230033

REPORT NO. UMTA-MA-06-0048-75-4 PB256139

MORGANTOWN PERSONAL RAPID TRANSIT LONGITUDINAL CONTROL SYSTEM DESIGN SUMMARY

Boeing Aerospace Company
Automated Transportation Systems
Seattle WA 98124

DECEMBER 1975

FINAL REPORT

AUTOMATED GUIDEWAY TRANSIT TECHNOLOGY PROGRAM

U.S. DEPARTMENT OF TRANSPORTATION
Urban Mass Transportation Administration
Office of Research and Development
Washington DC 20590

Technical Report Documentation Page

1. Report No. UMTA-MA-06-0048-75-4	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle MORGANTOWN PERSONAL RAPID TRANSIT LONGITUDINAL CONTROL SYSTEM DESIGN SUMMARY		5. Report Date December 1975
		6. Performing Organization Code
7. Author(s) Robert P. Lang*		8. Performing Organization Report No. DOT-TSC-UMTA-75-23
9. Performing Organization Name and Address Boeing Aerospace Company Automated Transportation Systems P. O. Box 3999 Seattle WA 98124		10. Work Unit No. (TRAIS) UM533/R6709
		11. Contract or Grant No. DOT-TSC-994
12. Sponsoring Agency Name and Address U.S. Department of Transportation Urban Mass Transportation Administration Office of Research and Development Washington DC 20590		13. Type of Report and Period Covered Final Report April through October 1975
		14. Sponsoring Agency Code

15. Supplementary Notes

*Under Contract to: U.S. Department of Transportation
Transportation Systems Center
Kendall Square, Cambridge MA 02142

16. Abstract

Experience with the longitudinal control system used on each vehicle in the Morgantown Personal Rapid Transit System has shown that nonlinearities and variations in control system parameters can significantly affect performance if such characteristics are not adequately considered in the system design. A design summary is provided that documents this experience and emphasizes the important analysis and hardware design problems encountered. The performance capability of the final design is computed on the basis of analysis and test results. A description of the detailed nonlinear analytical model developed is included for possible use in future studies. Potential system improvements are described that may be the objects of future research and development.

This report was generated in support of the Automated Guideway Transit Technology program of the Office of Research and Development of the Urban Mass Transportation Administration.

17. Key Words Surface Transportation, Automated Vehicle Systems, Personal Rapid Transit, Longitudinal Control, Synchronous Point Follower System, Regulation	18. Distribution Statement Document is available to the U.S. public through the National Technical Information Service, Springfield, Virginia 22161

19. Security Classif. (of this report) UNCLASSIFIED	20. Security Classif. (of this page) UNCLASSIFIED	21.

PREFACE

Numerous studies have been conducted concerning the feasibility of a variety of control schemes for the longitudinal control of automated vehicle systems. This report documents the experience gained in the conversion of the Morgantown operational specifications into realizable and performance predictable longitudinal control system elements.

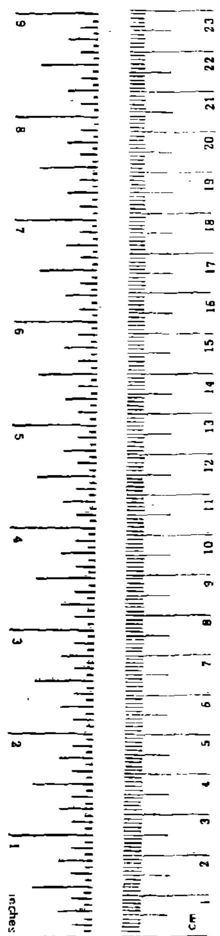
Experience with the Morgantown Personal Rapid Transit System has shown that use of simplified analytical models can lead to erroneous performance predictions. Hardware nonlinearities and variations in control system parameters may have a greater impact on system performance than do external disturbances such as winds and grades if such nonlinearities and parametric variations are not adequately considered in the system design. Since available information on actual automated vehicle systems hardware is limited, a description of the detailed nonlinear analytical model which has been developed for Morgantown is included for possible use in future studies.

The work described in this design summary was performed by the Boeing Aerospace Company for the U.S. Department of Transportation. The design of the Morgantown longitudinal control system includes contributions from a large number of individuals. The author wishes to acknowledge, in particular, the contributions made by Raymond C. Buckner, Raymond E. Hare, Milt A. Moorhead, Tom A. Owan, Curtiss W. Robinson, Dale G. Shellhorn, and George E. Swartz during the Phase IB design, analysis and test effort. The author also wishes to acknowledge the assistance of Robert C. Milnor in the review and editing of this report.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

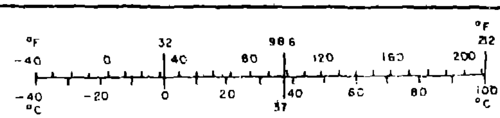


TABLE OF CONTENTS

TABLE OF CONTENTS
(Continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS
(Continued)

LIST OF TABLES

LIST OF TABLES
(Continued)

LIST OF ABBREVIATIONS

C&CS	Control and Communications System
CCCS	Central Control and Communications System
D/A	Digital to Analog
FSK	Frequency Shift Keying
GCCS	Guideway Control and Communications System
LCS	Longitudinal Control System
M-PRT	Morgantown Personal Rapid Transit
PD	Presence Detector
PRT	Personal Rapid Transit
SCCS	Station Control and Communications System
SCR	Silicon Controlled Rectifier
STTF	Surface Transportation Test Facility
VCCS	Vehicle Control and Communications System

1. INTRODUCTION

Numerous studies have been conducted concerning the feasibility of a variety of control schemes for the longitudinal control of automated vehicle systems. The majority of these studies use simplified analytical models of the hardware elements involved and are concerned primarily with system sensitivity to external forces such as wind and grades. Experience with the Morgantown Personal Rapid Transit (M-PRT) system has shown that use of simplified analytical models can lead to erroneous performance predictions. Hardware nonlinearities and variations in control system parameters can have a greater impact on system performance than do external disturbances such as winds and grades if such nonlinearities and parametric variations are not adequately considered in the system design.

This report provides a design summary for the longitudinal control system (LCS) used on each vehicle in the M-PRT system; where the LCS is defined as the vehicle system which converts speed commands from the guideway into the desired vehicle speed - position - time trajectory. The following material documents the experience gained in the conversion of the Morgantown Phase IB LCS operational specifications into realizable and performance predictable LCS elements. Since available information on actual automated vehicle systems hardware and the associated detailed design problems is limited, an emphasis is placed on the major analysis and hardware design problems encountered. This information is provided for possible use in the preparation of realistic system specifications, evaluation of proposed system designs and in conducting meaningful analytical studies.

The Morgantown Project, which began in 1969, is an Urban Mass Transportation Administration demonstration, to provide personal rapid transit (PRT) service between the central business district of Morgantown, West Virginia and the widely separated campuses of West Virginia University. The M-PRT system consists of a fleet of relatively small, automatically controlled vehicles which operate on a dedicated guideway, on a predetermined schedule basis or on a passenger demand self-service basis. The overall project is being built in phases with the first phase consisting of a Phase IA and a Phase IB. Phase IA, completed in September of 1973, resulted in a prototype system consisting of 2.1 miles of guideway, 3 passenger stations, a maintenance and central control facility and 5 test vehicles. Phase IB, which is the primary subject of this report, provides the additional facilities required for public service including a fleet of 45 vehicles. The system, of modular design, allows growth from the present configuration to an expanded configuration which could accommodate 70 to 100 vehicles, up to 6 passenger stations and the associated interconnecting guideway. Figure 1 shows the present guideway configuration and delineates the 3 basic system elements: the Control and Communications System; the Vehicle System; and the Structures and Power Distribution System. The LCS includes hardware elements of both the Control and Communications System and the Vehicle System.

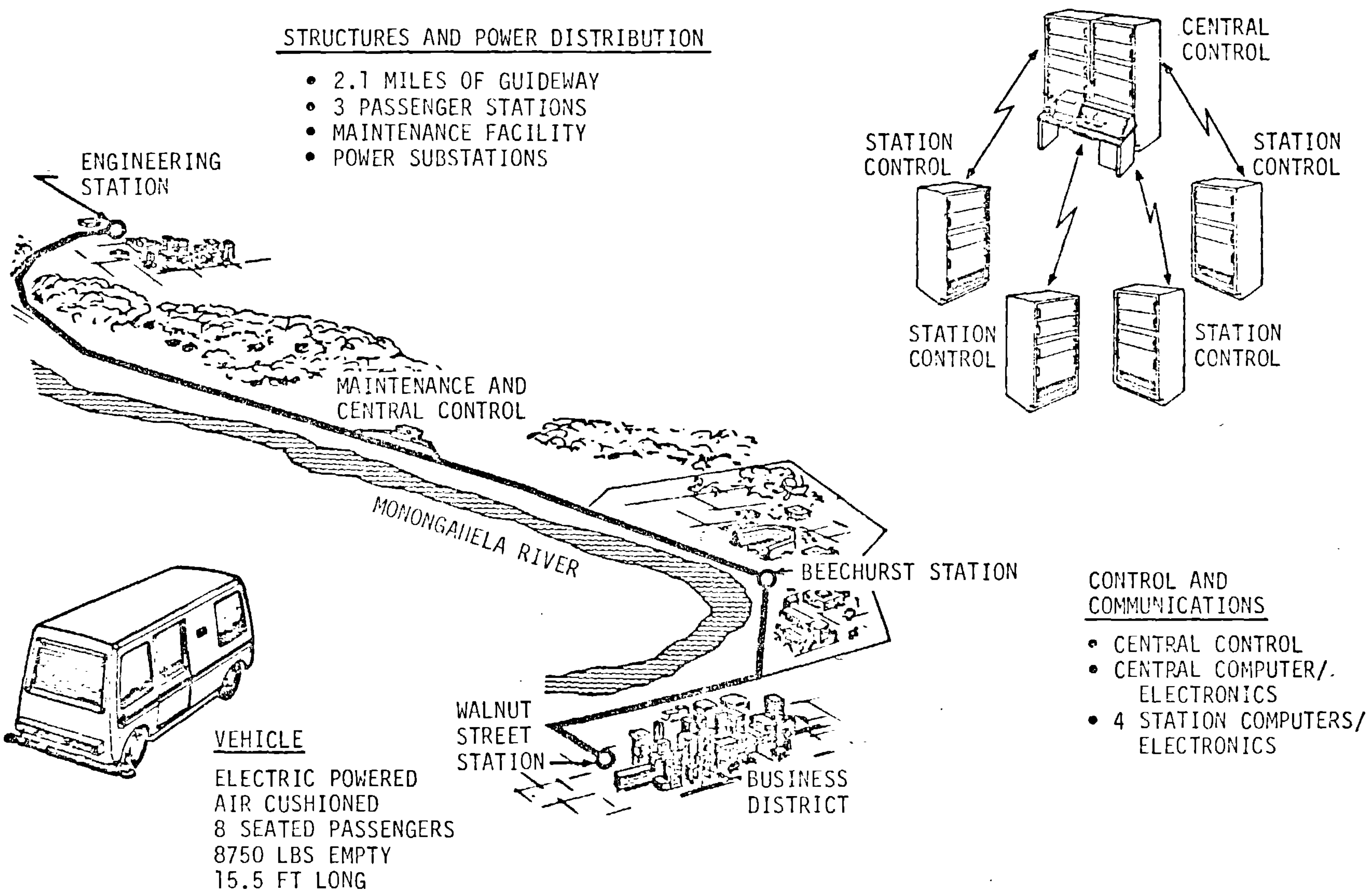


FIGURE 1. M-PRT SYSTEM ELEMENTS

2. GENERAL SYSTEM DESCRIPTION

2.1 Synchronous Point Follower System

The Control and Communications System (C&CS) automatically controls the position of each vehicle by means of a synchronous point follower system. The point follower system conceptually consists of a series of moving points or slots, referenced to a fixed time base, circulating in the C&CS computers. These imaginary or theoretical points, as viewed from a fixed point on the guideway, pass at intervals which are multiples of 15 seconds. The minimum nominal headway between vehicles is 15 seconds. Vehicle headway or position control is accomplished by assigning a vehicle to one of the imaginary points and designing the LCS such that the actual vehicle speed - position - time trajectory matches the trajectory of the theoretical point within prescribed tolerances. The vehicles are physically assigned to a theoretical point by control of their dispatch time. Once dispatched, vehicle control is "open-loop". The LCS generates an on-board point (defined by the physical location of the speed loops in the guideway and the corresponding time at which the vehicle receives a change in speed command) and issues the brake and motor commands required to follow the on-board point.

The function of the wayside computers, following dispatch, is to monitor vehicle performance by means of presence detectors (PD) located along the guideway. The system operator is notified if a vehicle is out-of-tolerance and has the option of stopping vehicles via normal rate braking if an emergency rate braking situation is imminent. No provisions are provided for modification of a vehicle's trajectory following dispatch other than the option to bring the vehicle to a stop. A hardwired check-in/check-out fixed block system, which is independent of the primary control system, is used to provide positive collision avoidance protection. Violation of the minimum safe headway, determined via the fixed block system, results in removal of a safe tone which brings the trailing vehicle to a stop via emergency rate braking.

Analytically, the theoretical point follower position reference is defined as the integral of the acceleration limited civil speed command or theoretical speed reference defined below. Physically, the vehicles receive discrete civil speed commands (4, 8, 22, 33 or 44 fps) from the guideway with changes in speed command occurring at fixed guideway locations. The theoretical speed reference is equal to the civil speed command trajectory except that changes in speed are made at an acceleration/deceleration rate of 2 fps^2. The theoretical time for the start of each speed transition is the time at which a perfect vehicle (one which exactly follows the theoretical speed reference) crosses the guideway location defining the start of a new speed zone. A rigorous analytical definition of the theoretical point follower position reference is given in Appendix A.

The vehicle longitudinal control system (LCS), which has the function of producing the specified speed - position - time trajectory, consists of four major elements: the Vehicle Control and Communications System (VCCS); an electric propulsion system; a hydraulic friction brake system; and the vehicle itself. A simplified block diagram of the LCS is given in Figure 2 which shows the basic components of each major element and the functional interfaces between components.

2.2 Vehicle Control and Communications System

The VCCS is that portion of the C&CS carried on-board the vehicle. It responds to guideway and vehicle inputs and controls vehicle doors and switching as well as controlling vehicle speed. A summary of the VCCS input/output characteristics is given in Figure 3; i.e., the functional interface between the VCCS, the Station Control and Communications System (SCCS) or the Guideway Control and Communications System (GCCS), and the vehicle.

Inductive coupling of electrical energy is the method used to transmit information between the S/GCCS guideway loops and the VCCS. Communications transmitted from the guideway to the VCCS consists of tones, to command

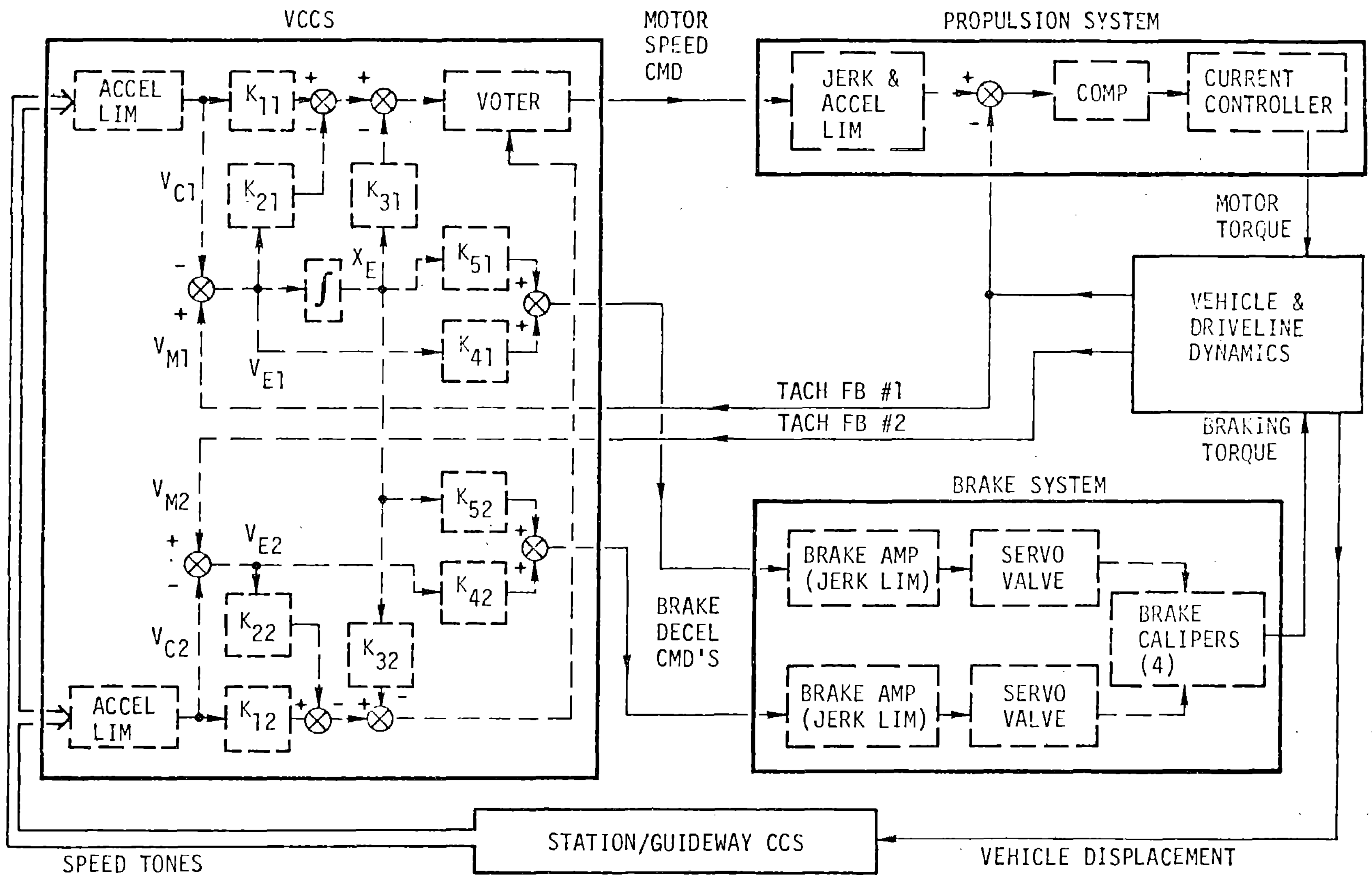
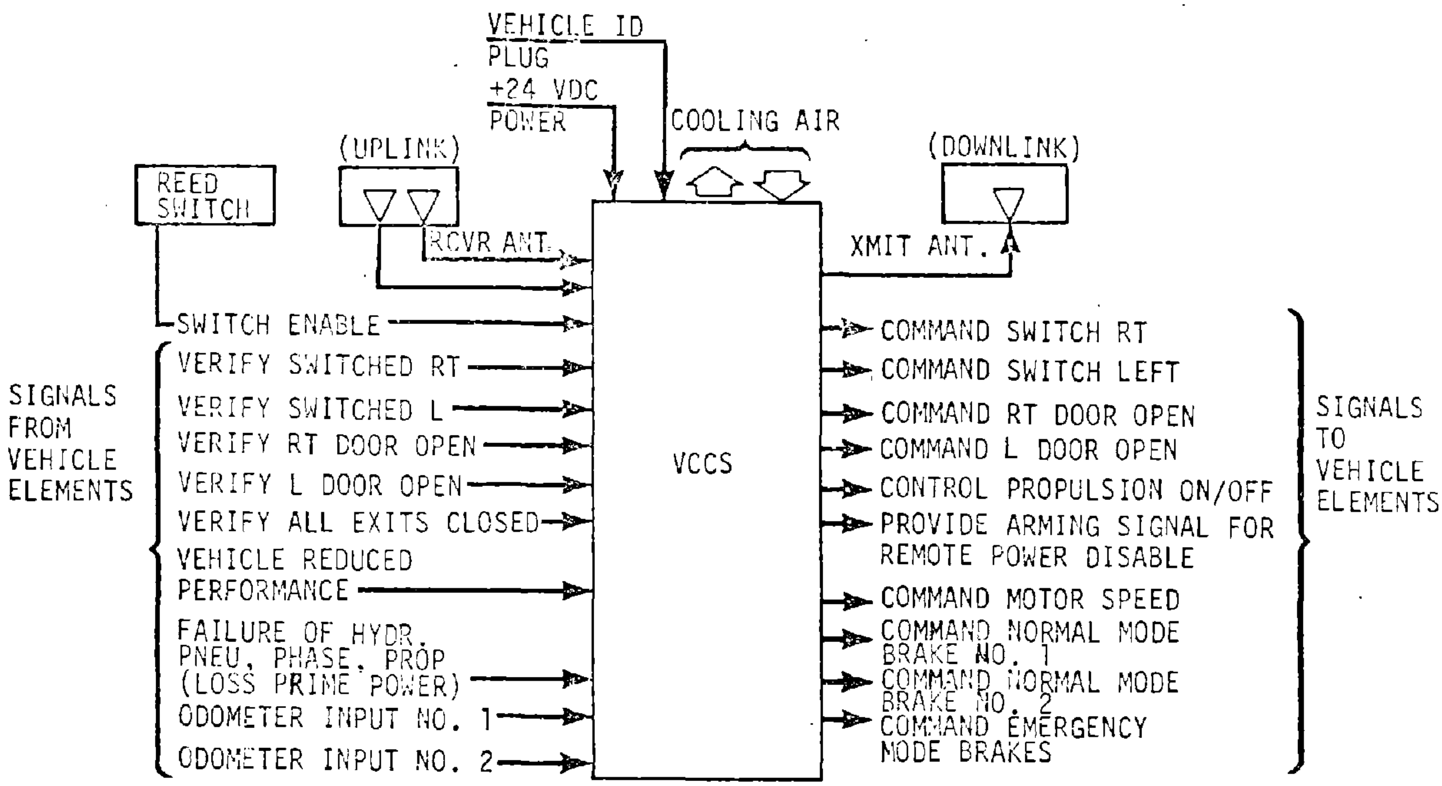


FIGURE 2. M-PRT LONGITUDINAL CONTROL SYSTEM



UPLINK

- SPEED TONES
- SAFE TONE
- SWITCH TONE
- STA STOP/CAL TONE
- FSK
  DOOR COMMANDS
  PERFORMANCE LEVELS
  ID REQUEST
  EMER BRAKE

DOWNLINK

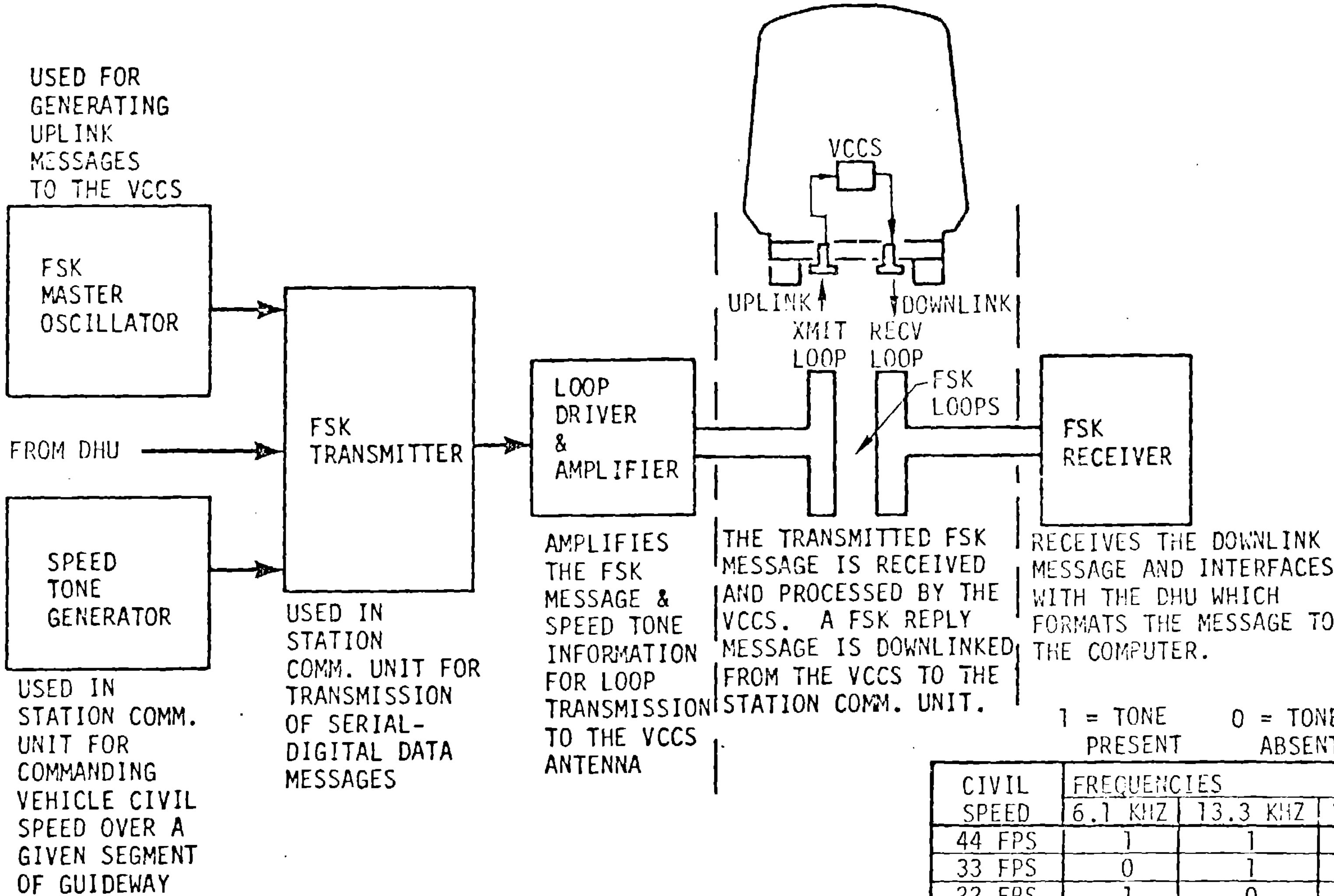
- SWITCH VERIFY
- FSK
  - VEHICLE ID
  - DOOR STATUS
  - FAULT STATUS

FIGURE 3. VCCS INPUT/OUTPUT CHARACTERISTICS

specific functions to all VCCS units on a non-selective basis, a Frequency Shift Keying (FSK) digital message to interrogate or command a VCCS function on a vehicle selective basis, and magnets which provide switch enable commands via vehicle mounted reed switches. Communications from the VCCS to the guideway consists of an FSK message, reporting vehicle status and switching tones verifying the response of the vehicle to a switching command. Transmission of speed commands, the major LCS interface between the VCCS and guideway is illustrated in Figure 4.

The major longitudinal control function of the VCCS is to generate the brake and motor commands required to produce the specified speed - position - time vehicle trajectory. This is accomplished by first computing an acceleration-limited speed command and by measuring speed and position error signals using digital circuitry to achieve the required accuracy. Analog circuitry is then used to generate the required differential 0 to 10 vdc analog brake and motor commands according to the control law shown in Figure 2. Two separate channels, driven by redundant antennas and tachometers are used to maximize safety. The redundant motor commands are compared and the lowest (safest) is sent to the propulsion system. The redundant brake commands are both sent to the brake system where the brake calipers ultimately vote the highest (safest) command. An exception is the position error computation circuitry which is single thread. Redundant position error circuits are not used because of the complexity of the circuit and the fact that a full scale position error failure does not constitute a safety hazard.

Measured speed and position error computations are based on inputs from two redundant tachometers which are located on the motor shaft and are part of the propulsion system. The tachometers, which can also be considered as odometers, are photoelectric digital devices which generate 76 pulses per motor revolution or, nominally, one pulse per 0.166 inches of vehicle travel. The incoming pulse train is converted, within the VCCS, to a calibrated pulse train with a scaling of 0.25 inches/pulse to compensate for variations in tire rolling radius. Required corrections in tachometer scale factor are calculated by the VCCS



1 = TONE PRESENT   0 = TONE ABSENT

CIVIL SPEED	FREQUENCIES 6.1 KHZ	13.3 KHZ	17.2 KHZ
44 FPS	1	1	0
33 FPS	0	1	1
22 FPS	1	0	1
8 FPS	1	0	0
6 FPS	0	1	0
4 FPS	0	0	1
4 FPS	0	0	0
4 FPS	1	1	1

FAILURE MODES ONLY INVALID COMBINATIONS COMMAND 4 FPS

FIGURE 4.
FSK/SPEED TONE LOOP CONTROL

from inputs provided by calibration tone loops which are 200 feet in length. Calibration loop spacing, where the distance between loops is specified in multiples of 100 feet, is used to compute automatic periodic updates in the on-board measure of position error.

Vehicle speed, which is proportional to tachometer pulse rate, is computed by counting the number of calibrated tachometer pulses during 0.1 second intervals. Position error is computed by feeding both the calibrated tachometer pulse train and a reference pulse train (having a pulse rate proportional to the acceleration limited speed command) to an up/down counter. The output or net difference in pulses from the two sources is proportional to position error or, functionally, the integral of speed error. Position error is computed directly, rather than as the difference between a commanded and measured total displacement, because of the round-off errors inherent in the later approach.

A special purpose speed versus position command profile is used during station stop sequences. Upon detection of a stop tone (in a 4 fps civil speed zone), the VCCS switches from its normal acceleration limited civil speed command reference to the speed command from the station stop profiler. The profiler reduces its output speed command from 4 to 0 fps solely on the basis of distance traveled. Because of the loop closure on position, the speed command and the actual speed reach zero at a precise point on the guideway, rather than at a specified point in time, which minimizes errors in final vehicle position.

Emergency stops are performed open-loop. The VCCS, in response to loss of a safe tone or in response to specified on-board anomalies, disables the propulsion system and issues a fail safe emergency brake command (removes 28 vdc) to the brake amplifiers. The brake amplifiers respond by profiling the braking force commands to their nominal full scale value of 3737 lbf at a rate of 3291 lbf/sec. The result is a deceleration rate of 0.3 g for a maximum weight vehicle with a 30 mph tailwind on a 0% grade. Because of the open-loop or constant brake force control, actual deceleration rates vary with changes in vehicle weight, grade, wind and brake system parameters.

The VCCS also has the capability, in response to anomalies not requiring an emergency rate stop, to stop the vehicle at the normal rate of 2 fps^2. This is accomplished by setting a performance level to zero. A zero performance level sets the velocity command input to the acceleration limiter to zero. The limiter responds by profiling the speed command reference to zero at the normal rate of 2 fps^2 and the vehicle is brought to a stop via the normal closed loop controller described in Figure 2.

2.3 Propulsion System

The electric propulsion system, which generates the torque required to maintain speed or accelerate the vehicle, consists of a DC drive motor, a motor controller, 2 redundant tachometers and a transformer. A simplified functional diagram, Figure 5, illustrates the functions of each element. The propulsion motor is a compound-wound DC motor rated at 70 hp at 2720 rpm with 420 volts on the armature and 12.3 amps on the shunt field. Field weakening is used to maintain a constant 70 hp from 2720 rpm up to the limit of 3168 rpm. The motor drives the rear wheels through a conventional differential having a gear ratio of 7.17:1. Included in the motor controller are a three-phase full wave Silicon Controlled Rectifier (SCR) AC/DC converter, current control circuits, speed control circuits, a jerk and acceleration limiter and a tachometer digital to analog converter. The tachometer drive units consist of motor shaft-mounted discs that spin through an optical transducer. The signals are conditioned in the tachometer enclosure and are fed to both the VCCS and motor controller. The transformer converts incoming 575 vac, 3 phase power to 355 vac, 61 vac and 120 vac voltage levels.

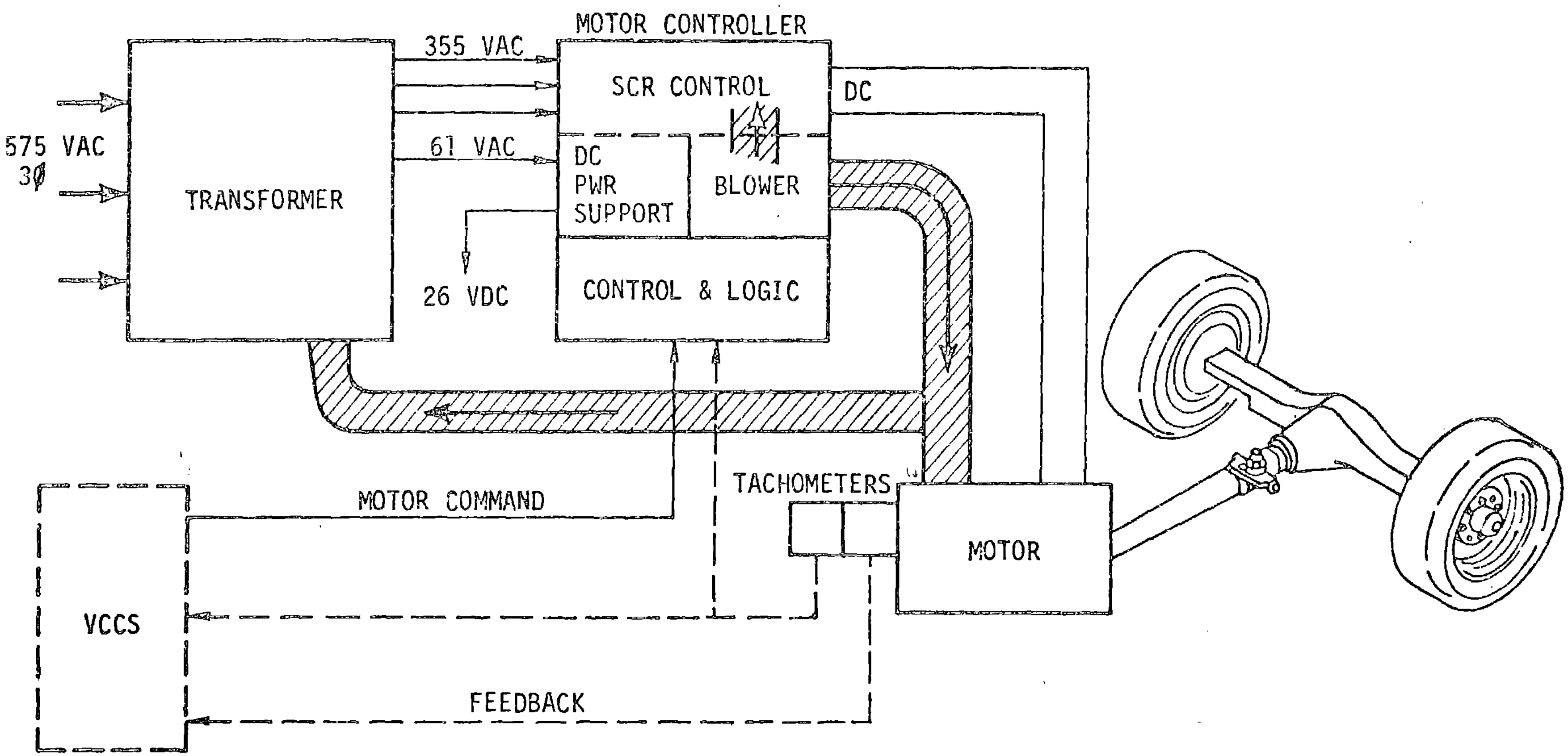


FIGURE 5. PROPULSION SYSTEM FUNCTIONAL DIAGRAM

The longitudinal control functions of the motor controller are illustrated in both Figures 2 and 5. The propulsion system receives a differential 0 to 10 vdc speed command from the VCCS having a scaling of 316.8 rpm/volt. The incoming command is first passed through a jerk and acceleration limiter with a jerk limit of 3.09 rev/sec^3 (0.1 g/sec) and an acceleration limit of 3.97 rev/sec^2 (0.125 g). The main function of the limiter is to guard against subjecting passengers to excessive jerks. The acceleration limit of 0.125 g is not reached during normal operation since changes in speed command occur at a nominal rate of 0.0625 g (2 fps^2). The output of the limiter becomes the command for an analog speed control loop. Speed feedback is obtained by passing one of the two redundant digital tachometer outputs through a D/A converter. Integral compensation provides the required steady-state speed control accuracy of $\pm$ 13.2 rpm. Dynamically, for a nominal weight vehicle, the speed loop has a natural frequency of 1.34 rad/sec (0.213 Hz) with a damping ratio of 1.0.

An analog current control loop provides the commands to the SCR's and regulates motor current (torque) as called for by the speed controller. Commutator arcing is prevented by limiting the current command rate to a maximum of 1400 amps/second. An adjustable upper current command limit of approximately 400 amps is used to prevent excessive motor currents and potential motor damage. An adjustable lower limit on current command of approximately 10 amps is used to maintain a bias current level at all times, thereby preloading the drive and minimizing time delays and the effects of backlash in the driveline. The nominal gain from current to torque is 0.917 ft lb/amp.

The main 28 vdc power supply and battery charger for the vehicle is located in the motor controller cabinet. This section furnishes all of the DC loads of the vehicle including the motor control circuits, and is on whenever power is applied to the vehicle. The propulsion system, in addition to the speed command, receives an on/off command from the VCCS. An "off" command is issued whenever the vehicle is stopped or is in an emergency stop sequence. The propulsion system responds to an "off" command by removing AC power from the vehicle (which maximizes passenger safety) and cages or initializes its control circuits. Motor control circuits remain on during these periods.

2.4 Brake System

The vehicle brake system, which generates the braking torque required to decelerate or stop the vehicle, is a dual system, either one of which can stop the vehicle safely. As shown in Figure 2, the major components consist of two brake amplifiers, two servo valves and four brake calipers (one per wheel). The system is redundant and independent up to the brake pads. The brakes are discs on all four wheels, with a single caliper and rotor at each wheel. A detailed schematic of the complete brake system is given in Figure 6.

Redundant braking signals come from the VCCS to the brake amplifiers. The brake amplifiers command the servo valves (hydraulic pressure regulators) to respond, and the servo valves apply the proper pressure (20 to 900 psig) to the calipers. There are two braking modes: normal and emergency. In the normal mode, the VCCS provides a differential 0 to 10 vdc analog signal to the brake amplifier and the servo valve responds with 20 to 700 psig. The nominal normal mode deceleration is 2 fps^2 (0.0625 g) with the brake system providing a brake force capability in excess of 0.2 g to compensate for grades and controller lags. The emergency mode is created by an absence of a 28 vdc signal to the brake amplifier which causes the servo valve to release up to 900 psig to the calipers and results in a nominal emergency rate deceleration of 0.3 g.

The brake amplifiers perform a number of control functions in addition to their prime function of converting from a voltage command to the current command required by the servo valves. These functions include:

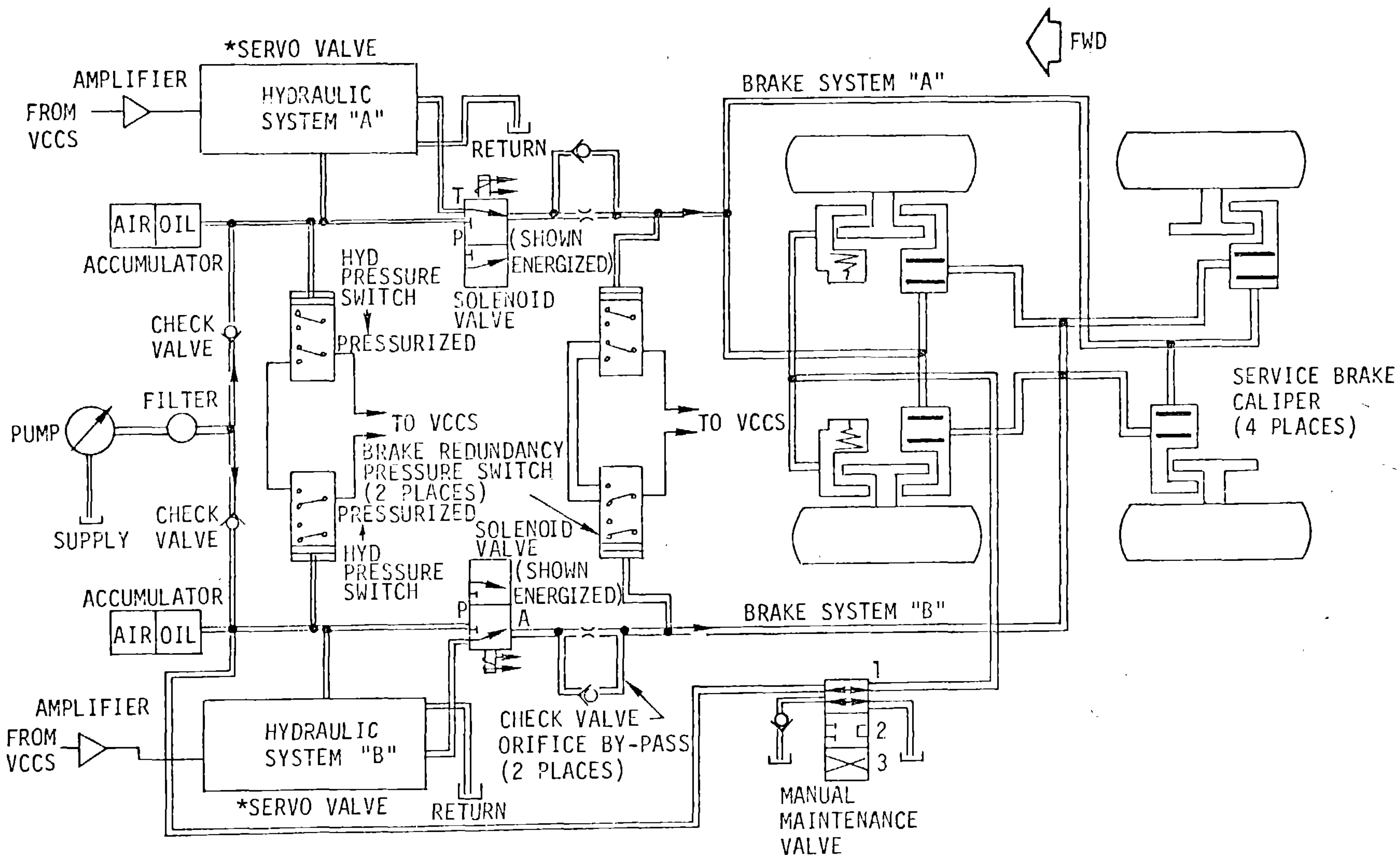


FIGURE 6. BRAKE SYSTEM SCHEMATIC

- Jerk limiting of the VCCS deceleration command to prevent excessive vehicle jerks and to provide accurate off time control during a startup sequence.

- Limiting of the minimum output to an adjustable bias level to minimize brake time delays.

- Shaping of the output command near null to optimize the tradeoff between brake drag and brake time delays.

- Limiting of the maximum output to specified normal and emergency values to prevent excessive deceleration levels.

The calipers contain tandem piston actuators with independent hydraulic actuation. Either piston in the caliper assembly is able to actuate the brakes at full capacity; but, when both pistons are actuated, which is normal, the braking results are not additive. Functionally, the tandem pistons perform the voting function for the redundant system, with braking torque being proportional to the highest (safest) of the two input pressures. Retractor springs are used to prevent brake drag below a specified pressure threshold. The brake pads, two with each caliper, and the brake rotors are of standard automotive design. The caliper design includes an automatic adjustment feature which compensates for brake pad wear and minimizes on-time delays.

Two orifices and bypass check valves are used to limit hydraulic fluid flow rates and minimize coupling between the hydraulic systems. These orifices limit flow and allow the servo valves to maintain the desired pressure during periods of floating (voting) caliper piston motion caused by a change in the dominant pressure.

Brake energy and control are provided by the hydraulic and the electrical systems respectively. In the absence of either or both, hydraulic energy is provided from the accumulators and energy for control is provided from the batteries. In an extreme case, when loss of power and failure of the batteries might occur, a special emergency braking system is activated by two solenoid valves in the system, which open upon absence of DC voltage, by-pass the servo valves, and dump all the energy in the accumulators directly into the brake calipers.

Two hydraulic pressure switches continuously monitor pressure in the accumulators and issue a fault signal to the VCCS if either pressure falls below a specified level. A second pair of pressure switches monitor the servo valve output control pressures and report a loss of brake redundancy to the VCCS if the control pressures differ by more than a specified tolerance.

Independent parking brake calipers are mounted on the front wheels and are spring loaded assemblies which are held off by hydraulic pressure. In the event hydraulic pressure decays to an unsafe level, the parking brakes automatically come on and provide a fail safe backup to the primary system. The parking brakes also serve to hold the vehicle in place during storage in a power-off condition.

2.5 Vehicle

The controlled vehicle is the final element of the longitudinal control system. The M-PRT vehicle, shown in Figure 7, is relatively small carrying up to 21 passengers - 8 seated and 13 standing. The vehicle size has been selected to provide economical service during both peak and low demand periods. Key physical characteristics are:

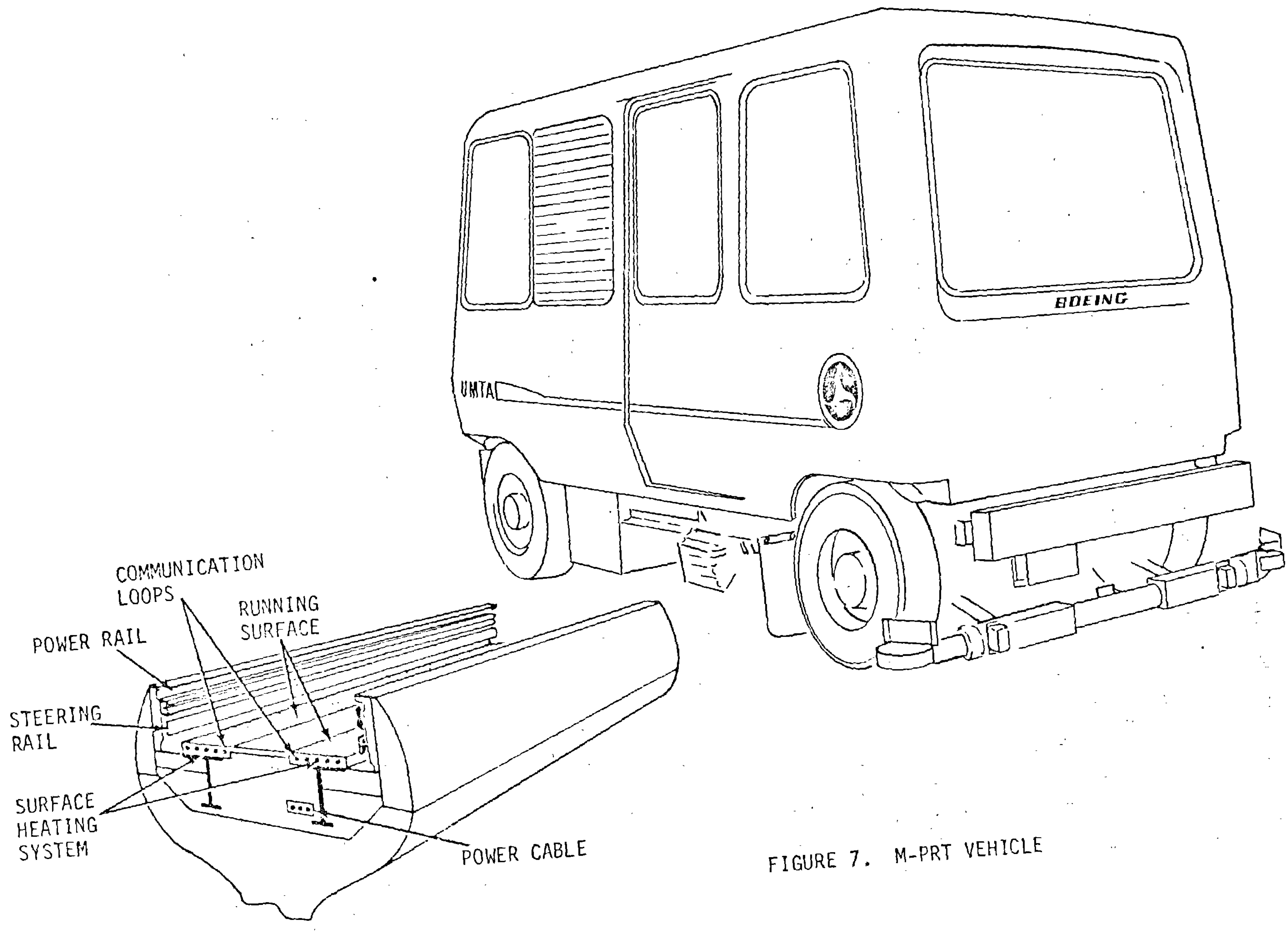


FIGURE 7. M-PRT VEHICLE

Length	15 ft. 6 in.
Height	8 ft. 9 in.
Width	6 ft. 8 in.
Weight	3750 lbm empty to 11,900 lbm maximum
Wheel Base	127 in.
Tread Width	62 in.

Conventional bias-ply rubber tires and an air spring suspension are used to provide a quiet and comfortable ride. Four-wheel steering is used, with the direction of the wheels controlled by a hydraulically-boosted mechanical linkage using guide wheels at either end of a steering arm. Steering guide wheels follow rails mounted on the side of the guideway. The system is biased to use either the left or right steering rail, depending on the desired route. The steering system provides the capability of down to a 30 foot turning radius. The propulsion system provides the capability for speeds up to 30 mph.

Figure 7 also shows a cross section of a typical section of guideway. Approximately 65% of the guideway is elevated, the remainder being at ground level. Both single and double lane guideway is used. The running surface is concrete containing distribution piping for guideway heating to allow all-weather operation. Inductive communication loops are also installed in the running surface. Steering and electrical power rails are mounted vertically along the side of the guideway. From a longitudinal control standpoint, the key parameter is a maximum grade value of $\pm$ 10%.

2.6 Overall Vehicle Control Task

This section illustrates operation of the longitudinal control system in the context of the overall vehicle control task. A general description is given of the sequence of events which are required for automatic vehicle control from dispatch to the vehicle's arrival at its destination. The control functions are performed by a combination of the Central Control and Communication System (CCCS) which is responsible for overall system scheduling, monitoring and fault reaction, four Station Control and Communication Systems (SCCS) which handle all vehicle operations within a localized area, and four Guideway Control and Communication Systems (GCCS). The latter act only to relay passively commands and signals between the VCCS and SCCS and will not be discussed further.

After the passengers have boarded and the allotted vehicle door open time has expired, the door is automatically closed and the vehicle is ready for dispatch. The SCCS requests a dispatch time from CCCS in the demand mode, or determines if the scheduled dispatch time can be met in the scheduled mode. The dispatch time is determined so that a vehicle following the point follower profile for that station and starting position will merge on the guideway with its assigned moving slot position. The SCCS clocks are synchronized with the CCCS clock so that the system operates relative to a common time standard. Dispatch is accomplished by removal of the stop tone from the stopping communication loop at the specified time. If a scheduled dispatch time cannot be met, a new time allocation is requested from CCCS.

The vehicle accelerates to 8 fps and proceeds at this speed to the acceleration ramp. Steering switching commands direct the vehicle from the platform channel to the acceleration ramp. On the ramp, the vehicle accelerates at 2 fps^2 until the main guideway speed of 22 or 33 fps is reached. The vehicle steers right on the acceleration ramp past the merge point on the main guideway and then is commanded to steer left.

The SCCS monitors the dispatched vehicle's movement via presence detector (PD) data to assure that guideway speed is reached and that the vehicle has followed the point follower control law. If the PD hit times are within tolerance the vehicle is committed to the main guideway. The collision avoidance systems on the acceleration ramp and on the appropriate section of main guideway are interlocked so that out-of-tolerance vehicles will initiate emergency braking.

Vehicle progress on the guideway is monitored by the SCCS via PD hit times. The SCCS also monitors vehicle status. A vehicle downlink status report includes: vehicle identification; current location; current destination and switch condition; current performance level; current civil speed command; door status; brake system status; and any current anomaly. Status data are periodically transmitted to CCCS for overall system monitoring and for control of handover from station to station.

Civil speed is 22, 33 or 44 fps on different sections of the main guideway. A speed change is commanded by the step change in speed tone occurring at two adjacent speed tone communication loops. This step change is detected by the vehicle VCCS which commands a speed transition at the point follower control law rate of 2 fps^2.

Responsibility for detailed vehicle management is transferred from one SCCS to the next at a designated guideway PD. CCCS informs the receiving SCCS of the enroute vehicle identification, destination, status, and assigned point follower slot. When the vehicle arrives at the PD, the receiving SCCS takes over vehicle control and fault report monitoring tasks.

As the vehicle approaches each enroute station, it is interrogated for its identification. At the destination station the identification is recognized and the availability of an open unloading berth is checked. If no space is available at a berth the vehicle is stopped on the ramp until a space opens. If no space is available at a ramp the station is bypassed and the central operator is notified to take appropriate action. Under normal conditions an unloading berth will be available and a switching command is sent to exit the vehicle from the main guideway. Switch verification is sent to the SCCS from the vehicle. Failure to receive switching verification results in a stop command from the SCCS.

Routing of an incoming vehicle to an unloading berth is based on: channel assignment and station fill policy, and the availability of an open berth. The routing logic decisions are implemented at station branch points by steering commands which direct the vehicle into the proper channel. Normal vehicle speed during channel switching is 8 fps. After the switching region is cleared, the vehicle is decelerated to 4 fps from which a vehicle will initiate a station stop sequence. The SCCS commands a station stop by energizing the stopping loop at the channel location at which the vehicle is scheduled to unload.

In unloading positions the door is commanded open for a preselected time to allow passengers to depart. The door is then automatically closed and the vehicle is commanded to "move up" to the forward position in the channel (loading position) and open its door (in the scheduled mode) or wait for a destination request (in the demand mode). The first empty car in a station channel may be sent to another station to meet demands if not required at the current station. During the scheduled mode, vehicles are commanded to have station dwell times sufficient to unload, move up, and load to meet their scheduled departure.

3. PHASE IB DESIGN TASK AND REQUIREMENTS

This section describes the key requirements on the longitudinal control system and the resulting design, analysis and development test program undertaken to meet these requirements. A major goal of the design effort was to improve on both the performance and reliability of the prototype Phase IA system design. The design effort was performed in conjunction with changes in VCCS, propulsion and brake system suppliers.

3.1 Requirements

The key LCS requirements are as follows:

- Station stop accuracy; $\pm$ 6 inches.
- Regulation (position control); $\pm$ 1.1 seconds.
- Speed control; + 3, -4 fps.
- Acceleration control during speed transitions; $2.0\,^{+2.4}_{-0.5}$ fps^2.
- Jerk control

  For time intervals equal to or greater than 0.2 seconds; $\pm$ 4.025 fps^3 ($\pm$ 0.125 g/sec).

  Over any 0.1 second time interval; $\pm$ 8.05 fps^3 ($\pm$ 0.25 g/sec).
- Maximum brake drag; 36 ft. lb. of wheel torque.

The major LCS requirement is a vehicle regulation or position control requirement. The specific requirement is to maintain vehicle position within $\pm$ 1.1 seconds of the moving point defined by the nominal vehicle trajectory. This requirement is derived from system level requirements to be able to run vehicles at nominal intervals or headways of 15 seconds and to be able to safely stop a vehicle which encounters another stopped vehicle on the guideway. These system requirements are met by an allocation of the 15 seconds into requirements on vehicle regulation, safe stopping distance and collision avoidance system block size. The point follower control law and regulation requirement are terminated at the beginning of a stopping sequence in the stations.

Vehicle speed errors are constrained by an LCS requirement to maintain vehicle speed within +3 fps, -4 fps of the acceleration limited civil speed command reference. A major factor leading to the 3 fps overspeed limit is the safe stopping distance requirement since stopping distance capability is strongly dependent on initial vehicle speed.

The acceleration and jerk control limits given are derived from passenger ride comfort considerations. All acceleration and deceleration ramps are designed on the basis of nominal acceleration/deceleration and jerk levels of 2.0 fps^2 and 3.22 fps^3, respectively.

The brake drag requirement is imposed by the Vehicle System for the purpose of preventing overheating of the motor and the possibility of brake fade due to excessive heating.

All of the above requirements (except for nominal values) are interpreted as 3σ limits on system performance capability.

3.2 Phase IB Design Task

The following paragraphs provide a summary of the overall Phase IB design task. This summary is in the form of a chronological description of the major tasks performed. They included several iterations of the basic development steps of simulation, analysis, design, fabrication and test.

The first step in the design effort was the development of a detailed non-linear analytical model and simulation of the final prototype Phase IA design used as an initial baseline. A preliminary design simulation developed early in the Phase IA program served as the starting point. Performance deficiencies not predicted by Phase IA analyses led to the initial requirement for a more detailed model.

A period of extensive testing of Phase IA vehicles followed, with tests conducted both at Morgantown and in Seattle at the Surface Transportation Test Facility (STTF). Time history data for all key LCS variables were recorded with the specific objective of determining the cause or causes of excessive station stop final position errors. A secondary objective was to verify the accuracy of the analytical model via comparison of simulation results against actual test data.

Test data results led to a station stop analysis and trade study effort ending in a basic change in the station stop control law. The test results also initiated a brake system servo valve analysis and test effort ending in a change in servo valve suppliers.

The foregoing simulation development, testing and analysis provided the initial baseline design for Phase IB and the analysis tool needed for sensitivity studies. The next step was to conduct a detailed analysis of the design. Position, speed, acceleration and jerk control sensitivity studies were performed to determine the subsystem and component requirements needed to insure compliance with LCS requirements. The results of these studies led to several minor changes in the initial baseline design and the formal requirements on the Phase IB hardware.

The sensitivity studies were followed by a second simulation development effort to account for minor differences in hardware characteristics resulting from changes in VCCS, propulsion system and brake caliper suppliers.

Initial design verification was accomplished by closed-loop testing of all non-moving portions of the first Phase IB hardware units. Loop closure was achieved via an analog computer simulation of the vehicle and motor dynamics. Test results were in good agreement with analysis predictions.

Preliminary testing of the first complete Phase IB vehicle at STTF uncovered two major problems not discovered in the initial design verification integration testing. The first of these problems was a propulsion speed controller instability. The second problem was a combination of brake caliper performance deficiencies and a servo valve flow capability problem.

The propulsion instability proved to be a classical linear flexible-body stability problem involving the driveline dynamics. The problem had not been predicted by simulation studies because of inaccuracies in the model in the frequency range of interest. An update of model parameters, based on the results of detailed testing, led to simulation instabilities of the type observed in testing. Subsequent analyses resulted in a solution consisting of a phase compensation network and changes in several motor controller parameters.

Test data showed the major brake problem to be an inability of the servo valve to maintain commanded pressure under some transient conditions due to a limited hydraulic fluid flow capability of the servo valve. An analysis and design trade study effort was conducted leading to installation of two orifices and bypass check valves which externally limit fluid flow rates to within a range where the servo valve can maintain adequate pressure regulation. A secondary brake problem was caliper gain nonlinearities and differences in gain between the redundant channels. This problem was solved by revising the brake system analyses defining the brake amplifier settings and using existing brake amplifier adjustment capabilities to compensate for known caliper nonlinearities.

Final design verification tests were then run at STTF with an extensive instrumentation package recording time history data for all key LCS variables. Test data analyses showed performance was within required limits in all areas, except for excessive short term jerk levels, which did not result in a noticeable ride comfort problem.

Final testing at Morgantown was directed primarily at vehicle regulation. The objectives were to establish a data base for monitoring vehicle performance and to verify that regulation errors were within the $\pm$ 1.1 second requirement. A data base representing nominal vehicle performance rather than the theoretical point follower control law is used in the operational C&CS system as a reference for computing regulation errors. The result of using a nominal vehicle trajectory in computing regulation errors is to provide increased design margins by removing systematic or bias errors which do not affect headway control capability.

4. ANALYSIS AND TEST RESULTS

A major finding in both the Phase IA and Phase IB LCS design efforts is that hardware nonlinearities and variations in control system parameter values have a significant impact on system performance. This section describes the results of the analyses and tests conducted in Phase IB with an emphasis on: the key nonlinearities and parameter variations affecting system performance; the major problems encountered along with their solutions; and the estimated performance capability of the final design. The material is organized under the following headings:

- Regulation (Position Control)
- Station Stop
- Station Start
- Brake Amplifier Settings
- Speed Control (Overspeed/Underspeed)
- Acceleration and Jerk Control
- Propulsion Stability Studies.

The analyses described were conducted at various times during the design cycle and in some cases by different analysts. As a result, minor differences in assumed parameter values and the analytical models used occur. Unless otherwise noted, these differences do not have a significant impact on the results presented.

4.1 Regulation (Position Control)

The regulation requirement is to maintain vehicle position within ± 1.1 seconds of the moving point defined by the speed-position-time trajectory for a nominal vehicle. Use of a nominal trajectory as a reference for computing regulation error, rather than the theoretical point follower control law defined in Appendix A, removes systematic or bias errors, which do not affect system operation, from the error budget. This change from Phase IA is made possible by use of a data base (which corresponds to a nominal vehicle trajectory) for monitoring vehicle performance in the operational system. The results of the Phase IB analyses are summarized in Table 1 which shows the error sources involved and the allocation of the 1.1 seconds between error sources.

TABLE 1. REGULATION ERROR BUDGET

ERROR SOURCE CATEGORY	ERROR SOURCE	ERROR ALLOCATION (SEC)
LCS Servo Loop	Steady-state position error	± 0.43
	Vehicle dynamic response	± 0.60
	Stopping position	± 0.125
VCCS	Odometer calibration	± 0.71
	Duration of illegal speed command	± 0.25
	VCCS clock accuracy	± 0.08
Wayside controller	Software dispatch tolerance	± 0.20
	Central clock accuracy	± 0.28

RSS Total = ±1.1 seconds

The error source category termed LCS servo loop includes all errors in following the on-board generated reference trajectory. VCCS errors are those in the calculation of the on-board trajectory which are caused by error sources on the vehicle. Wayside controller/guideway errors are those in the calculation of the on-board trajectory which are caused by error sources in the guideway or wayside controller.

A description of each error source and the derivation of the error allocations are given below followed by a discussion of system test results.

Steady-State Position Error

The LCS control loop configuration is such that a steady-state position error is required to compensate for long term variations in VCCS motor speed command, motor scale factor and tire rolling radius. A non-zero steady-state position error also will occur under nominal constant speed operation causing the vehicle to run a nominal value of 0.24 second behind point. The purpose of this intentional bias is to minimize brake/motor interaction by depressing the brake command. The steady-state position error entry in Table 1 refers only to variations in the intentional position error bias.

Actual vehicle speed for a perfectly calibrated vehicle in a steady-state condition (where measured speed equals the VCCS speed command reference) is given by

$$V_A = V_{CS} = (R_{WRR}/N_{GR})(K_M)(K_{VC}V_{CS} - K_{XE}X_E), \quad (1)$$

where

V_A = actual vehicle speed

V_{CS} = VCCS speed command reference (acceleration limited civil speed command)

R_{WRR} = Tire rolling radius

N_{GR} = Differential gear ratio

K_M = Motor scale factor

K_{VC} = Gain from V_{CS} to VCCS motor speed command

K_{XE} = Position error gain

X_E = Position error relative to on-board point.

Solving Equation 1 for the regulation error due to a steady-state position error gives:

$$T_{RE} = X_E/V_{CS} = \left[(K_{VC}K_MR_{WRR}/N_{GR})-1\right]/(K_{XE}K_MR_{WRR}/N_{GR}), \qquad (2)$$

$$= 0.24 \text{ second .}$$

This error is included in the nominal trajectory and does not enter into the error budget.

The tolerance limits on K_{VC}, K_M and R_{WRR} are $\pm$ 3%, $\pm$ 0.42% and $\pm$ 1.6%, respectively. Random variations in T_{RE} due to K_{VC}, K_M and R_{WRR} variations are computed as follows:

$$\Delta T_{RE} = \frac{\sqrt{\left[\frac{K_{VC}K_MR_{WRR}}{N_{GR}}\right]^2 \mathsf{x} \left[\frac{\Delta K_{VC}}{K_{VC}}\right]^2 + \left[\frac{\Delta K_M}{K_M}\right]^2 + \left[\frac{\Delta R_{WRR}}{R_{WRR}}\right]^2}}{K_{XE}K_MR_{WRR}/N_{GR}}, \qquad (3)$$

$$= \sqrt{0.91 \times (0.03)^2 + 0.0042^2 + 0.016^2}/0.194 = 0.170 \text{ second .}$$

Calculation of the limits on variations in tire rolling radius is given in Table 2. The estimated variations are for the conventional bias ply type of tire design used on the M-PRT vehicles.

TABLE 2. VARIATIONS IN TIRE ROLLING RADIUS

PARAMETER	PARAMETER LIMITS	VARIATION IN ROLLING RADIUS
Tire pressure	67.3 to 90 psig	$\pm$ 0.42%
Tire wear	0.0 to 0.4 inch	$\pm$ 1.39%
Loading	2188 to 2975 lbm/wheel	$\pm$ 0.59%

RSS Total = $\pm$ 1.6% = ΔR_{WRR}

The random variation in steady-state error of 0.17 second relative to the on-board point results in an additional error (in the on-board point) at the start of each speed transition because the transition starts at the wrong time. The position error before and after a transition remains constant in terms of feet of error. The regulation or time error, which is the position error divided by the commanded speed, however, changes as a function of the initial and final commanded speeds. This change represents a permanent error in the on-board point. This error mechanism is included in the steady-state position error entry of Table 1 and is computed as follows:

Error at 4-8 transition	=	0.5 sec/sec
Error at 8-33 transition	=	0.76 sec/sec
Error at 33-44 transition	=	0.25 sec/sec
Total		1.51 sec/sec.
Total error = 0.17 (1 + 1.51)	=	0.43 second.

Vehicle Dynamic Response

Dynamic response errors are those relative to the on-board point over and above the steady-state errors. These transient servo loop errors occur primarily in response to guideway grades and speed transitions. The error allocation for dynamic errors was initially arrived at by first determining capability relative to the remaining error sources and then assigning the remainder of the 1.1 seconds (0.6 second), on an RSS basis, to vehicle dynamic response. The following paragraphs describe the analysis subsequently performed to verify that this allocation could be met by the Phase IB design.

The first step taken in determining dynamic performance capability was to make a run from Engineering to Walnut using the nonlinear LCS simulation. As shown in Figure 1, a trip from Engineering to Walnut covers the entire Phase IB guideway. All parameter values were set at nominal or baseline values in this run. The objective was to locate the worst guideway point in terms of regulation capability.

Figure 8 shows the regulation error computed by the simulation for a nominal vehicle. Also shown are the corresponding grade profile and location of speed transitions, both as a function of distance from Engineering. The data shows an initial negative regulation error of -1.3 seconds which is a consequence of the VCCS/motor/brake phasing sequence used during station start and discussed in a later section. At the 8-33 fps speed transition, the regulation error has decreased to -0.3 second which is close to the steady-state error value of -0.24 second. The initial -1.3 second error does not present a significant problem as it is quickly corrected for and there are no moving vehicles directly behind a vehicle which has just been dispatched.

Figure 8 shows two areas where significant dynamic regulation errors occur: the first is on the 10% downgrade leaving Engineering where T_{RE} = 0.43 second; and the second is at the 4.5% downgrade approaching Walnut where T_{RE} = + 0.3 second. Higher regulation errors occur on downgrades because of the different and somewhat lower performance control loop used during braking.

The errors of concern are the variations from the nominal trajectory shown in Figure 8, as the nominal errors relative to the theoretical trajectory are accounted for in the software data base. The 10% downgrade area was chosen for further study to obtain a 3σ estimate of these variations from nominal. The specific variations considered and their impact on regulation are given in Table 3. The RSS total of the individual pertubations from nominal is 0.12 second, well within the 0.6 second allocated in Table 1.

Stopping Position

Errors in stopping a vehicle at the prescribed location in a berth are also errors in starting position which result in an initial error in the on-board point. The stopping requirement of ± 6 inches (± 0.5 foot) divided by 4 fps, the initial civil speed command, gives the maximum allowable initial regulation error of 0.125 second.

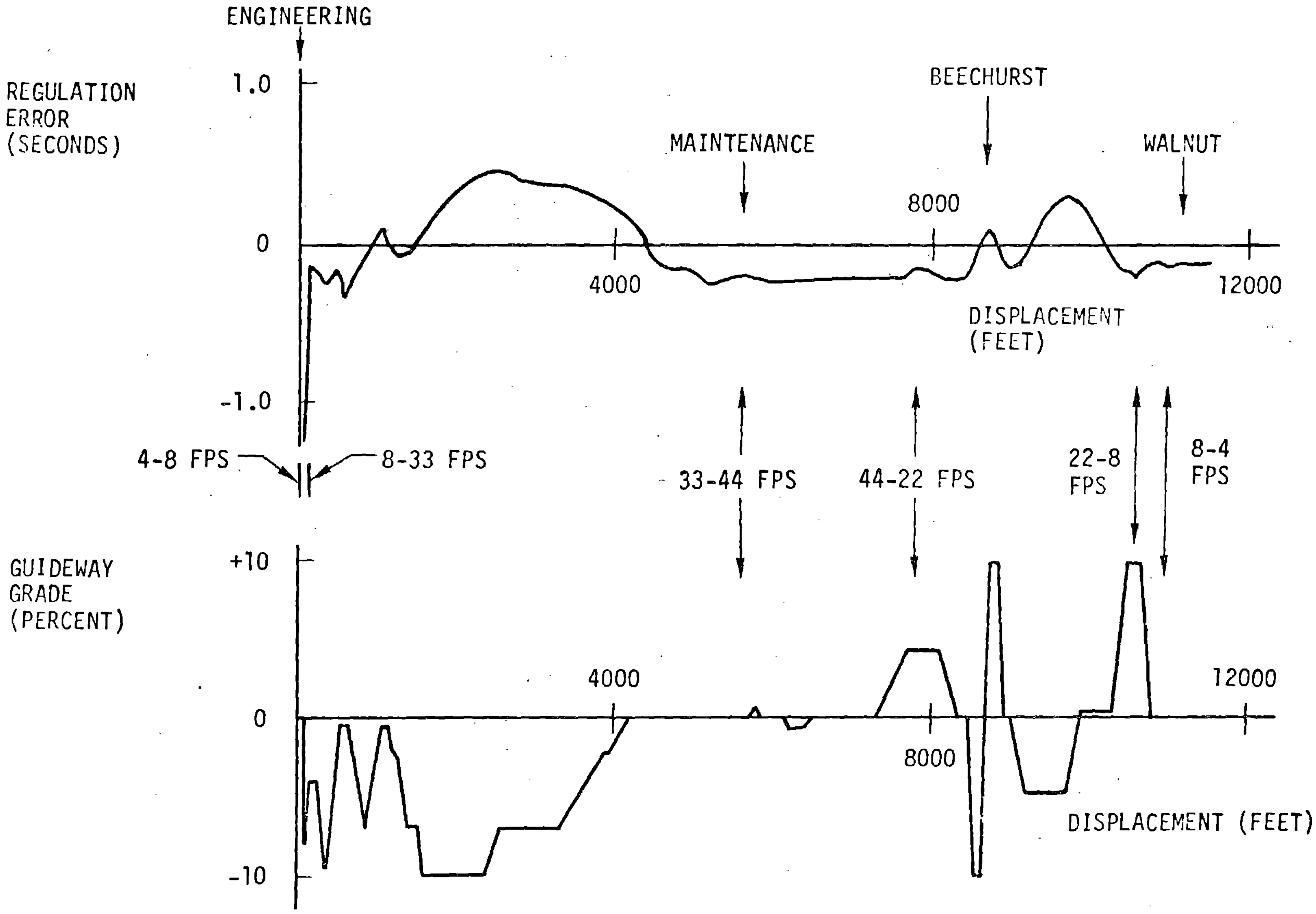


FIGURE 8. DYNAMIC REGULATION ERROR

TABLE 3. DYNAMIC REGULATION SENSITIVITY ANALYSIS

ERROR SOURCE PARAMETER	VARIATION FROM NOMINAL	INCREASE IN REGULATION ERROR (SEC)
Jerk limit on increasing brake command	-5%	0.009
Jerk limit on decreasing brake command	-10%	0.008
Magnitude of brake amplifier static compensation	+3%	0.007
Brake amplifier static compensation break point	-20%	0.010
Servo valve hysteresis	-0.6 ma	0.013
Servo valve gain	-10%	0.056
Servo valve null pressure	-25 psi	0.062
Brake caliper pressure threshold	+10%	0.017
Brake caliper gain	-10%	0.047
Propulsion speed scale factor	+0.042%	0.012
Motor torque command bias	+45%	0.020
Aerodynamic drag coefficient	-25%	0.013
Rolling resistance	-40%	0.034
Vehicle weight	+13.2%	0.045

RSS Total = + 0.12 second

Odometer Calibration

Required corrections in tachometer/odometer scale factor are calculated by the VCCS based on inputs provided by 200 foot calibration tone loops. The tachometers, which generate 76 pulses per motor shaft revolution, are also used as odometers to determine measured position by counting the number of scaled pulses rather than the pulse rate. The method of calibration is to count the number of scaled pulses during the presence of a calibration tone. If the measured displacement is greater or less than 200 feet, the scale factor is adjusted accordingly. The VCCS requirement is to calibrate to within ± 0.5% of the true scale factor.

The distance between calibration loops is nominally 800 feet with a requirement that spacing be a multiple of 100 feet. Automatic position updates are made at the end of each calibration loop encountered during a vehicle trip. The first update is computed by counting the number of scaled odometer pulses between the start and end of the first loop. The counting process is restarted at the end of each loop and the second and subsequent updates are based upon the number of pulses counted between the ends of calibration loops.

Because of the position update function, the only place where calibration errors result in a significant regulation error is from dispatch to the start of the first calibration loop. Table 4 gives the maximum time to the start of the first calibration loop for each station including Maintenance.

TABLE 4. MAXIMUM TIME BEFORE CALIBRATION

DISPATCH STATION	DIRECTION	TIME BEFORE CALIBRATION (SEC)
Engineering	Southbound	21
Maintenance	Southbound	30
Maintenance	Northbound	29
Beechurst	Southbound	28
Beechurst	Northbound	55
Walnut	Northbound	32

The factors contributing to regulation error are VCCS calibration accuracy, changes in tire rolling radius with passenger loading/unloading and errors in calibration loop length. The errors in loop length are compensated for by adjustments in dispatch times and are not considered further here.

The maximum regulation error due to odometer calibration is based on a northbound dispatch from Beechurst which gives the longest time (55 seconds) to the start of the first calibration loop. A $\pm$ 0.5% calibration error for the 55 seconds gives a regulation error of $\pm$ 0.27 second. The maximum change in tire rolling radius with loading (Table 2) of 1.2% gives a corresponding regulation error of 0.66 second. The error allocation of 0.71 seconds is the RSS sum of the two component error sources.

Duration of Illegal Speed Command

In crossing from one FSK guideway loop to the next it is possible for the VCCS to detect speed tones from both loops resulting in an illegal combination of the three speed tones for a short period. The VCCS interprets an illegal speed command (presence of all three tones) as a 4 fps command. The possibility of receiving all three tones exists at the 8-33 fps, 22-33 fps and 33-44 fps speed transitions. The VCCS responds to the assumed 4 fps command by starting to profile the limited speed command down at a rate of 2 fps^2. Since the limited speed command must first be profiled back to the initial value before proceeding to the new, higher speed command, the effect is to delay start of the speed transition for a time equal to twice the duration of the illegal command.

A worst case trip on the Phase IB guideway will have additive illegal speed command errors at an 8-33 fps and a 33-44 fps speed transition. The hardware requirement is to limit duration of an illegal command to a maximum of 0.125 second for a total delay of 0.25 seconds in the start of the speed transition. The regulation error due to an illegal speed command is the total delay in the start of the transition times the difference in initial and final speeds divided by the final speed.

Regulation error due to illegal speed commands, for a worst case trip, is computed as follows:

Error at 8-33 transition	=	0.76 sec/sec
Error at 33-44 transition	=	0.25 sec/sec
Total		1.01 sec/sec.

$$\text{Total error} = 0.125 \times 2 \times 1.01 = 0.25 \text{ second.}$$

VCCS Clock Accuracy

The autonomous VCCS clock establishes the time base for computation of the on-board point follower command trajectory. A VCCS clock error, therefore, translates directly into an error in the on-board point which increases with trip time and is greatest at the end of a vehicle trip. The longest possible trip time is 427 seconds for a run from Engineering to Walnut. The regulation error allocation of 0.08 second is computed by multiplying the 427 seconds by the VCCS clock accuracy requirement of 0.02%.

Software Dispatch Tolerance

A major function of the station software is to dispatch vehicles at the specific time prescribed by central. The requirement is to issue the dispatch command within $\pm$ 0.2 second which covers software cycle time, quantization and communication delay errors.

Central Clock Accuracy

All of the wayside computing equipment, including central, uses the incoming power line frequency as a time base. The central clock accuracy error source, therefore, refers to the accuracy or stability of the power line frequency.

The impact of central clock errors on vehicle regulation is in the accuracy of dispatching two vehicles which later merge on the main guideway. The worst case Phase IB condition is a vehicle from Beechurst merging with a vehicle from

Engineering (See Figure 1) where the time between dispatches is at a maximum of 285 second. The regulation error allocation of 0.28 seconds is computed by multiplying the 285 seconds by an assumed power line frequency accuracy of 0.1%.

Typically, a power company will only guarantee $\pm$ 0.83% ($\pm$ 0.5 Hz) on power line frequency which obviously is not adequate. Short term accuracy which is the parameter of interest is, however, typically much better. Realistic limits on frequency variations were arrived at by an analysis of actual frequency variation data at Morgantown over a period from 19 November 1973 to 27 November 1973.

The largest deviations observed in the analysis, considering both magnitude and duration, are given in Table 5. The maximum variation of 0.067% is well within the 0.1% assumed limit, leading to the conclusion that use of power line frequency as a time base will, in fact, give acceptable performance.

TABLE 5. POWER LINE FREQUENCY VARIATIONS

MAXIMUM DEVIATION (Hz)	DURATION (SEC)	PERCENT ERROR
+0.037	600	+0.062
+0.025	3600	+0.042
-0.025	1800	-0.042
+0.035	600	+0.058
+0.04	600	+0.067

Estimated Performance Capability

Each error allocation in the regulation error budget of Table 1, with the exception of vehicle dynamic response, represents the estimated capability relative to the applicable error source. Using 0.12 second as the 3σ limit for vehicle dynamic response gives a total RSS estimated system performance capability of 0.95 second, just under the requirement of 1.1 seconds.

Design Verification

Design verification testing was performed both at STTF (Surface Transportation Test Facility) and at Morgantown. Overall, the data obtained shows good vehicle repeatability with average errors relative to the theoretical trajectory being larger than expected. Variations in tire rolling radius and errors accumulated at the start of speed transitions were found to be the dominant error sources, rather than errors due to wind and grades as assumed in many analyses. Total error was found not to be a strong function of trip length, but, rather of the number and type of speed transitions encountered.

An example of the test results obtained is given in Figures 9 and 10. Figure 9 shows the 3σ variations for 12 minimum weight and 12 maximum weight trips at STTF. These variations from average, which have a maximum value of 0.55 second, illustrate the good trip to trip repeatability observed in all of the data. A comparison of the average vehicle trajectory, for the STTF data of Figure 9, to the theoretical trajectory is given in Figure 10. The relatively large deviations from the theoretical trajectory (0.9 second maximum) illustrates the motivation for using the nominal trajectory as a basis for computing regulation error.

In Phase IA, the theoretical trajectory was used in the operational software for dispatching vehicles and monitoring performance. Regulation errors relative to the theoretical trajectory, as defined in Appendix A, were, therefore, the parameter of interest.

In Phase IB, a data base representing a nominal or average vehicle trajectory is used in the operational software for dispatching vehicles and monitoring performance. As a consequence, only regulation errors relative to the nominal trajectory are of interest and these random errors are shown by the test data to be within the $\pm$ 1.1 second requirement. Large systematic errors of the type shown in Figure 10 are of interest only in the definition of the required data base.

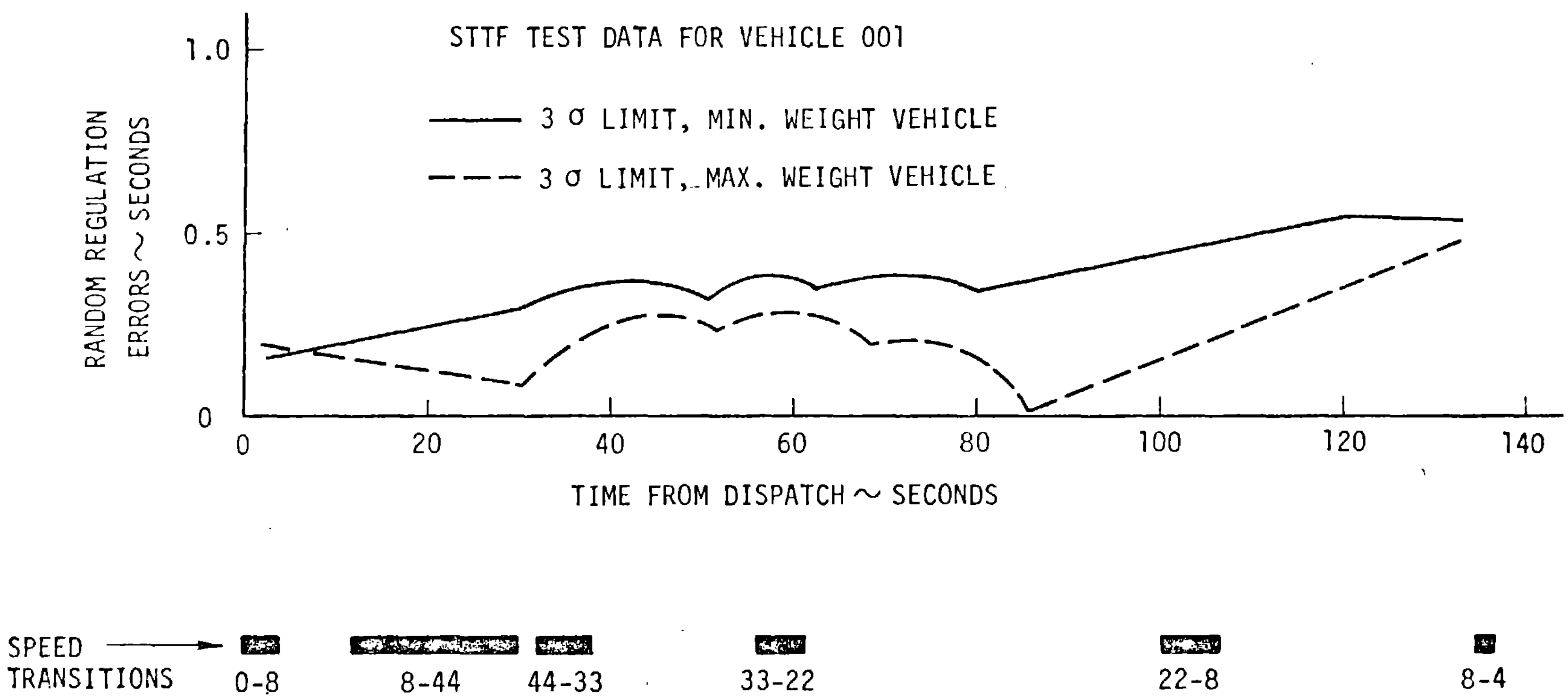


FIGURE 9. RANDOM VEHICLE DEVIATIONS FROM AVERAGE TRAJECTORY

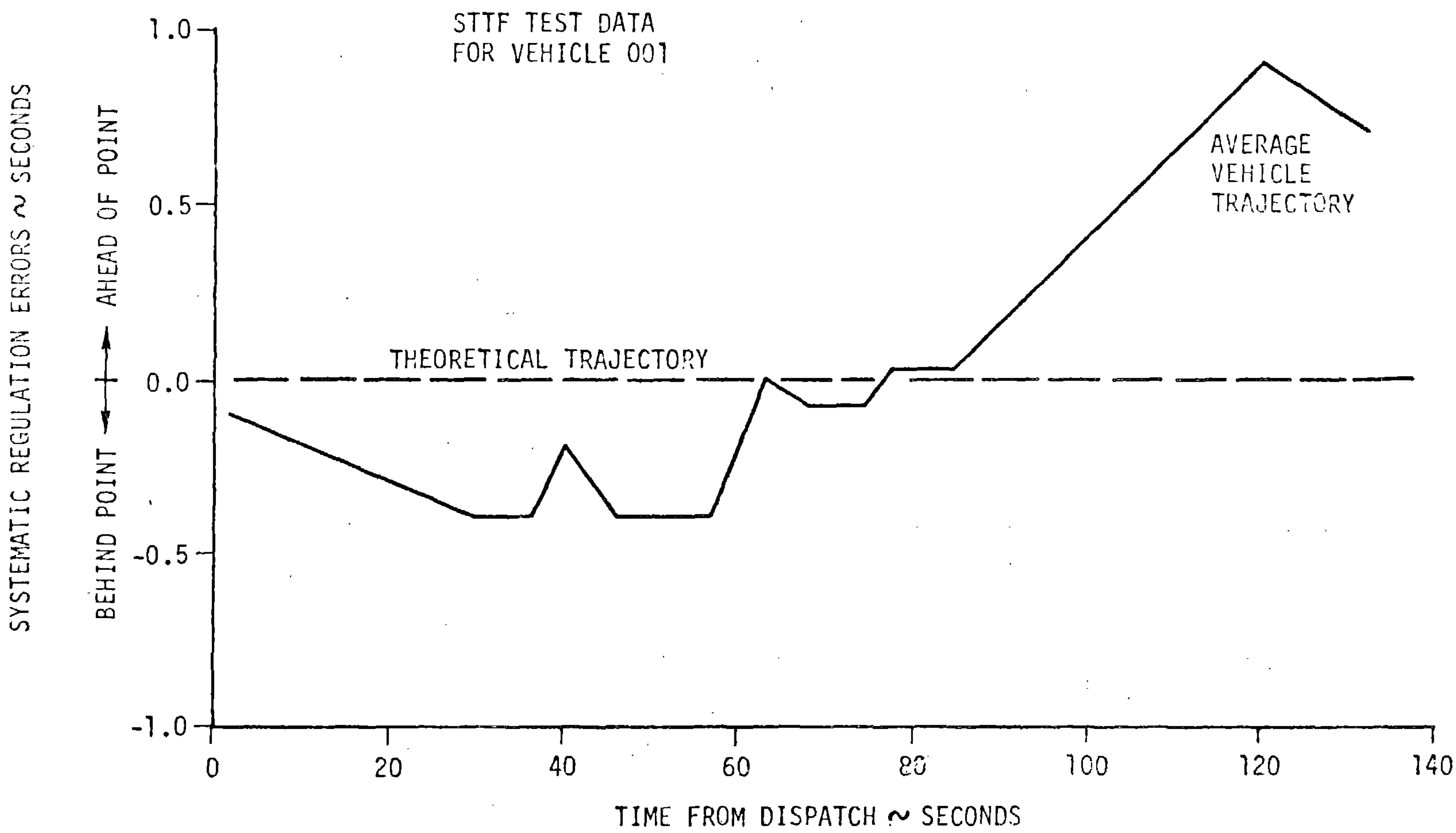


FIGURE 10. DEVIATION OF AVERAGE VEHICLE TRAJECTORY FROM THEORETICAL

4.2 Station Stop

The station stop requirement is to stop the vehicle within $\pm$ 6 inches of its assigned position in a station berth. Two station stopping control concepts are discussed: the Phase IA control concept which did not meet the requirements; and the Phase IB control concept used in the operational Phase IB system, which does meet the requirement.

Phase IA Control Concept

Figure 11 illustrates the Phase IA station stop concept. The vehicle is commanded to a speed of 4 fps prior to reaching the stop tone loop. Upon detection of the leading edge of the stop tone signal, the speed command is profiled down to 2 fps at the normal rate of 2 fps^2. Closed-loop control is continued until a short time after detection of the stop tone null resulting from the loop crossover shown in Figure 11. A "forced brake" signal or a full scale normal brake command is issued by the VCCS following receipt of a specified number of scaled odometer pulses after detection of the stop tone null. In this final portion of the sequence, open-loop braking is used to bring the vehicle to a stop, i.e., the brake amplifier responds to the "forced brake" command by profiling the servo valve current command to full scale at the specified jerk rate. The brakes are left in their full scale "on" condition until dispatch to insure no vehicle movement after it has been brought to a stop.

Actual performance of the Phase IA concept is illustrated in Figure 12. This histogram is a compilation of data for 649 stops at both Morgantown and STTF. As shown, final position errors exceed the requirement by approximately a factor of three. A second problem with the concept is that the LCS had difficulty providing adequate speed control at 2 fps. The specific problem is excessive underspeeds following the 4-2 fps speed transition which gave very poor ride quality, i.e., a roller coaster type of ride.

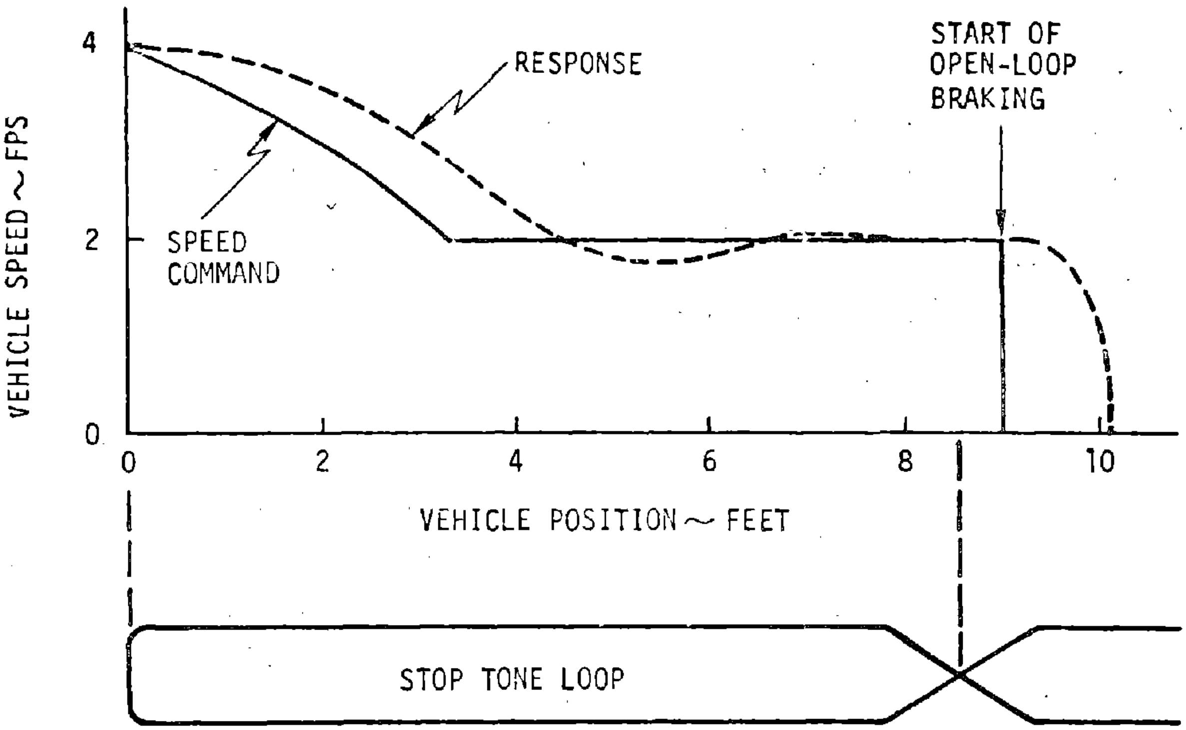


FIGURE 11. PHASE IA STATION STOP CONTROL CONCEPT

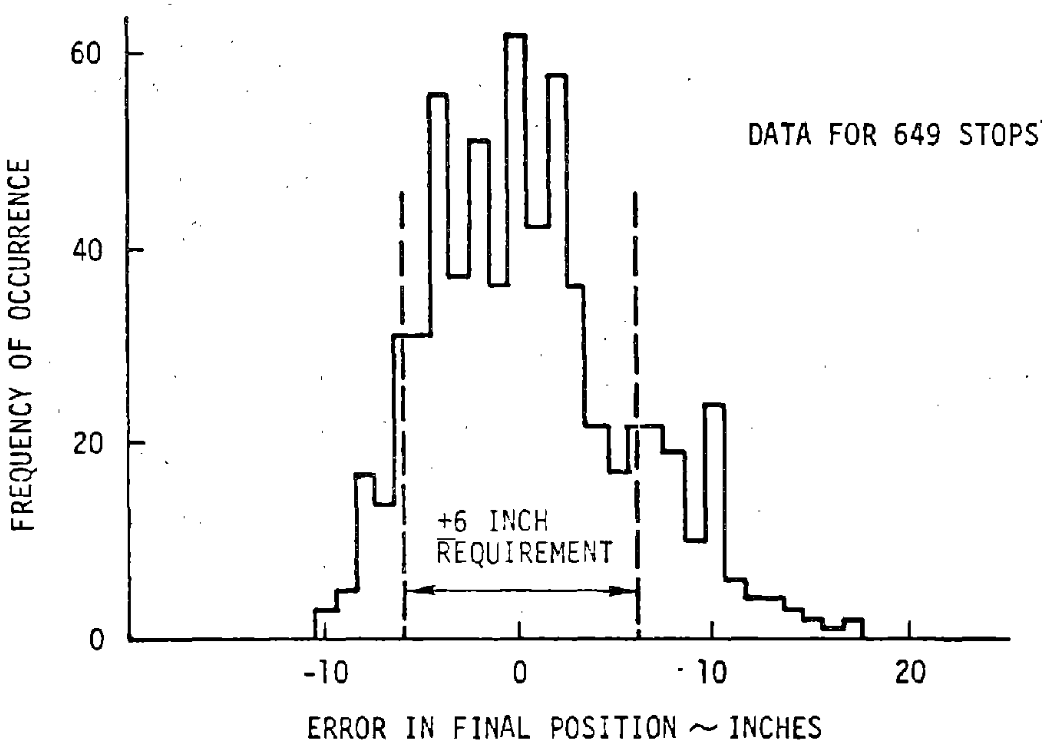


FIGURE 12. PHASE IA STATION STOP TEST RESULTS

Special tests were conducted to understand and identify the cause or causes of the excessive stopping errors. Time history data for 20 key LCS variables were recorded under a variety of conditions. Of particular interest was the final "forced brake" portion of the stopping sequence, because of the inherent accuracy problem associated with open-loop control.

A detailed analysis of the open-loop braking portion of the test data resulted in the identification of six error sources which have a significant impact on stopping accuracy. Table 6 summarizes the results of a statistical error analysis conducted to determine the net impact on stopping accuracy capability of the six error sources identified. This analysis covers only the final open-loop braking portion of the stopping sequence.

Table 6 shows two 3σ or "RSS total" estimates of stopping accuracy. The first estimate of + 27.4, -16.2 inches is for Phase IA error source limits derived from the test data. The reason these estimates of stopping accuracy are slightly higher than the measured errors of Figure 12 is believed to be a correlation between initial velocity and acceleration errors not accounted for in the analysis. The second estimate of + 10.6, -8.3 inches is for estimated minimum hardware limits used in conjunction with the Phase IA braking concept. These minimum limits are estimates of the lowest variations which can be achieved within realistic hardware and cost constraints.

The conclusion of the analysis is that meeting the $\pm$ 6 inch requirement with the open-loop braking concept requires extremely close and expensive control of error source parameter variations. The low probability of meeting the requirement with open-loop braking led to a decision to discard the Phase IA concept and adopt the closed-loop control concept described in the next section.

TABLE 6. PHASE IA STATION STOP ERROR ANALYSIS

ERROR SOURCE PARAMETER	UNITS	NOMINAL VALUE	ESTIMATED PHASE IA 3σ LIMITS*	IMPACT ON FINAL POSITION ERROR (INCHES)
Initial velocity	fps	2.0	2.73 (2.5) 1.27 (1.5)	+10.6 (+7.1) - 8.8 (-6.2)
Initial acceleration	fps^2	0.0	+1.8 (+0.64) -1.8 (-0.64)	+23.0 (+5.8) -10.9 (-3.9)
Brake time delay	sec	0.0	+0.26 (+0.2) -0.0 (-0.0)	+6.2 (+4.8) -0.0 (-0.0)
Initial brake pressure	psig	0.0	-0.0 (-0.0) +26.9 (+10.0)	+0.0 (+0.0) -7.0 (-3.2)
Jerk limit variations				
Brake pressure rate	psig/sec	65.0	31.0 (58.5) 99.0 (71.5)	+8.0 (+1.0) -3.4 (-0.8)
Brake pad friction coefficient	lbf/psig	1.6	1.34 (1.34) 1.86 (1.86)	+1.6 (+1.6) -1.3 (-1.3)
Vehicle weight	lbm	10390	11965 (11965) 8815 (8815)	+1.3 (+1.3) -1.4 (-1.4)
Oscillation in vehicle deceleration (half amplitude)	fps^2	0.0	-0.0 (-0.0) +1.29 (+0.32)	+0.0 (+0.0) -0.9 (-0.2)

RSS Total = +27.4 (+10.6) inches
-16.2 (-8.3) inches

*Numbers in ( ) are estimates of lowest achievable variations.

Phase IB Control Concept

Figure 13 illustrates the Phase IB station stop control concept. As in the Phase IA concept, the vehicle is commanded to a speed of 4 fps prior to reaching the stop tone loop. Upon detection of the leading edge of the stop tone signal, the VCCS switches from its normal acceleration limited civil speed command reference to the special purpose speed versus position command profile shown in Figure 13. During the station stop sequence, this command to the closed-loop speed servo is reduced from 4 to 0 fps solely on the basis of distance traveled. Because of this loop closure on position, the speed command and the actual speed reach zero at a precise point on the guideway, rather than at a precise point in time, which minimizes errors in final vehicle position.

The closed-loop nature of the Phase IB station stop speed command eliminates the open-loop braking feature of the Phase IA concept and its attendant sensitivity to error sources. The square law profile chosen produces an effective constant deceleration command of 0.84 fps^2 (0.026 g) in the time domain which maximizes ride quality and eliminates the two stage deceleration characteristic of the Phase IA concept. Stop tone loop crossovers and the need to detect a null in the stop tone signal are eliminated since the entire profile is referenced to the leading edge of the stop tone. Use of the leading edge of the stop tone as the final position reference requires an accurate calibrated odometer which, in the M-PRT LCS design, is available as a result of the normal point follower controller requirements.

The total length of the station stop profile stored in the VCCS is 127.5 inches. A preset or initial value capability is included in the odometer pulse counter which provides the input to the profiler. Upon detection of the stop tone, the counter (for the channel driven by the forward antenna) starts from an initial value of 11.5 inches giving an effective profile length of 116 inches as shown in Figure 13. This preset function provides the capability to compensate for any bias or systematic errors uncovered in initial testing and also allows compensation for the difference in longitudinal location of the two redundant vehicle uplink antennas. Length of the total profile was chosen to allow use of existing stop tone loops.

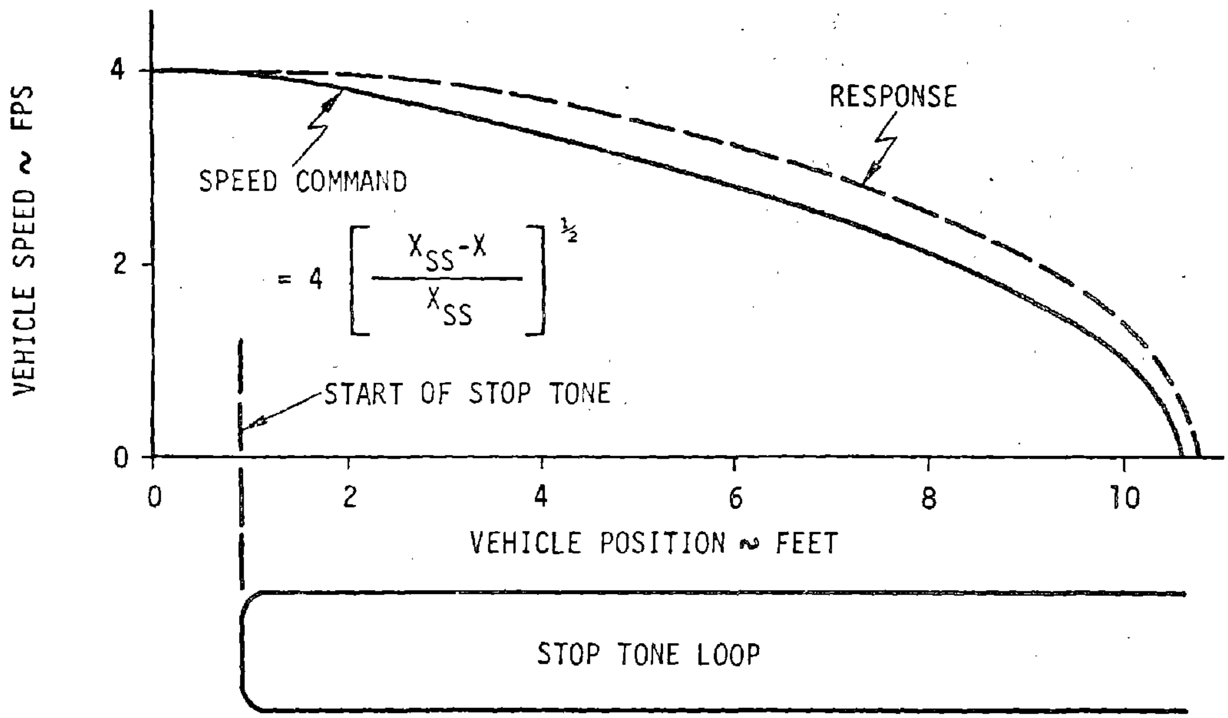


FIGURE 13. PHASE IB STATION STOP CONTROL CONCEPT

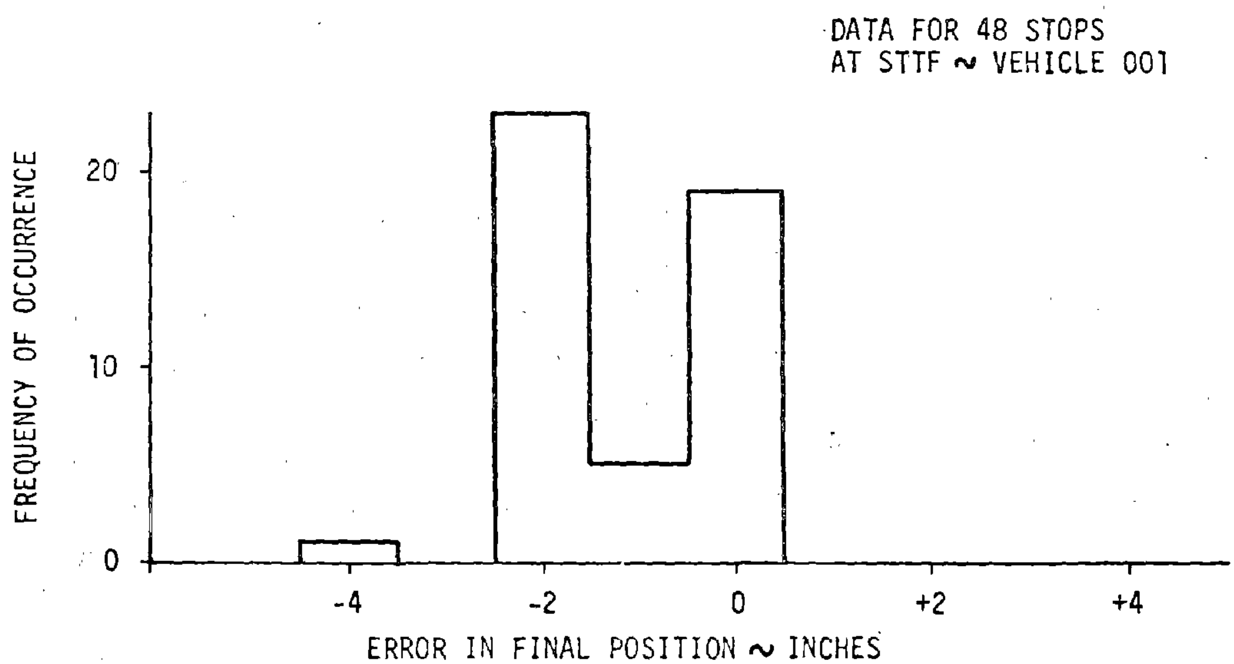


FIGURE 14. PHASE IB STATION STOP TEST RESULTS

Existing LCS speed control circuits are used, with one modification, to provide control of actual vehicle speed during the stopping sequence. The one modification is to disable the VCCS position error feedback loop used during point follower control. The position error feedback is not needed because of the position dependent speed command and could degrade stopping accuracy if large position errors were present at the start of the stop sequence.

Vehicle motion, once stopped, is prevented by a "forced brake" signal or a full scale normal brake command from the VCCS, as was the case in the Phase IA concept. The difference is that the "forced brake" command is issued by the VCCS upon detection of a zero speed command, just prior to the actual vehicle stop, at which point good stopping accuracy is assured.

A detailed sensitivity analysis of the Phase IB concept verified that it was, in fact, insensitive to system parameter variations. The results of this analysis are summarized in Table 7. The predicted 3σ stopping accuracy limits of +3.5, -2.8 inches are well within the $\pm$ 6 inch requirement.

The stopping errors shown in Table 7 were computed using the nonlinear LCS simulation where each error source parameter was varied, one at a time, to determine its impact on stopping accuracy. Of the error sources considered, three are dominant, namely:

- o Errors in tachometer/odometer calibration,
- o Initial overspeeds at the start of the stop sequence, and
- o Location and detection of the leading edge of the stop tone.

TABLE 7. PHASE IB STATION STOP ERROR ANALYSIS

ERROR SOURCE PARAMETER	NOMINAL VALUE & UNITS	ESTIMATED PHASE IB 3σ VARIATIONS	IMPACT ON FINAL POSITION ERROR (INCHES)
Tachometer/odometer scale factor	0.1675 ft/rad	-0.0033 +0.0033	+2.40 -2.16
Gain from velocity error to brake command	3.9 v/fps	-0.125 +0.125	+0.12 -0.12
Brake command null offset	0.0 v	-0.05 +0.05	+0.12 -0.12
Motor command null offset	0.0 v	+0.10 -0.10	+0.24 -0.0
Jerk limit on increasing brake command	4.425 v/sec	-0.221 +0.221	+0.0 -0.12
Magnitude of brake amplifier static compensation	2.4 ma	-0.07 +0.07	+0.0 -0.12
Brake amplifier static compensation break point	0.25 v	+0.05 -0.05	+0.12 -0.12
Brake amplifier steady state gain	2.51 ma/v	-0.13 +0.13	+0.12 -0.24
Servo valve hysteresis	0.6 ma	+0.6 -0.6	+0.48 -0.24
Servo valve gain	22.42 psig/ma	-2.24 +2.24	+0.48 -0.36
Servo valve null pressure	8.3 psig	-25.8 +46.7	+0.72 -1.20
Brake caliper pressure threshold	50 psig	+5.0 -5.0	+0.12 -0.24
Brake caliper gain	5.757 ft lb/psig	+0.576 -0.576	+0.36 -0.36
Propulsion speed loop gain resistor	250,000 ohms	+150,000 -150,000	+0.0 -0.12
Vehicle weight	10,325 lbm	+ 1,575 - 1,575	+0.48 -0.60

TABLE 7. PHASE IB STATION STOP ERROR ANALYSIS
(Continued)

ERROR SOURCE PARAMETER	NOMINAL VALUE & UNITS	ESTIMATED PHASE IB 3σ VARIATIONS	IMPACT ON FINAL POSITION ERROR (INCHES)
Rolling resistance	152 lbf	-61 +91	+0.12 -0.12
Initial velocity	4.0 fps	+1.0 -1.0	+2.04 -0.0
Stop Tone acquisition delay	0.001 sec	+0.001 -0.001	+0.05 -0.05
Stop tone loop leading edge location	0.0 in	+0.5 -0.5	+0.5 -0.5
Detection of stop tone leading edge	1.57 in	+0.78 -0.78	+0.78 -0.78

RSS Total = +3.50 inches
-2.80 inches

Actual performance of the Phase IB concept is illustrated in Figure 14. This histogram is a compilation of measured stopping accuracy for the 12 maximum weight and 12 minimum weight runs made at STTF to obtain the regulation data presented in Figures 9 and 10. Two stops, one at a rear berth and one at the forward berth, were made for each run giving a total of 48 data points. The two groupings of data represent the difference between berths rather than variations in stopping accuracy at a given berth. In summary, this data verifies analysis predictions that stopping errors will be significantly less than the $\pm$ 6 inch requirement.

4.3 Station Start

Regulation and ride comfort requirements dictate close control of VCCS, brake and motor phasing during startup from zero speed. The design task is somewhat complicated by the fact that, for safety reasons, the brakes are left in a full scale "on" state and the motor is disabled or turned off when the vehicle is at rest. The requirements during startup can be summarized as follows:

- To avoid excessive jerks, the motor must not begin producing torque before the brakes are off.
- To avoid excessive overspeeds, the motor must begin producing torque as soon as possible following start of the speed command profile.
- To avoid excessive regulation errors, the vehicle must reach the first speed transition at a consistent time relative to dispatch.

Figure 15 illustrates the startup phasing adopted for M-PRT. The startup sequence is normally initiated by removal of the stop tone which corresponds to a dispatch command. (Startups on the guideway, following an anomaly, are initiated by issuing a 100% performance level command.) In response to the stop tone removal, the VCCS begins profiling its acceleration limited speed command reference from 0 to 4 fps, removes the "forced brake" or full scale brake command, and issues a propulsion system "on" command.

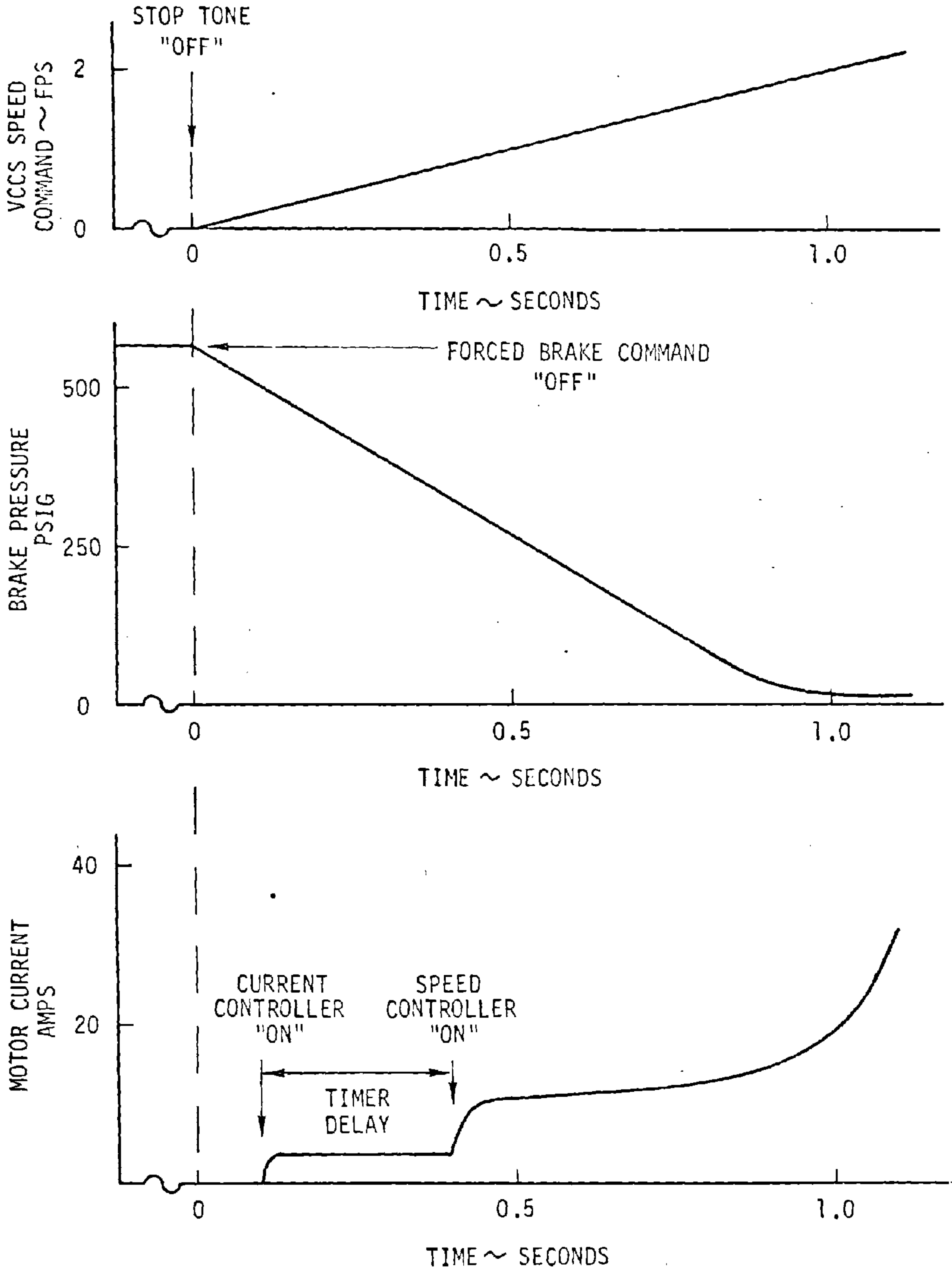


FIGURE 15. VCCS, BRAKE AND MOTOR PHASING AT STARTUP

The brake system responds to the removal of the "forced brake" signal by reducing the hydraulic brake pressure to bias levels. Brake "off" time is governed by the jerk limiter used in the brake amplifier to limit vehicle jerks during normal closed-loop control. To insure repeatability, an "off" time requirement of 0.9 $\pm$ 0.1 second is imposed on the brake system.

The propulsion system requirement is to reach an output current value of 10 amps above bias level within 1.0 $\pm$ 0.1 seconds after receipt of an "on" command. One amp equals approximately one ft lb of output torque at the motor shaft. As shown in Figure 15, there are three basic series components in the 1.0 second delay. The initial delay of approximately 0.1 second is the time required to activate the relays or contactors which apply power to the system. The second delay of approximately 0.3 second is due to a timer which provides the adjustment capability needed to insure meeting the accuracy requirement on the total delay. The timer inhibits the speed controller circuits, but allows the current controller to turn on resulting in the residual current level shown in Figure 15. The final delay of approximately 0.6 second is the dynamic response lag of the jerk and acceleration limiter and speed loop control circuits.

The delay in motor torque, without a corresponding delay in the speed command profiler, causes some increase in speed overshoot at the end of the speed transition. This increase, however, is small compared to the 3 fps overspeed limit and does not justify the circuitry required to delay the speed command. The delay also means each vehicle will cross the first presence detector approximately 1 second late relative to the theoretical trajectory. This bias error affects system performance only in that it must be included in the data base used to monitor vehicle performance at the first presence detector. The error accumulated initially is eventually corrected for by the position error circuitry.

4.4 Brake Amplifier Settings

The primary function of the brake amplifiers is to generate a servo valve current command proportional to the 0 - 10 vdc VCCS analog brake command issued during normal operation. The brake amplifiers also have a significant number of control functions which include:

- Jerk limiting of the VCCS deceleration command to prevent excessive vehicle jerks and to provide accurate off time control during a startup sequence.
- Limiting of the minimum output to an adjustable bias level to minimize brake time delays.
- Shaping of the output command near null to optimize the tradeoff between brake drag and brake time delays.
- Limiting of the maximum output to specified normal and emergency values to prevent excessive deceleration levels.

These control functions result in a total of seven specific parameters which must be assigned values, namely:

- Output bias level (for 0.0 volt brake command)
- Off time (time to go from full scale normal rate braking output to static compensation break point)
- Initial static compensation step
- Full scale emergency rate braking output
- Full scale normal rate braking output
- Maximum emergency rate braking "on" rate
- Maximum normal rate braking "on" rate.

The design task is to assign values to the above parameters. This section describes the rationale and computations used in deriving values for these adjustable parameters and illustrates their use in compensating for caliper nonlinearities and servo valve flow limit problems discovered late in the design cycle. Included are sections describing the brake system analytical model used and an analysis of brake drag performance.

Simplified Brake System Model

The elements of the brake system and the interconnections between elements are defined in Figure 6. Figure 16 provides a simplified analytical model for this brake system during normal operation. The model, which does not account for system redundancy, is the model used in the LCS simulation for the majority of the analyses conducted and, specifically, forms the basis of the analyses described in this section. A significantly more detailed model, required to describe the caliper nonlinearities and servo valve flow limit problems encountered, is presented in Section 5.

Figure 16 also provides a simplified schematic of the brake caliper design. An understanding of this design, which has the important function of voting the higher of the two input pressures, is required to follow many of the calculations made in choosing amplifier settings. In terms of the brake schematic of Figure 6, Brake System "A" refers to the brake amplifier and servo valve controlling the inboard pressure input of each caliper. Brake System "B", in turn, controls the outboard pressure input of each caliper. Voting is accomplished by movement of the floating caliper piston. If the outboard pressure is high, the floating piston moves to the right on Figure 16 and allows the outboard system to supply all of the force on the primary piston which produces the brake torque. If the inboard pressure is high, the floating piston moves to the left and allows the inboard system to supply all of the force on the primary piston via contact with the floating piston. The caliper design also has retractor springs and a pad wear compensator mechanism not shown in Figure 16. The retractor springs result in a pressure threshold which must be exceeded to produce braking torque and

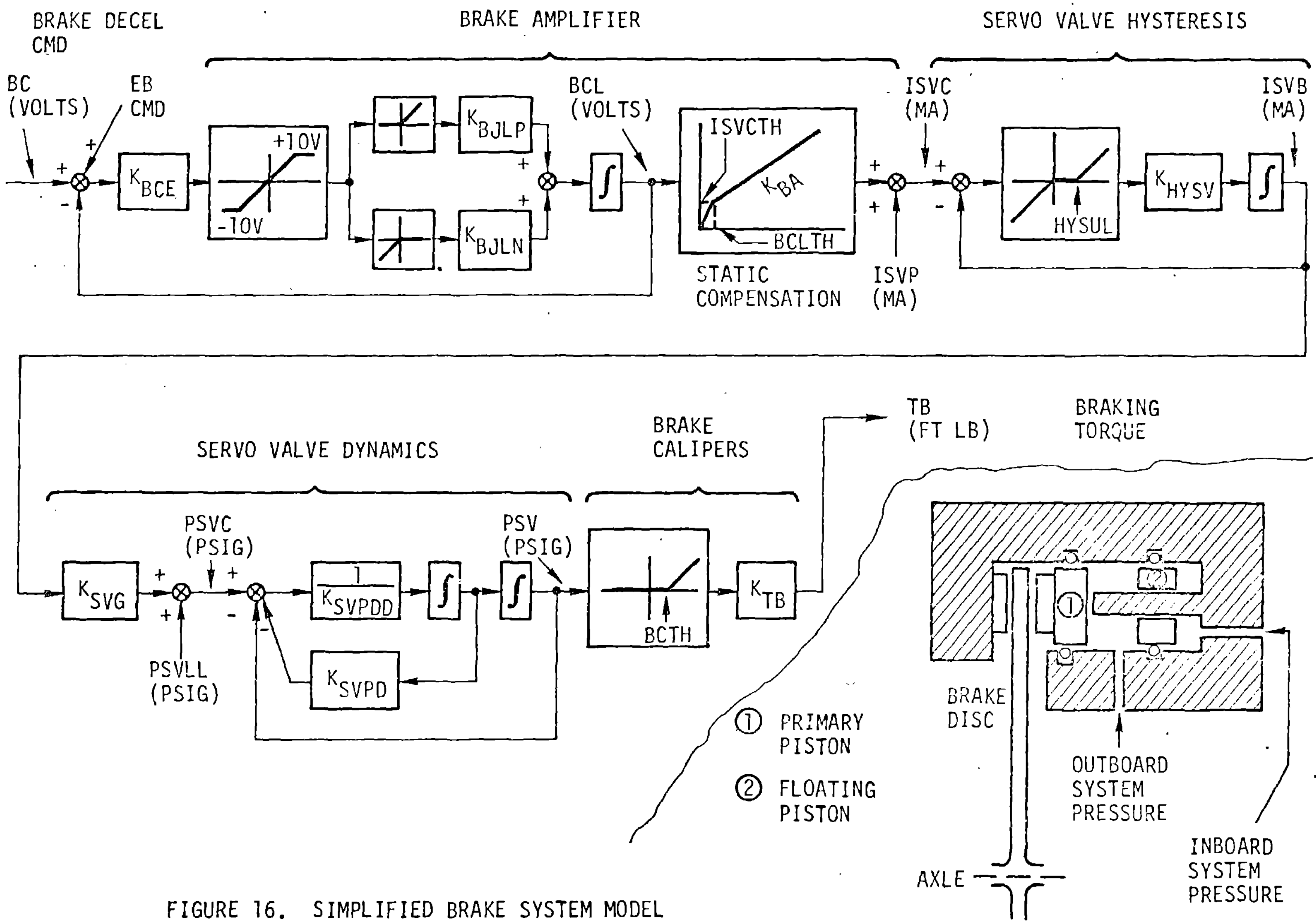


FIGURE 16. SIMPLIFIED BRAKE SYSTEM MODEL

thereby minimizes brake drag. The compensator mechanism has the function of reducing time delays by minimizing the distance the primary piston must travel to accomplish pad to rotor contact.

Output Bias Level

The reason for requiring a non-zero servo valve bias level, which was not part of the original design, is to reduce excessive initial servo valve time delays. Test data show that use of a small bias significantly reduces delays. Large brake system time delays, in excess of 0.2 second, are of concern because of their adverse effect on speed control, jerk control and emergency brake stopping distances.

The bias levels selected are 20 psig for the outboard system and 35 psig for the inboard system. To reduce the impact of valve-to-valve variations, the brake amplifier bias levels are set (after installation) to give the actual pressure bias levels specified as measured by a pressure gauge.

The major factor in choosing bias levels is their value relative to the brake caliper pressure threshold. If the bias exceeds the threshold, brake drag will occur. If the bias is significantly less than the threshold, excessive time delays will occur since the brake pressure can increase no faster than the amplifier jerk limit setting. The optimum value for the difference between the caliper threshold and the bias level is 45 psig, based on expected parameter variations and their impact on brake drag and brake time delays.

Use of a non-zero bias level required an increase in the caliper threshold relative to the original design value. This was accomplished by adding a second retractor spring to each of the calipers. Threshold measurements on the final caliper design provided the following estimates of the nominal threshold value and limits:

- Outboard system (front and rear calipers), 65 $\pm$ 15 psig
- Inboard system (front calipers), 80 $\pm$ 15 psig
- Inboard system (rear calipers), 105 $\pm$ 15 psig.

These estimates and the desired pressure differential between bias and threshold values of 45 psig form the basis of the bias level requirements stated above.

Setting the inboard system bias level higher than that of the outboard system has two additional advantages, namely:

- o Because of differences between caliper inboard and outboard pressure - torque gains and the different bias levels, the inboard system (which is the preferred system) will normally start out in control and stay in control. Movement of the caliper floating piston and the resulting fluid flow rates is thereby minimized.

- o During brake applications, when the inboard system is in control, the front calipers will normally start producing torque before the rear calipers engage. This built in phasing has the benefit of reducing short term jerk at the start of braking.

Off Time

The brake system "off" time requirement is to respond to the removal of a "forced brake" full scale normal brake command by reducing brake pressure from its full scale level to its bias level in a time period of 0.9 $\pm$ 0.1 second. The reason for this requirement, described in Section IV-C, is to insure proper VCCS/brake/motor phasing during startup. "Off" time is controlled by the brake amplifier, specifically the parameter which sets the amplifier jerk limit during a decreasing output condition.

A major factor in setting the brake amplifier "off" time requirement is the impact of the two orifices and check valves shown in Figure 6. The function of these orifices is to limit fluid flow rates in the reverse direction. The reason for this function, which was not part of the original design, is an inability of the servo valves to maintain commanded pressure during periods of high fluid flow rates. Without the orifices, high flow rates caused by movement of the caliper floating pistons resulted in time delays on the order of one second under some dynamic conditions.

The impact of the orifices on "off" time is to cause the pressure "off" time to be longer than that commanded by the amplifiers. During the final portion of the pressure reduction, the caliper retractor springs force the primary piston away from the rotor resulting in a reverse fluid flow in both pressure systems. Because of the restriction imposed by the orifices, the pressure decay is exponential and slower than that commanded by the amplifiers which results in the increase in "off" time.

The brake amplifier "off" time requirement selected is 0.7 ± 0.1 second and was arrived at empirically via a trial and error test procedure. A 0.7 second amplifier "off" time results in a pressure level, 0.9 second after removal of the full scale brake command, which is approximately 45 psig above the bias level or right at the caliper pressure threshold point.

Initial Static Compensation Step

The function of the brake amplifier static compensation circuit (See Figure 16) is to optimize the tradeoff between brake drag and brake time delays through shaping of the servo valve command near null. The compensation circuit provides a two stage amplifier gain characteristic. Specifically, the output stage gain near null is seven times higher than the gain (K_{BA} in Figure 16) over the remainder of the operating range. Without this compensation, the time needed to achieve the initial 45 psig change in pressure (required to reach the caliper threshold and obtain braking torque) would be excessive. The reason is the limit imposed by the jerk limiter on the output command rate. With the compensation, a very small change in jerk limiter output puts the amplifier output and valve pressure at the threshold point due to the initial high gain. This allows use of the pressure differential (45 psig) needed to meet the brake drag requirement and still maintains a minimum time delay characteristic.

The design task is to compute the value of the initial static compensation step or break point (ISVCTH in Figure 16) in the two stage gain curve. The criteria used is to select a value for ISVCTH which exactly cancels the intentional pressure bias level to caliper threshold separation under nominal conditions. The value selected for ISVCTH is 2.6 ma and is computed by the equation

$$ISVCTH = HYSUL + (BCTH - PSVB)/K_{SVG}, \tag{4}$$

where

HYSUL	=	Servo valve hysteresis
	=	0.6 ma
BCTH	=	Brake caliper pressure threshold
	=	65 psig for outboard system
	=	80 psig for inboard system
PSVB	=	Servo valve control pressure bias level
	=	20 psig for outboard system
	=	35 psig for inboard system
K_{SVG}	=	Steady-state servo valve gain
	=	22.36 psig/ma.

Full Scale Emergency Rate Braking Output

Emergency stops are performed open-loop and are initiated by the VCCS which removes a 28 vdc logic signal to the brake amplifiers. The amplifiers respond by replacing the 0 - 10 vdc analog input to the jerk limiter with a constant voltage input (See Figure 16) having a value which gives the required full scale emergency rate braking output. The jerk limit (K_{BJLP} in Figure 16) is also changed to the higher level required during emergency rate braking. The amplifiers respond by profiling their output to the full scale emergency level, at the specified jerk rate, and hold this level until a reset signal is received. The result is an open-loop constant braking force stop. Except for the change in inputs and jerk limits, operation of the brake system is the same as during normal operation.

The emergency deceleration requirement is to generate a nominal full scale braking force of 0.3 g under 0% grade, 30 mph tailwind and maximum weight conditions. The design task is to choose a full scale pressure level and corresponding amplifier output level which meets this requirement.

The required full scale braking force is computed by the equation

$$F_B = 1.06 \times \text{WEIGHT} \times \ddot{X}_g - 0.01 \times \text{WEIGHT} \pm 0.037\, V_{WIND}^{\;2} \qquad (5)$$

$$= 1.06 \times 11900 \times 0.3 - 0.01 \times 11900 + 0.037\, (30 \times \frac{88}{60})^2$$

$$= 3737 \text{ lbf},$$

where

WEIGHT = Vehicle weight in lbm

$\ddot{X}_g$ = Vehicle deceleration in g's

V_{WIND} = Wind velocity in fps.

The initial factor of 1.06 accounts for the momentum of rotating parts such as the motor armature and wheels. The corresponding full scale braking torque required is 4410 ft lb and is based on a nominal axle height of 1.18 ft.

Figure 17 shows the results of dynamometer testing of the final caliper design. This data illustrates 2 deficiencies observed in the caliper performance characteristics: the outboard and inboard system gains are not equal as called for in the original design; and the inboard system gain is lower than the minimum level specified. Rather than redesign the calipers to meet the requirements, the solution adopted was to compensate for the difference in gain by the choice of brake amplifier gains and corresponding full scale servo valve commands.

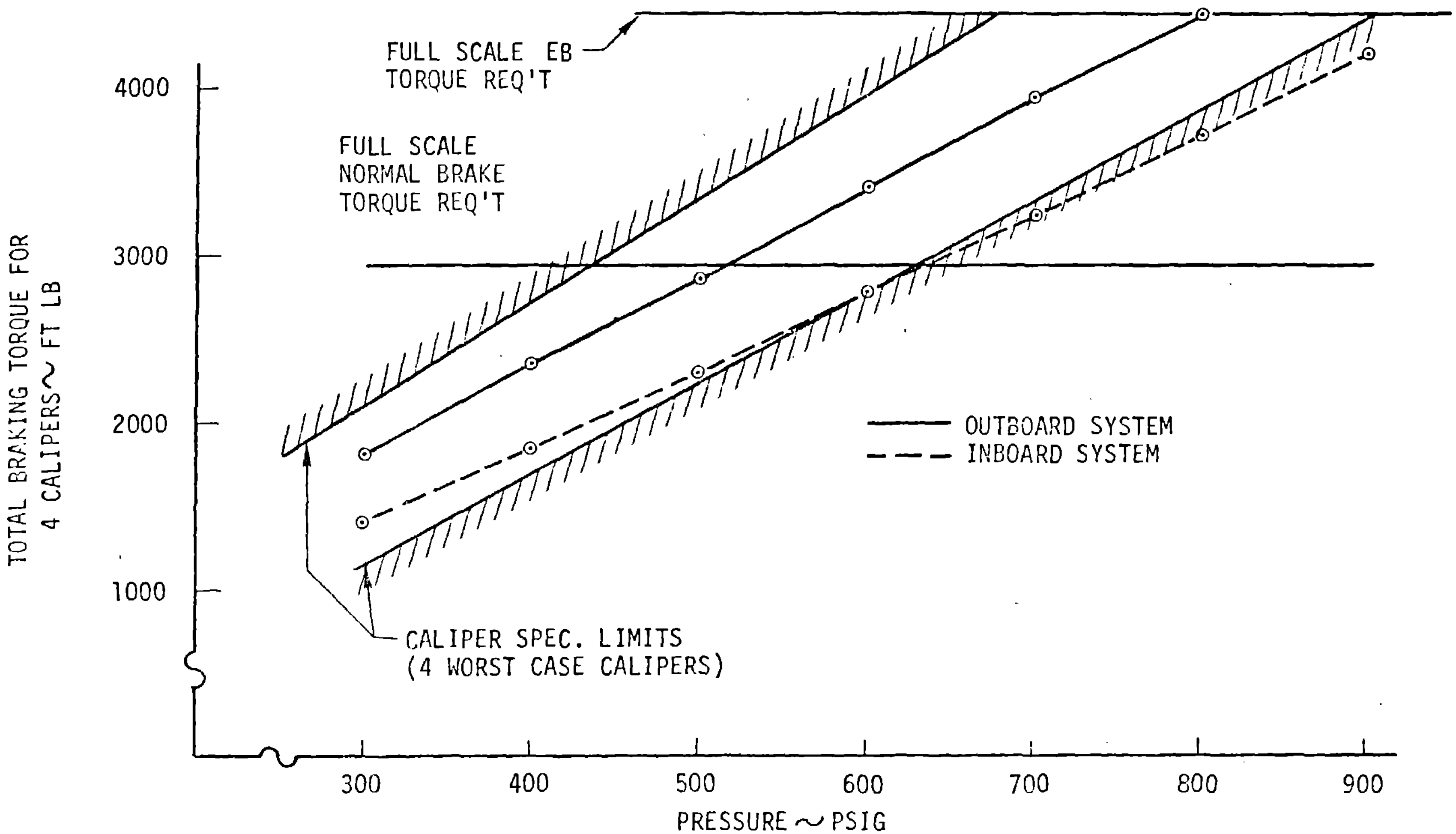


FIGURE 17. MEASURED BRAKE CALIPER PRESSURE - TORQUE GAIN

The full scale emergency braking pressure levels selected, based on the data of Figure 17, are:

- o 790 psig for the outboard system
- o 900 psig for the inboard system.

The 900 psig level, which is slightly lower than the level called for by Figure 17, reflects a measured characteristic of the calipers to produce slightly more torque when both systems are pressurized than when only one side is pressurized. This modification was not made on the outboard system as the inboard system is normally in control.

To reduce the impact of valve-to-valve variations, the full scale emergency rate amplifier outputs are specified in psig rather than ma and are set (after installation) to give the actual pressure outputs specified as measured by a pressure gauge.

Full Scale Normal Rate Braking Output

Deceleration during normal operation is performed in a closed-loop manner using the VCCS measured speed signal to close the loop. The brake system requirement is to provide sufficient braking force capability to follow a 2 fps^2 deceleration command ramp on a 10% downgrade. The derived requirement is that the brake system shall provide a minimum full scale deceleration capability of 0.2 g under 0% grade, 30 mph tailwind and maximum weight conditions. An 0.2 g force capability provides 1.2 fps^2 (0.0375 g) of reserve capability which is needed by the servo to compensate for initial dynamic response lags during a downspeed transition on a 10% downgrade.

The required minimum full scale braking force is 2475 lbf, based on Equation 5, with a corresponding torque requirement of 2922 ft lb. The full scale pressure levels required are:

- o 510 psig for the outboard system and
- o 635 psig for the inboard system

based on the measured caliper data of Figure 17.

The full scale amplifier outputs required to achieve the above pressure levels are

- o 23 ma for the outboard system and
- o 28.6 ma for the inboard system

and are computed by the equation

$$ISVC_{FS}^{NB} = HYSUL + (PSV_{FS}^{NB} - PSVLL)/K_{SVG}, \tag{6}$$

where

$HYSUL$ = Servo valve hysteresis
= 0.6 ma
PSV_{FS}^{NB} = Full scale normal rate braking pressure level
$PSVLL$ = Servo valve null offset
= 8.3 psig
K_{SVG} = Steady-state servo valve gain
= 22.36 psig/ma.

Tolerances on the full scale normal rate braking force are not critical and tuning of $ISVC_{FS}^{NB}$ by means of a pressure gauge is not required as was the case for the emergency braking full scale set point.

The above pressure settings resulted in a minor problem concerning the pressure switches used to verify redundancy when a vehicle is stopped and the pressures are at their full scale levels. These on/off switches are set to come on at a maximum pressure of 500 psig. A full scale outboard system pressure of 510 psig, which is lower than the original design value, means that normal parameter variations could result in a failure of one pressure switch to come on leading to unnecessary vehicle fault messages. The solution adopted is to raise each amplifier setting by 3.0 ma for a pressure increase of 67 psig. This avoided replacement of the pressure switches with no significant impact on system performance. The final full scale amplifier output settings resulting are:

- o 26 ma for the outboard system and
- o 31.6 ma for the inboard system.

Maximum Emergency Rate Braking "On" Rate

The emergency braking jerk requirement is to limit the average jerk due to braking to a maximum of 0.33 g/sec under any allowable operating conditions. After accounting for grade, weight and wind variations, the braking torque rate required is 3291 lbf/sec. The brake amplifier jerk limit or emergency braking "on" rate is computed by the equation

$$\dot{ISVC}_{EB} = (\dot{TB}_{EB}) (PSV_{FS}^{EB} - BCTH)/(TB_{FS}^{EB} \times K_{SVG}) , \quad (7)$$

where

$\dot{TB}_{EB}$ = Emergency rate braking torque rate limit
= 3291 ft lb/sec

PSV_{FS}^{EB} = Full scale emergency rate braking pressure level
= 790 psig for outboard system
= 900 psig for inboard system

TB_{FS}^{EB} = Full scale emergency rate braking torque level
= 4410 ft lb

BCTH and K_{SVG} are as previously defined.

The emergency braking "on" rate limits given by Equation 7 are:

- o 24.2 ma/sec for the outboard system, and
- o 27.4 ma/sec for the inboard system.

Maximum Normal Rate Braking "On" Rate

The long term jerk requirement during normal operation is to limit vehicle jerk to a maximum of $\pm$ 4.025 fps^3($\pm$ 0.125 g/sec). The resulting jerk limit under nominal conditions is $\pm$ 3.22 fps^3 ($\pm$ 0.1 g/sec). Table 8 summarizes the analysis conducted to verify that a nominal limit of 3.22 fps^3 meets the requirement of $\pm$ 4.025 fps^3.

TABLE 8. LONG TERM JERK DURING BRAKING

ERROR SOURCE PARAMETER	NOMINAL VALUE & UNITS	WORST CASE VALUE	INCREASE IN JERK (FPS^3)
Braker amplifier "on" rate limit	11.3 ma/sec	11.9	0.161
Servo valve gain	22.36 psig/ma	24.86	0.361
Brake caliper gain	4.65 ft lb/psig	5.35	0.483
Wheel radius (axle height)	1.18 ft	1.22	0.100
Vehicle weight	10325 lbm	8750	0.491

RSS Total	=	+0.800 fps^3
+ Nominal	=	+3.220 fps^3
= 3σ Jerk Limit		4.02 fps^3 (0.125 g/sec)

The braking torque rate required to achieve a nominal jerk level of 3.22 fps^3 is 1293 ft lb/sec. The brake amplifier jerk limit or normal braking "on" rate limit is computed by Equation 7 using normal rate braking rather than emergency rate braking parameter values. The resulting normal braking "on" rate limits are:

- o 8.8 ma/sec for the outboard system and
- o 11.3 ma/sec for the inboard system.

Brake Drag

The brake drag requirement is to limit the total braking torque (in the absence of a brake command) to a maximum of 36 ft lb. Analysis of brake drag is complicated by the nonlinear nature of the equations: which is largely due to the intentional caliper pressure threshold; and which makes the output of any conventional linearized sensitivity analysis meaningless. The analysis approach taken is to compute brake drag statistics using a Monte-Carlo approach where computer random number generation techniques are used to determine brake drag values for a large number of parameter value combinations.

The analytical model used in computing brake drag is given by the equation

$$TB_{DRAG} = K_{TB1}\left[PSVB1 + PSVC1 - BCTH1\right] \tag{8}$$

for all parameter value combinations where the outboard system is in control and the caliper threshold is exceeded

$$= K_{TB2F}\left[PSVB2 + PSVC2 - BCTH2F\right] + K_{TB2R}\left[PSVB2 + PSVC2 - BCTH2R\right]$$

for all parameter value combinations where the inboard system is in control and the caliper threshold is exceeded

$$= 0 \qquad \text{otherwise,}$$

where

K_{TB} = Brake caliper gain

PSVB = Servo valve control pressure bias level

PSVC = Servo valve control pressure due to VCCS brake command null offset

BCTH = Brake caliper pressure threshold

F, R refers to front and rear calipers, respectively

1, 2 refers to outboard and inboard pressure systems, respectively.

Servo valve control pressure due to VCCS brake command null offset is given by

$$PSVC = K_{SVG} \times 7 \times K_{BA} \times BC_0 , \tag{9}$$

where

K_{SVG} = Steady-state servo valve gain

K_{BA} = Brake amplifier steady-state gain

BC_0 = VCCS brake command null offset.

The individual parameter statistics used in the analysis are given in Table 9. Results of the analysis are shown in Figure 18. Interpreting the requirement as a 3σ limit means the probability of exceeding 36 ft lb of drag must not exceed 0.0027.

A major finding of the study is the sensitivity of brake drag to the maximum value of VCCS brake command null offsets. Figure 18 shows the results for null offset limits of 0.1 and 0.167 volts. Results were also computed for a null offset limit of 0.055 volts showing a negligible probability of incurring any brake drag.

The reason for the sensitivity to null offsets is a change in the brake command voltage required to place the amplifier output at the knee of the static compensation two slope gain curve. The original requirement was for the knee to occur at an input command of 0.25 ± 0.05 volts. This characteristic was achieved by adjusting the amplifier null output level as required. In the present design, the amplifier null adjustment is used to provide a positive servo valve bias. As a result, the input voltage at the knee of the curve is determined solely by the magnitude of the static compensation and is significantly less than 0.25 volts.

TABLE 9. MONTE-CARLO BRAKE DRAG ANALYSIS INPUTS

ERROR SOURCE PARAMETER *	NOMINAL VALUE & UNITS	LOWER 3σ LIMIT	UPPER 3σ LIMIT
BC_{01} , BC_{02}	0.0 v	- 0.167	+ 0.167
K_{BA1}	2.34 ma/v	2.22	2.46
K_{BA2}	2.90 ma/v	2.76	3.04
K_{SVG1} , K_{SVG2}	22.36 psig/ma	19.86	24.86
PSVB1	20.0 psig	0.0	45.0
PSVB2	35.0 psig	10.0	60.0
BCTH1	65.0 psig	50.0	80.0
BCTH2F	80.0 psig	65.0	95.0
BCTH2R	105.0 psig	90.0	120.0
K_{TB1}	6.6 ft lb/psig	5.3	7.9
K_{TB2F}	3.7 ft lb/psig	3.0	4.4
K_{TB2R}	1.6 ft lb/psig	1.3	1.9

* See text for definition of parameter symbols.

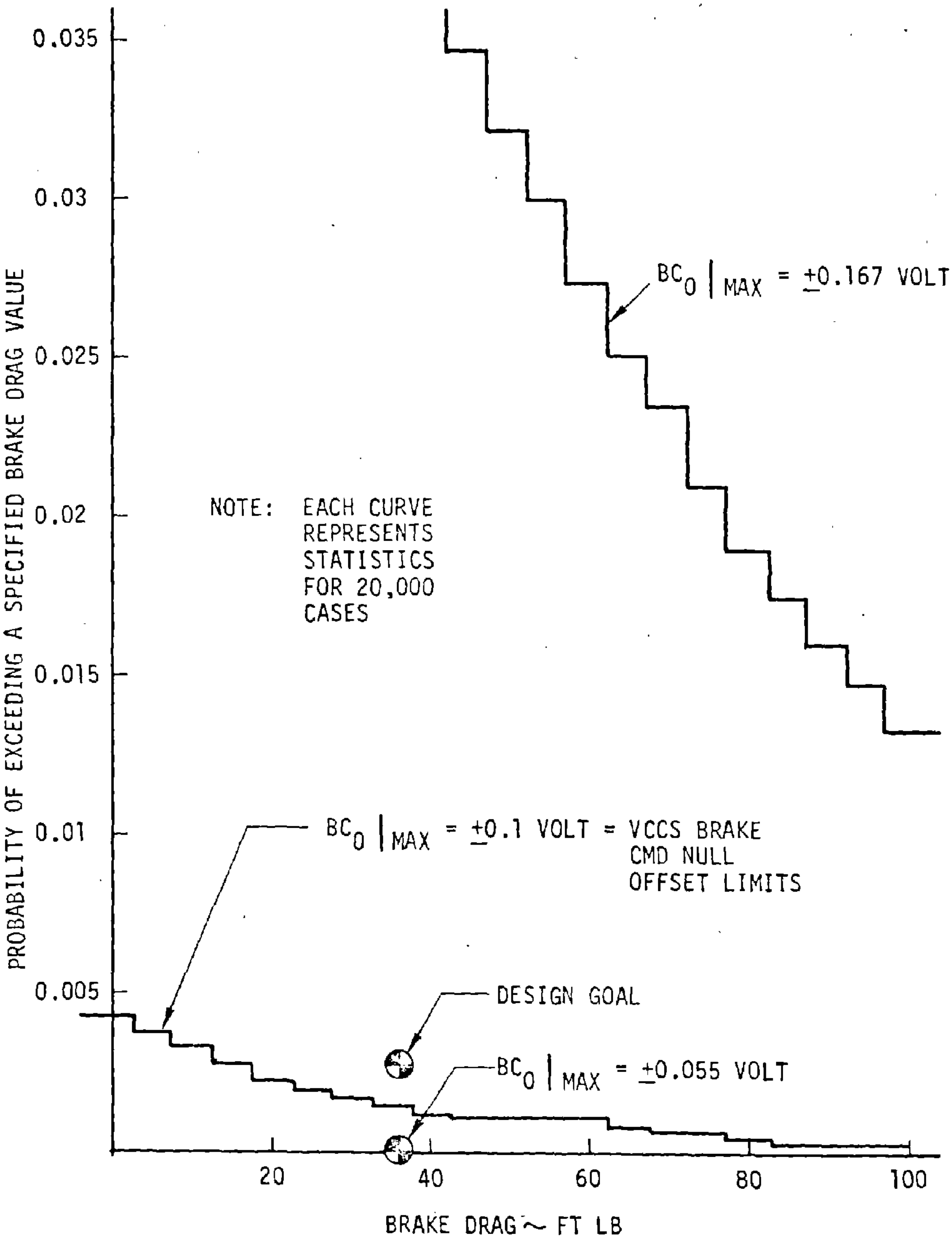


FIGURE 18. MONTE-CARLO BRAKE DRAG ANALYSIS RESULTS

The brake command null offset limits of $\pm$ 0.167 volts quoted in Table 9 are for VCCS temperature limits of 25 $\pm$ 60 degrees Centigrade and assume a failure of the vehicle Environmental Control Unit.

VCCS temperature limits under normal operation are 25 $\pm$ 20 degrees Centigrade with corresponding estimated null offset limits of $\pm$ 0.055 volts. The analysis shows, therefore, that the brake drag requirement is met except during extreme anomaly conditions.

4.5 Speed Control (Overspeed/Underspeed)

The speed control requirement is to maintain overspeeds within 3 fps and underspeeds within 4 fps of the acceleration limited speed command during normal operation. Overspeeds greater than 3 fps result in an automatic emergency rate stop. This response requirement is a result of the sensitivity of safe stopping distance to overspeed errors. Underspeeds greater than 4 fps, which indicate a sick vehicle, result in a downlink fault message and removal of the vehicle from service at the next opportunity. The speed control requirement is of concern primarily during speed and grade transients, as the regulation or position control requirement limits speed errors in steady-state operation to a much lower value. The problem is to insure that transient conditions do not trip the overspeed or underspeed limits.

A worst case test of speed control capability, based on Phase IA experience, is the Beechurst underpass segment of the Morgantown guideway, which contains a rapid change in grade at a speed of 22 fps. The grade profile consists of an 80 foot 10 percent downgrade followed by an 80 foot 10 percent upgrade, with grade changes made at a rate of 0.167 percent per foot. Total length of the grade profile is 400 feet.

A detailed sensitivity analysis of speed control performance through the Beechurst underpass was conducted using the nonlinear computer simulation. The parameter variations considered and the impact of each are given in Table 10. 3σ limits for both overspeed and underspeed are 1.98 fps, well within the requirements.

The above analysis assumes a limit of 0.2 second on brake system time delays, the original design requirement. Actual delays are significantly in excess of 0.2 second, due to limited servo valve flow capability relative to caliper flow requirements. Following is a description of an analysis conducted to determine the impact on vehicle overspeed control.

The first step involved in determining the impact of large time delays is to estimate the maximum value of the expected delays. Table 11 shows the results of a brake time delay sensitivity analysis. Estimated limits on time delay are 0.312 second during normal operation or for the case where the outboard system is inactive and 0.586 second for the case where the inboard system is inactive. Delays are larger when the inboard system is inactive for two basic reasons:

- Lower outboard system amplifier gains increase the time required to reach the caliper threshold; and

- Larger caliper volumes on the outboard system side increase flow requirements and magnify the impact of servo valve flow limits.

TABLE 10. SPEED CONTROL SENSITIVITY ANALYSIS
(Beechurst Underpass Simulation Results)

ERROR SOURCE PARAMETER	NOMINAL VALUE & UNITS	WORST CASE VALUE	INCREASE IN MAXIMUM OVERSPEED (FPS)	INCREASE IN MAXIMUM UNDERSPEED (FPS)
VCCS tachometer/odometer scale factor	0.1675 ft/rad	0.169 max 0.166 min	0.32	 0.12
VCCS brake command null offset	0.0 v	-0.22	0.09	0.08
VCCS motor command null offset	0.0 v	-0.1	0.15	0.20
Magnitude of brake amplifier static compensation	2.6 ma	1.3	0.14	0.17
Servo valve gain	22.36 psig/ma	20.5	0.14	0.12
Brake caliper pressure threshold	40.0 psig	50.0	0.10	0.09
Brake caliper gain	5.68 ft lb/psig	5.12	0.14	0.09
Motor tachometer scale factor	0.025 v/rad/sec	0.02512	0.04	0.04
Motor integral compensation lead term	1.25 sec	0.5	0.0	0.38
Rolling resistance	200 lbf	100	0.14	0.12
Vehicle weight	10325 lbm	11900	0.20	0.22
		RSS Total =	0.51 fps	0.57 fps
		+ Nominal =	1.47 fps	1.41 fps
		= 3σ Limit	1.98 fps	1.98 fps

TABLE 11. BRAKE TIME DELAY SENSITIVITY ANALYSIS
(Normal Rate Braking Mode)

ERROR SOURCE PARAMETER	UNITS	NOMINAL VALUE		WORST CASE VALUE		INCREASE IN BRAKE TIME DELAY (SEC)	
		#1	#2	#1	#2	#1	#2
VCCS brake command null offset	v	0.0	0.0	-0.167	-0.167	0.049	0.047
Brake amplifier static compensation break point	v	0.143	0.118	0.193	0.168	0.015	0.014
Brake amplifier "on" rate limit	ma/sec	8.8	11.3	8.36	10.7	0.002	0.002
Servo valve hysteresis	ma	0.6	0.6	1.2	1.2	0.068	0.053
Servo valve control pressure bias level	psig	20	35	0	10	0.102	0.099
Servo valve gain	psig/ma	22.36	22.36	19.86	19.86	0.029	0.022
Hydraulic system dynamic response lag	sec	0.3	0.1	0.5	0.2	0.200	0.100
Brake caliper pressure threshold	psig	65	80	80	95	0.076	0.059

RSS Total =	0.253 second	0.170 second
+ Nominal =	0.333 second	0.142 second
= 3σ Limit	0.586 second	0.312 second

NOTES:

1) Column#1 shows results when inboard system is inactive.

2) Column #2 shows results when outboard system is inactive; and also applies for normal case where both systems are active.

The second step consisted of determining the impact of large time delays via the nonlinear computer simulation for the Beechurst underpass case and for a 33-22 fps downspeed transition. Runs were made using a nominal time delay value of 0.1 second and a near worst case value of 0.5 second. Other parameter values were held at nominal values. The impact of a 0.5 second delay is an increase of 0.5 fps in maximum overspeed relative to the nominal value. Extrapolating, a worst case delay of 0.586 second would increase maximum overspeed by 0.73 fps. Adding this 0.73 fps to the original estimate of 1.98 fps gives a revised overspeed limit of 2.7 fps or a design margin of 0.3 fps.

Design verification test data taken at STTF during regulation testing shows that, with two exceptions, speed errors are less than $\pm$ 2.0 fps. The first exception is on station starts where underspeeds are typically on the order of 2.5 fps. The reason is the VCCS/brake/motor phasing used during startup and described in Section 4.3. The 2.5 fps underspeed is larger than the analysis 3σ prediction of 1.98 fps, which did not consider a station start situation, but still well below the 4.0 fps requirement. The second exception is on the first two runs following a change from minimum to maximum vehicle weight. An overspeed of just under the limit of 3.0 fps and an overspeed of 2.4 fps occurred at the start of the 44-33 fps downspeed transition. The reason for these high overspeeds are a combination of a position correction, a major calibration update and the start of a speed transition all occurring at essentially the same point in time. The severity of this inadvertent test of speed control, which is not a normal condition at Morgantown, is a good indication of the system's capability to meet the overspeed requirement.

4.6 Acceleration and Jerk Control

All acceleration and deceleration ramps are designed on the basis of nominal acceleration/deceleration and jerk levels of 2.0 fps^2 and 3.22 fps^3, respectively. In addition to these nominal values, limits are given on peak acceleration/deceleration and jerk to insure passenger ride comfort. The specific requirements are to:

- Limit peak acceleration/deceleration to a maximum of 4.4 fps^2 (0.137 g),
- Limit peak long term jerk to a maximum of $\pm$ 4.025 fps^3 ($\pm$ 0.125 g/sec) where long term jerk is defined as the average over any time interval of 0.2 second or greater, and
- Limit peak short term jerk (average over any 0.1 second interval) to a maximum of $\pm$ 8.05 fps^3 ($\pm$ 0.25 g/sec).

Control of peak acceleration is accomplished by setting the acceleration limit in the propulsion system jerk and acceleration limiter (See Figure 2) at 3.83 rev/sec^2 which gives a nominal limit on acceleration of 4.0 fps^2. In normal operation this limit will not be reached, as the VCCS speed and position loops will maintain the rate of change (acceleration) of the motor speed command close to the nominal value of 2.0 fps^2 during speed transitions.

Control of peak deceleration during use of the brakes is provided by the dynamics of the speed and position loops used. Specifically, the dynamics of the control system used during braking can be characterized by a second-order system having a natural frequency of 0.286 rad/sec and a damping ratio of 2.45. The overdamped characteristic of the dynamics result in peak deceleration values close to the nominal commanded value of 2.0 fps^2.

Control of peak long term jerk is accomplished by setting the jerk limit in the propulsion system jerk and acceleration limiter and in the brake system jerk limiter at 3.22 fps^3 (0.1 g/sec) under nominal conditions. As shown in Table 8 (Section 4.4), a nominal brake system jerk limiter setting of 3.22 fps^3 gives a 3σ vehicle long term jerk limit of 4.02 fps^3 which is equal to the allowed maximum. Peak long term jerk when using the motor will be less due to the speed and current feedback loops which reduce variations downstream of the limiter.

Short term jerk is a problem primarily during transitions between braking and use of the motor where nonlinearities and differences in the dynamics of the control systems used have the greatest impact. Table 12 summarizes the results of a computer simulation study to determine a 3σ limit on short term jerk. These results are for the Beechurst underpass segment of the Morgantown guideway where the grade profile requires a transition from use of the motor to use of the brakes and back to use of the motor. The estimated 3σ limit of 5.08 fps^3 (0.158 g/sec) is significantly below the requirement of 8.05 fps^3.

An analysis of test data taken at STTF gave the measured range of maximum jerk levels shown in Figure 19. As shown, the measured peak jerk levels exceed both the requirement and analysis predictions by a substantial amount. The major reason for the high jerk levels is the servo valve flow limit problem discussed in Section 4.4. The analyses performed prior to STTF testing used the brake system model shown in Figure 16 which did not account for fluid flow dynamics and assumed the servo valve would meet its requirements and maintain commanded pressure independent of actual flow levels. The final brake system model, developed after the completion of STTF testing, illustrates the level of detail required to adequately simulate the nonlinearities of the as-built brake system and to accurately predict actual vehicle jerk performance.

TABLE 12. SHORT TERM JERK SENSITIVITY ANALYSIS
(Beechurst Underpass Simulation Results)

ERROR SOURCE PARAMETER	NOMINAL VALUE & UNITS	WORST CASE VALUE	INCREASE IN MAXIMUM JERK (FPS^3)
VCCS tachometer/odometer scale factor	0.1675 ft/rad	0.166	0.193
VCCS brake command null offset	0.0 v	-0.22	0.129
VCCS motor command null offset	0.0 v	-0.1	0.483
Magnitude of brake amplifier static compensation	2.6 ma	1.3	0.483
Servo valve gain	22.36 psig/ma	20.5	0.129
Brake caliper pressure threshold	40.0 psig	50.0	0.161
Brake caliper gain	5.68 ft lb/psig	5.12	0.129
Motor tachometer scale factor	0.025 v/rad/sec	0.02512	0.097
Motor integral compensation lead term	1.25 sec	2.5	1.51
Rolling resistance	200 lbf	100	0.097
Vehicle weight	10325 lbm	8750	0.322

RSS Total = 1.727 fps^3
+ Nominal = 3.349 fps^3

= 3σ Limit 5.076 fps^3
(0.158 f/sec)

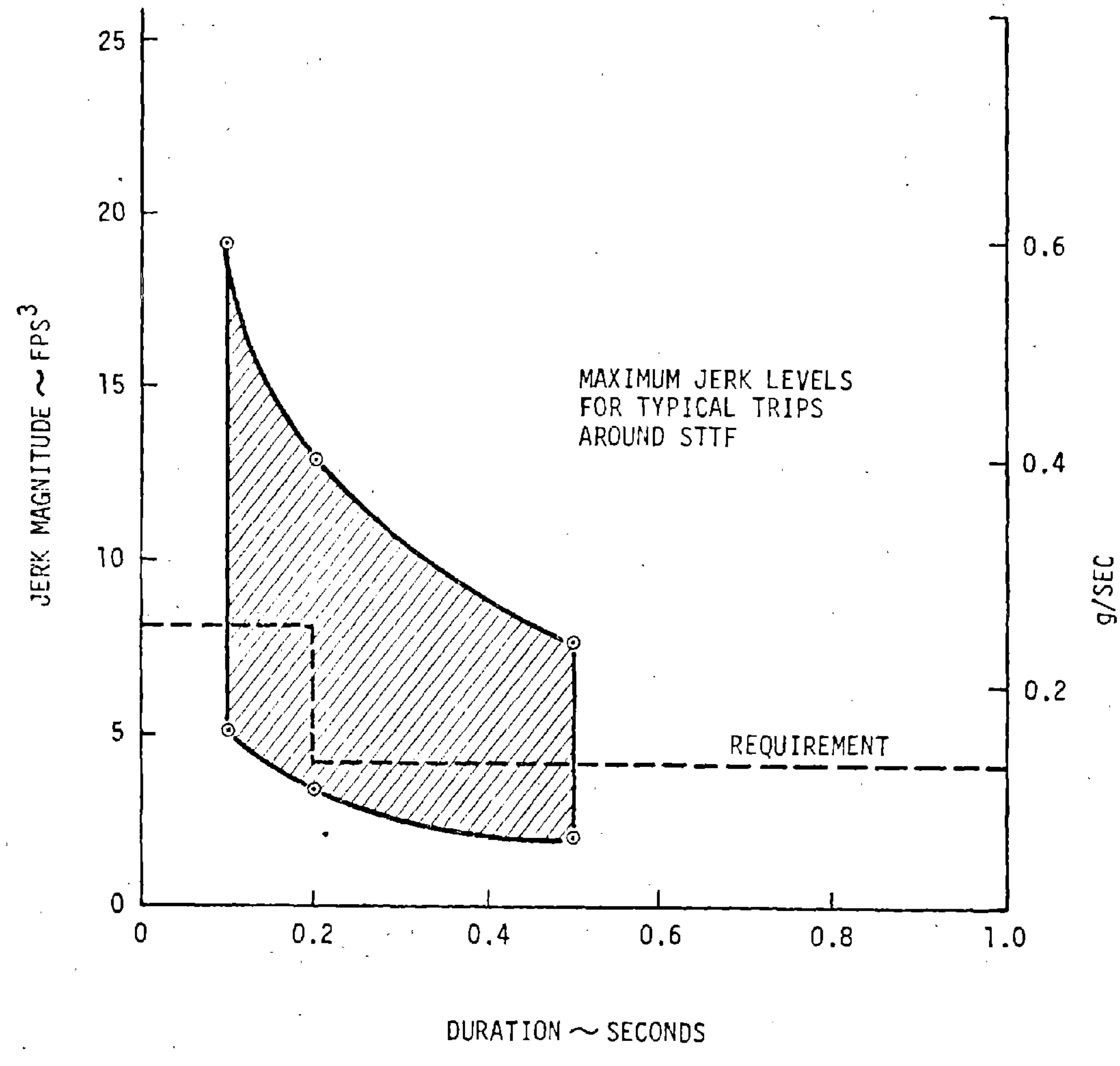


FIGURE 19. MEASURED VEHICLE JERK PERFORMANCE

Excessive jerk levels are caused by the servo valve sensitivity to flow levels in the following manner. During a typical brake application the servo valve command is a ramp command having a slope equal to the nominal jerk limit amplifier setting. The servo valve responds by initially increasing its output pressure to a level just under the caliper threshold. The pressure remains at this level for 0.1 to 0.4 second, as the primary caliper piston moves toward the brake rotor, rather than following the commanded pressure ramp. The reason is the inability of the servo valve to maintain commanded pressure during the fluid flow condition caused by the caliper piston motion. When the brake pad makes contact with the rotor, piston motion and fluid flow levels are reduced to negligible levels and the control pressure quickly rises to the commanded level at a rate significantly above the commanded jerk rate. It is this final pressure transient and the resulting brake force transient which is the primary cause of the excessive jerk levels shown in Figure 19.

The above results illustrate the sensitivity of short term vehicle jerk to component nonlinearities. Because of this sensitivity, very detailed analytical models along with corresponding data on detailed hardware performance characteristics are required to accurately predict actual peak jerk levels.

An interesting fact is that the peak jerk levels of Figure 19 are, at worst, barely noticeable by passengers and do not significantly impact ride quality. This conclusion is based on observations by numerous project engineers who tend to be more critical of vehicle performance than would the general public. The conclusion is that the impact of high jerk levels is strongly dependent on their duration and that research is needed in the area of deriving realistic specifications on vehicle jerk.

Since reduction of the high jerk levels required major hardware modifications and since these levels do not significantly impact ride quality, a decision was made to accept the as-built final design with no further modifications.

4.7 Propulsion Stability Studies

Initial testing of the first complete Phase IB vehicle uncovered a propulsion speed loop instability with a frequency of oscillation close to the computed natural driveline frequency of 38.7 rad/sec (6.2 Hz). A factor of 10 reduction in loop gain, from the level required to meet performance requirements, was required to regain stability and allow initial vehicle movement under manual control. Analysis of preliminary test data identified the problem as a classical flexible-body stability problem caused by the dynamics or flexibility of the driveline.

A simplified block diagram of the propulsion speed control loop is given in Figure 20 and shows how the dynamics of the driveline affect operation of the control loop. Modeling driveline dynamics by single stiffness and damping parameters leads to the second-order transfer function shown. The effect of this transfer function is to introduce a large peak in loop gain at the natural frequency of the driveline. Because of higher phase lags in the electronics and an increase of the driveline natural frequency relative to the Phase IA design, total phase lag in the frequency range of interest exceeds 180 degrees resulting in the observed divergent oscillation or instability.

The failure of earlier propulsion system testing to uncover the stability problem is a consequence of using test loads with significantly different dynamic characteristics from those seen in actual operation. Simulation studies also failed to predict the problem due to insufficient data on actual hardware phase lag characteristics. The current rate limiter, in particular, exhibits nonlinear operation and significantly higher phase lags than expected due to tachometer pulse ripple originating at the output of the D/A converter.

The first step taken in deriving a solution to the stability problem was to concurrently obtain time history test data on all key motor control system variables under various conditions and to update the nonlinear analytical

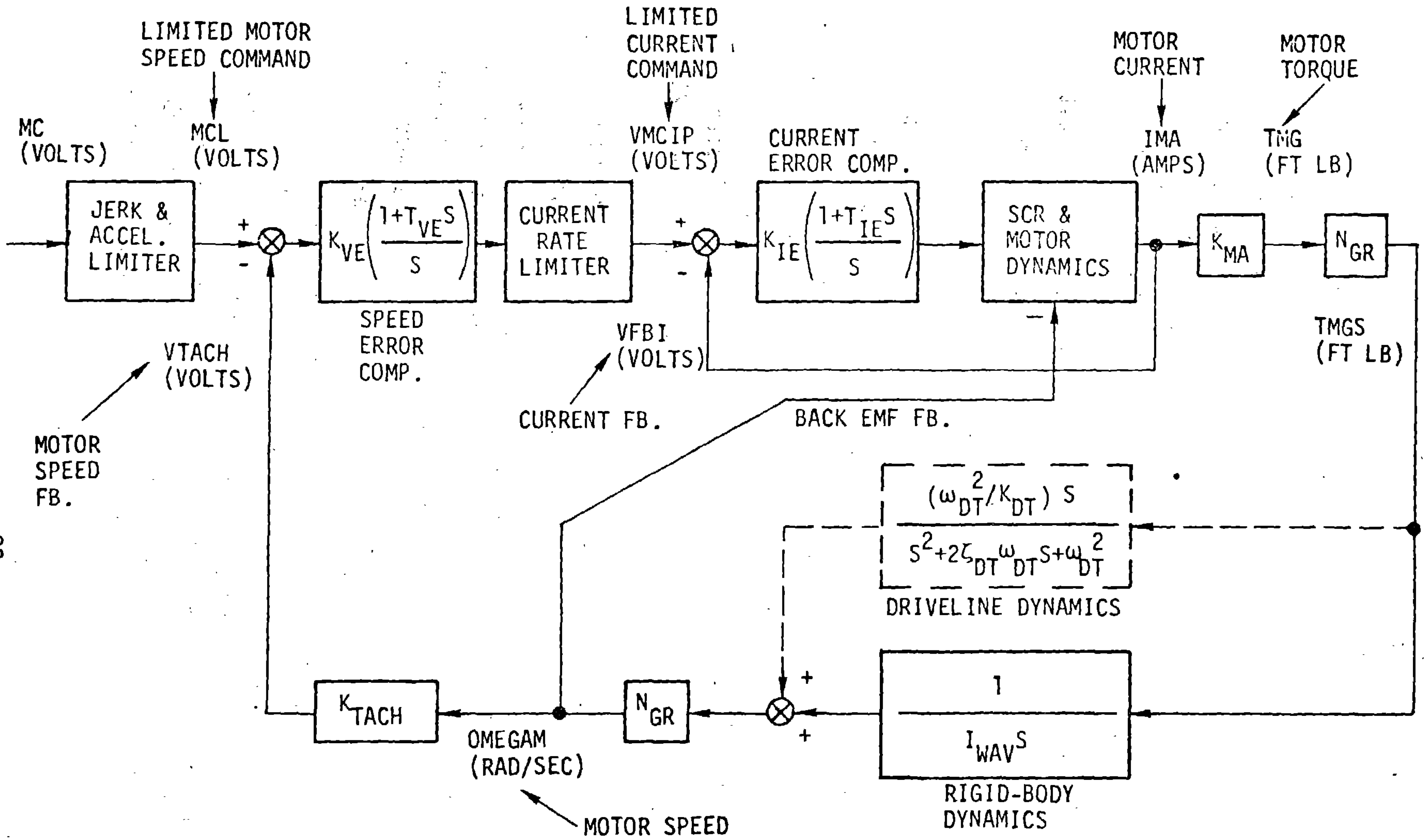
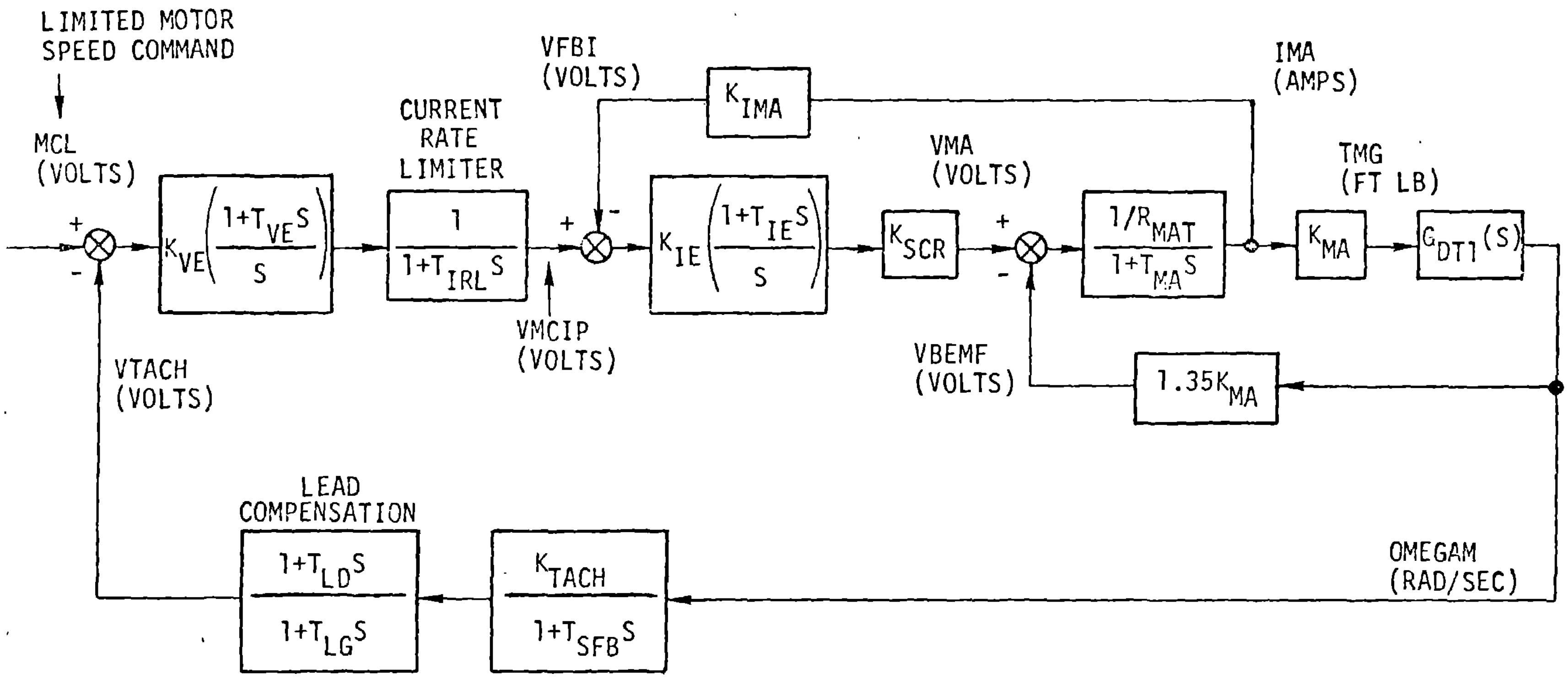


FIGURE 20. PROPULSION SPEED CONTROL LOOP BLOCK DIAGRAM

model to agree with measured characteristics. The specific test data case selected for the analytical model verification effort is a case where the motor speed loop oscillation is excited by external forces and decays very slowly. Gains selected for this test case of close to neutral stability give K_{VE} and T_{VE} values of 4 sec^{-1} and 1.25 seconds, respectively. Corresponding design values are 14.7 sec^{-1} and 1.496 seconds. The frequency of the measured oscillation is 7.12 Hz and the damping or time to reach half amplitude is 2.1 seconds or 15 cycles. Comparable simulation results were subsequently obtained with the simulation giving an oscillation of 7.0 Hz and a 40% reduction in oscillation magnitude after 2.1 seconds. Matching test data results required numerous parameter value changes but no major changes in the model structure, which is described in Section 5.

A linearized model of the updated nonlinear model was subsequently developed to allow use of classical linear design techniques. The analysis approach taken is to define potential parameter value changes and compensation circuits on the basis of the linear model and to verify their effect on system operation using the nonlinear simulation. Figure 21 shows the linear model developed and represents the final design configuration. Speed loop gain and phase characteristics of the final design are shown in Figure 22.

Both gain and phase stabilization approaches were considered. The objective of gain stabilization is to attenuate the gain peak shown in Figure 22 via compensation networks such that the loop gain at the driveline frequency never exceeds 0 db under worst case conditions. Attenuating gain via first order lag filters was rejected because of the adverse effect on control system performance,i.e., such filters also affect gain and phase in the frequency region of interest. Use of notch filters, which reduce gain only near a specified frequency, was rejected because of concern over variations in driveline frequency between vehicles. A third approach, which will be discussed later, is a change in tachometer location. The objective of phase stabilization, which is the approach selected, is to maintain total loop phase lag below 180 degrees for all frequencies where the gain exceeds 0 db. In this manner, a negative feedback situation is assured. Phase stabilization



$$G_{DT1}(S) = \frac{N_{GR}^{2}(I_{WAV}S^{2}+C_{DT}S+K_{DT})}{S\left[I_{MR}I_{WAV}S^{2}+C_{DT}(I_{MR}+I_{WAV})S+K_{DT}(I_{MR}+I_{WAV})\right]}$$

$$= \frac{3.56(S^{2}+0.024S+6.826^{2})}{S(S^{2}+0.8S+39.27^{2})}$$

FIGURE 21. LINEAR MODEL OF FINAL PROPULSION SPEED LOOP DESIGN

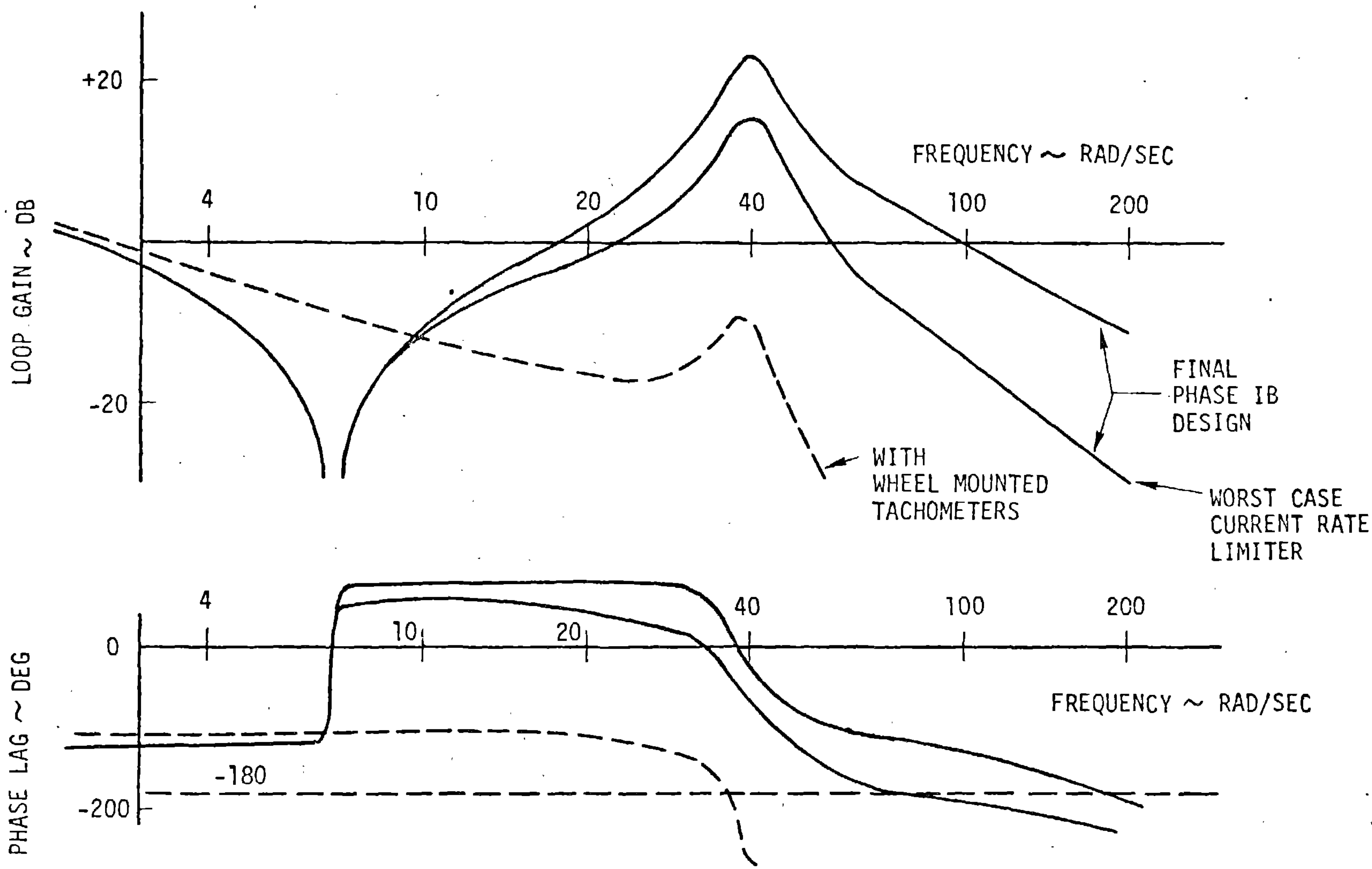


FIGURE 22. MOTOR SPEED LOOP GAIN AND PHASE CHARACTERISTICS

of the speed loop required use of the lead compensation shown in Figure 22 as well as a number of control circuit parameter value changes. Each of the changes made are discussed in detail in the following paragraphs.

Test results, with the speed feedback loop open, showed the current control loop to have negligible damping in the frequency range of interest. Two changes were made to improve current loop dynamic characteristics. The first change was removal of a positive feedback loop from measured back emf to SCR firing angle command. The purpose of this feedback, included by the supplier, is to maintain firing angle or current command close to nominal values during a power interrupt condition and, thereby, minimize vehicle jerk due to the transient caused by reapplication of power. During normal operation, the positive feedback has the adverse effect of effectively canceling any inherent damping provided by the internal back emf feedback within the motor and, thereby, adding to the speed loop stability problem. The second modification made, to reduce current loop phase lag, was to replace the original K_{IE} and T_{IE} values of 10.0 sec^{-1} and 0.0056 second with values of 2.5 sec^{-1} and 0.08 second.

An important characteristic of the current control loop is the fact that large nonlinear variations in the gain K_{SCR} (Figure 21) occur as a function of current level and speed. Low values of K_{SCR} result in the largest current control loop phase lags and are the critical case for stability. The minimum K_{SCR} value, and the worst case for stability, occurs at speeds of 4 to 8 fps with current levels around 30 amps. The gain/phase characteristics shown in Figure 22 are for a worst case value of K_{SCR}.

The function of the current rate limiter is to limit current rates to a maximum value of 1400 amps/sec. The reason for this function is to prevent commutator arcing and a corresponding potential fire hazard. This function is included in the Phase IB design and not in the Phase IA design due to the higher operating voltage level of the Phase IB propulsion system. The impact of the current rate limiter on stability is to introduce significant phase lags under some conditions.

Design of the current rate limiter is essentially identical to the brake amplifier jerk limiter design illustrated in Figure 16. It is a high gain first order feedback circuit with a nonlinear clamp on output rate. For inputs with rates less than the specified maximum, the circuit behaves as a first order filter with the transfer function shown in Figure 21. For inputs with rates greater than the specified maximum, the output is a ramp having a slope equal to the specified maximum. The problem with this circuit is its susceptibility to high frequency noise, specifically ripple from the tachometer D/A converter. The high input rates associated with the noise result in limiter saturation such that its response to low frequency signals is similar to that of a low frequency filter. Attenuations of up to 50 percent and phase lags of up to 90 degrees have been observed in test data for frequencies near that of the driveline. The loop gain of the limiter circuit was reduced by a factor of 8.3 from its original value to reduce its sensitivity to noise. To further insure stability, an overall speed loop design has been selected which gives positive stability margins for a worst case current rate limiter characteristic. The lower of the gain/phase curves shown in Figure 22 are for this worst case limiter condition.

The digital tachometer pulses are converted to an analog speed signal by passing the pulses through a first order filter with the time constant, T_{SFB}, shown in Figure 21. A large value of T_{SFB} reduces tachometer pulse ripple on the measured speed signal and, thereby, reduces current rate limiter phase lags, but increases the phase lag of the D/A converter. A low value of T_{SFB} reduces phase lags of the converter but increases limiter phase lag. The optimum value for T_{SFB}, as determined by a trial and error test procedure, is 0.011 second and is a change from the original design value of 0.0052 second.

The final modification to the original design is the addition of the lead compensation shown in Figure 21. The function of this lead/lag circuit is to reduce phase lag at frequencies between 40 and 100 rad/sec where the loop gain falls below 0 db for the final time. With the lead compensation, the design provides 6 db of gain margin and 20 degrees of phase margin under worst case conditions. Without the lead circuit, a small negative (unstable) phase margin condition would occur. Under most operating conditions, phase margins of the final design meet or exceed the design goal of 45 degrees.

Design verification tests included a gain margin check where the loop gain was set at twice its design value. The analysis results of Figure 22 predict that this 6 db gain increase will result in a condition just short of neutral stability for worst case track conditions. Test data, showing lightly dampened oscillations around 55 rad/sec (8.6 Hz), confirmed this prediction.

The lead compensation required to insure stability results in a minor side effect at speeds of 4 fps or less. Because lead compensation is a differentiating process, the circuit amplifies tachometer pulse ripple at low speeds where the ripple has the largest magnitude. The speed compensation circuit includes a clamp on negative current commands, allowing only positive current commands at the output. This clamp, in conjunction with the amplified tachometer pulse ripple, results in an increase in average current command or bias level over that required to maintain constant speed. The result is a brake/motor interaction problem due to the high current command which causes the vehicle to run faster than the commanded speed. Since constant low speed operation of 4 fps or less is not a normal situation at Morgantown, this side effect does not significantly impact system operation.

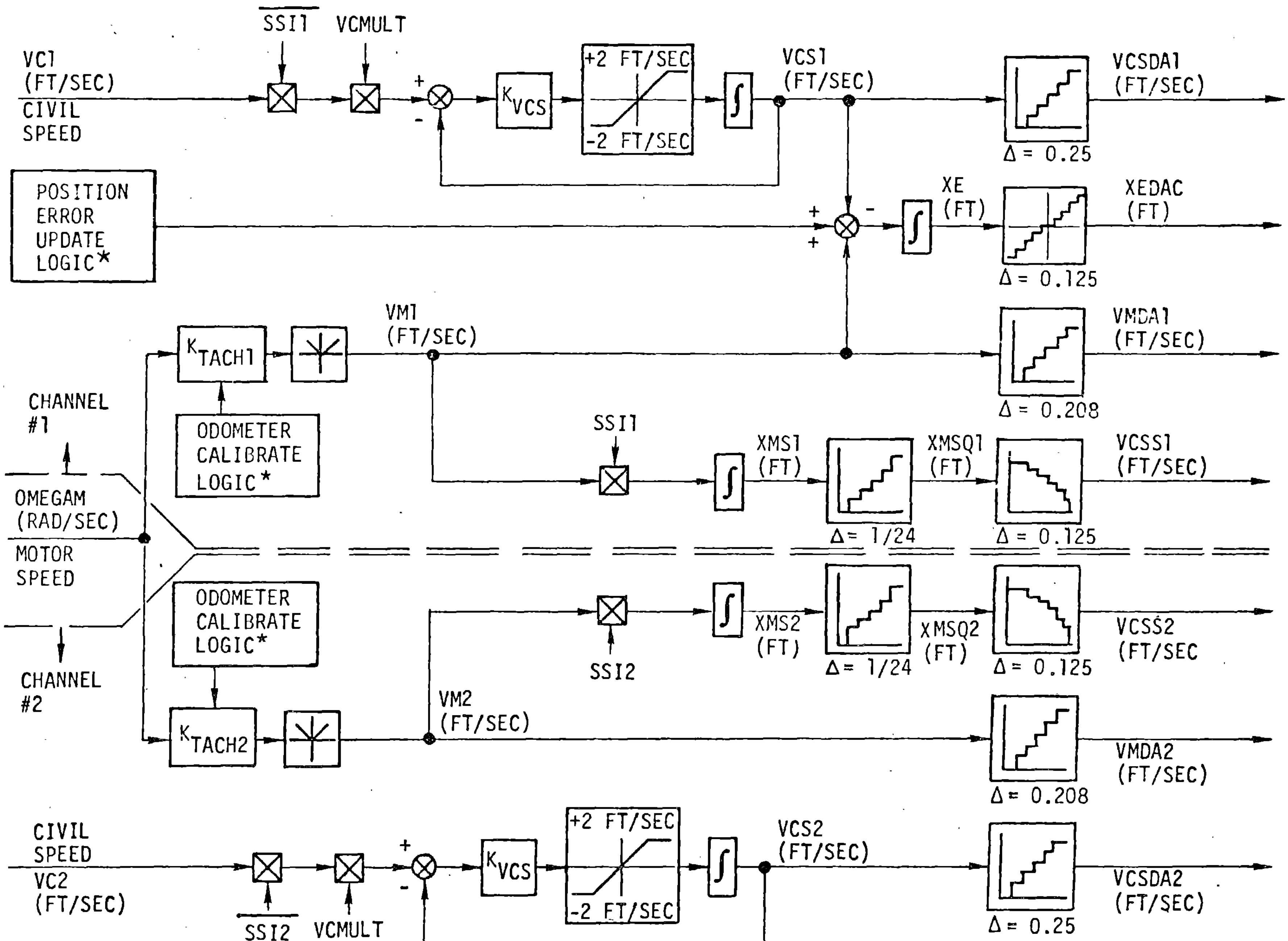
An alternative stabilization approach, which was studied but rejected on the basis of cost, is to move the motor tachometer from its motor shaft location to a wheel mounted location. (Regarding costs, the phase stabilization solution adopted has the advantage of requiring only minor changes to existing circuitry such as changes in existing resistor and capacitor values.) A key aspect of the stability problem is that motion due to driveline oscillations is greatest at the motor shaft where the tachometers are located. Mounting the tachometer at the wheels results in a significant reduction (30 db) in loop gain at the driveline frequency due to the reduction in actual oscillation magnitude. Loop gain and phase characteristics for a wheel-mounted tachometer are shown in Figure 22. The change in tachometer location allows use of gain stabilization techniques and eliminates the need for careful control of phase lags, along with the associated problems due to tachometer pulse ripple. The conclusion is that wheel-mounted tachometers should be used in any new design, to minimize the impact of driveline dynamics.

5. ANALYTICAL MODEL

A detailed nonlinear analytical model was developed as a result of the LCS design effort. This section, in conjunction with Appendix B, describes the final model configuration in sufficient detail to allow use of all or portions of the model in the analysis and evaluation of future system designs. Included in this section are: a block diagram description of the model (Figures 23 through 32); definition of the variables used (Table 13); and a list of nominal parameter values along with estimated limits on parameter value variations where available (Tables 14 through 17). For easier reference, Tables 13 through 17 are presented together at the end of this section. Appendix B provides a listing of the problem dependent Fortran code used in the digital computer simulation of the analytical model. The coding supplied can be used to implement the model in a variety of general purpose simulation programs.

5.1 Vehicle Control and Communications System

Figures 23 and 24 are a block diagram description of the VCCS model. Figure 23 represents the digital portion of the VCCS and Figure 24 represents primarily the analog portion. In Figure 23, VC1 and VC2 are the redundant discrete civil speed commands received from the guideway. VCMULT is the performance level multiplier where a value of 1.0 represents a 100 percent performance level command. Analog models of the digital acceleration limiters are used to obtain the acceleration limited speed commands, VCS1 and VCS2, which define the commanded vehicle speed-position-time point follower trajectory. The tachometers and the VCCS tachometer output processing circuitry are modeled by simple gains in conjunction with an absolute value function giving the redundant measured speed outputs, VM1 and VM2. Position error, XE, is obtained in the model by integrating speed error and is single thread as is the case in the VCCS hardware. The parameters SSI1 and SSI2 are switching variables having a value of 0 or 1 and illustrate the actual switching which takes place during a station stop



*NOT INCLUDED IN SIMULATION

FIGURE 23. PHASE IB VCCS ANALYTICAL MODEL (PART 1)

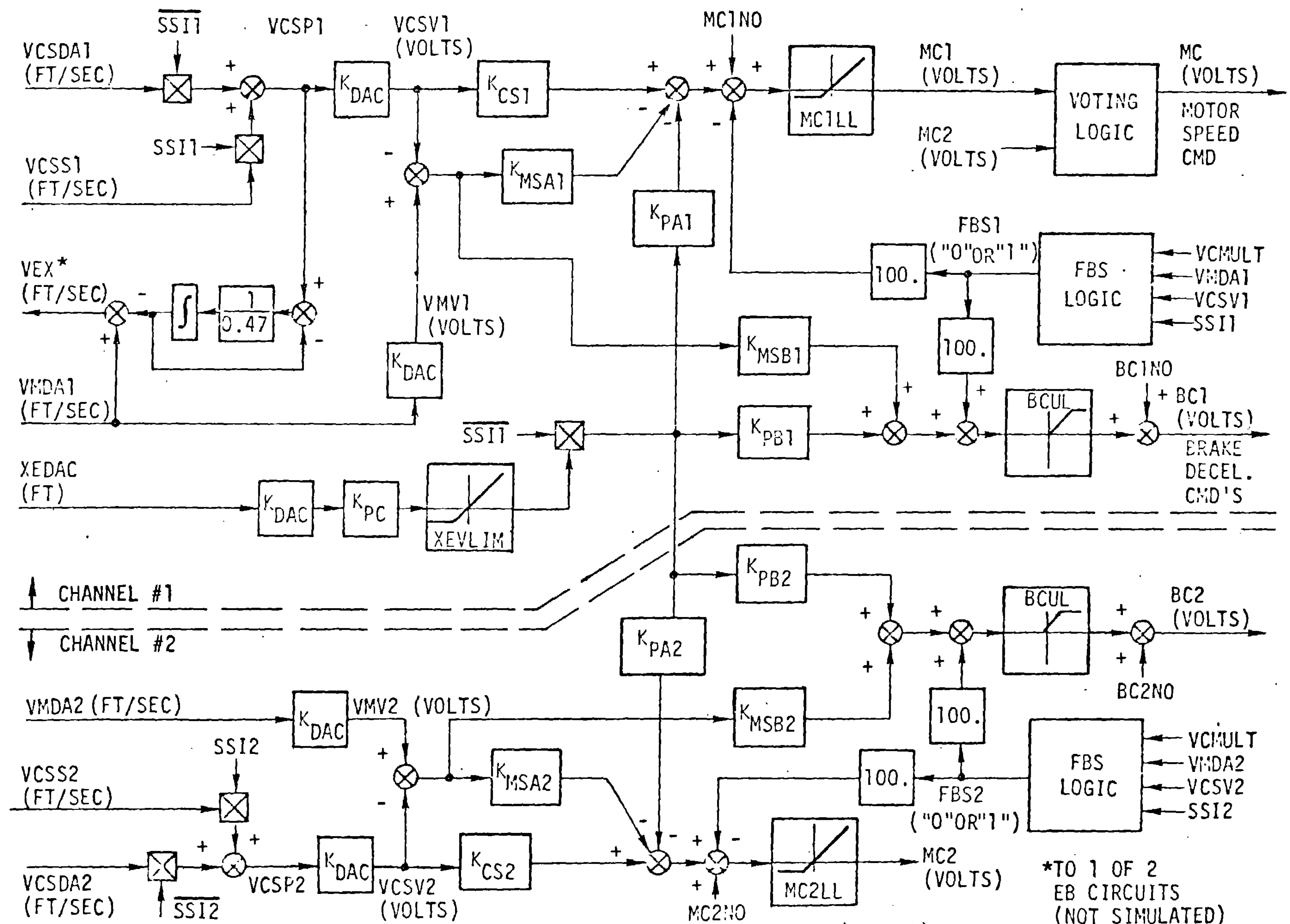


FIGURE 24. PHASE IB VCCS ANALYTICAL MODEL (PART 2)

sequence. A table lookup block is used to generate the position dependent station stop speed command profile as is the case in the actual hardware. VCCS quantization is modeled by truncating the outputs on the right of Figure 23 to the next lowest quantum level. The value of one VCCS least significant bit is shown under each quantization block.

The D/A converters are modeled by the gain parameter, K_{DAC}, as shown in Figure 24. A total of five D/A converters are used. The blocks downstream of the converters represent the analog VCCS implementation of the separate control laws used to generate the motor speed command, MC, and the redundant brake commands, BC1 and BC2. The function of the position error clamp, XEVLIM, is to minimize the possibility of a large negative (behind point) error causing a 3 fps overspeed and an emergency stop. Output null offsets are modeled by the parameters MC1NO, MC2NO, BC1NO and BC2NO. MC1LL and MC2LL are lower motor speed command limits with BCUL being the upper full scale brake command limit. FBS1 and FBS2 are the redundant Forced Brake logic signals which generate the full scale brake commands used to hold a vehicle in place when stopped. For safety reasons, a Forced Brake signal also sets the motor speed command at its lower limit. VEX represents the input to one of two redundant overspeed detectors used to initiate emergency braking in the event of an overspeed of 3 fps or greater. Prior to computing VEX, the acceleration limited speed command, VCSP1, is passed through a first-order filter to approximate the intentional effect of the jerk limiters located external to the VCCS.

5.2 Propulsion System

The propulsion system consists of a motor and an associated current controller plus a speed controller which generates the required current commands. Figure 25 provides a block diagram description of the speed controller model. The model of the motor and current controller is described in Figures 26 and 27.

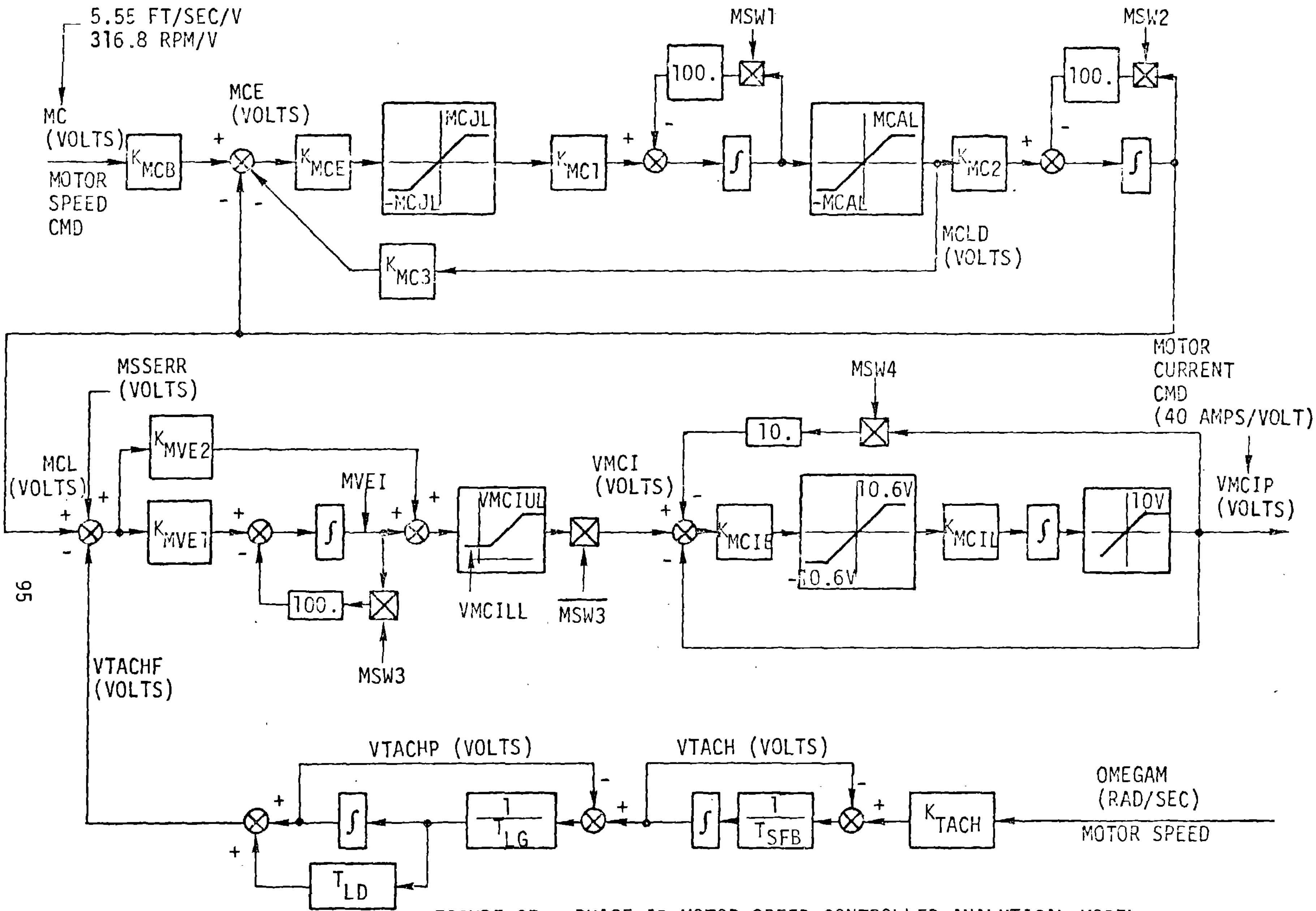


FIGURE 25. PHASE IB MOTOR SPEED CONTROLLER ANALYTICAL MODEL

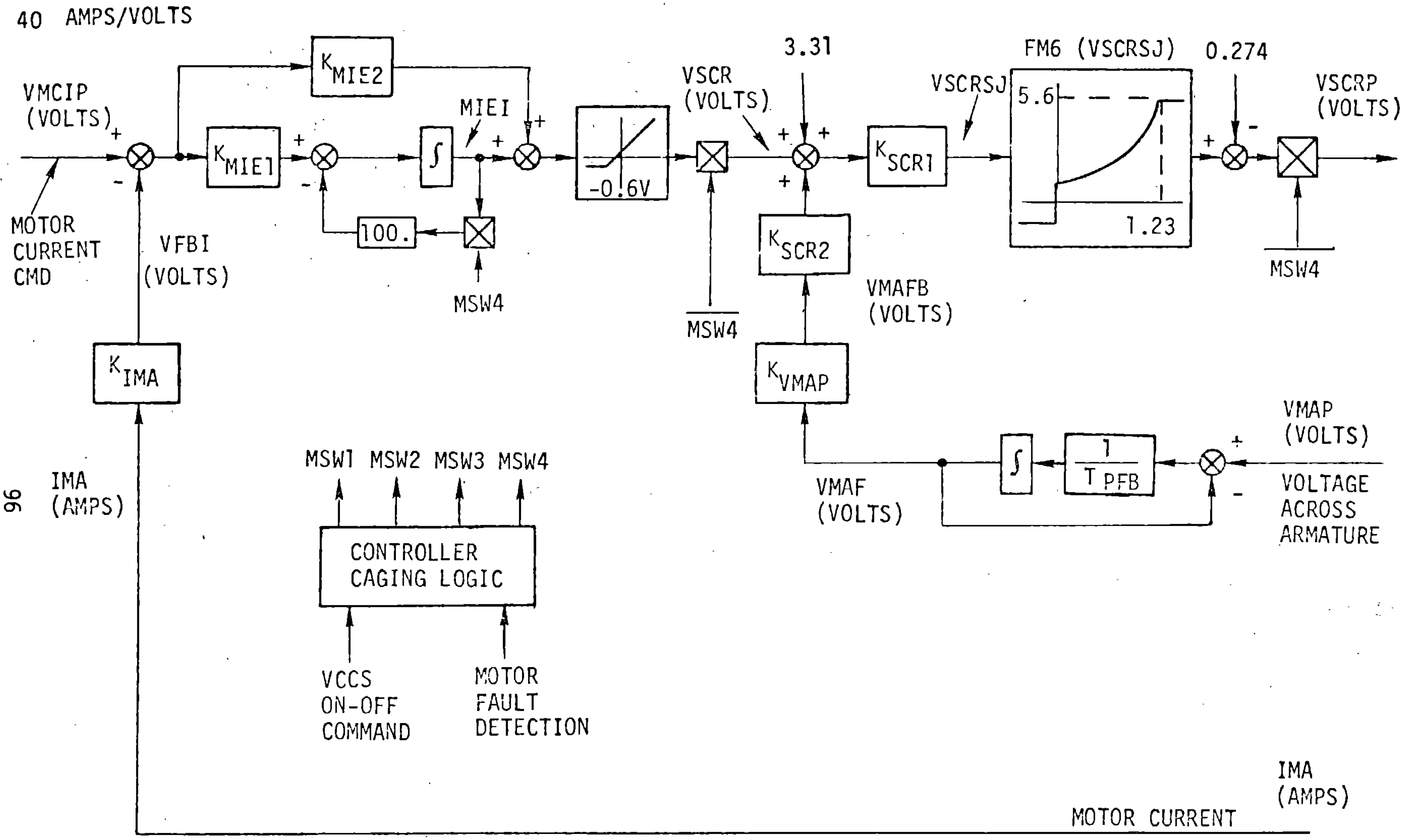


FIGURE 26. PHASE IB MOTOR CURRENT CONTROLLER ANALYTICAL MODEL (PART 1)

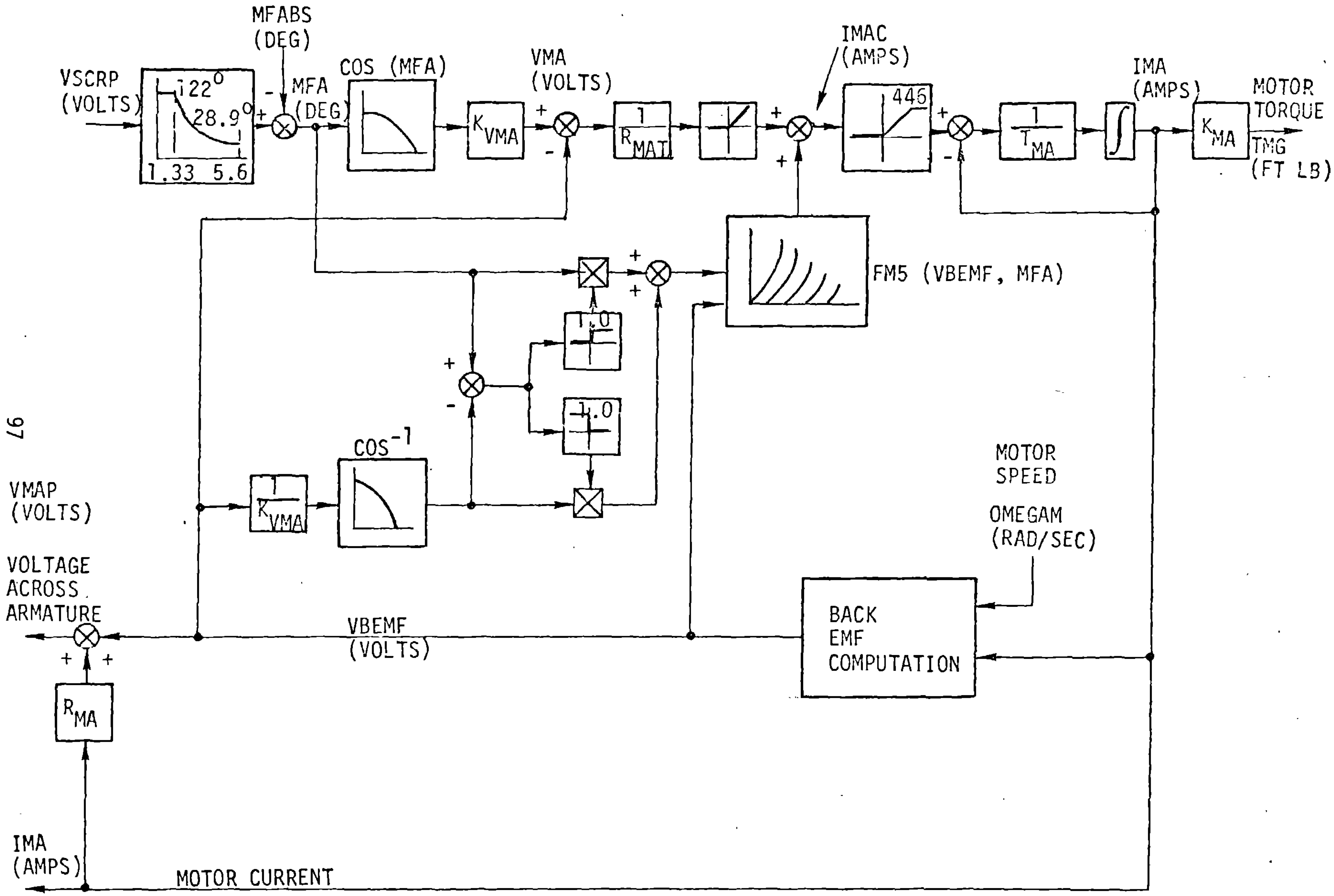


FIGURE 27. PHASE IB MOTOR CURRENT CONTROLLER ANALYTICAL MODEL (PART 2)

The first function performed by the speed controller is to jerk and acceleration limit the VCCS speed command, MC, giving the limited speed command, MCL, shown in Figure 25. The analog limiter used is a second-order high gain filter with clamps on the first and second derivatives of the output to provide the desired limits. The limiter is followed by a speed loop similar to that used within the VCCS. The elements required to compute the measured speed feedback signal are: the scale factor, K_{TACH}; the first order filter characteristic of the D/A converter with time constant T_{SFB}; and the lead compensation required for stability with time constants T_{LD} and T_{LG}. The current command signal, VMCI, is proportional to speed error and its integral, MVEI, i.e., integral compensation is used to provide good steady-state speed control accuracy. The variable, MSSERR, is used to model the small speed errors which can occur. The upper limit on current command is used to prevent excessive motor current levels. The lower limit, VMCILL, provides a small positive torque bias, even when braking. This small bias maintains a preload on the driveline thereby eliminating any impact of backlash and minimizes initial time delays by keeping the current controller "on" at all times. The final current command, VMCIP, is rate limited as discussed in Section 4.7 to minimize the possibility of commutator arcing. The discrete variables MSW1, MSW2, MSW3 and MSW4, which have a value of 0 or 1, are used to model the caging or initialization circuitry which provides the required sequencing during startup and fault response conditions.

Figure 26 describes the current controller circuitry which generates the SCR pulse generator commands. The initial SCR command, VSCR, is proportional to current error and its integral, MIEI. This command is subsequently shaped as shown to partially compensate for downstream nonlinearities. The inner positive feedback loop from back emf to SCR command is included in the model although no longer used for the reasons presented in Section 4.7; i.e., $K_{VMAP}= 0$ in the present design.

Characteristics of the motor and SCR's are described in Figure 27. As shown, firing angle of the SCR's is inversely proportional to the command in this specific design. Armature current is a nonlinear function of firing angle and back emf and is computed in one of two ways depending upon the actual current level. For currents above about 30 amps a continuous current flow through the armature occurs and current level is computed via the upper chain of blocks shown in Figure 27. For low current levels where discontinuous conduction occurs, current level is computed via the table lookup block shown. A table lookup approach is used in this region because of the complexity of the equations involved. The first order filter with time constant, T_{MA}, represents the effect of motor inductance. This filter has a high frequency and negligble effect on system operation. It is included only to prevent an algebraic loop in the model and thereby prevent the computational difficulties associated with algebraic loops in a computer simulation. A series of empirical tables are used in computing back emf, VBEMF, with the details described in the coding presented in Appendix B. Motor torque is assumed to be proportional to motor current.

The parameters, K_{SCR1} and K_{SCR2}, shown in Figure 26 are derived parameters and are computed from the input parameters listed in Table 15 by the following equations:

$$K_{SCR1} = \begin{bmatrix} 4.51 + (35.1 \times 10^3/R_{77}) + (20.0 \times 10^3/R_{28P}) \\ + (150.2 \times 10^6/\ (R_{77} \times R_{28P})\ ) \end{bmatrix}^{-1} \tag{10}$$

$$K_{SCR2} = (35.1 \times 10^3/R_{77}) + (150.2 \times 10^6/\ (R_{77} \times R_{28P})). \tag{11}$$

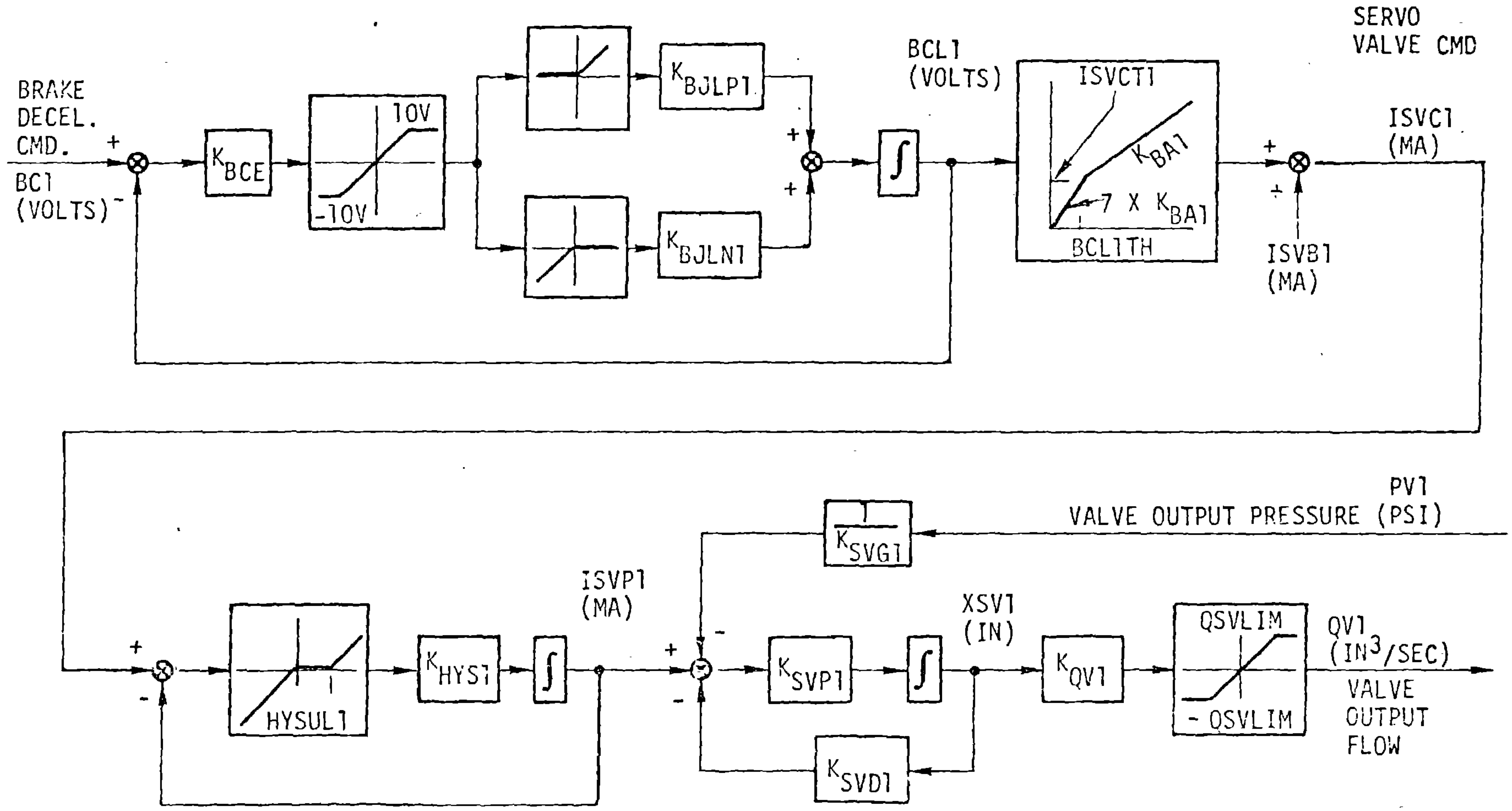
5.3 Brake System

The original single-thread model used to describe brake system operation is illustrated in Figure 16. In this model, the brake calipers were represented by a gain term and a pressure threshold. Servo valve flow limits and coupling between pressure systems were not included as the servo valves were expected to maintain the desired pressures independent of the caliper flow requirements. It was assumed that the two brake channels had identical characteristics which allowed use of a single-thread (non-redundant) model to generate the required performance predictions.

The analytical model required to adequately describe the characteristics of the as-built brake system is given in block diagram form in Figures 28 through 30. Key features of this significantly more complex model are:

- o Unique parameter values for each brake amplifier (required to compensate for unequal gains within the brake calipers)
- o Servo valve flow characteristics and limits (required to simulate time delays and transients shown in test data)
- o Check valve models (required to account for the check valves installed to minimize the flow requirements on the servo valves)
- o Separate front and rear brake caliper models (required to account for measured differences in gains and pressure thesholds)
- o Brake caliper floating and primary piston dynamics (required to generate the fluid flow characteristics resulting from various pressure conditions)
- o Brake caliper hysteresis (required to account for measured hysteresis effects).

Figure 28 provides a block diagram description of the outboard pressure system brake amplifier and servo valve models. The inboard pressure system models are identical except for parameter name and value changes. The brake amplifier model is unchanged from Figure 16 except that both amplifiers are now modeled independently. Servo valve flow limits, ±QSVLIM, are accounted for by modeling the valve in terms of its output spool displacement, XSV1. Output of the valve model is now flow, QV1, which is assumed proportional to spool displacement. The pressure of the fluid between the servo valve and check valve, PV1, is obtained by



NOTES: MODEL SHOWN IS FOR OUTBOARD PRESSURE SYSTEM
INBOARD PRESSURE SYSTEM MODEL IDENTICAL EXCEPT FOR PARAMETER NAME AND VALUE CHANGES, E.G., K_{BA1} BECOMES K_{BA2}, ETC.

FIGURE 28. PHASE IB BRAKE AMPLIFIER AND SERVO VALVE ANALYTICAL MODEL

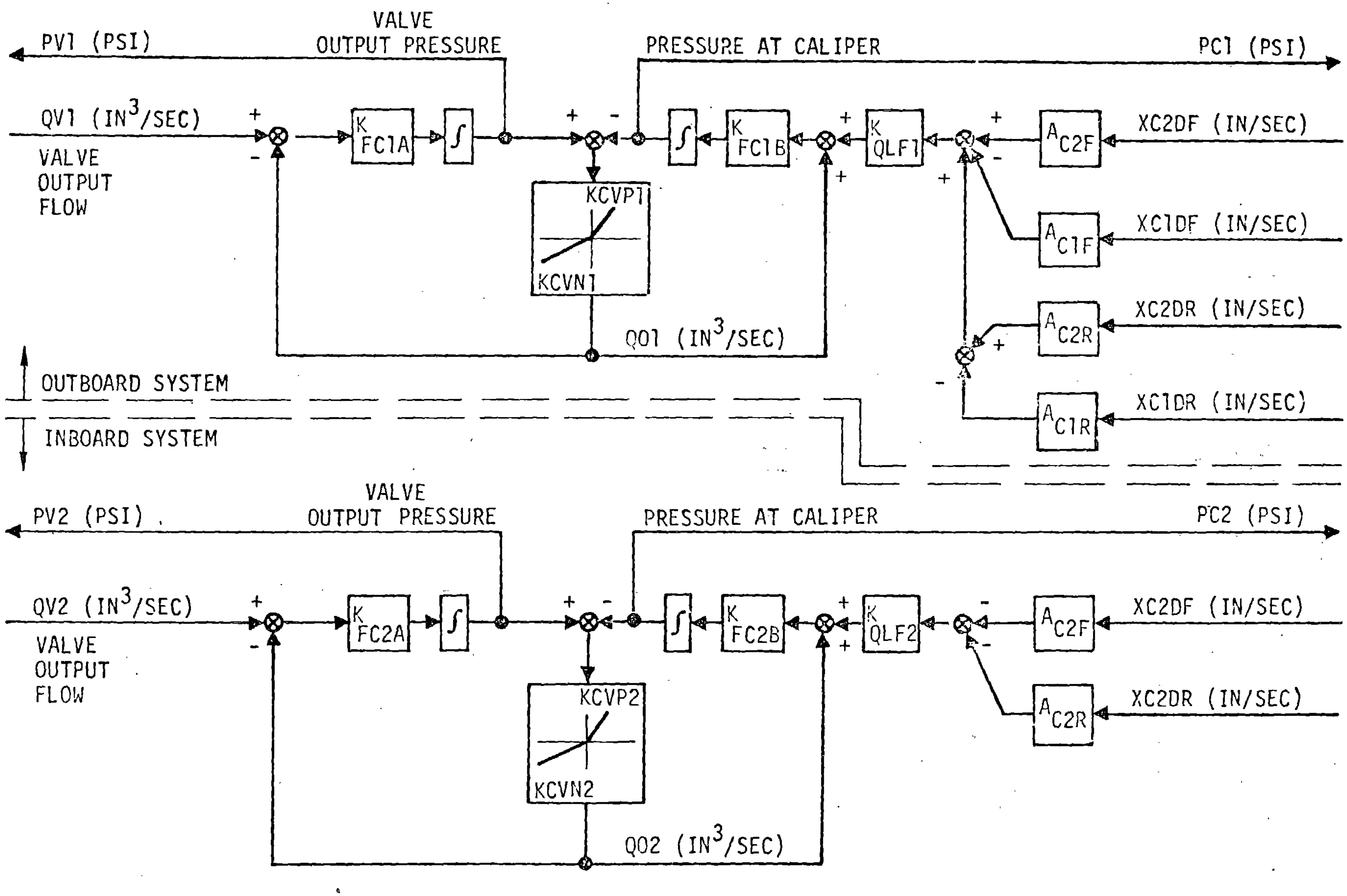
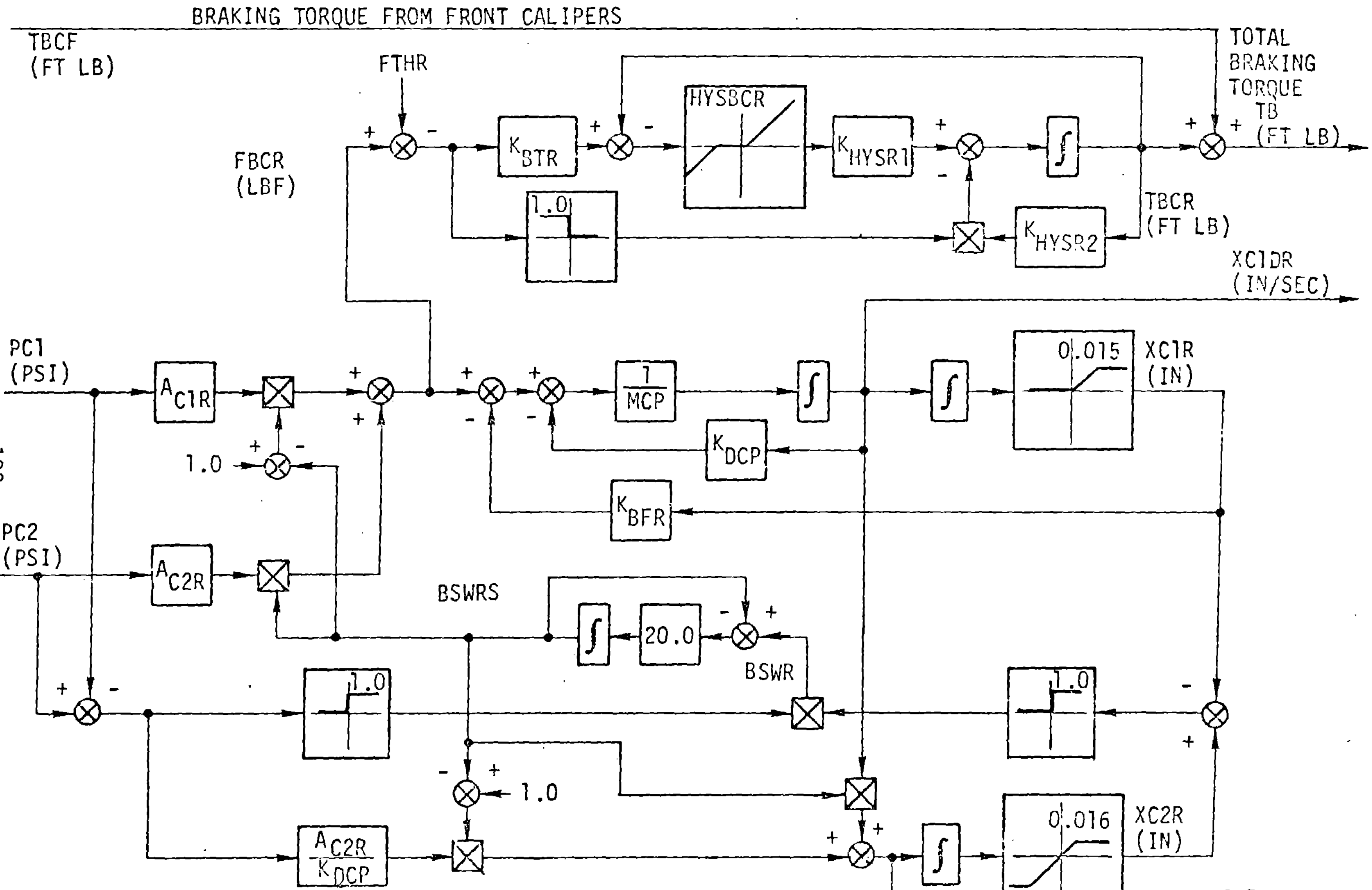


FIGURE 29. PHASE IB ANALYTICAL MODEL FOR BRAKE CHECK VALVES



NOTE: MODEL FOR FRONT BRAKE CALIPERS IDENTICAL EXCEPT FOR PARAMETER NAME AND VALUE CHANGES, E.G., A_{C1R} BECOMES A_{C1F}, ETC.

FIGURE 30. PHASE IB ANALYTICAL MODEL FOR REAR BRAKE CALIPERS

integrating the net flow into this volume as shown in Figure 29. Pressure regulation is accomplished by control of the flow into the volume upstream of the check valve and is a more realistic description of how the valve actually operates than was the simplified model originally used (Figure 16).

The dynamics introduced by the orifices and bypass check valves used to limit reverse fluid flow are modeled as shown in Figure 29. Fluid flow through the check valves, Q01 and Q02, is assumed proportional to the pressure differential across the valves, PV1 - PC1 and PV2-PC2. The check valves are open and offer little resistance to flow when the flow is toward the calipers giving the high gain terms KCVP1 and KCVP2. Reverse flow, toward the servo valves, is restricted to pass through the orifices giving the low gain terms, KCVN1 and KCVN2. Pressures are obtained by integrating the net flows into the volumes upstream and downstream of the check valves. Fluid flow within a caliper is a function of piston velocities and areas and is modeled as shown on the right hand side of Figure 29.

The rear brake calipers are modeled as shown in Figure 30. The front brake caliper model is identical except for parameter name and value changes. The final output gains, K_{BTR} and K_{BTF}, include a factor of two to account for the presence of two calipers on each end of the vehicle. Separate models are used for front and rear calipers to account for measured differences in gains and pressure thresholds.

Models of the dynamics of both the primary and floating caliper pistons are included for the purpose of computing piston velocities XC1DF, XC1DR, XC2DF and XC2DR which are required in computing fluid flow rates. The flow rates, in turn, are used to establish the pressure levels, PC1 and PC2, as a function of time and to determine which pressure is in control. The two pressures are multiplied by the appropriate areas to obtain corresponding force levels. Voting is accomplishing by the switching variable BSWRS which has a steady-state value of 0 or 1. The filter between BSWR and BSWRS is included to smooth the switching transients and, thereby, eliminate problems encountered with the numerical integration algorithm used in the simulation. The force selected by this voting logic is used as the forcing function for the primary piston dynamics and in computing the output torque, TBCR. A hysteresis model is used in computed output torque to account for measured hysteresis effects. A force threshold, FTHR, is used in this calculation to account for measured caliper pressure threshold characteristics. Dynamics of the primary piston and the associated retractor springs are modeled as a second-order system. Primary piston displacement is constrained to a total range of 0.015 inch. Floating piston velocity, XC2DR, is assumed proportional to the pressure differential across the piston with displacement, XC2R, constrained to stay within the limits of -0.12 and +0.016 inches.

Nominal brake system parameter values and limits, where available, are given in Table 16. Because of limited availability of detailed information on the servo valves and brake calipers, many of the parameter values are the result of matching the simulation to available input/output test data by a trial and error procedure. Therefore, while the model accurately describes overall operation of the as-built system, discrepancies between specific parameter values and actual component characteristics may occur. Also, some parameter values, such as hydraulic fluid compliance, were purposely kept low to keep the model frequencies low enough to prevent excessive computer run times.

A simulation/test data comparison is shown in Figure 31 for one of the test cases used to establish parameter values. This specific test case represents brake system operation during the onset of braking at the end of a 0 to 4 fps speed transition. The commanded pressure trajectory consists of a 45 psig step at 0.2 second followed by a constant rate pressure ramp. The initial pressure flat spot, starting at 0.3 second, is caused by the flow required as the caliper pistons travel the distance needed to achieve pad to rotor contact. This flat spot is followed by a high pressure rate as the pressure jumps to the commanded value and occurs

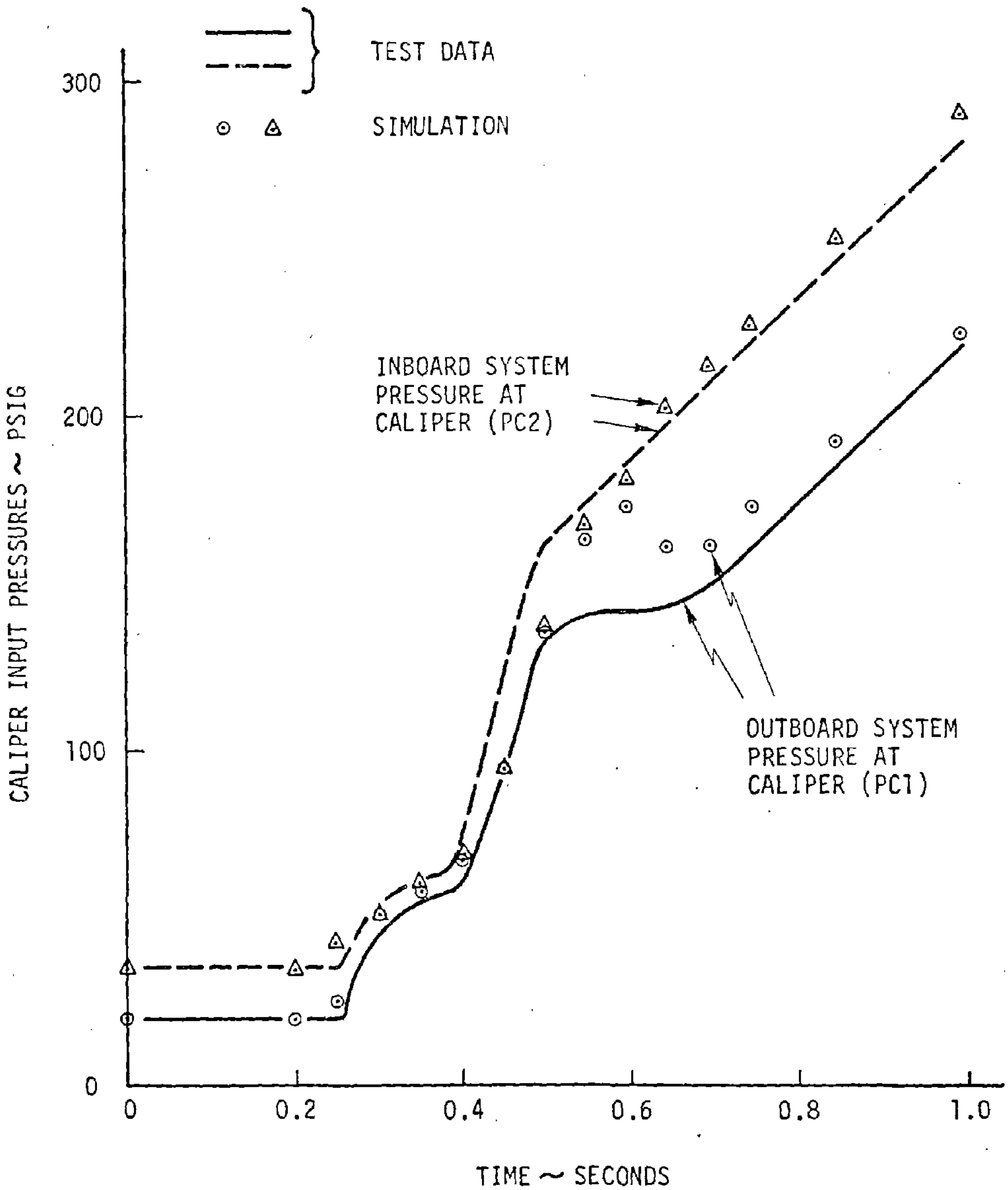


FIGURE 31. BRAKE PRESSURE SIMULATION/TEST DATA COMPARISON

after contact is made and the fluid flow goes to zero. The second flat spot in the outboard system pressure is caused by the check valves which allow the inboard pressure to force the outboard pressure above its commanded value.

The parameters BCL1TH, BCL2TH, ISVB1 and ISVB2 shown in Figure 28 are derived parameters and are computed from the input parameters listed in Table 16 by the following equations:

$$BCL1TH = ISVCT1/(7 \times K_{BA1}) \qquad (12)$$

$$BCL2TH = ISVCT2/(7 \times K_{BA2}) \qquad (13)$$

$$ISVB1 = PSVB1/K_{SVG1} \qquad (14)$$

$$ISVB2 = PSVB2/K_{SVG2}. \qquad (15)$$

5.4 Vehicle

Figure 32 provides a block diagram description of the controlled element (the vehicle) of the system. With the exception of driveline flexibility, the vehicle is modeled as a rigid body. Vehicle mass and motion are modeled in rotational terms with OMEGAW being the average rotational wheel speed and I_{WAV} representing the effective total inertia of the vehicle and wheels. Actual vehicle displacement (DISP), speed (RATE), and acceleration (ACCEL) are computed in translational units for simulation output purposes from the rotational variables used to represent vehicle dynamics. Driveline dynamics are modeled in terms of a stiffness parameter, K_{DT}, and a damping term, C_{DT}. I_{MR} is the inertia of the motor armature as seen at the wheel side of the differential. A major output of the driveline model is motor speed, OMEGAM, which is the sensed variable used to provide closed-loop control. Torques or forces on the vehicle include motor torque, TMG, forces due to grades, FGRAV, and the total braking torque, TBF. Polarity of the total braking torque variable, TBF, is equal to wheel speed polarity, i.e., the sign is always such that the TBF feedback will cause a deceleration or reduction in speed. Components of TBF include the output torque of the brake system, TB, aerodynamic drag, $K_{AERO} \times OMEGAW^2$, and

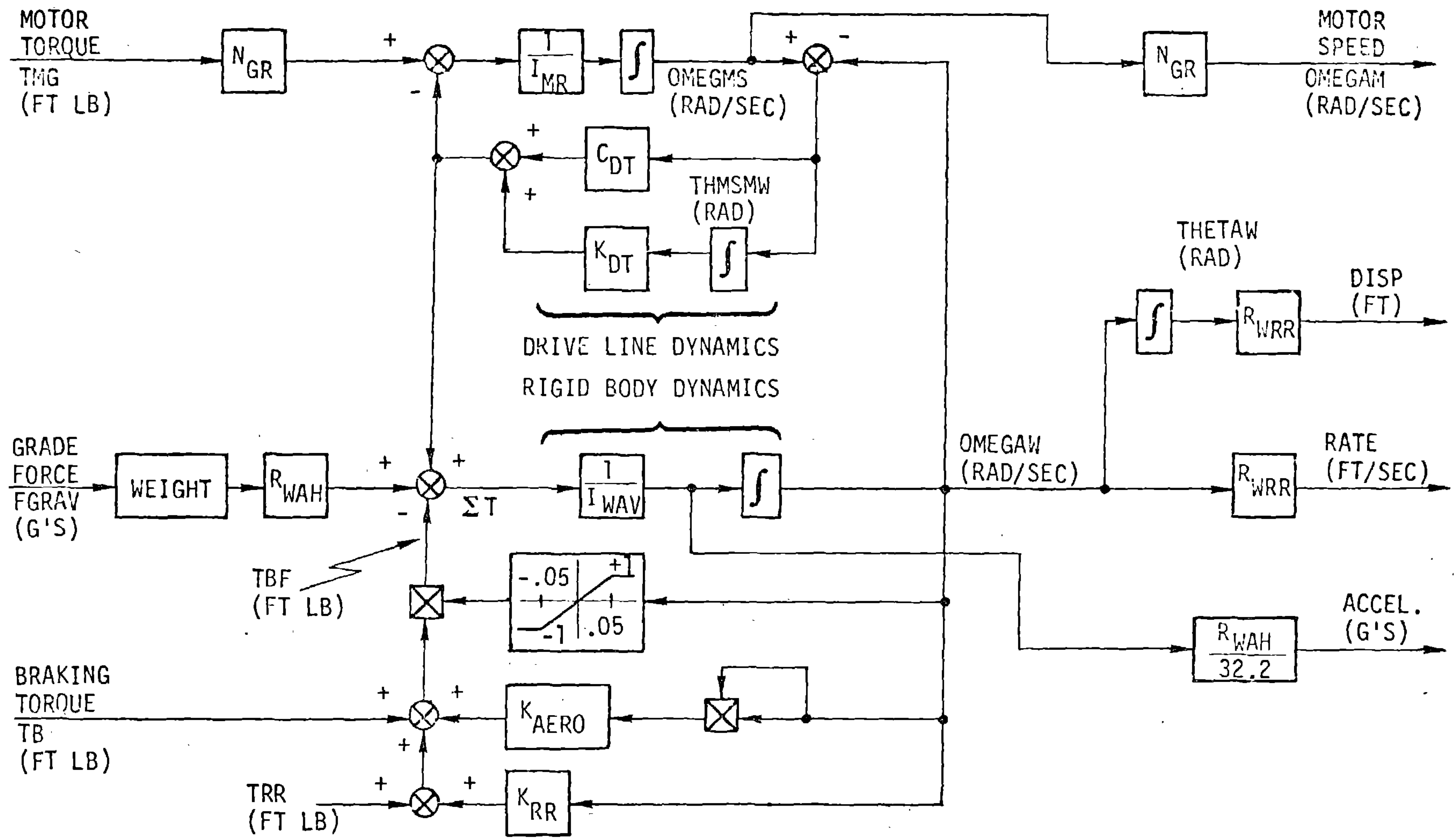


FIGURE 32. PHASE IB VEHICLE DYNAMICS ANALYTICAL MODEL

rolling resistance, TRR + K_{RR} x OMEGAW.

Nominal vehicle parameter values and limits are given in Table 17. The parameters I_{MR}, I_{WAV}, K_{AERO}, TRR, and K_{RR} shown in Figure 32 are derived parameters and are computed by the following equations:

$$I_{MR} = (I_M + I_p) \times N_{GR}{}^2 + I_G \tag{16}$$

$$I_{WAV} = 2 \times I_W + (R_{WAH}{}^2)\ (\mathrm{WEIGHT}/32.2) \tag{17}$$

$$K_{AERO} = C_{AERO} \times R_{WAH}{}^3 \tag{18}$$

$$TRR = C_{TRR} \times R_{WAH} \times \mathrm{WEIGHT} \tag{19}$$

$$K_{RR} = C_{KRR} \times \mathrm{WEIGHT} \times R_{WAH}{}^2. \tag{20}$$

5.5 Definitions and Parameter Values

A list of the variables used in the analytical model is given in Table 13, to aid in the understanding and use of the model. Nominal values for each constant or parameter used in the model are given in Tables 14 through 17. Estimated parameter value tolerances (3σ limits on random variations) are also given for those cases where a reasonable basis for limits has been established. These tolerances provide the basis for computing performance limits via sensitivity studies.

TABLE 13. DEFINITION OF VARIABLES USED IN ANALYTICAL MODEL

VARIABLE	UNITS	DEFINITION
ACCEL	g's	Vehicle acceleration
BC1, 2	v	VCCS brake commands
BCL1,2	v	Jerk limited VCCS brake commands
BSWF,R		Logic variables used to establish brake caliper mode of operation
BSWF, RS		Filtered versions of BSWF and BSWR
DISP	ft	Vehicle displacement
FBCF, R	lbf	Brake caliper force output
FBS1, 2		VCCS "Forced Brake" logic signals
FGRAV	g's	Force due to guideway grade
IMA	amps	Motor current
IMAC	amps	Input to motor current filter
ISVC1,2	ma	Servo valve commands
ISVP1,2	ma	Outputs of servo valve hysteresis models
MC	v	Final VCCS motor speed command
MC1,2	v	VCCS motor speed commands prior to voting
MCE	v	Motor jerk and acceleration limiter error signal
MCL	v	Limited motor speed command
MCLD	v	Limited motor speed command rate
MFA	deg	Motor SCR firing angle
MIEI	v	Integral of motor current error
MSW1,2,3,4		Logic variables used to cage motor control circuits
MVEI	v	Integral of motor speed error
OMEGAM	rad/sec	Motor speed
OMEGAW	rad/sec	Average wheel speed
OMEGMS	rad/sec	Motor speed with gear ratio factor
PC1,2	psig	Brake caliper input pressures
PV1,2	psig	Servo valve output pressures
QO1,2	$in.^3$/sec	Brake caliper input fluid flow rates

TABLE 13. DEFINITION OF VARIABLES USED IN ANALYTICAL MODEL
(Continued)

VARIABLE	UNITS	DEFINITION
QV1,2	$in.^3/sec$	Servo valve output fluid flow rates
RATE	ft/sec	Vehicle speed
SSI1,2		Logic variables used in station stop switching
$\overline{SSI1,2}$		Logical inverses of SSI1 and SSI2
TB	ft lb	Total brake system output torque
TBCF, R	ft lb	Output torque from front/rear brake calipers
TBF	ft lb	Total braking torque
THMSMW	rad	Driveline deflection angle
TMG	ft lb	Motor torque
VBEMF	v	Motor back emf
VC1,2	ft/sec	VCCS civil speed command signals
VCS1,2	ft/sec	Acceleration limited civil speed commands
VCSDA1,2	ft/sec	Quantized acceleration limited civil speed commands
VCSP1,2	ft/sec	Final VCCS reference speed commands
VCSS1,2	ft/sec	Quantized station stop speed commands
VCSV1,2	v	Outputs of speed command D/A converters
VEX	ft/sec	Speed error signal used to set emergency brakes
VFBI	v	Measured motor current
VM1,2	ft/sec	Calibrated measured speed signals
VMA	v	Effective dc motor source voltage
VMAF	v	Measured motor armature voltage
VMAP	v	Effective dc voltage across motor armature
VMAFB	v	Measured motor armature voltage after scaling
VMCI	v	Motor current command
VMCIP	v	Final rate limited motor current command
VMDA1, 2	ft/sec	Quantized calibrated measured speed signals
VMV1,2	v	Outputs of measured speed D/A converters
VSCR	v	Initial motor SCR controller command

TABLE 13. DEFINITION OF VARIABLES USED IN ANALYTICAL MODEL

(Continued)

VARIABLE	UNITS	DEFINITION
VSCRP	v	Final motor SCR controller command
VSCRSJ	v	Intermediate motor SCR controller command
VTACH	v	Measured motor speed
VTACHF	v	Final motor speed feedback signal
VTACHP	v	Intermediate speed feedback compensation variable
XC1F,R	in.	Brake caliper primary piston displacements
XC1DF, R	in./sec	Brake caliper primary piston velocities
XC2F, R	in.	Brake caliper floating piston displacements
XC2DF,R	in./sec	Brake caliper floating piston velocities
XE	ft	Position error
XEDAC	ft	Quantized position error
XEV	v	Output of position clamp circuit
XMS1,2	ft	Inputs to station stop command profilers
XMSQ1,2	ft	Quantized inputs to station stop command profilers
XSV1,2	in.	Servo valve output spool displacements

TABLE 14. NOMINAL VCCS PARAMETER VALUES AND TOLERANCES

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE LOWER LIMIT	TOLERANCE UPPER LIMIT
BC1,2NO--Brake deceleration command null offset	0.0 v	-0.055	+0.055
BCUL--Brake deceleration command upper limit	10.0 v	9.5	13.0
$K_{CS1,2}$--Motor control law gain	1.095	1.075	1.115
K_{DAC} -- D/A converter gain	0.156 v/ft/sec or 0.156 v/ft		
$K_{MSA1,2}$--Motor control law gain	0.775	0.760	0.789
$K_{MSB1,2}$-- Brake control law gain	25.0	24.2	25.8
$K_{PA1,2}$ -- Motor control law gain	0.23	0.226	0.234
$K_{PB1,2}$ -- Brake control law gain	1.51	1.48	1.54
K_{PC} -- Brake control law gain	0.97		
$K_{TACH1,2}$ -- Tachometer/odometer scale factor	0.1674 ft/rad	0.1666	0.1682
K_{VCS} -- Loop gain of acceleration limiter model	10.0 sec^{-1}		
MC1,2LL--Motor speed command lower limit	-0.8 v	-1.0	-0.6
MC1,2NO -- Motor speed command null offset	0.0v	-0.1	+0.1
VCMULT--Performance level	1.0		
XEVLIM--Position error clamp	-3.78 v		

TABLE 15. NOMINAL PROPULSION PARAMETER VALUES AND TOLERANCES

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE	
		LOWER LIMIT	UPPER LIMIT
K_{IMA} -- Current sensor scale factor	0.025 v/amp		
K_{MA} -- Motor torque gain	0.917 ft lb/amp		
K_{MC1} -- Jerk and accel. limiter gain	0.8	0.72	0.88
K_{MC2} -- Jerk and accel. limiter gain	0.07038		
K_{MC3} -- Jerk and accel. limiter gain	0.0303		
K_{MCB} -- Speed scale factor adjustment	1.0		
K_{MCE} -- Jerk and accel. limiter gain	1650. sec^{-2}		
K_{MCIE} -- Current rate limiter gain	60.5		
K_{MCIL} -- Current rate limiter gain	3.31 sec^{-1}		
K_{MIE1} -- Current control law gain	2.5 sec^{-1}		
K_{MIE2} -- Current control law gain	0.2		
K_{MVE1} -- Speed control law gain	14.7 sec^{-1}	13.2	16.2
K_{MVE2} -- Speed control law gain	22.0	20.9	23.1
K_{TACH} -- Tachometer fb. scale factor	0.03 v/rad/sec		
K_{VMA} -- Maximum DC voltage across motor armature	472.6 v		
K_{VMAP} -- Bach emf sensor gain	0.0		
MCAL -- Acceleration limit	10.3 v		
MCJL -- Jerk limit	10.3 v		
MFABS -- Firing angle bias level	1.64 deg		
MSSERR-- Steady state speed error	0.0 v	-0.042	+0.042
R_{28P} -- Circuit gain resistor	22600. ohms		
R_{77} -- Circuit gain resistor	82000. ohms		
R_{MA} -- Armature resistance	0.079 ohms		

TABLE 15. NOMINAL PROPULSION PARAMETER VALUES AND TOLERANCES (Continued)

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE LOWER LIMIT	TOLERANCE UPPER LIMIT
R_{MAT} -- Resistance of armature and line	0.158 ohm		
T_{LD} -- Lead compensation time constant	0.024 sec		
T_{LG} -- Lead compensation time constant	0.008 sec		
T_{MA} -- Armature time constant	0.002 sec		
T_{PFB} -- Back emf sensor time constant	0.025 sec		
T_{SFB} -- Tachometer D/A converter time constant	0.011 sec		
VMCILL -- Current command lower limit	0.2455 v	0.136	0.355
VMCIUL -- Current command upper limit	9.25 v	8.8	9.7

TABLE 16. NOMINAL BRAKE PARAMETER VALUES AND TOLERANCES

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE LOWER LIMIT	TOLERANCE UPPER LIMIT
$A_{C1F,\ R}$ -- Primary caliper piston area	6.5 $in.^2$		
A_{C2F} -- Floating front caliper piston area	5.7 $in.^2$		
A_{C2R} -- Floating rear caliper piston area	4.08 $in.^2$		
FTHF,R -- Caliper force threshold	423. lbf	325.	520.
HYSBCF -- Front caliper output hysteresis	-113. ft lb		
HYSBCR -- Rear caliper output hysteresis	-66.7 ft lb		
HYSUL1,2 -- Servo valve hysteresis	0.6 ma	0.0	1.2
ISVCT1,2 -- Magnitude of brake amplifier static compensation	2.6 ma	2.34	2.86
K_{BA1} -- Amplifier steady-state gain -- outboard system	2.34 ma/v	2.22	2.46
K_{BA2} -- Amplifier steady-state gain -- inboard system	2.90 ma/v	2.76	3.04
K_{BCE} -- Jerk limiter error gain	25.0		
$K_{BFF,R}$ -- Caliper retractor spring gain	28200. lb/in.		
$K_{BJLN1,2}$ -- Jerk limit for decreasing output -- both systems	1.43 sec^{-1}	1.22	1.64
K_{BJLP1} -- Jerk limit for increasing output -- outboard system	0.376 sec^{-1}	0.357	0.395
K_{BJLP2} -- Jerk limit for increasing output -- inboard system	0.390 sec^{-1}	0.371	0.410
K_{BTF} -- Torque gain of front calipers	0.513 ft lb/lbf	0.436	0.590
K_{BTR} -- Torque gain of rear calipers	0.423 ft lb/lbf	0.350	0.486

TABLE 16. NOMINAL BRAKE PARAMETER VALUES AND TOLERANCES (Continued)

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE LOWER LIMIT	TOLERANCE UPPER LIMIT
$K_{CVN1,2}$ -- Check valve reverse flow gain	0.04 $in.^3$/sec/psig		
$K_{CVP1,2}$ -- Check valve forward flow gain	1.0 $in.^3$/sec/psig		
K_{DCP} -- Caliper piston damping term	564. lb sec/in		
$K_{FC1A,B}$ -- $K_{FC2A,B}$ -- Fluid compliance	400. psig/$in.^3$		
$K_{HYSF,R1}$ -- Gain in caliper hysteresis model	100. sec^{-1}		
$K_{HYSF,R2}$ -- Gain in caliper hysteresis model	10. sec^{-1}		
K_{HYSV} -- Gain in servo valve hysteresis model	100.		
$K_{QLF1,2}$ -- Check valve/caliper flow gain	15.0		
$K_{QV1,2}$ -- Servo valve flow gain	1.5 in^3/sec/in.		
$K_{SVD1,2}$ -- Servo valve gain	0.214 ma/in.	0.107	0.321
$K_{SVG1,2}$ -- Steady-state servo valve gain	22.36 psig/ma	19.86	24.86
$K_{SVP1,2}$ -- Servo valve gain	294 in/sec/ma	220.	368.
M_{CP} -- Primary caliper piston mass	11.28 lb sec^2/in.		
QSVLIM--Servo valve flow limit	6.0 $in.^3$/sec	4.8	7.2
PSVB1 -- Servo valve control pressure bias level--outboard system	20.0 psig	0.0	45.0
PSVB2--Servo valve control pressure bias level--inboard system	35.0 psig	10.0	60.0

TABLE 17. NOMINAL VEHICLE PARAMETER VALUES AND TOLERANCES

PARAMETER	NOMINAL VALUE AND UNITS	TOLERANCE LOWER LIMIT	TOLERANCE UPPER LIMIT
C_{AERO} -- Aerodynamic drag coefficient	0.0495 lb sec^2/ft^2	0.037	0.062
C_{DT} -- Driveline damping coefficient	13.2 ft lb sec/rad		
C_{KRR} -- Rolling resistance coefficient	3.41×10^{-5} sec/ft		
C_{TRR} -- Rolling resistance coefficient	0.015	0.009	0.024
I_G -- Differential gear inertia	0.0932 slug ft^2		
I_M -- Motor armature inertia	0.248 slug ft^2		
I_P -- Differential pinion inertia	0.0311 slug ft^2		
I_W -- Inertia of wheels	0.932 slug ft^2		
K_{DT} -- Driveline stiffness	21600 ft lb/rad		
N_{GR} -- Differential gear ratio	7.17		
R_{WAH} -- Axle height	1.18 ft	1. 143	1.217
R_{WRR} -- Tire rolling radius	1.2 ft	1.18	1.22
WEIGHT -- Vehicle weight	10325 lbm	8750	11900

6. SUMMARY AND POTENTIAL SYSTEM IMPROVEMENTS

Estimated performance capability of the M-PRT longitudinal control system along with its requirements are summarized in Table 18. The primary performance requirements on speed and position control are met although design margins, the ability to handle out-of-tolerance parameter variations, are low. The only requirement not met by the Phase IB design is the jerk control requirement. The high jerk values are primarily a consequence of brake performance problems. The fact that these high jerk values are, at worst, barely noticeable by passengers indicates the need for further research in the area of deriving realistic ride comfort specifications.

TABLE 18. ESTIMATED DESIGN CAPABILITY VS REQUIREMENTS

PERFORMANCE PARAMETER	REQUIREMENT	ESTIMATED 3σ LIMITS FOR PHASE IB DESIGN
Regulation (position control)	± 1.1 seconds	± 0.95 second
Station stop accuracy	± 6 inches	+3.5, -2.8 inches
Speed control	+3, -4 fps	+3.0, -2.5 fps
Peak acceleration/deceleration	± 4.4 fps^2 (± 0.137g)	± 4.0 fps^2 (± 0.125g)
Jerk control		
Long term ($\Delta t > 0.2$ second)	± 4.025 fps^3 (± 0.125g/sec)	± 12.9 fps^3 (± 0.4g/sec)
Short term ($\Delta t = 0.1$ second)	± 8.05 fps^3 (± 0.25g/sec)	± 19.3 fps^3 (± 0.6g/sec)
Maximum brake drag	36 ft lb *	0.0 ft lb

*derived requirement

The M-PRT LCS can be characterized as a relatively complex and highly nonlinear control system. A major finding is the large number of hardware parameter variations and nonlinearities which have a significant impact on system performance. This fact is evidenced by the magnitude of the design and analysis effort required to define ways of accommodating both known and unexpected non-linearities and parameter variations within the framework of the existing Phase IA control system structure. Because of cost and schedule constraints, major changes in control system structure to reduce sensitivity, such as the change in station stop control law described in Section 4.2, were allowed only as a last resort.

Much of the design and analysis effort was devoted to the task of developing the detailed nonlinear analytical models required to produce meaningful analysis results. Difficulties in this area, as evidenced by the number of model updates required and the motor stability and brake performance problems encountered late in the design cycle, were encountered for a number of reasons. The initial problem was the absence of any detailed models to use as a starting point and the time which had to be spent on this initial model development. A major reason for including a description of the final M-PRT model in this report is to provide an improved starting point for future design efforts and, thereby, shorten the time required to generate meaningful analysis results. A second problem was in determining the level of detail required. A major benefit of the M-PRT experience described in this report is a substantial improvement in the knowledge of the relative importance of the various hardware characteristics. This information has considerable future value in the development of analytical models, in the preparation of hardware specifications, in the evaluation of proposed designs, and in defining future studies to be performed. A third problem was obtaining data on detailed hardware characteristics early in the program. The final model is based in large part on detailed circuit schematics and on system and subsystem test data. This problem, which is by no means unique to the M-PRT effort, was compounded by the need for very detailed models, the short schedule (the first production vehicle provided the first opportunity to test all portions of the LCS), and the failure in some cases of hardware suppliers to deliver hardware which met its requirements. This problem can be reduced, but probably not eliminated, in future efforts by improvements in hardware specifications and enforcement of these requirements and by providing for extensive detailed subsystem and system testing at the earliest opportunity.

A number of ways of improving system performance capability and reducing sensitivity to hardware parameter variations and nonlinearities have been identified as a result of experience with the M-PRT LCS design. These potential system improvements, which are the subject of the remainder of this section, represent logical candidates for future research and development. The specific changes proposed are:

- Single point torque control
- A jerk and acceleration limited speed command
- Consolidation of control functions within the VCCS
- Wheel mounted tachometers
- Steel belted radial tires
- Closed-loop emergency braking
- Dynamic or regenerative braking.

Single Point Torque Control

The M-PRT design uses significantly different control laws to compute the required brake and motor torque commands as shown in Figure 2. The proposed approach is to use a single control law or speed loop to generate a single torque command which is sent either to the propulsion or brake subsystems depending on its polarity. The proposed concept, exclusive of redundancy, is illustrated in Figure 33. A dual redundancy concept, similar to M-PRT, could be used with redundant torque commands computed by identical control laws. In this approach the lowest (safest) command would be sent to the propulsion subsystem and both commands would be sent to a dual brake system with the calipers voting the safest (highest) of the two input pressures.

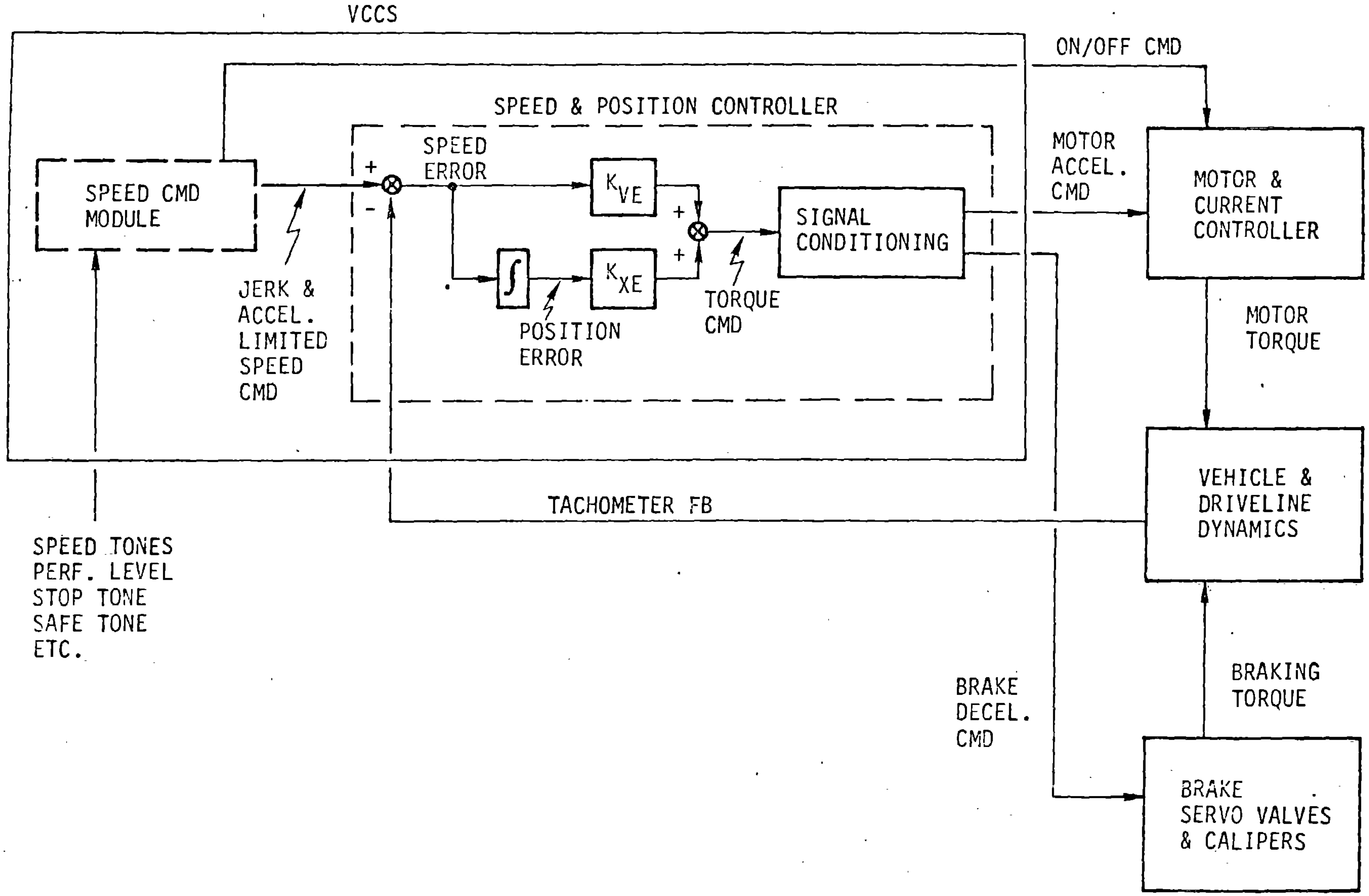


FIGURE 33. PROPOSED LONGITUDINAL CONTROL SYSTEM DESIGN

Advantages of the proposed change are:

- o Reduced brake/motor interaction

  Because of the different brake and motor control laws used in the M-PRT design, it is extremely difficult to prevent simultaneous non-zero brake and motor torque commands under all operating conditions. This problem is significantly reduced in the proposed design since both brake and motor commands are derived from the same torque command.

- o Reduced sensitivity and improved performance potential

  Much of the M-PRT sensitivity to parameter variations is a consequence of interaction between the independent and different brake and motor control laws. The proposed design reduces sensitivity by eliminating the source of these interactions and opens up the possibility of improved speed and position control via an increase in control loop gain.

- o Design and analysis simplification

  Use of a single control law provides a design simplification which reduces the number of components required, eliminates unnecessary duplication of control functions, and simplifies the interface between subsystems thereby simplifying the task of developing adequate hardware specifications. The hardware simplification also provides a corresponding simplification in the analysis task.

Jerk and Acceleration Limited Speed Command

Jerk limiting of the motor speed and brake deceleration commands is used in the M-PRT design to provide vehicle jerk control. The proposed approach is to jerk and acceleration limit the civil speed command and rely on the speed loop to follow the limited command with sufficient accuracy to prevent excessive jerks. This jerk and acceleration limiter would be located within the speed command module shown in Figure 33. Advantages of the proposed change are:

- Increased speed loop bandwidth
  The M-PRT approach of placing jerk limiters within the speed loop severely constrains the bandwidth or response time capability of the servo. This constraint is eliminated by locating the limiters external to the servo.
- Reduced speed errors
  The proposed change reduces speed errors for several reasons. Use of a jerk and acceleration limited command as a basis for computing speed errors, in place of the M-PRT reference which is only acceleration limited, provides a more accurate measure of the trajectory the vehicle is expected to follow. Jerk limiting of the command eliminates rapid changes or discontinuities in the command resulting in a command which is easier for the servo to follow. Finally, elimination of the jerk limiters within the speed loop improves its capability to accurately follow the command.
- Design simplification and improved flexibility
  The proposed change simplifies the design by replacing three jerk and/or acceleration limiters with a single limiter. Design flexibility is increased as jerk and/or acceleration limits can be changed with no impact on the speed control system design.
- Analysis simplification
  Elimination of the major nonlinearity within the speed loop, assuming use of a single speed loop, significantly simplifies the analysis task by allowing use of linear design techniques in initial design studies. The highly nonlinear M-PRT design required extensive and time consuming nonlinear simulation studies in all phases of the design effort.

Changing the location of the jerk limiting function requires the proposed change to single point torque control, to obtain the servo performance necessary to prevent excessive jerks. Conversely, to obtain the full benefit of the change to single point torque control requires the proposed change in jerk limiter location.

Consolidation of Control Functions within the VCCS

An undesirable characteristic of the M-PRT LCS design is a wide distribution of control functions among the VCCS, brake, and propulsion subsystems. Disadvantages of this approach are a duplication of many control functions, complex interfaces between subsystems, and the lack of a single control system design focal point. The proposed approach is to consolidate, to the maximum extent possible, all control functions within the VCCS. The proposed LCS design shown in Figure 33 illustrates the type of control function distribution envisioned. In this example, the M-PRT propulsion speed controller and the brake amplifiers are eliminated and their control functions consolidated within the VCCS. Both the propulsion and brake subsystems receive torque commands with the brake servo valves driven directly by the VCCS. Both the propulsion and brake subsystems now have the identical functions of producing torques proportional to the command with similar requirements on response time, linearity, null offsets, etc. The VCCS is further organized into two basic elements: a speed and position controller; and a speed command module. The speed and position controller generates the torque commands required to follow the commanded speed-position-time trajectory and operates in the same manner in all modes of operation. The primary function of the speed command module is to generate the jerk and acceleration limited speed command which defines the desired vehicle trajectory. To simplify interfaces, many of the logic functions required in an automated vehicle system are consolidated within the speed command module. Examples are control of start up phasing, generation of the motor on/off command, and commanding braking while stopped. In addition, the M-PRT station stop speed command profile and associated logic is generated within the speed command module.

Wheel Mounted Tachometers

The major factor impacting speed loop stability is the location of the tachometers as indicated in Figure 22 and discussed in Section 4.7. A wheel mounted tachometer location is strongly recommended in any future design, rather than the M-PRT motor shaft location, to minimize the impact of driveline dynamics on speed loop stability. The phase stabilization approach used to stabilize the M-PRT motor speed loop represents, at best, a difficult design problem. The 30 db reduction in the gain of the driveline mode dynamics obtained by moving the tachometers to a wheel location allows gain stabilization of the speed loop and eliminates the need for careful control of the phase lag of each element in the loop. The proposed change also eliminates any possibility of a runaway vehicle due to a broken axle. The major impact is a reduction in the rate of tachometer pulses since wheel speed is considerably slower than the speed of the motor shaft. This reduction will require either a change in the method used to process tachometer data or a change in tachometer design to obtain adequate measured speed resolution and sample rate levels.

Steel Belted Radial Tires

In the M-PRT design, variations in tire rolling radius are a major contributor to regulation error. To reduce the magnitude of these variations, use of steel belted radial tires is strongly recommended. Conventional bias ply tires are used in the M-PRT design. Available data indicate that use of steel belted radial tires would reduce variations in rolling radius by approximately an order of magnitude.

Closed Loop Emergency Braking

Emergency stops are performed via an open-loop constant-force braking scheme in the M-PRT design. Because of the open-loop nature of the control, variations in brake system gain, vehicle weight and guideway grade result in a large variation in stopping distance. A closed-loop speed control emergency brake system is proposed as an inexpensive method of significantly reducing the variations. In terms of the proposed LCS design shown in Figure 33, the concept would be as follows:

- o Upon receipt of an emergency stop command, the position error signal would be reset at zero and the output of the jerk and acceleration limiter would be profiled from its current value to zero at emergency rates.

- o The speed and position controller would respond by stopping the vehicle at the commanded emergency rate.

The advantage of the proposed concept, given the design in Figure 33, is that it uses existing LCS elements and requires only the addition of a very small amount of logic functions.

The concept can also be implemented within the framework of the M-PRT design without the substantial rework required to achieve the changes described in Figure 33. This alternate implementation would require a brake amplifier redesign and a method of providing measured speed signals to the brake amplifiers, but would have no impact on other system components. The reduced variations in stopping distance could be used to allow shorter headways by a reduction in the nominal stopping distance, and/or to reduce maximum emergency rate deceleration levels, thereby reducing the possibility of passenger discomfort and injury.

Dynamic or Regenerative Braking

A final potential system improvement is to use the motor to generate all, or a portion, of the braking torques required. The energy produced during deceleration may either be dissipated in on-board resistors (dynamic braking) or fed back into the power distribution system (regenerative braking). The potential benefits of this frequently proposed approach are not as compelling as those of the other proposed changes. The purpose of including the subject in this discussion is to identify the pros and cons involved.

Potential advantages of using the propulsion system to meet all or a portion of the braking torque requirements are:

- o Reduced brake system maintenance

  Reduced use of friction brakes would reduce the need for relatively frequent replacement of brake pads or linings and other brake system maintenance.

- o Reduced power cost

  Use of regenerative propulsion system braking would reduce power demands by allowing recovery of a portion of the power used.

- o Smoother control

  The relatively high short term jerk values observed in the M-PRT test data are primarily due to brake system nonlinearities. While improved brake system performance is technically feasible, experience to date indicates that is easier to obtain smooth, accurate control with electrical systems. Use of the propulsion system to provide all of the torques during normal operation would also eliminate the phasing problem associated with switching from one actuator to another. Experience with the M-PRT design, however, raises the question as to whether tight control of short term jerk is really needed to provide acceptable ride quality.

Potential problems and disadvantages associated with dynamic or regenerative braking are:

- o Power dissipation

  If all of the braking torques are to be provided by the propulsion system, some means of dissipating recovered energy on board may be required as it may not be possible to pump energy back into the power line under all conditions.

- o Driveline design

  Dynamic or regenerative braking requires a driveline design which can tolerate the resulting frequent torque reversals and still maintain required component reliability. Also, successful use of the motor to perform the total normal braking task will require a drive configuration with a minimum amount of backlask. Any significant amount of backlash would present a serious speed loop stability and performance problem.

- o Propulsion system design

  If all of the braking torques are to be provided by the propulsion system, a propulsion system design must be selected which maintains time delays below a value of about 0.2 second when switching between positive and negative torques and which provides full reverse torque capability at all speeds.

- Brake system design
  A friction brake system, with performance requirements similar to those of the M-PRT design, is required in any case to meet emergency braking requirements. Additional propulsion system cost is, therefore, not offset by comparable brake system cost savings.

The dynamic problems associated with driveline backlash and potential propulsion time delays could be largely eliminated if dynamic or regenerative braking were used only to meet long term braking requirements. The concept would be to use slow motor response times, when in a braking mode, to avoid dynamic control problems and to use a friction brake system to provide the short term torques required. This approach would result in a significant reduction in required brake system maintenance and would result in power savings if regenerative rather than dynamic braking were employed. It would not, however, provide any improvement in position, speed, acceleration or jerk control capability.

APPENDIX A
THEORETICAL POINT FOLLOWER CONTROL LAW

I Symbol Definition

A_R - Reference acceleration during speed transitions = 2.0 fps^2 ($V_R \neq V_C$).

K_{PL} - Performance level.

t - Time.

t_F - Time at beginning of a stopping maneuver.

t_T - Time at beginning of a speed transition.

T_R - Maximum vehicle position error (in seconds).

V_a - Actual vehicle speed.

V_C - Commanded guideway civil speed (speed tones).

V_R - Theoretical point follower speed reference.

X_a - Actual vehicle position.

X_R - Theoretical point follower position reference.

II Development of Point Follower Reference Point

A Initial Conditions

From stop tone:

$t = 0$ at instant of stop tone state change from "ON" to "OFF".

$V_R(0) = 0$

$X_R(0) = 0.$

Preceding page blank

From any other stopped position on guideway:

t = 0 at instant of nonzero performance level command to vehicle.

$V_R(0) = 0$

$X_R(0) = X(t_f) + X_{TRANS}$,

where $X(t_f)$ is the vehicle position at the time the vehicle began its stop on the guideway (indicated by a fault message downlink or a zero performance command by the computer)

and X_{TRANS} is the distance gained during the downspeed transition from $V(t_f)$ to zero speed.

B Speed Transitions

A speed transition occurs at any time when $V_R \neq V_C$. During speed transitions, V_R is determined by the following equations:

For $V_C > V_R \quad (t_T)$

$$V_R(t) = V_R(t_T) + A_R(t - t_T).$$

For $V_C < V_R \quad (t_T)$

$$V_R(t) = V_R(t_T) - A_R(t - t_T).$$

These equations remain valid until V_R reaches the value of V_C ($V_R = V_C$).

A typical speed profile generated by this scheme is shown in Figure A-1.

C Point Reference

From t = 0 until the point follower scheme ends at station stop, the reference point is determined by:

$$X_R(t) = \int_0^t V_R(t)\, dt.$$

D Point Follower Requirements

The functions:

$$X_R = X_R(t)$$

$$X_a = X_a(t)$$

have inverse functions:

$$t_R = t_R(X_R)$$

$$t_a = t_a(X_a)\ .$$

At any point on the guideway, the maximum allowable vehicle position error is:

$$T_R = \left|t_R - t_a\right| = 1.1 \text{ sec } (3\sigma).$$

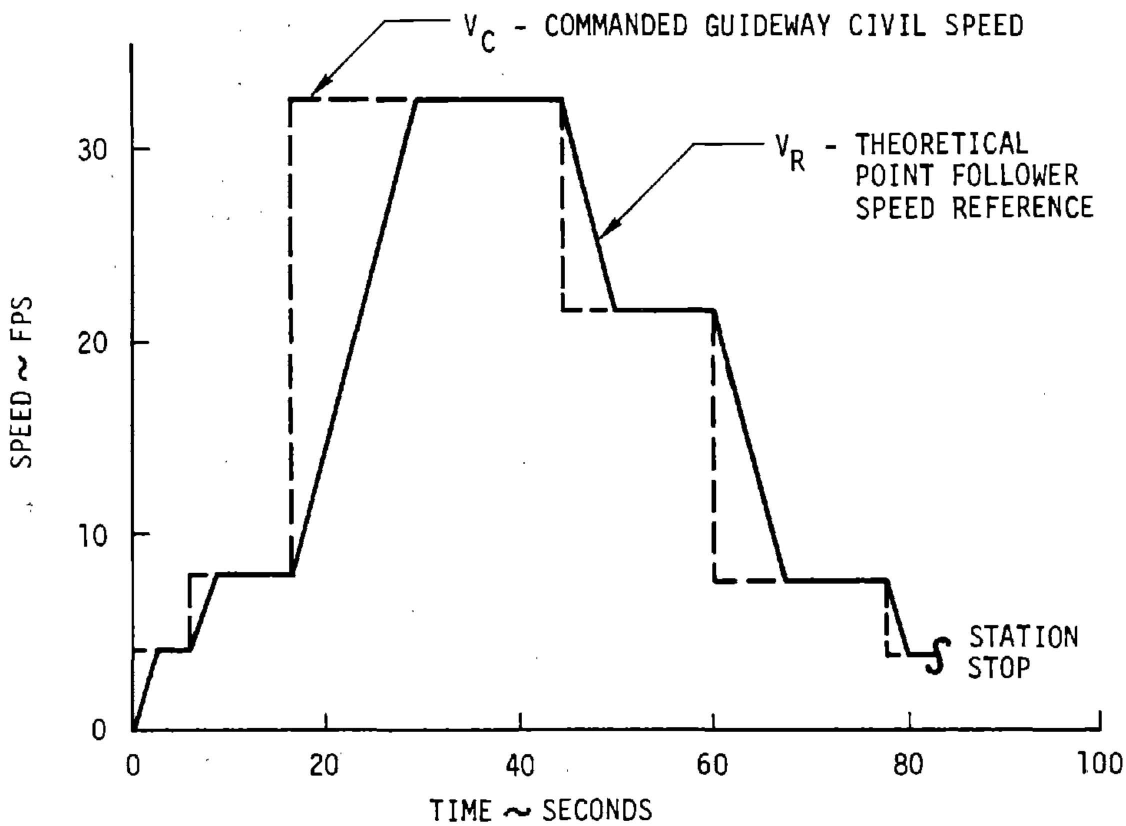


FIGURE A-1. SPEED TRANSITIONS (TYPICAL)

APPENDIX B

COMPUTER SIMULATION

This appendix provides a partial listing of the LCS digital computer simulation which has been developed based on the analytical model described in Section 5. Included are:

- o A listing of the subroutine SYSDEF, the code which computes values for the derivatives of each state variable as required by the numerical integration algorithm.
- o A listing of the Fortran code developed to compute initial condition values for the state variables based upon the values of seven input parameters.
- o A listing of nominal values for all input parameters including tables.

Other portions of the coding are not included as they are unique to the specific simulation program used and would be of little value in incorporating the analytical model into other simulation programs.

A listing of the subroutine SYSDEF, the major element of the simulation, is given in Section I of this appendix. The function of SYSDEF is to compute values for the derivatives of each state variable given state variable input values. Communication with the numerical integration routine is accomplished via the state variable and corresponding derivative arrays, X and XD. An equivalence statement is used to allow writing the equations in terms of the state variable names given in Section V of the main text rather than in terms of X and XD array elements. A similar approach is used to transmit parameter values to SYSDEF via the array SD. Tabular data are transmitted to SYSDEF as indicated in the /SDTBL/ "common" statement. The common block /SDOUT/ provides a means for transmitting the values of intermediate variables to the output routine for printout purposes. Comment cards are used frequently in the coding and identify the hardware elements represented by each block of code.

Section II of this appendix provides a listing of the code used to compute initial conditions for all of the state variables, if desired, from the following input parameters:

- o BCLIC -- Initial jerk limited brake command
- o DISPIC -- Initial displacement
- o ICCODE -- Program mode control variable
- o VMIC -- Initial speed
- o XEIC -- Initial position error
- o XMS1IC, XMS2IC -- Station stop profiler preset values.

This code eliminates the need to compute initial conditions by hand for each case studied. The control parameter, ICCODE, specifies the simulation mode to be used and provides considerable flexibility in using the simulation. Comment cards at the beginning of the initial condition code listing describe the mode associated with each allowable ICCODE value.

Section III of this appendix is a complete list of the input data required by the simulation to run a nominal station start test case. Deletion of the last six lines of data would result in the data required to run a nominal station stop test case. A free field Fortran "Namelist" input format is used for convenience. Values for both tables and single parameters are included.

```
I      SUBROUTINE SYSDEF
C***SYSTEM DEFINITION - PHASE IB MORGANTOWN LCS DESIGN
C
C
C***STATEMENT NO'S USED ARE 1 THRU 14 & 19 THRU 22
C
      SUBROUTINE SYSDEF
C
C
C***SPECIFICATION STATEMENTS
      IMPLICIT REAL(I-N)
      COMMON /TRA1/ TIME,TZERO,KSTATE,NPRDEL,NTERM,TFINAL,DT,NPRINT,
     *              NPLOT
      COMMON /TRA2/ PROPT,PLOPT,SCOPT,INOPT
      COMMON /TRA3/ NP1,NP2,NP3,NPT,MAXNPT,NPLDEL,CASE(20),NCASE
      COMMON /TRA4/ N,XD(100),X(100)
      COMMON /TRA5/ T(2000)
      COMMON /SDPARM/ SD(20,10),XIC(30)
      COMMON /SDTBL/ NVC1,XVC1(10),YVC1(10),NVC2,XVC2(10),YVC2(10)
     *              ,NSSI1,XSSI1(10),YSSI1(10),NSSI2,XSSI2(10),YSSI2(10)
     *              ,NFM1,XFM1(4),YFM1(4),NFM2,XFM2(10),YFM2(10)
     *              ,NFM3,XFM3(10),YFM3(10),NFM4,XFM4(7),YFM4(7)
     *              ,NRFM5,XRFM5(25),NCFM5,XCFM5(7),YFM5(25,7),NSAV(2)
     *              ,NFM6,XFM6(27),YFM6(27)
     *              ,NGRADE,XGRADE(10),YGRADE(10)
     *              ,NVCSS,XVCSS(70),YVCSS(70)
     *              ,XBC1(6),YBC1(6),XBC2(6),YBC2(6)
      COMMON /SDOUT/ MC,VMCI,VFBI,VSCR,VSCRP,KBEMF,VBEMF,MFA,ICR
     *              ,VM1,VM2,DISP,OMEGAM,TMG,BC1,BC2,TB,TBF,TGRAV,FGRAV
     *              ,VCSS1,VCSS2,VCSP1,VCSP2,VCSV1,VCSV2,VMV1,VMV2,XEV
     *              ,VEX,SSI1,SSI2,FBS1,FBS2,ISVC1,ISVC2,VTACHF,VELERR
     *              ,QV1,QO1,QV2,QO2,BSWF,BSWR,FBCF,FBCR,FTERR,RTERR
     *              ,XC1FER,XC1RER,XC2DF,XC2DR
      EQUIVALENCE
     *    (XD(01),DVCS1)  ,(X(01),VCS1)   ,(XD(02),DVCS2)  ,(X(02),VCS2)
     *   ,(XD(03),DXMS1)  ,(X(03),XMS1)   ,(XD(04),DXMS2)  ,(X(04),XMS2)
     *   ,(XD(05),DXE)    ,(X(05),XE)     ,(XD(06),DVCSX)  ,(X(06),VCSX)
     *   ,(XD(07),DMCLD)  ,(X(07),MCLD)   ,(XD(08),DMCL)   ,(X(08),MCL)
     *   ,(XD(09),DVTACH),(X(09),VTACH)  ,(XD(10),DVTACP),(X(10),VTACHP)
     *   ,(XD(11),DMVEI)  ,(X(11),MVEI)   ,(XD(12),DVMCIP),(X(12),VMCIP)
     *   ,(XD(13),DMIEI)  ,(X(13),MIEI)   ,(XD(14),DVMAF)  ,(X(14),VMAF)
     *   ,(XD(15),DIMA)   ,(X(15),IMA)    ,(XD(16),DOMGMS),(X(16),OMEGMS)
     *   ,(XD(17),DOMEGW),(X(17),OMEGAW),(XD(18),DTHMSW),(X(18),THMSMW)
     *   ,(XD(19),DTHETW),(X(19),THETAW),(XD(20),DBCL1)  ,(X(20),BCL1)
     *   ,(XD(21),DBCL2)  ,(X(21),BCL2)
      EQUIVALENCE
     *    (XD(22),DISVP1),(X(22),ISVP1)  ,(XD(23),DISVP2),(X(23),ISVP2)
     *   ,(XD(24),DPV1)   ,(X(24),PV1)    ,(XD(25),DPV2)   ,(X(25),PV2)
     *   ,(XD(26),DXSV1)  ,(X(26),XSV1)   ,(XD(27),DXSV2)  ,(X(27),XSV2)
     *   ,(XD(28),DPC1)   ,(X(28),PC1)    ,(XD(29),DPC2)   ,(X(29),PC2)
     *   ,(XD(30),DXC1F)  ,(X(30),XC1F)   ,(XD(31),DXC1R)  ,(X(31),XC1R)
     *   ,(XD(32),DXC2F)  ,(X(32),XC2F)   ,(XD(33),DXC2R)  ,(X(33),XC2R)
     *   ,(XD(34),DXC1DF),(X(34),XC1DF)  ,(XD(35),DXC1DR),(X(35),XC1DR)
     *   ,(XD(36),DBSWFS),(X(36),BSWFS)  ,(XD(37),DBSWRS),(X(37),BSWRS)
     *   ,(XD(38),DTBCF)  ,(X(38),TBCF)   ,(XD(39),DTBCR)  ,(X(39),TBCR)
      EQUIVALENCE (SD(1,1),ICCODE),(SD(2,1),DISPIC)
```

```
*     ,(SD(03,1),VMIC)   ,(SD(04,1),XEIC)   ,(SD(05,1),BCLIC)
*                                           ,(SD(11,1),MCX1)
*     ,(SD(12,1),MCX2)   ,(SD(13,1),MCSL)   ,(SD(14,1),TGO)
*     ,(SD(15,1),VMCIX1),(SD(16,1),VMCIX2),(SD(17,1),VMCISL)
 EQUIVALENCE (SD(1,2),VCMULT),(SD(2,2),KTACH1)
*     ,(SD(03,2),KTACH2),(SD(04,2),KVCS)   ,(SD(05,2),KDAC)
*     ,(SD(06,2),KCS1)   ,(SD(07,2),KCS2)   ,(SD(08,2),KMSA1)
*     ,(SD(09,2),KMSA2)  ,(SD(10,2),KMSB1)  ,(SD(11,2),KMSB2)
*     ,(SD(12,2),KPA1)   ,(SD(13,2),KPA2)   ,(SD(14,2),KPB1)
*     ,(SD(15,2),KPB2)   ,(SD(16,2),KPC)    ,(SD(17,2),XEVLIM)
*     ,(SD(18,2),MC1LL)  ,(SD(19,2),MC2LL)  ,(SD(20,2),BCUL)
 EQUIVALENCE (SD(1,3),MC1NO),(SD(2,3),MC2NO)
*     ,(SD(03,3),BC1NO)  ,(SD(04,3),BC2NO)  ,(SD(05,3),XMS1IC)
*     ,(SD(06,3),XMS2IC)
 EQUIVALENCE (SD(1,5),KMCIE),(SD(2,5),KMCIL)
*     ,(SD(03,5),KIMA)   ,(SD(04,5),KMIE1)  ,(SD(05,5),KMIE2)
*     ,(SD(06,5),R77)    ,(SD(07,5),KVMAP)  ,(SD(08,5),TAUPFB)
*     ,(SD(09,5),R28P)   ,(SD(10,5),MFABS)  ,(SD(11,5),KVMA)
*     ,(SD(12,5),RMAT)   ,(SD(13,5),TAUMA)  ,(SD(14,5),KMA)
*     ,(SD(15,5),RMA)    ,(SD(16,5),TSTART)
 EQUIVALENCE (SD(1,6),KMCB),(SD(2,6),KMCE)
*     ,(SD(03,6),MCJL)   ,(SD(04,6),KMC1)   ,(SD(05,6),MCAL)
*     ,(SD(06,6),KMC2)   ,(SD(07,6),KMC3)   ,(SD(08,6),MSSERR)
*     ,(SD(09,6),KTACH)  ,(SD(10,6),TAUSFB),(SD(11,6),TAULG)
*     ,(SD(12,6),TAULD)  ,(SD(13,6),KMVE1)  ,(SD(14,6),KMVE2)
*     ,(SD(15,6),VMCILL),(SD(16,6),VMCIUL)
 EQUIVALENCE (SD(1,7),HYSUL1),(SD(2,7),HYSUL2)
*     ,(SD(03,7),KHYSV)  ,(SD(04,7),KSVG1)  ,(SD(05,7),KSVG2)
*     ,(SD(06,7),KSVD1)  ,(SD(07,7),KSVD2)  ,(SD(08,7),KSVP1)
*     ,(SD(09,7),KSVP2)  ,(SD(10,7),KQV1)   ,(SD(11,7),KQV2)
*     ,(SD(12,7),QSVLIM),(SD(13,7),KFC1A)  ,(SD(14,7),KFC2A)
*     ,(SD(15,7),KFC1B)  ,(SD(16,7),KFC2B)  ,(SD(17,7),KCVP1)
*     ,(SD(18,7),KCVP2)  ,(SD(19,7),KCVN1)  ,(SD(20,7),KCVN2)
 EQUIVALENCE (SD(1,8),KBCE),(SD(2,8),KBJLP1)
*     ,(SD(03,8),KBJLP2),(SD(04,8),KBJLN1),(SD(05,8),KBJLN2)
*     ,(SD(06,8),KBA1)   ,(SD(07,8),KBA2)   ,(SD(08,8),ISVCT1)
*     ,(SD(09,8),ISVCT2)                    ,(SD(11,8),KHYSF1)
*     ,(SD(12,8),KHYSF2),(SD(13,8),KHYSR1),(SD(14,8),KHYSR2)
*     ,(SD(15,8),MCP)    ,(SD(16,8),KDCP)   ,(SD(17,8),KBFF)
*     ,(SD(18,8),KBFR)   ,(SD(19,8),PSVB1)  ,(SD(20,8),PSVB2)
 EQUIVALENCE (SD(1,9),RWRR),(SD(2,9),RWAH)
X     ,(SD(03,9),WEIGHT),(SD(04,9),NGR)    ,(SD(05,9),IM)
X     ,(SD(06,9),IP)     ,(SD(07,9),IG)     ,(SD(08,9),IW)
X     ,(SD(09,9),CDT)    ,(SD(10,9),KDT)    ,(SD(11,9),CAERO)
*     ,(SD(12,9),CTRR)   ,(SD(13,9),CKRR)   ,(SD(14,9),KQLF1)
*     ,(SD(15,9),KQLF2)  ,(SD(16,9),AC1F)   ,(SD(17,9),AC2F)
*     ,(SD(18,9),AC1R)   ,(SD(19,9),AC2R)   ,(SD(20,9),FTHF)
 EQUIVALENCE (SD(1,10),KSCR1),(SD(2,10),KSCR2)
*                        ,(SD(04,10),IMR)    ,(SD(05,10),IWAV)
*     ,(SD(06,10),KAERO) ,(SD(07,10),TRR)    ,(SD(08,10),KRR)
*                                            ,(SD(11,10),ISVB1)
*     ,(SD(12,10),ISVB2) ,(SD(13,10),BCL1TH),(SD(14,10),BCL2TH)
*                        ,(SD(16,10),FTHR)   ,(SD(17,10),KBTF)
*     ,(SD(18,10),KBTR)  ,(SD(19,10),HYSBCF),(SD(20,10),HYSBCR)
 INTEGER KSTATE,NPRDEL,NTERM,NPRINT,NPLOT,NP1,NP2,NP3,NPT,MAXNPT
```

```
     *         ,NPLDEL,NCASE,N,PROPT,PLOPT,SCOPT,INOPT,I,J,K,L,M
     *         ,NVC1,NVC2,NSSI1,NSSI2,NLSB,NGRADE,NVCSS
     *         ,NFM1,NFM2,NFM3,NFM4,NRFM5,NCFM5,NFM6,NSAV,IERR
C
C
C***COMPUTE VEHICLE POSITION & MOTOR SPEED
      DISP=RWRR*THETAW
      OMEGAM=NGR*OMEGMS
C
C
C***PROGRAM CONTROL LOGIC
      FBS1=0.
      FBS2=0.
      IF(ICCODE.LT.2.5.OR.ICCODE.GT.5.5) GO TO 4
      TB=0.
      IF(ICCODE.GT.3.5) GO TO 9
      MC=MCX1
      IF(TIME.GE.TGO) MC=MCX1+MCSL*(TIME-TGO)
      IF(MC.GT.MCX2) MC=MCX2
      GO TO 5
 9    IF(ICCODE.GT.4.5) GO TO 22
      MSW1=1.
      MSW2=1.
      MSW3=1.
      MSW4=0.
      VMCI=VMCIX1
      IF(TIME.GE.TGO) VMCI=VMCIX1+VMCISL*(TIME-TGO)
      IF(VMCI.GT.VMCIX2) VMCI=VMCIX2
      GO TO 8
 22   BC1=TABUP1(YBC1,TIME,XBC1,6)
      BC2=TABUP1(YBC2,TIME,XBC2,6)
      GO TO 20
C
C
C***VCCS MODEL
 4    CONTINUE
C  SPEED TONE PROFILERS & STOP TONE INDICATORS
      VC1=TABUP1(YVC1,DISP,XVC1,NVC1)
      VC2=TABUP1(YVC2,DISP,XVC2,NVC2)
      SSI1=TABUP1(YSSI1,DISP,XSSI1,NSSI1)
      SSI2=TABUP1(YSSI2,DISP,XSSI2,NSSI2)
      IF(SSI1.GT.0.5) VC1=0.
      IF(SSI2.GT.0.5) VC2=0.
      IF(VC1.GT.4.1) VC1=VCMULT*VC1
      IF(VC2.GT.4.1) VC2=VCMULT*VC2
      DVCS1=KVCS*(VC1-VCS1)
      DVCS2=KVCS*(VC2-VCS2)
      IF(ABS(DVCS1).GT.2.) DVCS1=SIGN(2.,DVCS1)
      IF(ABS(DVCS2).GT.2.) DVCS2=SIGN(2.,DVCS2)
      IF(VCS1.LT.0.0.OR.VCS1.GT.63.75) DVCS1=0.
      IF(VCS2.LT.0.0.OR.VCS2.GT.63.75) DVCS2=0.
C   MEASURED SPEED & POSITION ERROR
      VM1=KTACH1*ABS(OMEGAM)
      VM2=KTACH2*ABS(OMEGAM)
      DXE=VM1-VCS1
```

```
      IF(SSI1.GT.0.5) DXE=-20.0*XE
C  STATION STOP SPEED COMMANDS
      DXMS1=0.
      IF(SSI1.LE.0.5) GO TO 7
      DXMS1=VM1
      NLSB=XMS1/0.04167
      XMSQ1=NLSB*0.04167*12.
      VCSS1=TABUP1(YVCSS,XMSQ1,XVCSS,NVCSS)
      XMSQ1=XMSQ1/12.
 7    CONTINUE
      DXMS2=0.
      IF(SSI2.LE.0.5) GO TO 10
      DXMS2=VM2
      NLSB=XMS2/0.04167
      XMSQ2=NLSB*0.04167*12.
      VCSS2=TABUP1(YVCSS,XMSQ2,XVCSS,NVCSS)
      XMSQ2=XMSQ2/12.
 10   CONTINUE
C  COMMAND SPEED, MEASURED SPEED, & POSITION ERROR DAC'S
      NLSB=VCS1/0.25
      VCSDA1=NLSB*0.25
      VCSP1=VCSDA1
      IF(SSI1.GT.0.5) VCSP1=VCSS1
      VCSV1=KDAC*VCSP1
      NLSB=VCS2/0.25
      VCSDA2=NLSB*0.25
      VCSP2=VCSDA2
      IF(SSI2.GT.0.5) VCSP2=VCSS2
      VCSV2=KDAC*VCSP2
C
      NLSB=VM1/0.208
      VMDA1=NLSB*0.208
      VMV1=KDAC*VMDA1
      NLSB=VM2/0.208
      VMDA2=NLSB*0.208
      VMV2=KDAC*VMDA2
C
      NLSB=XE/0.125
      XEDAC=NLSB*0.125
      XEV=KPC*KDAC*XEDAC
      IF(XEV.LT.XEVLIM) XEV=XEVLIM
      IF(SSI1.GT.0.5) XEV=0.
C  SPEED MONITOR
      DVCSX=(VCSP1-VCSX)/0.47
      VCX=VMDA1-VCSX
C  FORCED BRAKE LOGIC
      IF(VCSV1.LE.0.0.OR.VCMULT.LE.0.0) FBS1=1.
      IF(VCSV2.LE.0.0.OR.VCMULT.LE.0.0) FBS2=1.
      IF(VMDA1.LE.0.0.AND.SSI1.GT.0.5) FBS1=1.
      IF(VMDA2.LE.0.0.AND.SSI2.GT.0.5) FBS2=1.
      IF(ICCODE.GT.1.5.AND.TIME.LT.TSTART) FBS1=0.
      IF(ICCODE.GT.1.5.AND.TIME.LT.TSTART) FBS2=0.
C  BRAKE & MOTOR CONTROLLER
      MC1=KCS1*VCSV1-KMSA1*(VMV1-VCSV1)-KPA1*XEV+MC1NO
      IF(MC1.LT.MC1LL.OR.FBS1.GT.0.5) MC1=MC1LL
```

```
      MC2=KCS2*VCSV2-KMSA2*(VMV2-VCSV2)-KPA2*XEV+MC2NO
      IF(MC2.LT.MC2LL.OR.FBS2.GT.0.5) MC2=MC2LL
      MC=MC1
      IF(MC2.LT.MC1) MC=MC2
C
      BC1=KMSB1*(VMV1-VCSV1)+KPB1*XEV
      IF(BC1.LT.0.) BC1=0.
      IF(BC1.GT.BCUL.OR.FBS1.GT.0.5) BC1=BCUL
      BC1=BC1+BC1NO
      BC2=KMSB2*(VMV2-VCSV2)+KPB2*XEV
      IF(BC2.LT.0.) BC2=0.
      IF(BC2.GT.BCUL.OR.FBS2.GT.0.5) BC2=BCUL
      BC2=BC2+BC2NO
C
C
C***MOTOR SYSTEM MODEL
C  CONTROLLER CAGING LOGIC
 5    MSW1=0.
      MSW2=0.
      MSW3=0.
      MSW4=0.
      IF(ICCODE.LT.1.5.OR.ICCODE.GT.2.5.OR.TIME.GE.TSTART) GO TO 3
      MSW1=1.
      MSW2=1.
      MSW3=1.
 3    IF(FBS1.LT.0.5.AND.FBS2.LT.0.5) GO TO 19
      MSW1=1.
      MSW2=1.
      MSW3=1.
      MSW4=1.
 19   CONTINUE
C  JERK & ACCEL. LIMITER
      MCE=KMCE*(KMCB*MC-MCL-KMC3*MCLD)
      IF(ABS(MCE).GT.MCJL) MCE=SIGN(MCJL,MCE)
      DMCLD=KMC1*MCE
      IF(MCLD.GT.MCAL.AND.DMCLD.GT.0.) DMCLD=0.
      IF(MCLD.LT.-MCAL.AND.DMCLD.LT.0.) DMCLD=0.
      IF(MSW1.GT.0.5) DMCLD=-100.*MCLD
      DMCL=KMC2*MCLD
      IF(MSW2.GT.0.5) DMCL=-100.*MCL
C  SPEED CONTROLLER (LIMITER OUTPUT TO CURRENT COMMAND)
      DVTACH=(KTACH*OMEGAM-VTACH)/TAUSFB
      DVTACP=(VTACH-VTACHP)/TAULG
      VTACHF=VTACHP+TAULD*DVTACP
      VELERR=MCL-VTACHF+MSSERR
      DMVEI=KMVE1*VELERR-100.0*MVEI*MSW3
      VMCI=(1.0-MSW3)*(MVEI+KMVE2*VELERR)
      IF(VMCI.LT.VMCILL.AND.VELERR.LT.0.0.AND.MSW3.LT.0.5)
     *     DMVEI=(VMCILL-MVEI)/.484
      IF(VMCI.LT.(VMCILL+0.01).AND.MSW3.LT.0.5) VMCI=VMCILL
      IF(VMCI.GT.VMCIUL.AND.VELERR.GT.0.0.AND.MSW3.LT.0.5)
     *     DMVEI=(VMCIUL-MVEI)/.484
      IF(VMCI.GT.(VMCIUL-0.01).AND.MSW3.LT.0.5) VMCI=VMCIUL
C  CURRENT COMMAND RATE LIMITER
 8    CONTINUE
```

```
      VMCIE=KMCIE*(VMCI-VMCIP-10.*MSW4*VMCIP)
      IF(ABS(VMCIE).GT.10.6) VMCIE=SIGN(10.6,VMCIE)
      DVMCIP=KMCIL*VMCIE
      IF(VMCIP.GT.10.0.AND.VMCIE.GT.0.0) DVMCIP=0.
C  CURRENT CONTROLLER (LIMITED CURRENT CMD TO SCR COMMAND)
      VFBI=KIMA*IMA
      VMIERR=VMCIP-VFBI
      DMIEI=KMIE1*VMIERR-100.0*MIEI*MSW4
      VSCR=(1.0-MSW4)*(MIEI+KMIE2*VMIERR)
      IF(VSCR.LT.-0.6.AND.VMIERR.LT.0.0.AND.MSW4.LT.0.5)
     *    DMIEI=(-0.6-MIEI)/0.01
      IF(VSCR.LT.-0.59.AND.MSW4.LT.0.5) VSCR=-0.6
C  MOTOR BACK EMF
      XIMA=IMA
      IF(IMA.LT.0.) XIMA=0.
      IF(IMA.GT.446.) XIMA=446.
      OMEGM1=TABUP1(YFM2,XIMA,XFM2,NFM2)
      OMEGM2=TABUP1(YFM3,XIMA,XFM3,NFM3)
      OMEGMP=OMEGAM
      IF(OMEGAM.LT.0.0) OMEGMP=0.
      IF(OMEGAM.GT.331.0) OMEGMP=331.
      FSF=TABUP1(YFM1,OMEGMP,XFM1,NFM1)
      OMEGM3=OMEGM1+FSF*(OMEGM2-OMEGM1)
      KBEMF=9.55*(420.-0.158*IMA)/OMEGM3
      VBEMF=KBEMF*OMEGMP
C  FIRING ANGLE CONTROLLER (SCR COMMAND TO FIRING ANGLE COMMAND)
      DVMAF=(1.0/TAUPFB)*(VBEMF+RMA*IMA-VMAF)
      VMAFB=KVMAP*VMAF
      VSCRSJ=KSCR1*(3.31+VSCR+KSCR2*VMAFB)
      VSCRP=TABUP1(YFM6,VSCRSJ,XFM6,NFM6)-0.274
      IF(MSW4.GT.0.5) VSCRP=0.
      IF(ABS(VSCRP).LT.0.01) VSCRP=0.01
      MFA=162./VSCRP
      IF(MFA.GT.(120.+MFABS)) MFA=120.+MFABS
      MFA=MFA-MFABS
 1    CONTINUE
C  MOTOR & SCR PERF. (CONTINUOUS CURRENT CONDUCTION REGION)
      VMA=KVMA*COS(MFA/57.2958)
      IMAC=(VMA-VBEMF)/RMAT
      IF(IMAC.LT.0.) IMAC=0.
C  MOTOR & SCR PERF. (DISCONTINUOUS CURRENT CONDUCTION REGION)
      MFATEM=MFA
      XMFA=VBEMF/KVMA
      IF(ABS(XMFA).GT.1.) XMFA=SIGN(1.,XMFA)
      MFA=AMAX1(MFA,57.2958*ARCOS(XMFA))
      IMAC=TBLP2(XRFM5,XCFM5,YFM5,MFA,VBEMF,NRFM5,NCFM5,NRFM5,NSAV,IERR)
     1                +IMAC
      MFA=MFATEM
      IF(IMAC.LT.0.) IMAC=0.
      IF(IMAC.GT.446.) IMAC=446.
      DIMA=(IMAC-IMA)/TAUMA
 2    CONTINUE
C  TORQUE GAIN
      TMG=KMA*IMA
C
```

```
C
C***BRAKE SYSTEM MODEL
      IF(ICCODE.GT.2.5.AND.ICCODE.LT.4.5) GO TO 6
C  BRAKE AMPLIFIERS
 20   BCE=KBCE*(BC1-BCL1)
      IF(ABS(BCE).GT.10.) BCE=SIGN(10.,BCE)
      DBCL1=0.
      IF(BCE.GT.0.) DBCL1=KBJLP1*BCE
      IF(BCE.LT.0.) DBCL1=KBJLN1*BCE
      ISVC1=7.0*KBA1*BCL1
      IF(ISVC1.GT.ISVCT1) ISVC1=ISVCT1+KBA1*(BCL1-BCL1TH)
      ISVC1=ISVC1+ISVB1
      IF(ISVC1.LT.0.) ISVC1=0.
      BCE=KBCE*(BC2-BCL2)
      IF(ABS(BCE).GT.10.) BCE=SIGN(10.,BCE)
      DBCL2=0.
      IF(BCE.GT.0.) DBCL2=KBJLP2*BCE
      IF(BCE.LT.0.) DBCL2=KBJLN2*BCE
      ISVC2=7.0*KBA2*BCL2
      IF(ISVC2.GT.ISVCT2) ISVC2=ISVCT2+KBA2*(BCL2-BCL2TH)
      ISVC2=ISVC2+ISVB2
      IF(ISVC2.LT.0.) ISVC2=0.
C  10/74 SERVOVALVE MODELS
      ISVER1=ISVC1-ISVP1
      DISVP1=0.
      IF(ISVER1.GT.HYSUL1) DISVP1=KHYSV*(ISVER1-HYSUL1)
      IF(ISVER1.LT.0.) DISVP1=KHYSV*ISVER1
      SV1ERR=ISVP1-1.0/KSVG1*PV1-KSVD1*XSV1
      DXSV1=SV1ERR*KSVP1
      QV1=KQV1*XSV1
      IF(QV1.GT.QSVLIM.AND.SV1ERR.GT.0.) DXSV1=0.
      IF(QV1.LT.-QSVLIM.AND.SV1ERR.LT.0.) DXSV1=0.
      ISVER2=ISVC2-ISVP2
      DISVP2=0.
      IF(ISVER2.GT.HYSUL2) DISVP2=KHYSV*(ISVER2-HYSUL2)
      IF(ISVER2.LT.0.) DISVP2=KHYSV*ISVER2
      SV2ERR=ISVP2-KSVD2*XSV2-1.0/KSVG2*PV2
      DXSV2=SV2ERR*KSVP2
      QV2=KQV2*XSV2
      IF(QV2.GT.QSVLIM.AND.SV2ERR.GT.0.) DXSV2=0.
      IF(QV2.LT.-QSVLIM.AND.SV2ERR.LT.0.) DXSV2=0.
C  11/74 FRONT BRAKE CALIPERS MODEL
      BSWF=0.0
      IF((PC2-PC1).GT.0.0.AND.(XC2F-XC1F).GT.0.0) BSWF=1.0
      DBSWFS=20.0*(BSWF-BSWFS)
      FBCF=AC1F*PC1*(1.0-BSWFS)+AC2F*PC2*BSWFS
      FTERR=FBCF-FTHF
      FBERR=KBTF*FTERR-TBCF
      DTBCF=0.0
      IF(FBERR.GT.0.0) DTBCF=KHYSF1*FBERR
      IF(FBERR.LT.HYSBCF) DTBCF=KHYSF1*(FBERR-HYSBCF)
      IF(FTERR.LT.0.0) DTCBF=-KHYSF2*TBCF
      XC1FER=FBCF-KBFF*XC1F
      DXC1DF=(XC1FER-KDCP*XC1DF)/MCP
      DXC1F=XC1DF
```

```
      IF(XC1F.GE.0.0.AND.XC1F.LE.0.015) GO TO 11
      IF(XC1F.LT.0.0.AND.XC1FER.GT.0.0) GO TO 11
      IF(XC1F.GT.0.015.AND.XC1FER.LT.0.0) GO TO 11
      DXC1F=0.
      DXC1DF=-(KDCP/MCP)*XC1DF
 11   CONTINUE
      XC2DF=(AC2F/KDCP)*(PC2-PC1)*(1.-BSWFS)+XC1DF*BSWFS
      DXC2F=XC2DF
      IF(XC2F.GE.-0.12.AND.XC2F.LE.0.016) GO TO 12
      IF(XC2F.LT.-0.12.AND.XC2DF.GT.0.0) GO TO 12
      IF(XC2F.GT.0.016.AND.XC2DF.LT.0.0) GO TO 12
      XC2DF=0.0
      DXC2F=0.0
 12   CONTINUE
C  11/74 REAR BRAKE CALIPERS MODEL
      BSWR=0.0
      IF((PC2-PC1).GT.0.0.AND.(XC2R-XC1R).GT.0.0) BSWR=1.0
      DBSWRS=20.0*(BSWR-BSWRS)
      FBCR=AC1R*PC1*(1.0-BSWRS)+AC2R*PC2*BSWRS
      RTERR=FBCR-FTHR
      RBERR=KBTR*RTERR-TBCR
      DTBCR=0.0
      IF(RBERR.GT.0.0) DTBCR=KHYSR1*RBERR
      IF(RBERR.LT.HYSBCR) DTBCR=KHYSR1*(RBERR-HYSBCR)
      IF(RTERR.LT.0.0) DTBCR=-KHYSR2*TBCR
      XC1RER=FBCR-KBFR*XC1R
      DXC1DR=(XC1RER-KDCP*XC1DR)/MCP
      DXC1R=XC1DR
      IF(XC1R.GE.0.0.AND.XC1R.LE.0.015) GO TO 13
      IF(XC1R.LT.0.0.AND.XC1RER.GT.0.0) GO TO 13
      IF(XC1R.GT.0.015.AND.XC1RER.LT.0.0) GO TO 13
      DXC1R=0.
      DXC1DR=-(KDCP/MCP)*XC1DR
 13   CONTINUE
      XC2DR=(AC2R/KDCP)*(PC2-PC1)*(1.-BSWRS)+XC1DR*BSWRS
      DXC2R=XC2DR
      IF(XC2R.GE.-0.12.AND.XC2R.LE.0.016) GO TO 14
      IF(XC2R.LT.-0.12.AND.XC2DR.GT.0.0) GO TO 14
      IF(XC2R.GT.0.016.AND.XC2DR.LT.0.0) GO TO 14
      XC2DR=0.0
      DXC2R=0.0
 14   CONTINUE
C  10/74 ORIFICE MODELS
      PERR1=PV1-PC1
      IF(PERR1.GE.0.) QO1=KCVP1*PERR1
      IF(PERR1.LT.0.) QO1=KCVN1*PERR1
      QVERR1=QV1-QO1
      DPV1=KFC1A*QVERR1
      QLFER1=QO1+KQLF1*(AC2F*XC2DF-AC1F*XC1DF+AC2R*XC2DR-AC1R*XC1DR)
      DPC1=QLFER1*KFC1B
      PERR2=PV2-PC2
      IF(PERR2.GE.0.) QO2=KCVP2*PERR2
      IF(PERR2.LT.0.) QO2=KCVN2*PERR2
      QVERR2=QV2-QO2
      DPV2=KFC2A*QVERR2
```

```
      QLFER2=Q02-KQLF2*(AC2F*XC2DF+AC2R*XC2DR)
      DPC2=KFC2B*QLFER2
C   BRAKE TORQUE SUMMATION
      TB=TBCF+TBCR
      IF(TB.LT.0.) TB=0.
      IF(ICCODE.GT.4.5) GO TO 21
C
C
C***4-TH ORDER VEHICLE DYNAMICS MODEL
 6    CONTINUE
      DOMGMS=(NGR*TMG-CDT*(OMEGMS-OMEGAW)-KDT*THMSMW)/IMR
      DTHMSW=OMEGMS-OMEGAW
      TBF=TB+TRR+KRR*OMEGAW+KAERO*OMEGAW**2
      IF(ABS(OMEGAW).LT.0.05) TBF=20.*OMEGAW*TBF
      IF(OMEGAW.LE.-0.05) TBF=-TBF
      FGRAV=TABUP1(YGRADE,DISP,XGRADE,NGRADE)
      TGRAV=RWAH*WEIGHT*FGRAV
      DOMEGW=(-TBF+TGRAV+CDT*(OMEGMS-OMEGAW)+KDT*THMSMW)/IWAV
      DTHETW=OMEGAW
 21   RETURN
      END
```

II INITIAL CONDITIONS

```
C
C
C***INITIAL CONDITIONS
      ENTRY PROCS2
C   SET ICCODE=0. IF IC'S INPUT DIRECTLY
C   SET ICCODE=1. IF VEHICLE INITIALLY AT CONSTANT SPEED
C   SET ICCODE=2. IF VEHICLE INITIALLY AT REST, I.E. STATION START
C   SET ICCODE=3.  TO RUN MOTOR & VEHICLE OPEN-LOOP
C   SET ICCODE=4.  FOR OPEN-LOOP MOTOR TORQUE CONTROL MODE
C   SET ICCODE=5.  TO RUN BRAKES ALONE
C   IC PARAMETERS AREO  DISPIC, VMIC, XEIC, BCLIC, XMS1IC, XMS2IC
C   IC LOGIC FOR ICCODE=1. -- GOAL IS ZERO ACCEL. CONDITION
C        VCS1=VCS2=VCMULT*VC1, VCSX=VCS1, XMS1=XMS1IC, XMS2=XMS2IC
C        THETAW=DISPIC/RWRR,  SET THMSMW FOR ZERO ACCEL.
C        OMEGMS=OMEGAW=VMIC/NGR/KTACH1
C        MCL=VTACH=VTACHP=KTACH*OMEGAM, MCLD=0.
C        XE=XEIC,  IF XEIC=0. -- SET XE SUCH THAT MCL=MC
C        BCL1=BCL2=BCLIC,  IF BCLIC=0. -- SET BCL1=BC1, BCL2=BC2
C        X(22) THRU X(39) TO GIVE STEADY-STATE RESPONSE TO BCL1 & BCL2
C        MVEI, VMCIP, MIEI, VMAF, IMA TO GIVE ZERO ACCEL.
      IF(ICCODE.LT.0.5.OR.ICCODE.GT.4.5) GO TO 7
      IF(ICCODE.GT.3.2.AND.ICCODE.LT.3.3) GO TO 7
C
      VC1=TABUP1(YVC1,DISPIC,XVC1,NVC1)
      VCS1=VC1
      IF(VC1.GT.4.1) VCS1=VCMULT*VC1
      IF(ICCODE.GT.1.5) VCS1=0.
      VCS2=VCS1
```

```
      VCSX=VCS1
      XMS1=XMS1IC
      XMS2=XMS2IC
      THETAW=DISPIC/RWRR
      OMEGMS=VMIC/NGR/KTACH1
      OMEGAW=OMEGMS
      OMEGAM=NGR*OMEGMS
C   MOTOR J&A LIMITER & SPEED FB.
      VTACH=KTACH*OMEGAM
      VTACHP=VTACH
      MCL=VTACH
      IF(ICCODE.GT.1.5) VTACH=0.
      IF(ICCODE.GT.1.5) VTACHP=0.
      IF(ICCODE.GT.1.5) MCL=0.
      IF(ICCODE.GT.2.5.AND.ICCODE.LT.3.5) MCL=MCX1
      IF(ICCODE.GT.2.5.AND.ICCODE.LT.3.5) VTACH=MCX1
      IF(ICCODE.GT.2.5.AND.ICCODE.LT.3.5) VTACHP=MCX1
      IF(ICCODE.GT.2.5.AND.ICCODE.LT.4.5) GO TO 13
C   VCCS POSITION ERROR
      XE=XEIC
      IF(XEIC.NE.0.0.OR.ICCODE.GT.1.5) GO TO 8
      MC=MCL
      NLSB=VCS1/0.25
      VCSV1=KDAC*0.25*NLSB
      NLSB=VMIC/0.208
      VMV1=KDAC*0.208*NLSB
      XE=(KCS1*VCSV1-KMSA1*(VMV1-VCSV1)+MCINO-MC)/(KDAC*KPC*KPA1)
 8    CONTINUE
C   BRAKE SYSTEM IC'S
      BCL1=BCLIC
      BCL2=BCLIC
      IF(BCLIC.NE.0.) GO TO 9
      BC1=KMSB1*(VMV1-VCSV1)+KPB1*KPC*KDAC*XE
      IF(BC1.LT.0.) BC1=0.
      BCL1=BC1+BC1NO
      BC2=KMSB2*(VMV1-VCSV1)+KPB2*KPC*KDAC*XE
      IF(BC2.LT.0.) BC2=0.
      BCL2=BC2+BC2NO
 9    CONTINUE
      ISVC1=7.0*KBA1*BCL1
      IF(ISVC1.GT.ISVCT1) ISVC1=ISVCT1+KBA1*(BCL1-BCL1TH)
      ISVC1=ISVC1+ISVB1
      IF(ISVC1.LT.0) ISVC1=0.
      ISVC2=7.0*KBA2*BCL2
      IF(ISVC2.GT.ISVCT2) ISVC2=ISVCT2+KBA2*(BCL2-BCL2TH)
      ISVC2=ISVC2+ISVB2
      IF(ISVC2.LT.0.) ISVC2=0.
      ISVP1=ISVC1
      ISVP2=ISVC2
      PV1=KSVG1*ISVP1
      PV2=KSVG2*ISVP2
      PC1=PV1
      PC2=PV2
      BSWFS=0.
      IF(PC2.GT.PC1) BSWFS=1.
```

```
      BSWRS=0.
      IF(PC2.GT.PC1) BSWRS=1.
      FBCF=AC1F*PC1*(1.0-BSWFS)+AC2F*PC2*BSWFS
      FBCR=AC1R*PC1*(1.0-BSWRS)+AC2R*PC2*BSWRS
      XC1F=FBCF/KBFF
      IF(XC1F.GT.0.015) XC1F=.0151
      XC1R=FBCR/KBFR
      IF(XC1R.GT.0.015) XC1R=.0151
      XC2F=XC1F+0.001
      IF(PC1.GT.PC2) XC2F=-.12
      XC2R=XC1R+0.001
      IF(PC1.GT.PC2) XC2R=-.12
      TBCF=KBTF*(FBCF-FTHF)
      IF(TBCF.LT.0.0) TBCF=0.
      TBCR=KBTR*(FBCR-FTHR)
      IF(TBCR.LT.0.0) TBCR=0.
      TB=TBCF+TBCR
C  MOTOR TORQUE FOR ZERO ACCEL.
 13   IF(ICCODE.GT.2.5.AND.ICCODE.LT.4.5) TB=0.
      TBF=TB+TRR+KRR*OMEGAW+KAERO*OMEGAW**2
      IF(ABS(OMEGAW).LT.0.05) TBF=20.*OMEGAW*TBF
      FGRAV=TABUP1(YGRADE,DISPIC,XGRADE,NGRADE)
      TGRAV=RWAH*WEIGHT*FGRAV
      THMSMW=(TBF-TGRAV)/KDT
     / IF(ICCODE.GT.1.5.AND.ICCODE.LT.2.5) THMSMW=0.
      TMG=(KDT/NGR)*THMSMW
C  MOTOR VALUES FOR STEADY STATE
      IMA=TMG/KMA
      IF(ICCODE.GT.1.5.AND.ICCODE.LT.2.5) GO TO 7
      OMEGM1=TABUP1(YFM2,IMA,XFM2,NFM2)
      OMEGM2=TABUP1(YFM3,IMA,XFM3,NFM3)
      FSF=TABUP1(YFM1,OMEGAM,XFM1,NFM1)
      VBEMF=9.55*(420.-0.158*IMA)*OMEGAM/(OMEGM1+FSF*(OMEGM2-OMEGM1))
      VMAF=IMA*RMA+VBEMF
      ICR=TABUP1(YFM4,VBEMF,XFM4,NFM4)
      IF(IMA.LT.ICR) GO TO 10
      VMA=IMA*RMAT+VBEMF
      MFA=57.2958*ARCOS(VMA/KVMA)
      GO TO 11
 10   CONTINUE
      MFA=120.
 12   XIMA=TBLP2(XRFM5,XCFM5,YFM5,MFA,VBEMF,NRFM5,NCFM5,NRFM5,NSAV,IERR)
      DELIMA=IMA-XIMA
      IF(DELIMA.LT.0.) GO TO 11
      MFA=MFA-0.1
      IF(MFA.GT.100.) MFA=MFA-0.9
      GO TO 12
 11   CONTINUE
      VSCRP=162./(MFA+MFABS)+0.274
      VSCRSJ=0.5
 19   XVSCRP=TABUP1(YFM6,VSCRSJ,XFM6,NFM6)
      IF(XVSCRP.GT.VSCRP) GO TO 18
      VSCRSJ=VSCRSJ+0.02
      IF(VSCRSJ.GT.1.4) GO TO 18
      GO TO 19
```

```
18    VMAFB=KVMAP*VMAF
      MIE1=VSCRSJ/KSCR1-3.31-KSCR2*VMAFB
      IF(MIE1.LT.-0.6) MIE1=-0.6
      VMCIP=KIMA*IMA
      MVEI=VMCIP
      VMCIX1=VMCIP
7     CONTINUE
```

III INPUT DATA

```
STATION START TEST CASE -- BASELINE PHASE 1B M-PRT LCS
&NLPARM
      ICCODE=1.,DISPIC=0.,VMIC=8.,XEIC=0.,BCLIC=0.,
      MCX1=.789,MCX2=1.578,MCSL=.39457,TGO=.5,
      VMCIX1=0.,VMCIX2=0.,VMCISL=0.,
      VCMULT=1.,KTACH1=.1674,KTACH2=.1674,KVCS=10.,KDAC=.156,
      KCS1=1.095,KCS2=1.095,KMSA1=.775,KMSA2=.775,KMSB1=25.,KMSB2=25.,
      KPA1=.23,KPA2=.23,KPB1=1.51,KPB2=1.51,KPC=.97,XEVLIM=-3.78,
      MC1LL=-.8,MC2LL=-.8,BCUL=10.,
      MC1NO=0.,MC2NO=0.,BC1NO=0.,BC2NO=0.,XMS1IC=.958,XMS2IC=.958,
      KMCIE=30.0,KMCIL=3.31,KIMA=.025,KMIE1=2.5,KMIE2=.2,R77=82.E3,
      KVMAP=0.,TAUPFB=1.00,R28P=22.6E3,MFABS=1.64,KVMA=472.6,
      RMAT=.158,TAUMA=.010,KMA=.917,RMA=.079,TSTART=.4,
      KMCB=1.,KMCE=1650.,MCJL=10.3,KMC1=0.8,MCAL=10.3,KMC2=.07038,
      KMC3=.0303,MSSERR=0.,KTACH=.03,TAUSFB=.011,TAULG=.008,TAULD=.024,
      KMVE1=14.7,KMVE2=22.,VMCILL=.2455,VMCIUL=9.25,
      KBCE=25.,KBJLP1=.376,KBJLP2=.39,KBJLN1=1.43,KBJLN2=1.43,
      KBA1=2.34,KBA2=2.9,ISVCT1=2.6,ISVCT2=2.6,PSVB1=20.,PSVB2=35.,
      HYSUL1=0.6,HYSUL2=0.6,KHYSV=100.,KSVG1=22.36,KSVG2=22.36,
      KSVD1=.214,KSVD2=.214,KSVP1=294.,KSVP2=294.,KQV1=1.5,KQV2=1.5,
      QSVLIM=6.000,KFC1A=400.,KFC2A=400.,KFC1B=400.,KFC2B=400.,
      KCVP1=1.0,KCVP2=1.0,KCVN1=.04,KCVN2=.04,
      AC1F=6.5,AC2F=5.7,AC1R=6.5,AC2R=4.08,FTHF=423.,FTHR=423.,
      KBTF=.513,KBTR=.423,HYSBCF=-113.,HYSBCR=-66.7,KHYSF1=100.,
      KHYSF2=10.,KHYSR1=100.,KHYSR2=10.,MCP=11.28,KDCP=564.,
      KBFF=28200.,KBFR=28200.,KQLF1=15.,KQLF2=15.,
      RWRR=1.2,WEIGHT=10325.,NGR=7.17,IM=.248,IP=.0311,IG=.0932,IW=.932,
      CDT=11.2,KDT=21600.,RWAH=1.18,CAERO=.0495,CTRR=.015,CKRR=3.41E-5,
      DTMIN=1.E-7,REL=39*.001,
      ABSU=2*.01,2*.002,.005,.01,7*.002,2*.08,2*.008,2*.0002,2*.002,
           2*.006,2*1.,2*.00006,2*1.,4*.00001,2*.0001,2*.0002,2*.6,
      NVCSS=64,
      XVCSS=0.,23.,23.5,30.,30.5,36.5,37.,42.5,43.,48.5,49.,54.5,
            55.,60.5,61.,66.,66.5,71.,71.5,76.,76.5,80.5,81.,84.5,
            85.,89.,89.5,93.,93.5,96.5,97.,100.,100.5,103.5,104.,107.,
            107.5,109.5,110.,112.5,113.,115.,115.5,117.,117.5,119.,
            119.5,120.5,121.,122.5,123.,123.5,124.,124.5,125.,125.5,
            126.,126.49,126.5,126.99,127.,127.49,127.5,1000.,
      YVCSS=2*3.875,2*3.75,2*3.625,2*3.5,2*3.375,2*3.25,2*3.125,
             2*3.0,2*2.875,2*2.75,2*2.625,2*2.5,2* 2.375,2*2.25,
             2*2.125,2*2.0,2*1.875,2*1.75,2*1.625,2*1.5,2*1.375,
             2*1.25,2*1.125,2*1.0,2*0.875,2*0.75,2*0.625,2*0.5,2*0.375,
```

```
       2*.25,2*.125,2*0.0,
    NFM1=4,XFM1=0.,250.,300.,600.,YFM1=2*0.,2*1.,
    NFM2=10,XFM2=0.,40.,80.,120.,200.,250.,300.,340.,420.,500.,
     YFM2=3120.,2960.,2860.,2780.,2690.,2620.,2530.,2440.,2*2100.,
    NFM3=10,XFM3=0.,40.,80.,120.,200.,250.,300.,340.,420.,500.,
     YFM3=3600.,3600.,3360.,3230.,3050.,2950.,2840.,2720.,2*2300.,
    NFM4=7,XFM4=0.,100.,200.,300.,400.,450.,1000.,
    YFM4=41.8,39.9,40.7,39.9,32.26,27.2,0.,
    NRFM5=25,XRFM5=0.,16.,20.,31.,33.,40.,45.,49.6,50.5,57.5,61.,64.1,
      65.,71.,75.,77.,79.,81.,83.,89.2,90.,97.,100.,105.,120.,
    NCFM5=7,XCFM5=0.,100.,200.,300.,400.,450.,1000.,
    YFM5=0.,18*50.,41.8,37.5,15.,11.,5.,0.,
      0.,14*50.,39.9,36.,27.5,20.,10.,8.5,2.5,1.,2*0.,
      0.,10*50.,40.7,35.,20.,12.5,10.,7.5,5.,4.,0.5,5*0.,
      0.,6*50.,39.9,32.5,15.,10.,6.5,5.5,2.5,0.5,10*0.,
      0.,2*50.,32.26,21.5,12.,7.,4.,3.5,16*0.,
      0.,27.2,13.5,5.5,4.5,2.,0.5,18*0.,25*0.,
    NFM6=27,XFM6=-10.,0.,.05,.1,.15,.2,.25,.3,.35,.4,.45,.5,.55,.6,
         .65,.7,.75,.8,.85,.9,.95,1.0,1.05,1.1,1.15,1.23,10.
    YFM6=-0.6,-0.6,.7,.85,.95,1.07,1.17,1.27,1.35,1.45,1.53,1.62,1.7,
     1.8,1.9,2.,2.14,2.27,2.43,2.6,2.82,3.1,3.45,3.9,4.45,5.6,5.6,
    NVC1=4,XVC1=0.,1.,1.0001,100.,YVC1=2*8.,2*4.,
    NVC2=4,XVC2=0.,1.,1.0001,100.,YVC2=2*8.,2*4.,
    NSSI1=4,XSSI1=0.,24.99,25.,100.,YSSI1=2*0.,2*1.,
    NSSI2=4,XSSI2=0.,25.157,25.167,100.,YSSI2=2*0.,2*1.,
    NGRADE=4,XGRADE=0.,100.,500.,1000.,YGRADE=4*0.,
    ICCODE=2.,VMIC=0.,BCLIC=10.,
    NVC1=6,XVC1=-5.,0.,2.53,2.55,100.,1000.,YVC1=3*4.,3*8.,
    NVC2=6,XVC2=-5.,0.,2.53,2.55,100.,1000.,YVC2=3*4.,3*8.,
    NSSI1=4,XSSI1=-1.,0.,10.,1000.,YSSI1=4*0.,
    NSSI2=4,XSSI2=-1.,0.,10.,1000.,YSSI2=4*0.,
    NGRADE=4,XGRADE=-1.,0.,10.,1000.,YGRADE=4*0.,
&END
```

APPENDIX C
REPORT OF INVENTIONS

A diligent review of the work performed under this contract has revealed no innovation, discovery, improvement, or invention.