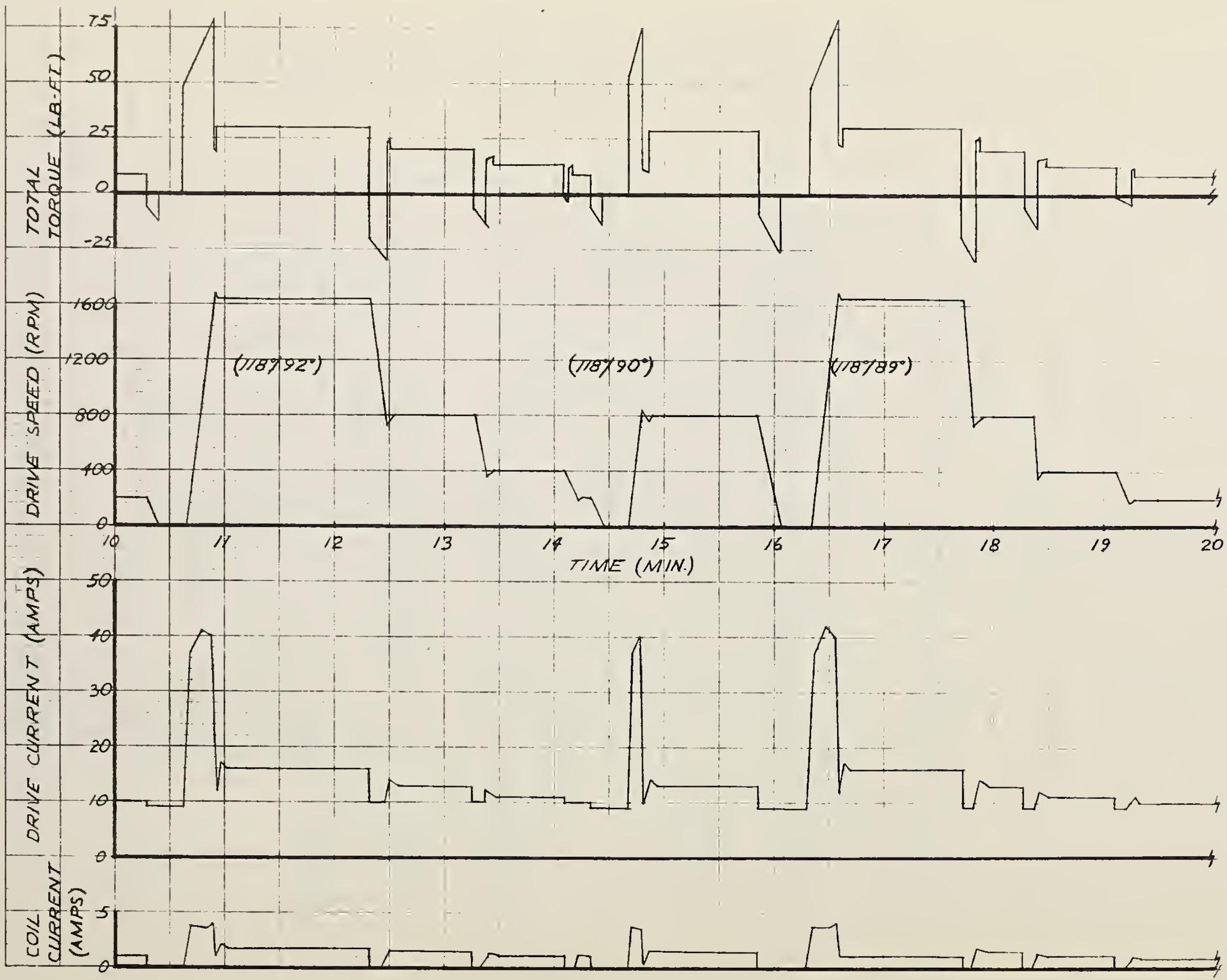


Figure F-1(2) TEST RUN 1 RECORDINGS

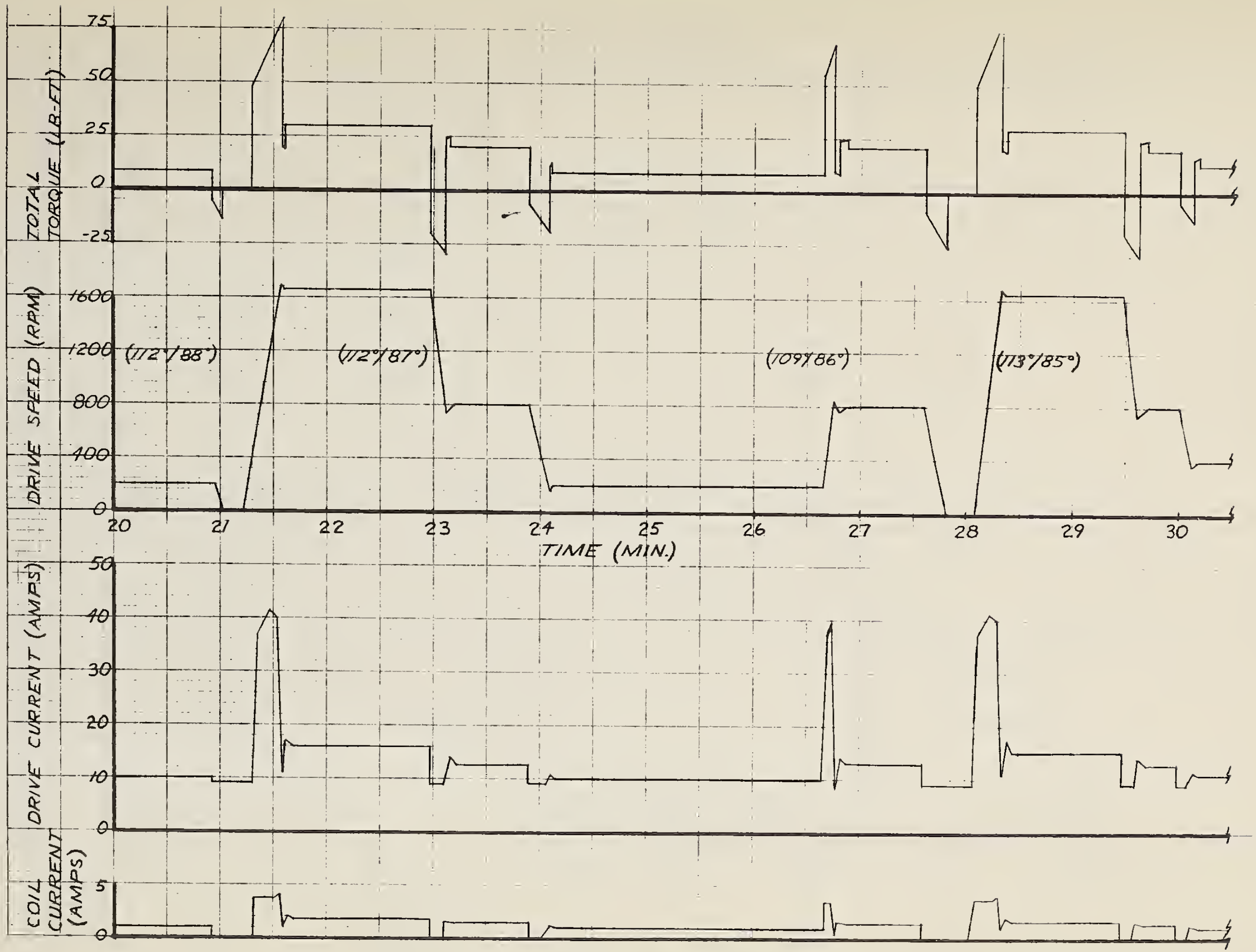


Figure F-1(3) TEST RUN 1 RECORDINGS

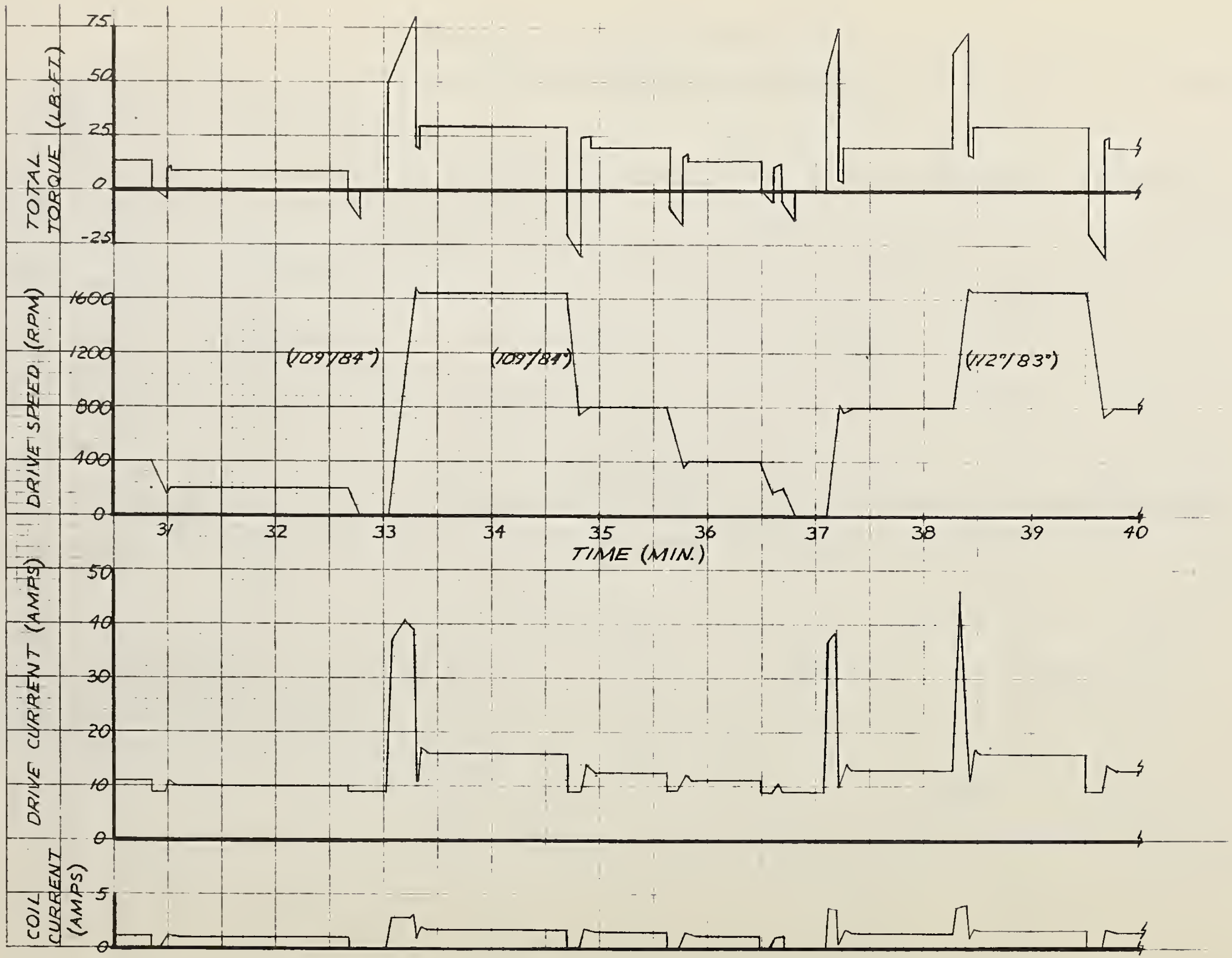


Figure F-1(4) TEST RUN 1 RECORDINGS

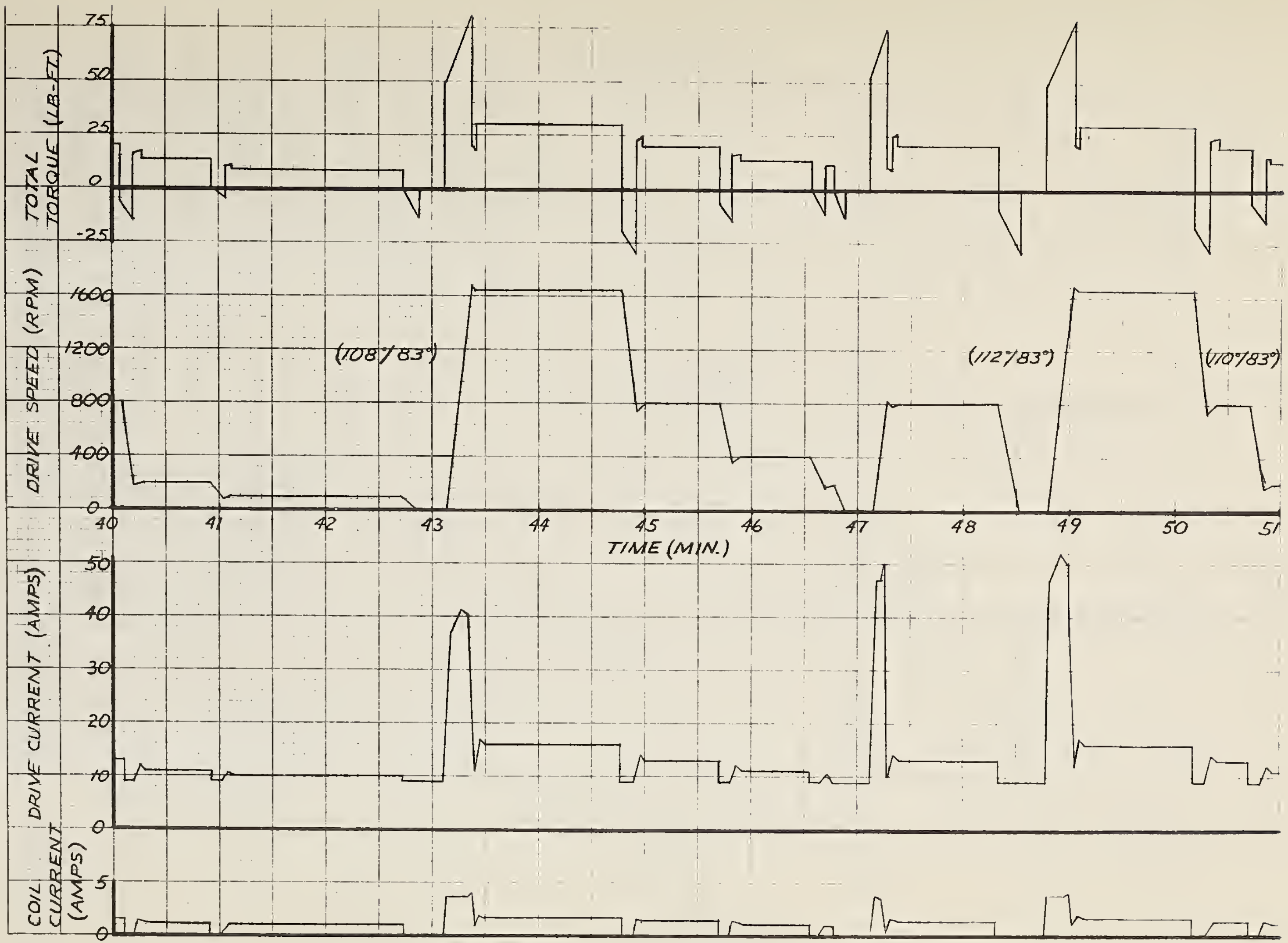


Figure F-1(5) TEST RUN 1 RECORDINGS

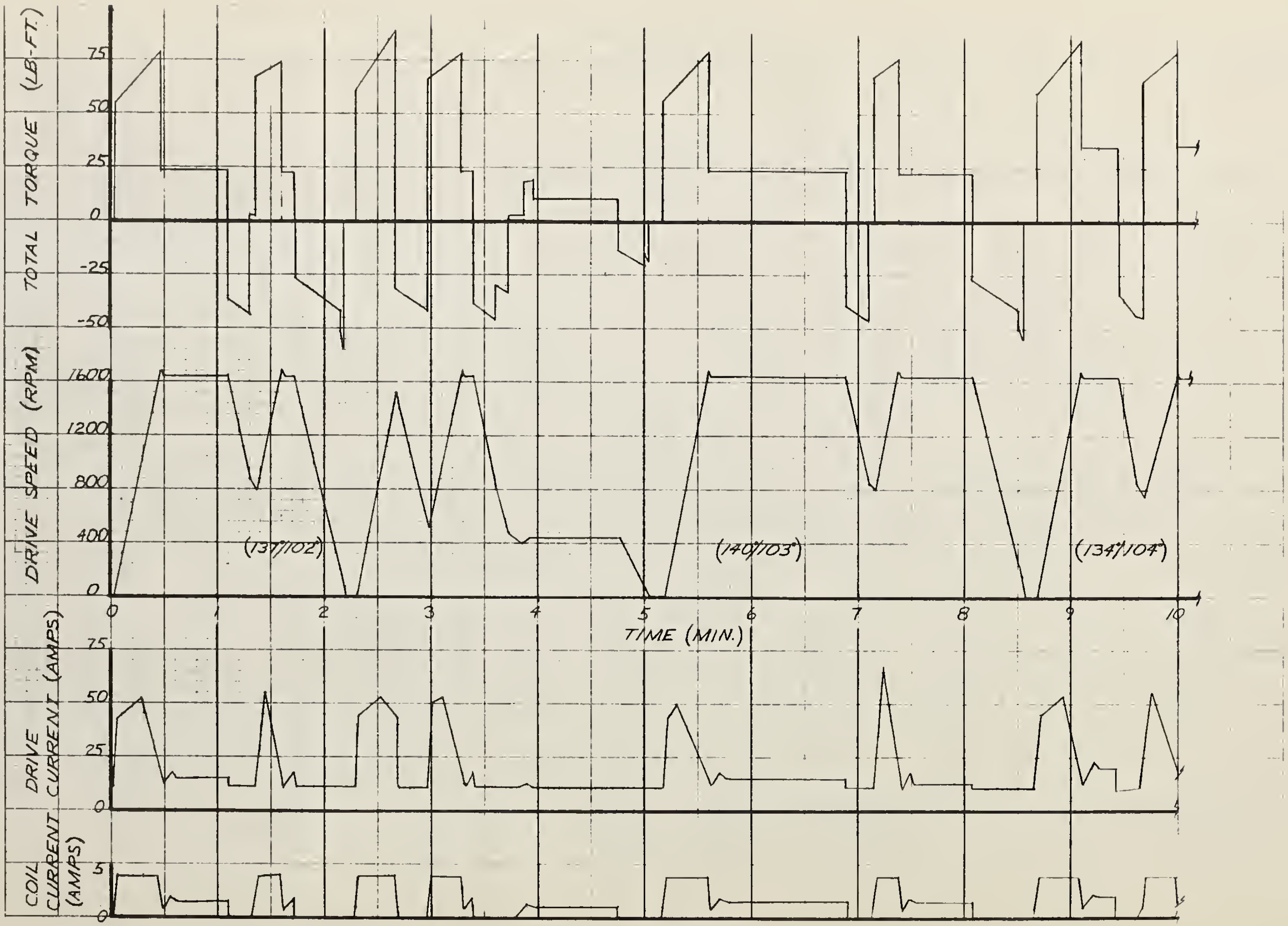


Figure F-2(1) TEST RUN 2 RECORDINGS

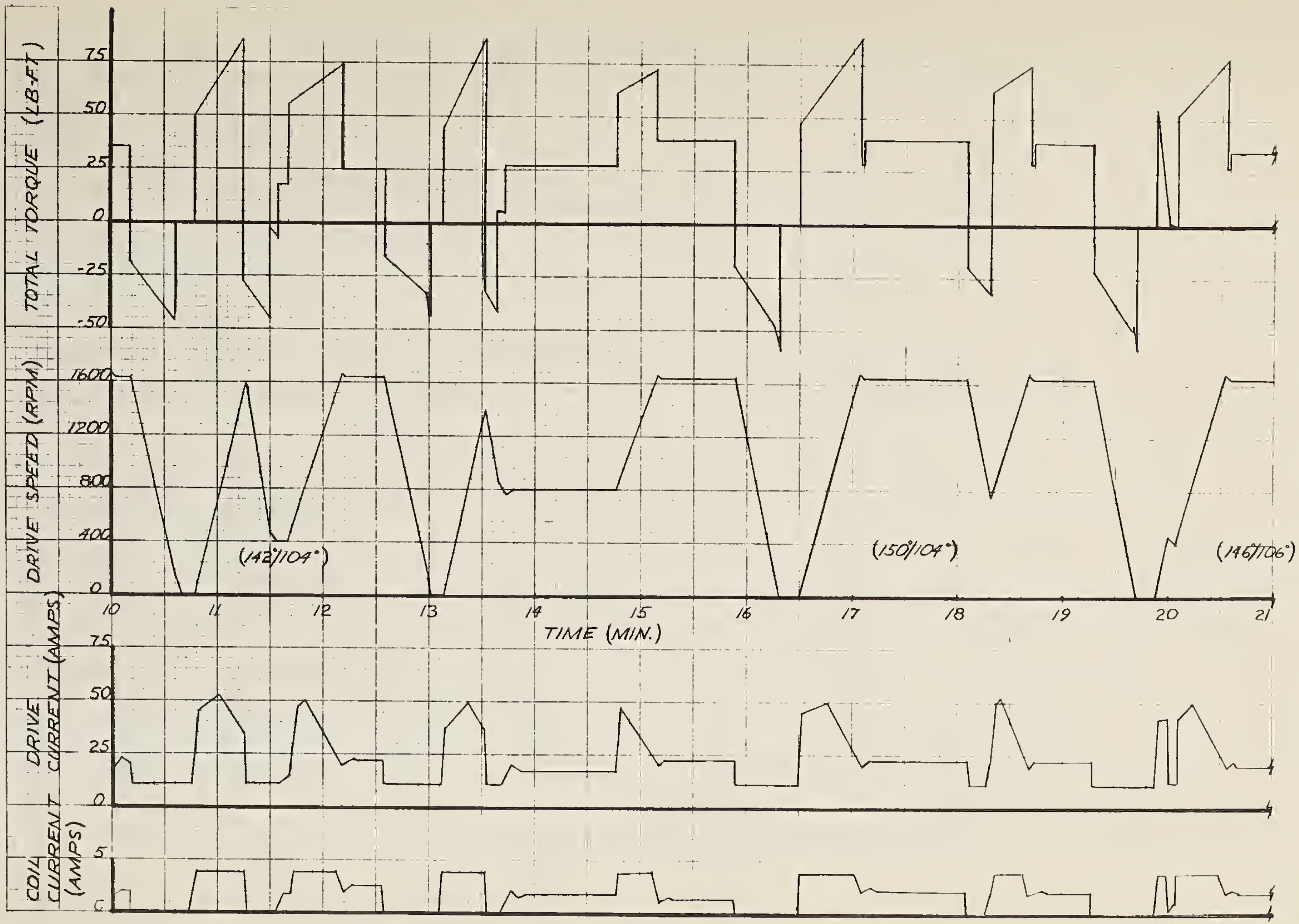


Figure F-2(2) TEST RUN 2 RECORDINGS

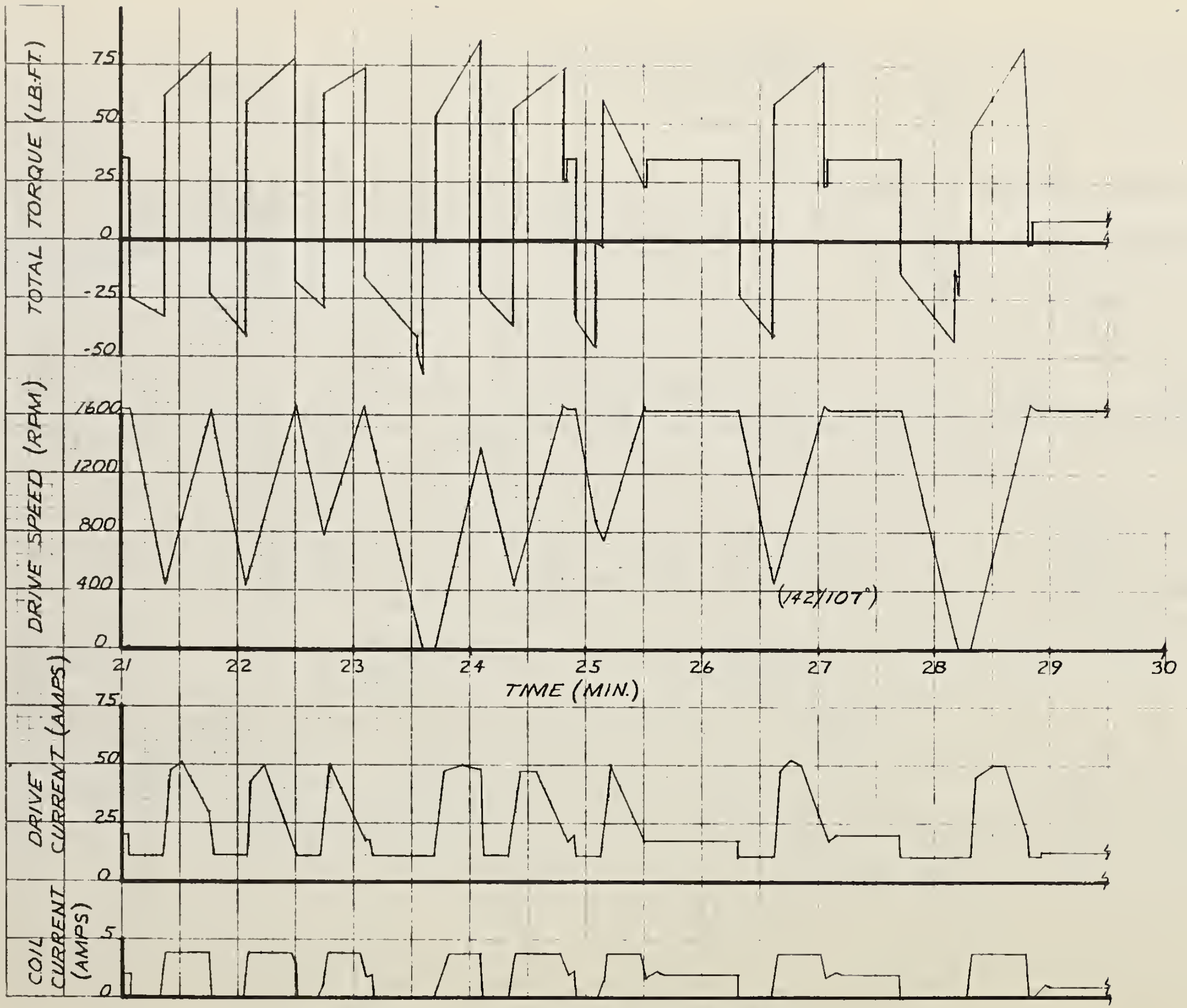


Figure F-2(3) TEST RUN 2 RECORDINGS

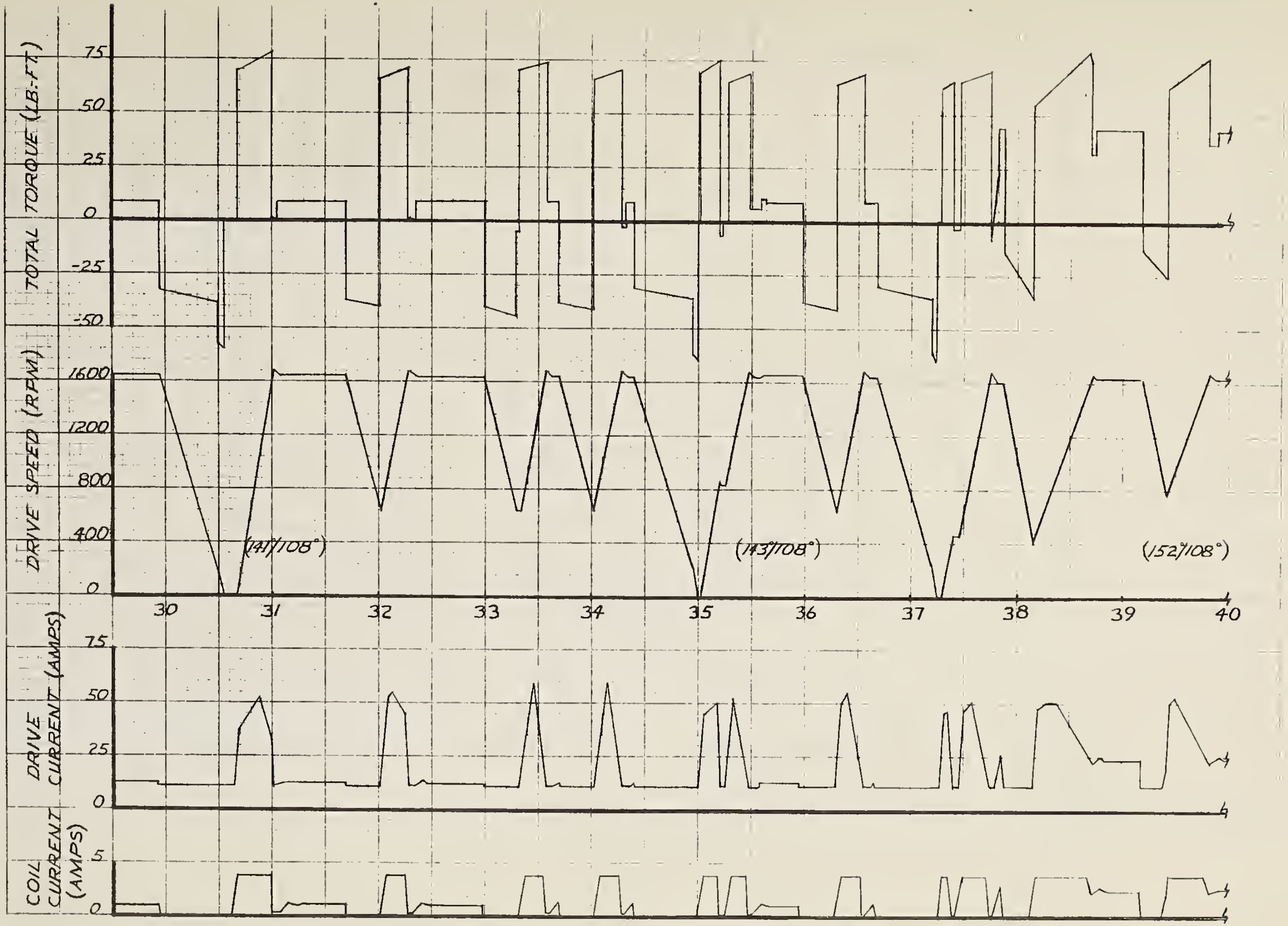


Figure F-2(4) TEST RUN 2 RECORDINGS

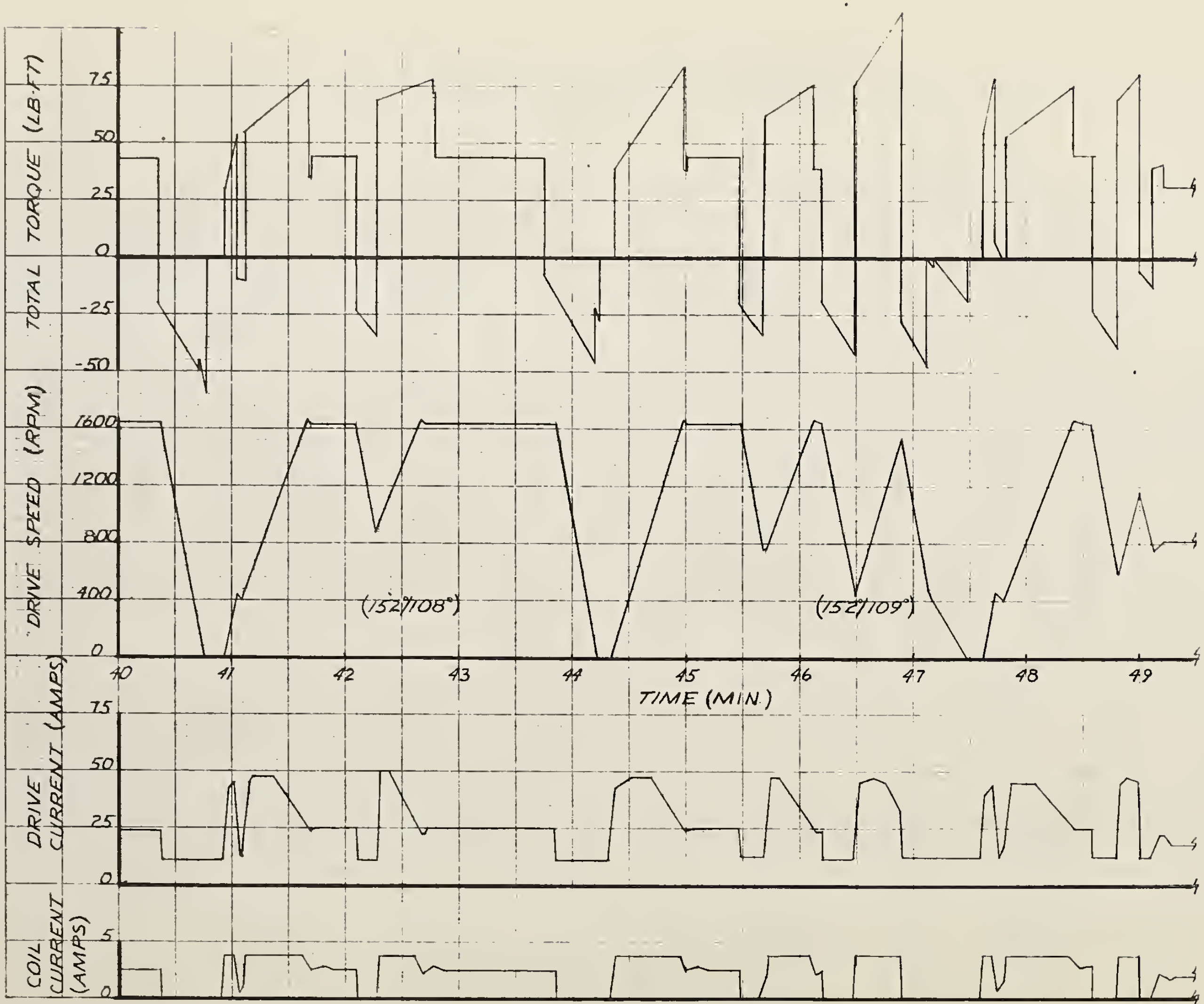


Figure F-2(5) TEST RUN 2 RECORDINGS

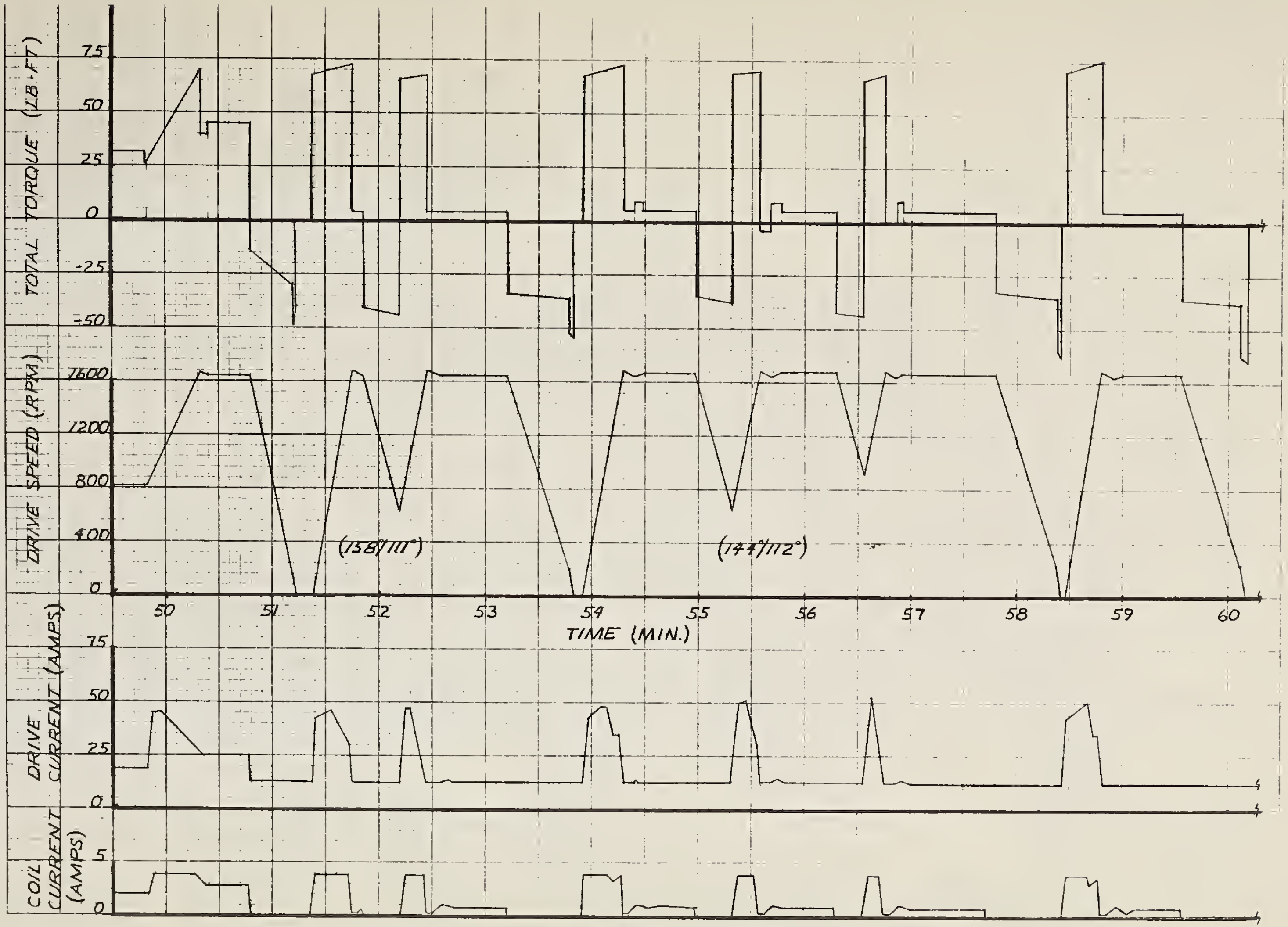


Figure F-2(6) TEST RUN 2 RECORDINGS

APPENDIX G

DRIVE UNIT COMPONENTS

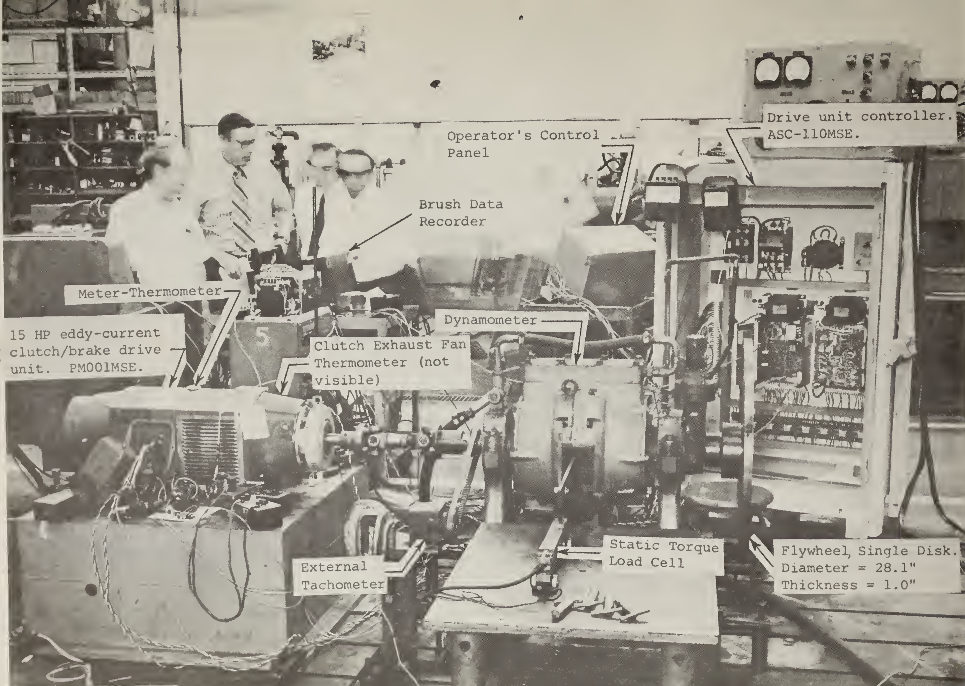


Figure G-1

DYNAMOMETER TESTING OF DRIVE SYSTEM WITH AC EDDY-CURRENT CLUTCH/BRAKE

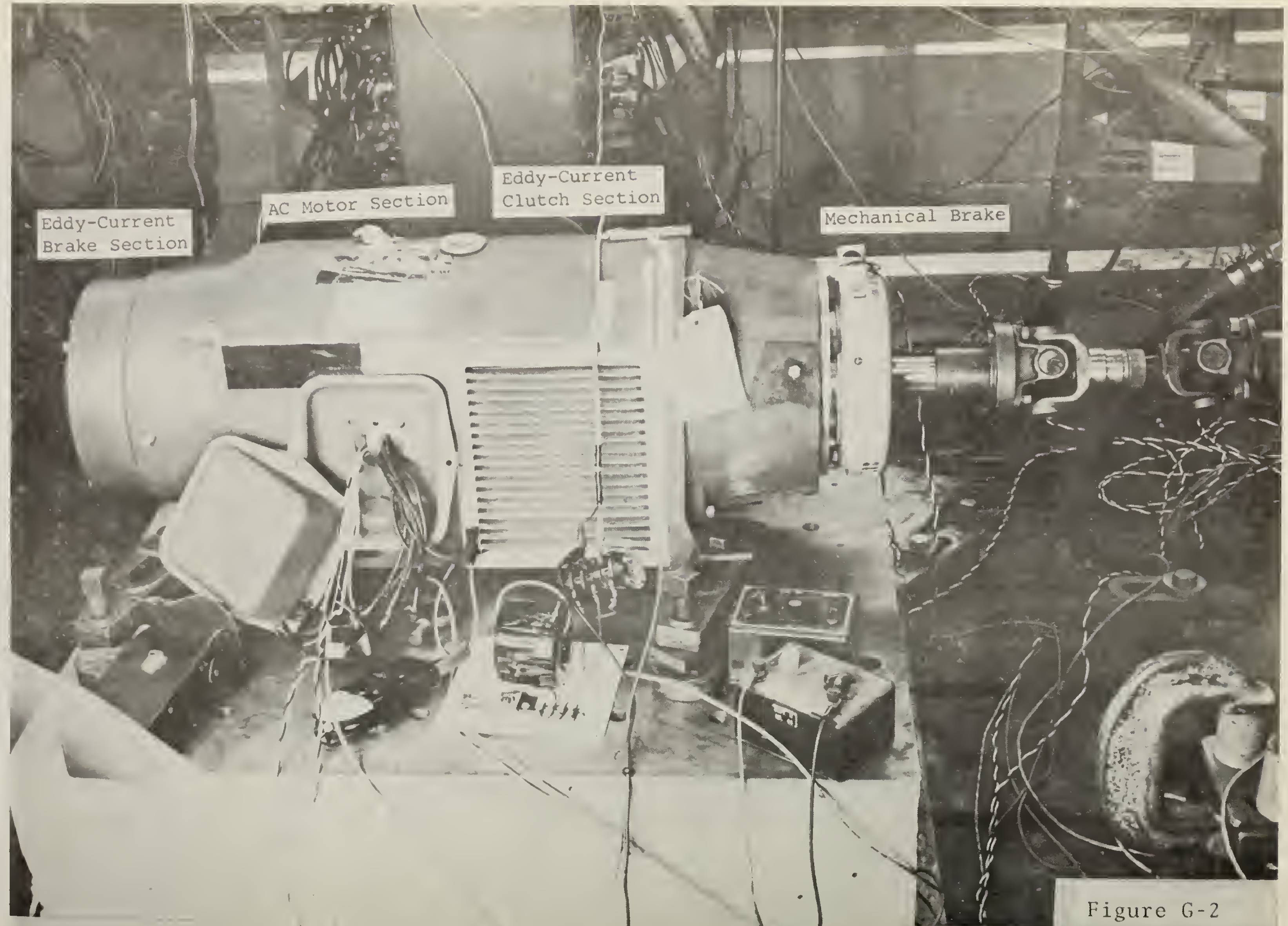


Figure G-2

MOBILITY SYSTEMS
AND
EQUIPMENT COMPANY

15 HORSEPOWER AC EDDY-CURRENT
CLUTCH/BRAKE DRIVE UNIT

CONTRACT NO. DOT-TSC-357