EXPERIMENTAL EVALUATION OF THE CONTROLLABILITY OF A DUAL PUMP/MOTOR HYDRAULIC ACCUMULATOR ENERGY STORAGE AUTOMOBILE

by

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ABSTRACT

Significant improvements in fuel economy and performance in automobiles can be achieved through the use of a hydrostatic transmission and an energy storage hydropneumatic accumulator. Operating the internal combustion engine at more efficient operating conditions with a continuously variable transmission and recovering braking energy provide the improvements. Based on previous analytical studies which have predicted the potential fuel economy, the required component sizes, and potential control procedures, a full scale, instrumented, and versatile test rig was built for experimental evaluation of the controllability, efficiency, and performance of the concept.

This thesis's main emphasis is the control aspects of the hybrid energy storage vehicle. Simulations of the system on the test rig under typical driving conditions were run. These simulations have confirmed that relatively simple open-loop control principles are adequate. The accuracy and speed of the controller required to provide the driver with acceptable control was easily obtained. The results of the driving simulations and an explaination of the data acquisition system are contained in this thesis.

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LIST OF ABBREVIATIONS

A/D	Analog to Digital
BSFC	Brake Specific Fuel Consumption
D/A	Digital to Analog
IC	Internal Combustion
LVDT	Linear Variable Differential Transducer
P/M	Pump/Motor

LIST OF NOMENCLATURE

A	Pump/Motor Angle
D	Pump/Motor Displacement
Dm	Maximum Pump/Motor Displacement
E	Energy
K1	Constant
K2	Constant
K3	Constant
K4	Constant
K5	Constant
K6	Constant
K7	Constant
P	Accumulator Pressure
Pc	Pump Angle Control Pressure
Pi	Precharge Accumulator Gas Pressure
Pm	Maximum Accumulator Gas Pressure
T	Torque
Tdes	Desired Pump Torque
Tm	Maximum Torque
Vc	Servovalve Control Voltage
Vdr	Driver Command Voltage
Vdrm	Driver Command Voltage at the Maximum Pump/Motor
Vi	Precharge Accumulator Gas Volume
Vlvdt	LVDT Voltage
Vlvdtm	LVDT Voltage at the Maximum Displacement of the Pump/Motor

1.0 INTRODUCTION

A hybrid energy storage vehicle that consists of a prime mover such as an internal combustion engine and an energy storage unit can potentially provide significant improvements in fuel economy for passenger cars and other motor vehicles. The improvement in fuel economy which can be achieved with these hybrid vehicles can be primarily attributed to two factors: the engine can be kept operating at, or close to, the point where its energy conversion efficiency is maximized, and the energy storage unit also allows the energy that is normally lost as heat during friction braking to be recovered, stored, and reutilized. Several types of hybrid energy storage vehicles have been proposed in recent years [1,2,3,4]. The vehicles differ in the size of the prime mover, the type of energy storage unit, the type of vehicle, and the proposed operating strategies.

1.1.0 Primary Energy Source

The hybrid energy storage vehicle examined in this thesis has an internal combustion engine as the prime mover. An internal combustion engine is a light weight and relatively efficient method of producing energy. The prime mover studied is large enough to handle all the performance requirements associated with the operation of the vehicle. Other hybrid vehicle systems have been proposed with smaller prime movers [5]. A system which uses a prime mover not capable of providing all the required performance may be undesirable, since the available power would drop to the maximum available power output of the prime mover if the energy stored was decreased to zero.

1.2.0 Energy Storage Devices

The types of energy storage devices which have been proposed in hybrid vehicles include a hydraulic energy storage device [2], a flywheel [1,6], and an electric battery [3,4]. A hydropneumatic accumulator was used as the energy storage unit on the test rig. The best energy storage device for a particular vehicle is dependent on criteria such as efficiency, performance, cost, and safety.

The technology used in constructing composite pressure vessels is well established, and provides a method to produce light-weight, safe accumulators. Studies have shown that an accumulator using a flexible, porous plastic foam material on the gas side of an accumulator allows the gas thermal process to be nearly isothermal [7]. With this improvement the thermodynamic losses associated with the operation of an accumulator are reduced. Computer studies have been done to determine the most efficient size and operating pressures for accumulators [8].

Although systems using a flywheel as the energy storage device have been previously built [6], a hydropneumatic accumulator has several advantages over a flywheel: a flywheel has bearing and windage losses, vehicle handling may be affected by the flywheel, and the flywheel speed is coupled to the road speed by a continuously variable transmission. The flywheel itself stores more energy per pound than does an accumulator; however, additional components for safety and confinement which are required for the flywheel could reduce this advantage depending on its design. In addition a vehicle utilizing an accumulator has the capability to eliminate the power steering pump, eliminate the starter, eliminate the current transmission, position the engine and other components in a manner which gives the vehicle a more aerodynamic shape, provide more interior car space, and obtain four wheel drive without additional gearing.

A hydropneumatic accumulator has several advantages over a battery: a battery storage unit including batteries and electric motors have much more mass than accumulators and hydraulic motors, current batteries have energy recovery efficiencies which are much lower than that of accumulators, and the rate at which present batteries can be charged and discharged would decrease the vehicle's performance.

1.3.0 Proposed Vehicle Types

The operating strategy used in an energy storage vehicle is dependent on many factors such as the vehicle size, the driving cycles of the vehicle, the size of the engine, the type and size of the energy storage device, and the relative efficiencies of all the components. The advantages and methods of energy storage are very dependent on the application.

Although research has been done on trucks, delivery vans, automobiles, and even bicycles, much of the past research on energy storage vehicles has been done using buses [9]. A number of prototypes have been built and tested using a single pump/motor hydraulic system. The buses which have been built emphasize the efficiency gains possible through the use of regenerative braking. The engine operation in the buses was not optimized for fuel economy. Although many of the same strategies are valid for both the operation of energy storage buses and passengers cars, there are significant differences in terms of necessary power to weight ratios, engine loads, and driving cycles. A bus is run under a more consistent driving cycle than a passenger car. Buses operate in a stop and go urban driving cycle which is ideal for the use of energy storage systems.

1.4.0 Research Scope

A test stand with all the necessary components was built to physically test some of the proposed energy storage concepts in a hybrid energy storage passenger automobile. The purpose of the study was to identify problems, verify computer simulation studies, measure component and system efficiencies, provide actual physical data, develop control strategies and algorithms, and evaluate system performance. The study was done on a test stand rather than in a vehicle because it allowed for more versatility in design and ease in obtaining physical data.

The hybrid energy storage vehicle system which was built and is being presented in this thesis is a dual pump/motor system. The dual pump/motor system is shown in Fig. 1.1. The system consists of the following components: a simulated internal combustion engine, a gear reduction box, a hydraulic pump, a hydraulic pump/motor, high pressure hydropneumatic accumulators, a low pressure accumulator, a flywheel, and a braking system for the flywheel. The flywheel was used to simulate the inertia of

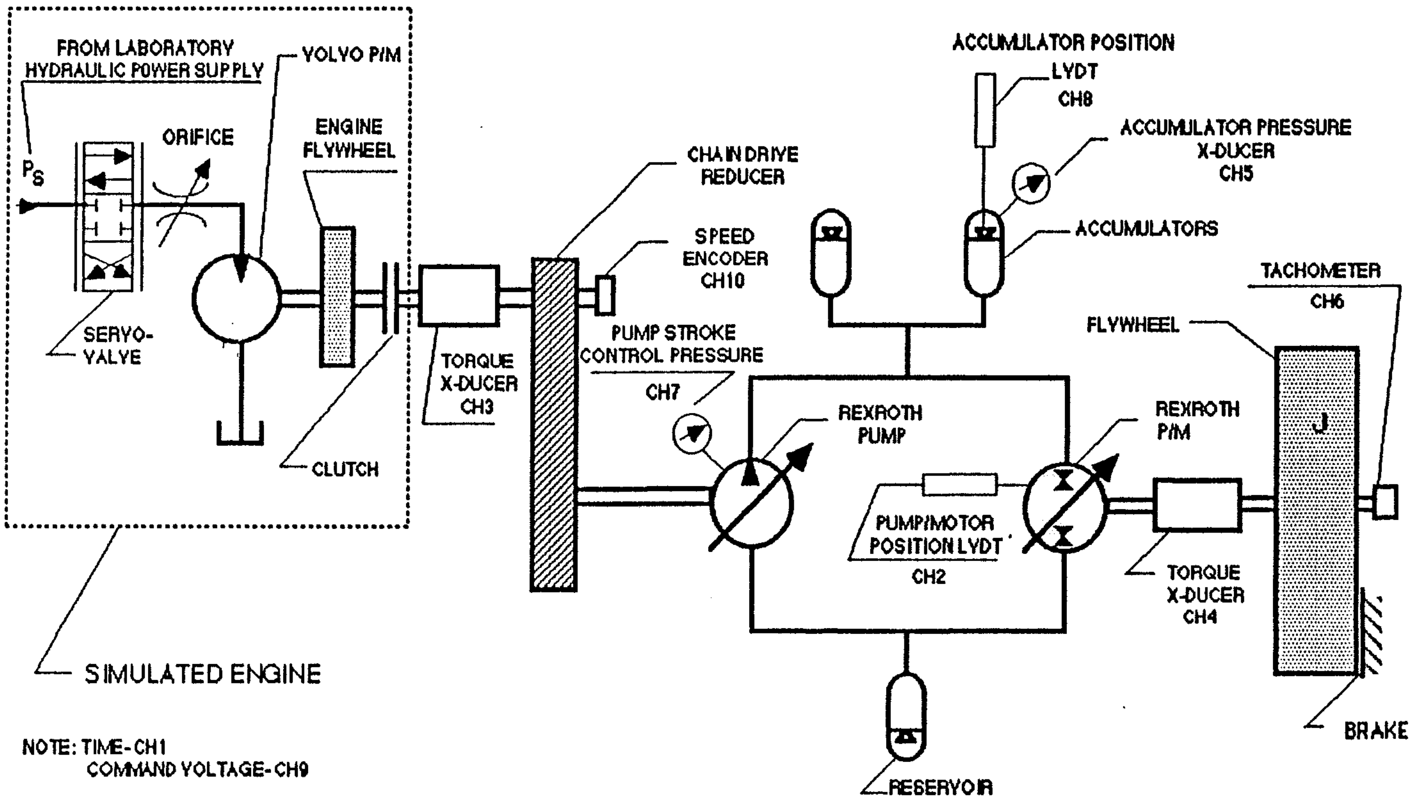

Fig. 1.1: Hydraulic Circuit-Hybrid Vehicle Simulation

the vehicle and not as an energy storage device. The system was also designed to permit ease in installation of data acquisition and control transducers such as torquemeters, pressure transducers, displacement transducers, and speed encoders.

The test program was divided into three main areas: (1) efficiency and performance testing of some of our system components such as the hydraulic pump/motors and the accumulators, (2) regenerative cycling tests to determine the efficiency of the system not including the engine, (3) driving cycle tests to determine the efficiencies and controllability of the entire system. This thesis will concentrate the last area.

2.0 HYDROPNEUMATIC ACCUMULATOR ENERGY STORAGE VEHICLE DESIGN

Two hybrid energy storage vehicle designs were considered for testing. One design utilized a single hydrostatic pump/motor unit, while the other design required two hydrostatic pump/motor units. This thesis contains results obtained from tests involving the two hydrostatic pump/motor unit design.

2.1.0 Comparison of Vehicle Designs

The single pump/motor design and the dual pump/motor design have many similarities. The design of the single pump/motor vehicle is shown in Fig. 2.1. The design of the dual pump/motor vehicle is shown in Fig. 2.2. In both designs the engine power is decoupled from the driving load. This decoupling allows the engine to be operated at or near its most efficient operating point during a normal driving cycle. In both cases the engine can be shut off while the accumulator is supplying the required power demand. Regenerative braking is performed in a similar manner in the two designs. During regenerative braking the energy is transferred to the high pressure accumulator through a single pump/motor unit. This energy in the accumulator is passed back through the pump/motor unit when it is reused at

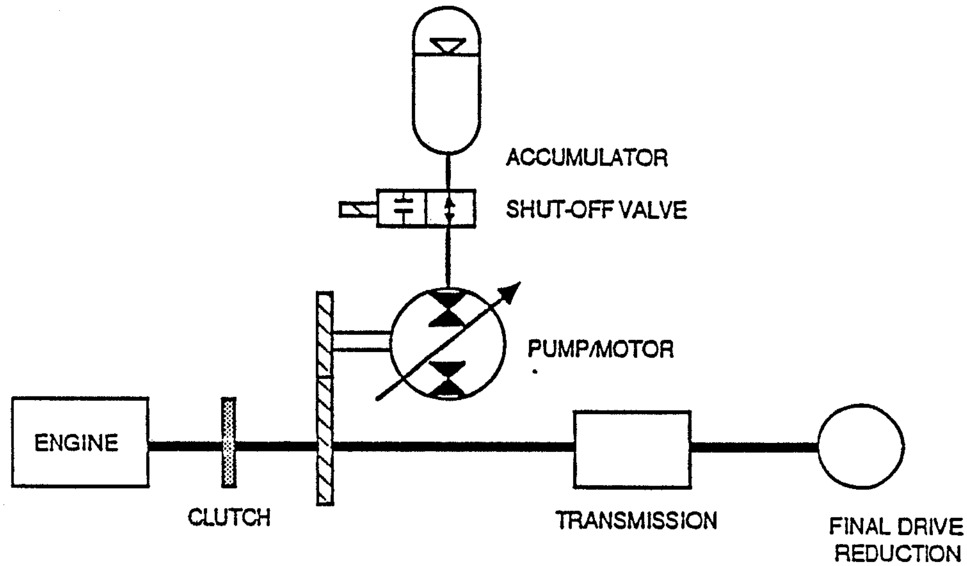

Fig. 2.1: Schematic Diagram of a Drivetrain with a One Pump/Motor Energy Storage System

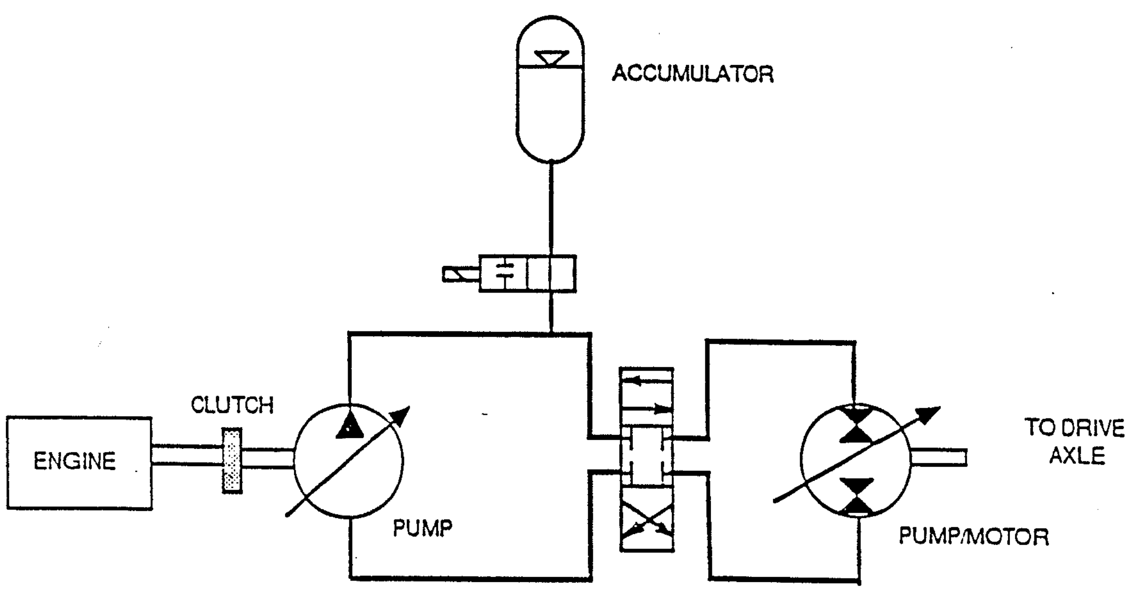

Fig. 2.2: Schematic Diagram of a Drivetrain with a Two Pump/Motor Energy Storage System

the drive axle.

Although the two designs operate using many of the same principles, there are relative advantages and disadvantages of each design. The dual pump/motor system converts all the engine's output power to be transmitted hydraulically, while the single pump/motor system can convert any percentage of the engine's output power to hydraulic energy. The dual pump/motor system can eliminate the need for a mechanical transmission. The single pump/motor system has a relative weight penalty since it requires a transmission in addition to the hydraulic components seen in Fig. 2.1. The dual pump/motor system's efficiency is more dependent on the efficiency of the system's hydraulic components. The potential fuel economy gains for the two designs appear comparable. The relative gains are dependent on the efficiencies of the components and the application. The relative costs between the two designs is dependent on several factors. The dual pump/motor system requires an additional hydrostatic pump/motor unit while the single pump/motor system requires a mechanical transmission in addition to the hydraulic components.

2.2.0 Operation of the Dual Pump/Motor System

Fig. 2.2 shows the hydraulic schematic for the dual pump/motor system. This system is composed of two subsystems isolated by an accumulator with virtually no interaction between the two subsystems. If the accumulator were isolated from the system during certain operating conditions, the subsystems would interact. The first subsystem is composed of the engine, a clutch, and a variable displacement hydrostatic pump. This subsystem converts the output of the engine to energy in the form of a pressurized hydraulic fluid which is stored in the accumulator. The variable displacement hydrostatic pump draws hydraulic fluid from the low pressure reservoir and pumps it to the high pressure accumulator. The second subsystem is composed of a variable displacement hydrostatic pump/motor. This pump/motor functions as a motor to power the drive wheels or functions as a pump during regenerative braking. The accumulator acts as a buffer between the two subsystems. The pressure in the accumulator and the driver's command signal determine the mode of operation.

There are four general modes of operation: (1) the car is driven by the energy in the accumulator and the engine is off, (2) the car is driven by the energy in the accumulator and the engine is on, (3) the car is driven by the engine-pump subsystem and the accumulator is closed off from the rest of the system, and (4) the accumulator is being charged during regenerative braking.

In the first mode the clutch between the engine and the pump is disengaged. The accumulator has a sufficient pressure to supply the driver's desired torque. In the second mode the accumulator's energy level is below a low prespecified level. The engine pump subsystem's output power increases the pressure in the accumulator and provides the required power at the wheels. The third mode is the only mode in which the accumulator does not act as a buffer between the two subsystems. A valve is closed to separate the high pressure accumulator from the rest of the system. All the power from the first subsystem is directly input to the second subsystem. In the fourth mode the pump/motor acts as a pump to charge the high pressure accumulator.

2.3.0 System Components

The components of the dual pump/motor energy storage vehicle test stands are shown in Fig. 2.3 and 2.4. The system components and the transducers are mounted on two test stands. The test stand shown in Fig. 2.3 contains a hydraulic motor, a clutch, a torque transducer, a chain drive box, a hydraulic pump, two high pressure accumulators, and a large low pressure accumulator. A large manifold valving system is also mounted on this test stand; however, it was not incorporated for the work described in this

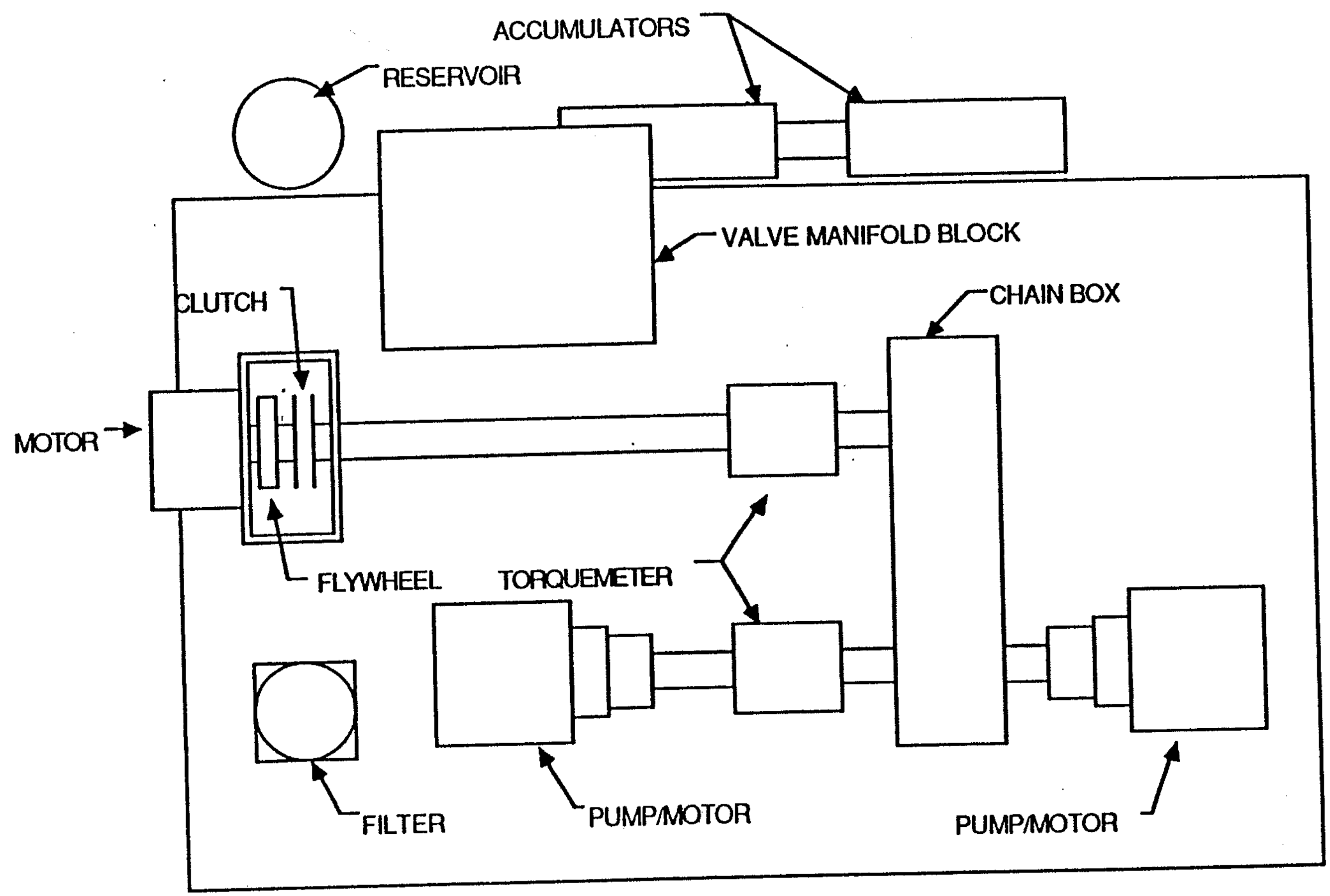

Fig. 2.3: Equipment Layout on the First Test Bench

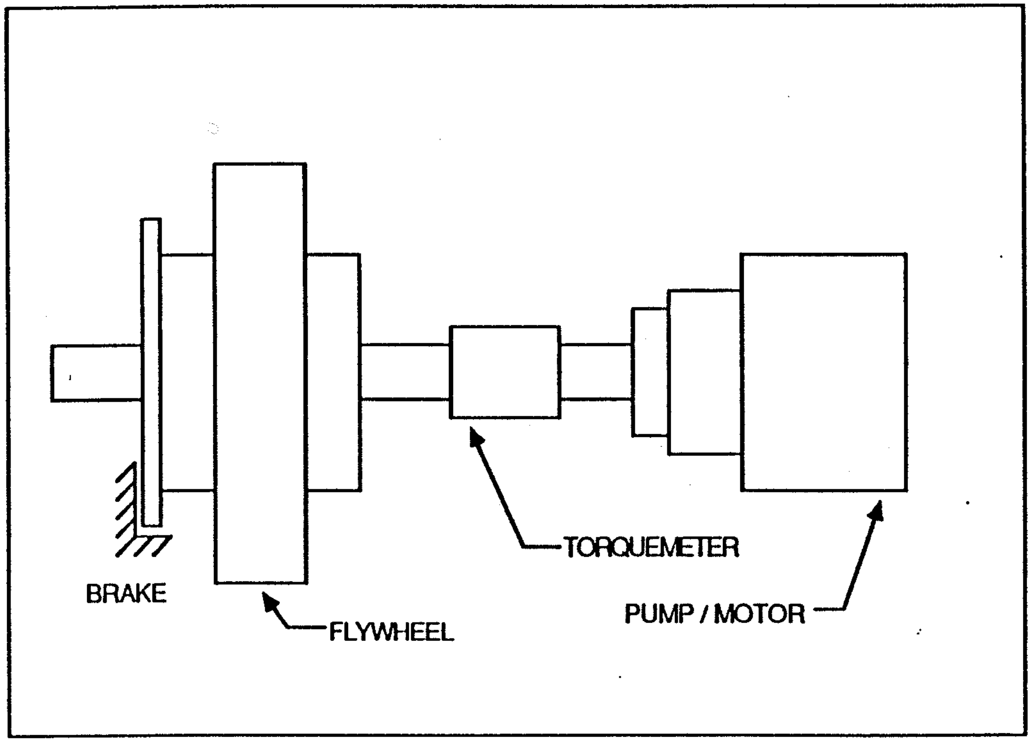

Fig. 2.4: Equipment Layout on the Second Test Bench

thesis. A hydraulic pump/motor, torque transducer, and flywheel are connected in line and are mounted on the test stand shown in Fig. 2.4. The flywheel braking system which consists of a hand pump, an accumulator, cylinders, and a brake pad are also mounted on this test stand. Protective guards were placed over the flywheel and torquemeter. In addition to the major system components listed are the hydraulic hoses, tubes, valves, fittings, and the electrical control and data acquisition components.

The following sections identity the test system's major components. The operating and control characteristics of the components are discussed in general terms. The component and system control actions taken during the actual test runs are discussed in section 4.

2.3.1 Simulated Engine

The simulated engine consists of a Vickers 50 horsepower hydraulic power supply, a Volvo hydraulic motor model F11-39, Moog flow control servovalve model 72-161, a variable orifice, an engine flywheel, and a clutch. These components were chosen to have the capability to simulate the operating characteristics of a typical 2.5 liter, 4 cylinder engine. The torque-speed curve for lines of constant throttle openings of an internal combustion engine as well as the lines of constant supply pressure for a simulated engine are shown in Fig. 2.5.

Fig. 2.6 shows a diagram of the simulated engine. An actual internal combustion engine could have been used rather than the simulated engine; however, the simulated engine has several advantages. The simulated engine can be easily modified to model the torque-speed curves of many actual engines. The relationship between torque and vehicle speed and throttle position determine the controllability and response of the engine. An internal combustion engine would also have the additional problems of fuel storage, emissions, and noise.

The design of the simulated engine is based on the principle that the pressure drop across an orifice is proportional to the square of the flow rate and proportional to the orifice area. The servovalve and variable orifice act as two orifices in series. A pressure transducer placed after the servovalve is used to provide feedback to a controller. The pressure to the hydraulic motor is proportional to the output torque when the motor is operating in a range where its mechanical efficiencies are near constant. The flow rate to the hydraulic motor is essentially proportional to its speed when the motor is operating in a range where its volumetric efficiencies are near constant. The simulated engine torque at zero speed is

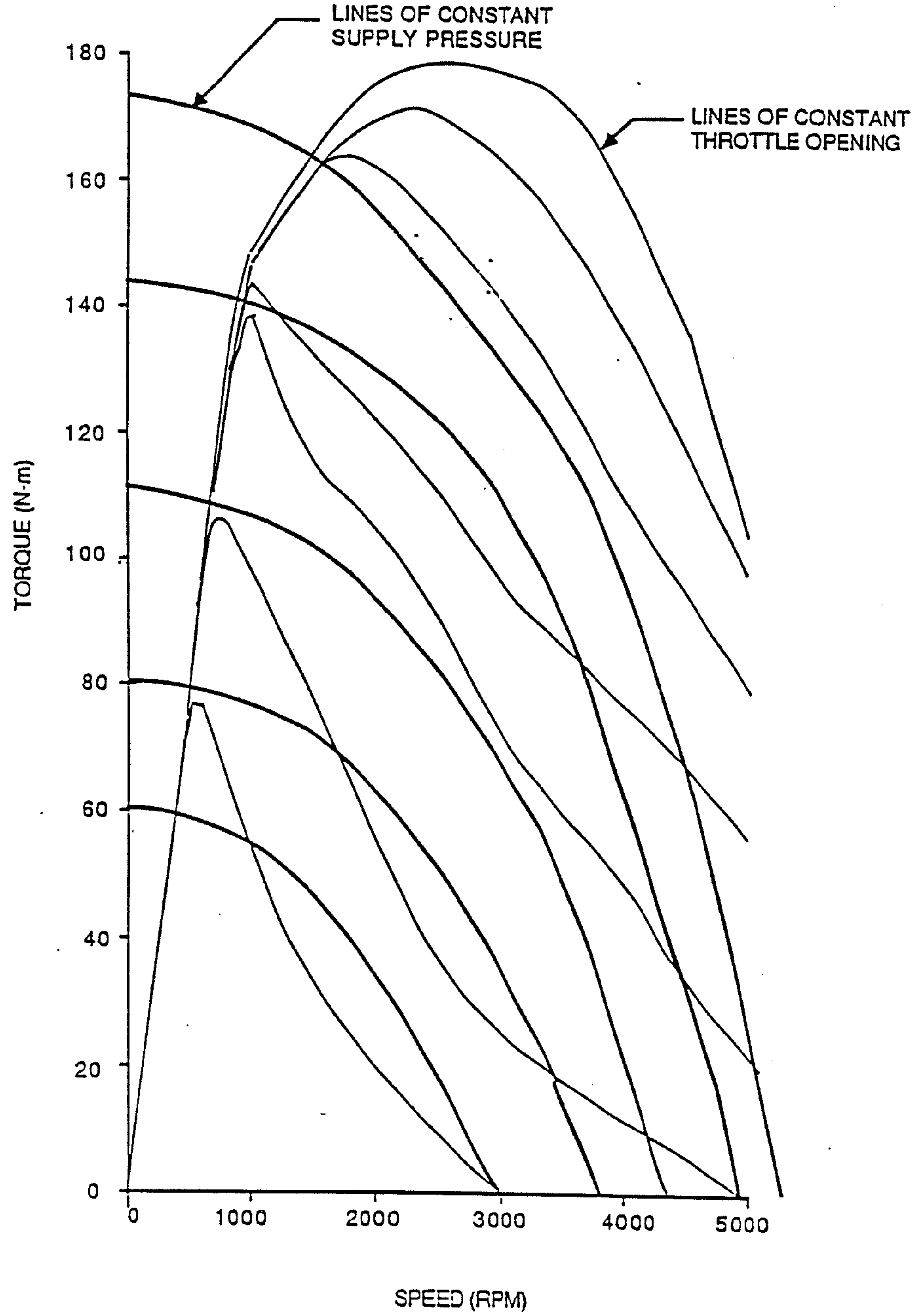

Fig. 2.5: Torque-Speed Curves of Actual and Simulated Engines

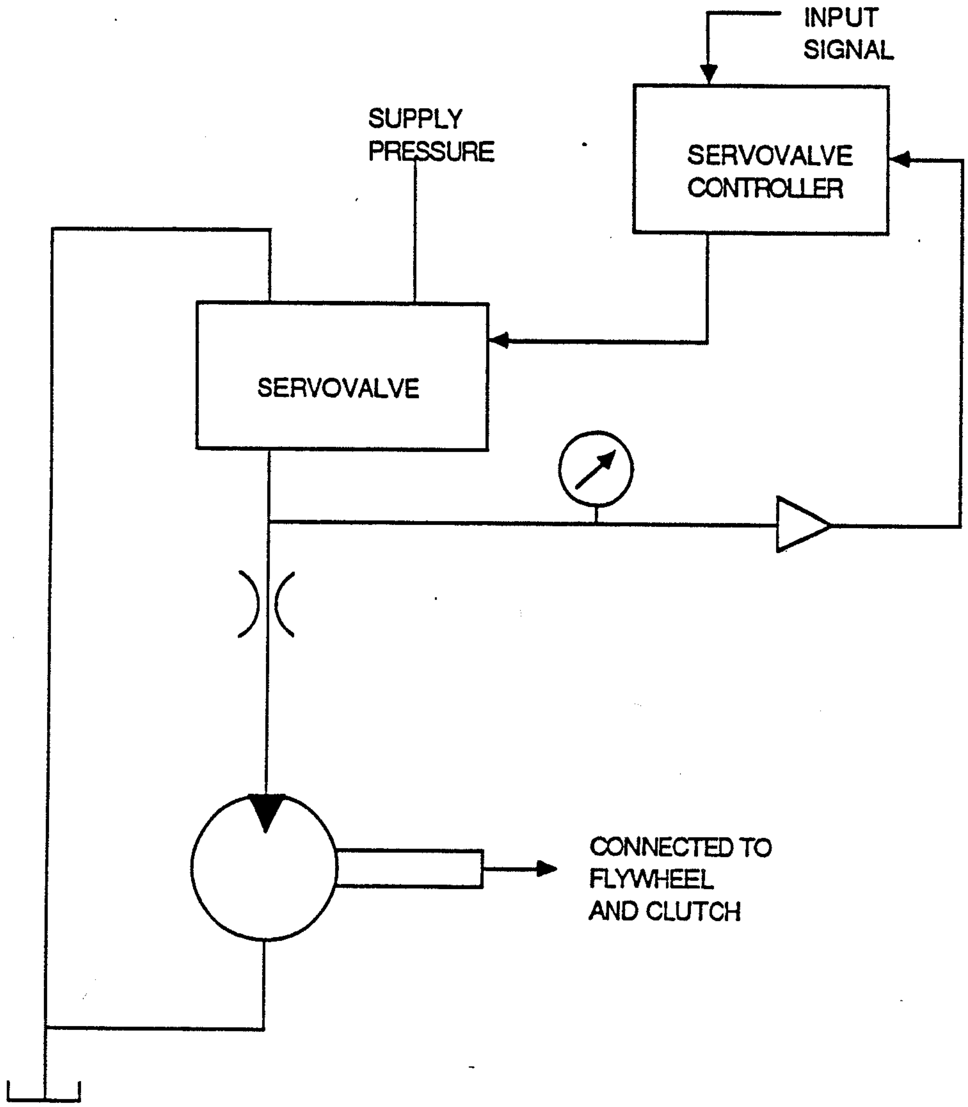

Fig. 2.6: Schematic of the Simulated Engine

proportional to the supply pressure from the power supply. As the servovalve opens the simulated engine speed increases and the torque drop is proportional to the square of the speed. The torque-speed relationships of any engine can be theoretically be matched by proper adjustments of the orifice opening, the servovalve opening, and supply pressure [10]. Accuracy in exactly matching a particular engine's performance in not necessary. A fairly close fit over the desired operating range is all that is necessary. Fig. 2.7 shows the simulated engine torque-speed curve used in the system test runs. Fig. 2.7 also shows the supply pressure as a function of the engine speed. The engine speed is limited to 1870 rpm, the speed at which the supply pressure drops off. This pressure drop is caused by the limitations of the hydraulic power supply used. The operating point of the engine is also identified on the torque-speed curve.

2.3.2 Hydraulic Engine Pump

The pump which is connected mechanically to the engine through a gear box was a Rexroth model A2V 107. The variable displacement, bent axis pump has a maximum displacement of 107 cm**3/rev. The pump's displacement is changed using a Rexroth model HD hydraulic control. The HD

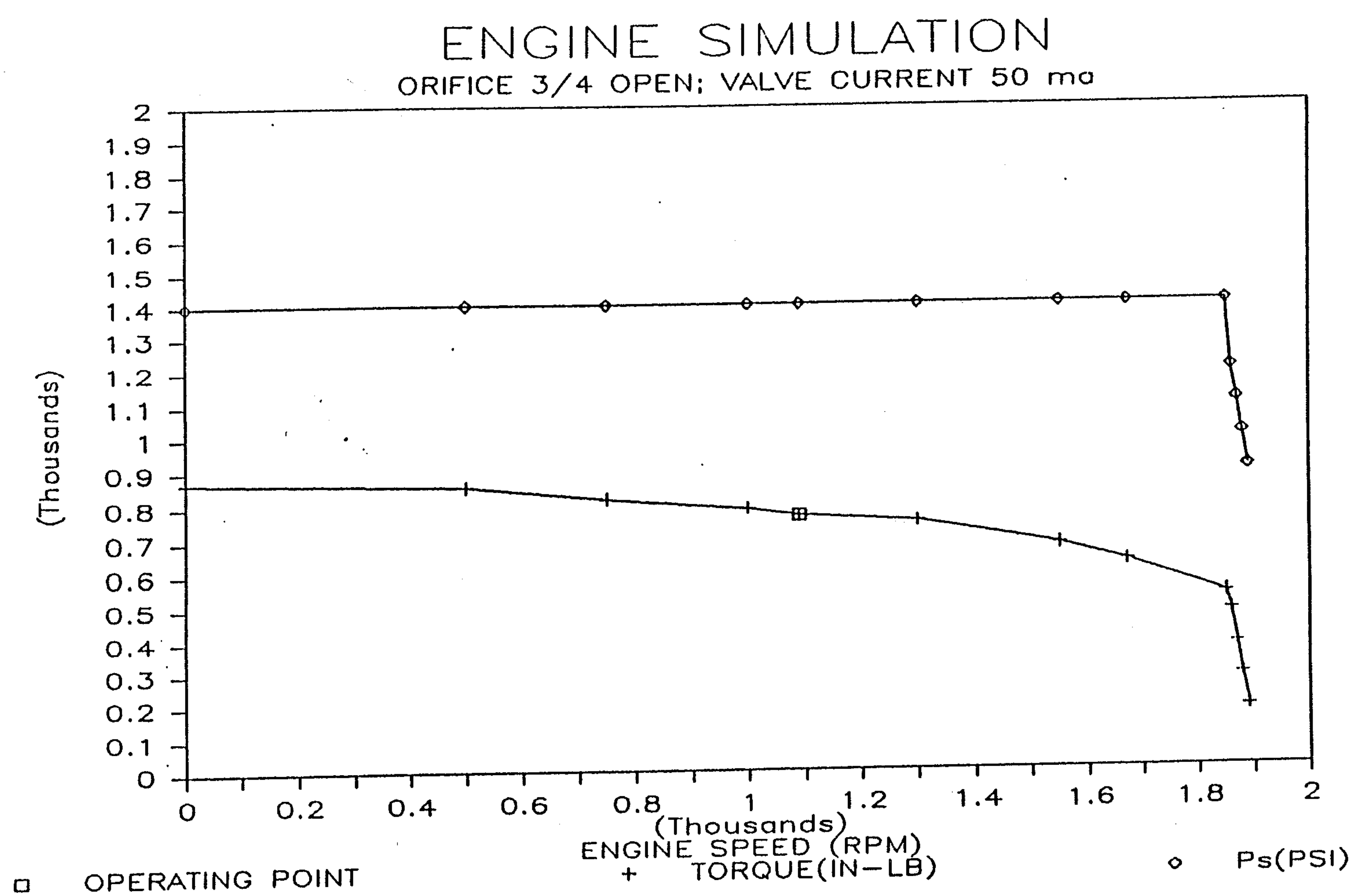

Fig. 2.7: Simulated Engine Torque-Speed Curve

controller consists of direction control valves and a control cylinder which adjust the pump displacement in relation to a pilot pressure. The pilot pressure controller consisted of a servovalve, a servovalve controller, and a pressure transducer. The servovalve used was a Moog flow control servovalve model 30S020. The controller was a Moog servovalve controller model 121-132. A diagram of the pump controller is shown in Fig. 2.8.

The pump displacement controller operates in the following manner. The displacement of the pump is a function of the pilot pressure. A voltage is input to the servovalve controller. The pilot pressure was calibrated to be a linear function of the voltage input. Since the Rexroth pump angle is also a linear function of the pilot pressure, the voltage input is equivalent to a Rexroth pump angle request. A pressure transducer provides a voltage feedback to the controller. The controller compares the input command voltage to the feedback voltage to determine the error. A current is sent to proportionally adjust the spool position of the flow control servovalve in relation to the pressure error. Changing the spool position of the flow control servovalve adjusts the pilot pressure.

2.3.3 Hydraulic Pump/Motor

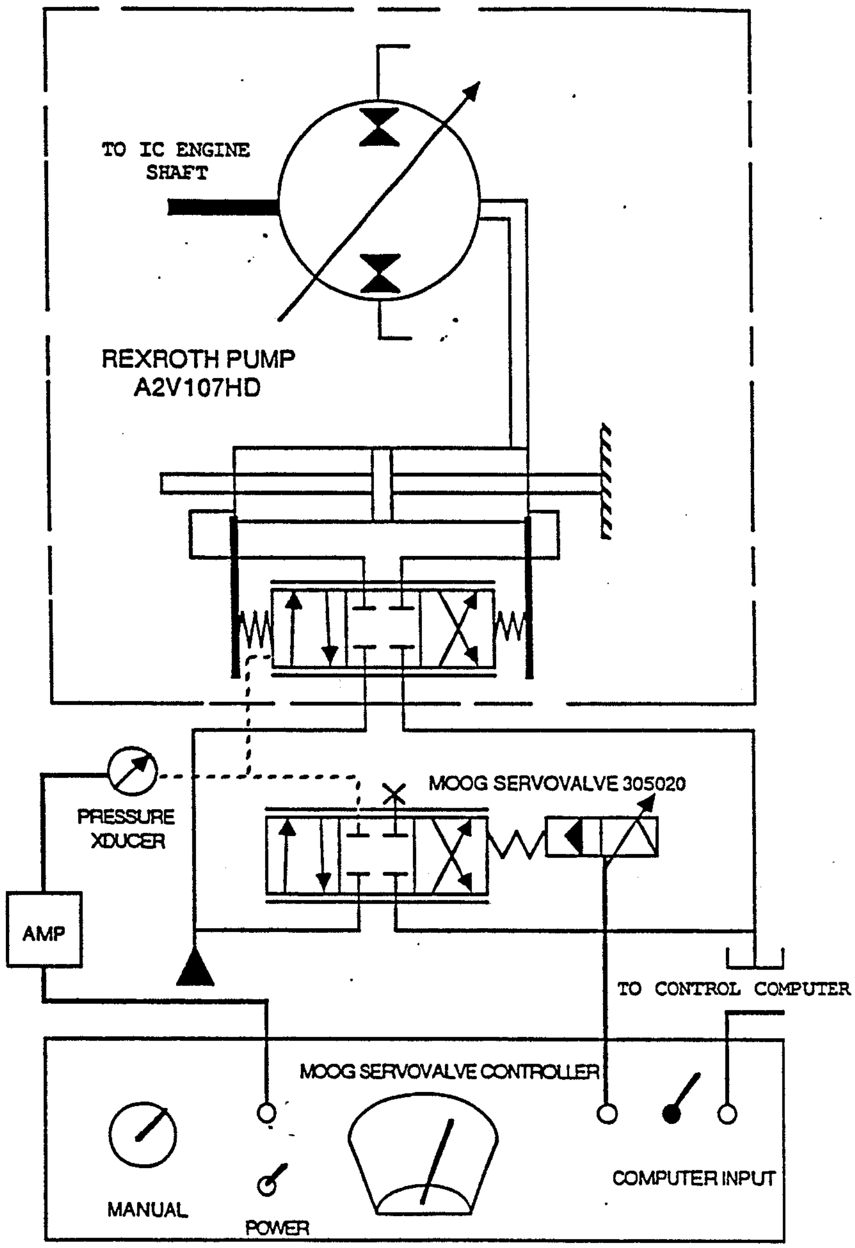

Fig. 2.8: Schematic of the Pump Displacement Controller

The hydraulic pump/motor mechanically connected to the inertial flywheel is a Rexroth model A2V 107. The variable displacement, bent axis pump/motor has a maximum displacement of 107 cm**3/rev. The pump/motor has the capability to go "over center", from positive to negative displacement. The displacement is controlled by a double acting piston which swivels the cylinder block. The pump/motor's function can also be changed by reversing the shaft's direction of rotation.

Control valves are used to control the pump/motor displacement. Fig. 2.9 shows the arrangement of these controlling valves. The first valves are two solenoid valves. The Orshansky HSV-3000 3-way,high speed solenoid valves have 1500 psig (10.34 MPa) operating limits. The valves are fast acting and can reach 90 percent of full flow in approximately 3 milliseconds after receiving a command signal. The valves close in approximately 1 millisecond. Each solenoid valve controls two second stage poppet valves. The Oilgear HSP800 pilot operated poppet values are capable of providing 7.5 gpm (30.69 liter/minute) at a pressure drop of 750 psi (5.17 MPa). The flow rate through the poppet valves was easily modified by adjusting a screw which limited the poppet travel. The pump/motor displacement control piston is moved by activating one of the solenoid valves. The solenoid valve provides a pilot pressure which

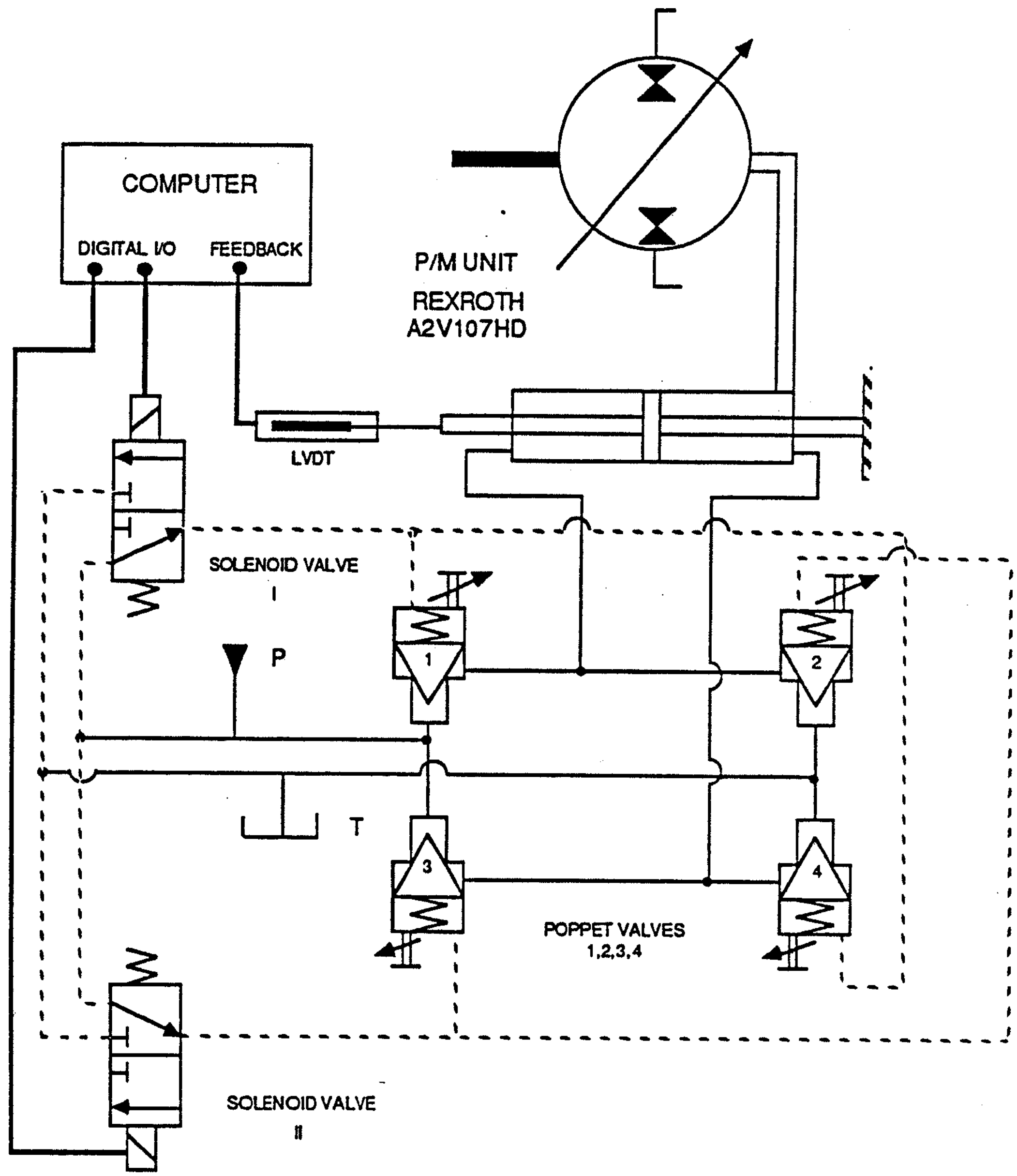

Fig. 2.9: Schematic of the Pump/Motor Displacement Controller

opens two poppet valves. One of the poppet valves allows flow into one side of the piston and the other poppet valve allows flow out of the other side.

The pump/motor displacement feedback was supplied by a Schaevitz linear variable displacement transformer, LVDT, model HR 2000. The core of the LVDT was attached to the pump/motor cylinder block.

The pump/motor's displacement was controlled digitally using solenoid valves, poppet values, a LVDT, and a computer controller rather than a servovalve, because servovalves and servovalve controllers currently are too expensive to be installed in a commercial automobile. Solenoid valves and poppet valves are much less expensive than servovalves. Although the LVDT used in the digital controller system used in the test is expensive, the LVDT could be replaced with an adaptive control system which uses inexpensive feedback devices such as limit switches or linear potentiometers. This type of control system is discussed in [11].

2.3.4 Accumulator Energy Storage Unit

The energy storage unit consisted of two two-gallon (8.185 liter) piston-type Parker accumulators, model A6RO462B69E. The accumulators were both teed off the high pressure line at the same location. The accumulators store energy by compressing an inert gas such as nitrogen on one side of the piston by adding oil on the other side of the piston. The gas side of the accumulator was filled with a high density plastic foam. This foam reduces the thermodynamic losses encountered when the accumulator is charged and discharged, by making the cycling process more isothermal. Current experimental results show that an accumulator thermodynamic efficiency of 98% is realistic [12].

The size of the energy storage accumulator depends on the operating strategy, the vehicle size, the engine size, as well as other factors. The energy which can be stored by an accumulator which operates isothermally can be calculated using the equation:

$$E = PiVi*\ln(Pm/Pi) \qquad (2-1)$$

where E = energy stored in the accumulator

Pi = the precharge gas pressure

Vi = the precharge gas volume

Pm = the maximum gas pressure

We precharged the accumulators to 1000 psig (6.89 MPa). The maximum gas pressure was limited by the 3000 psig (20.68 MPa) pressure rating of the accumulators. The maximum energy which could be stored in the accumulators with isothermal operation is 80058.5 ft-lbs (108.4 kJ).

An LVDT was attached to one of the accumulator's pistons to measure the accumulator piston position from which the flow rate into and out of the accumulator could be determined. The LVDT readings were not input to the computer controller, but were used as evaluate the system's performance after tests were run. The gas pressure of the accumulators was measured with pressure transducers. The pressure transducer values were used as inputs to the control system rather than the LVDT values since pressure transducers are less expensive and would be easier to mount than an LVDT in a vehicle.

2.3.5 Accumulator Reservoir

A 10 gallon (40.92 liter) low pressure bladder-type accumulator was connected to the system's low pressure line. The accumulator served as a hydraulic fluid reservoir. The accumulator also provided a boost pressure of between 50 and 100 psig (345 and 689 kPa) for the hydraulic pump/motor when it operated as a pump. The boost pressure was required to prevent cavitation in the pump.

2.3.6 Flywheel

A flywheel was used to simulate the inertia of the automobile rather than as an energy storage device. The flywheel used was designed by Industries Development Corporation, LTD. The inertia of the flywheel was determined by two methods. A value of 34.783 lb-in-sec**2 (3.929 kg-m**2) was calculated from the geometry of the flywheel. A value of 34.722 lb-in-sec**2 (3.922 kg-m**2) was calculated from the results of a spin down test. The spin down test consisted of the flywheel being accelerated up to 2500 rpm at which point the hydraulic pump/motor was disconnected from the flywheel. The flywheel then decelerated due to aerodynamic and bearing drag. The speed and torque at the flywheel was recorded during the acceleration and deceleration. The flywheel inertia was calculated from the recorded data. The two inertia values are within 0.2 percent. The kinetic energy of a 3000 lb (1360 kg) vehicle traveling at 30 mph (48.3 km/hr) is equivalent to the flywheel rotating at 2374 rpm.

The flywheel had brake rotors attached to its shaft. A disk brake was mounted on the test bench. Road loads on the vehicle were simulated by applying braking pressure on the brake rotor. In addition to the braking road load, there was a small drag on the flywheel due to the two SKF 1217 double-row oil-lubricated ball bearings on which the flywheel was mounted.

3.0 DATA ACQUISITION SYSTEM

An Isaac 41 12-bit Analog-Digital data acquisition system manufactured by Cyborg Corporation was used to sample the recorded parameters. This system was controlled by an IBM computer, model PC. The system was run using Labsoft assembly language functions called within compiled BASIC programs. There were 16 analog input channels and 16 bits of binary input available. The analog channels and binary bits had a maximum sampling rate of 10,000 Hz within the appropriate Labsoft function; however, if both analog and binary inputs were to be read a looping procedure between the analog function and the binary function was necessary. In this looping case the sampling rate was more a function of the compiled BASIC code than the Labsoft function. Maximum sampling rates for sequential analog and binary input programs were around 20 Hz.

During all of the complete system driving cycle tests, nine variables were recorded: driver's input command voltage, flywheel speed, flywheel torque, pump/motor stroke-control piston position, accumulator pressure, accumulator piston position, the simulated engine speed, the simulated engine torque, and the pump stroke control pressure. Fig. 3.1 shows the data acquisition equipment.

During the performance and efficiency testing of

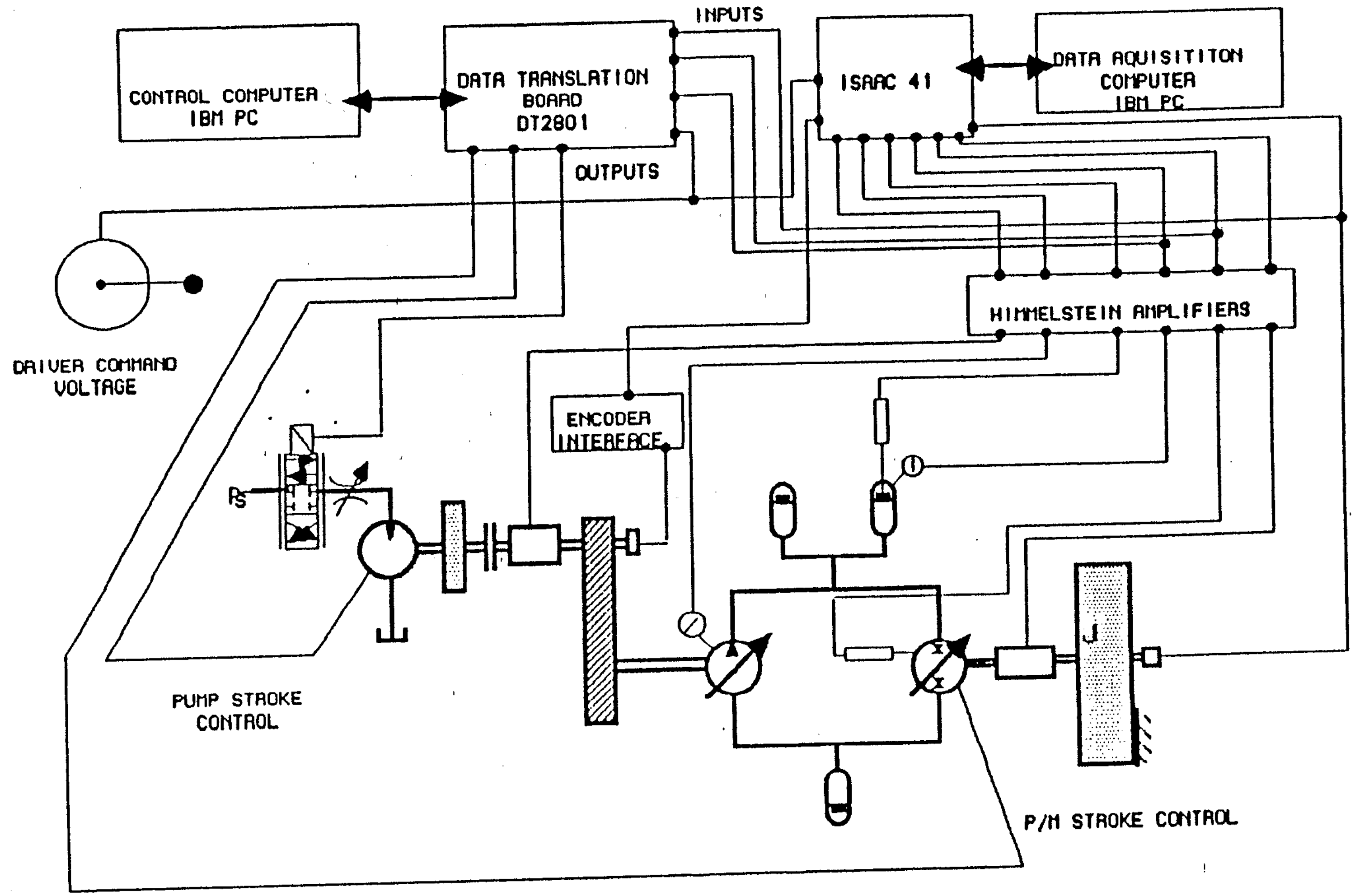

Fig. 3.1: Control and Data Acquisition Schematic

individual components some additional variables were recorded such as accumulator and reservoir temperature.

3.1.0 Instrumentation

The pressure transducers used were Schaevitz model number P2503-0001. The pressure transducers were rated at 0 to 6000 psig (41.4 MPa) and had less than .24% nonlinearity. Their outputs were amplified by Himmelstein System 6 transducer amplifiers model 6-201. The output range of the pressure amplifiers was -5 to +5 volts.

The torque was measured using a Himmelstein System 6 MCRT torquemeter model 9-02T. The transducer had a range of 4000 in-lbs and 0.1% nonlinearity. The torquemeter output signals were conditioned by Himmelstein transducer amplifiers model 6-201. The output range of the torquemeter amplifiers was -5 to +5 volts.

The accumulator was modified so that its piston displacement could be measured using a Schaevitz linear variable differential transformer (LVDT) model 10000HR. The LVDT had a range of +/- 10 inches (+/- .254 m) and 0.24% nonlinearity. A 1/4 inch (.00635 m) diameter rod was rigidly fastened to the accumulator piston. The rod extended out through a seal on the gas end of the accumulator. The rod was then connected to the LVDT core. The pump/motor displacement was measured using a Schaevitz LVDT model 2000HR. This LVDT had a range of +/- 2 inches (.0508 m) and 0.15% nonlinearity. A 1/4 inch (.00635 m) diameter rod connected the pump/motor cylinder block to the LVDT's core. The angle between the bent axis pump/motor's cylinder block and the drive shaft was calculated from the LVDT's output voltage. The LVDT output signals were conditioned using Himmelstein System 6 LVDT amplifier models 6-203A. The output range of the LVDT amplifiers was -5 to +5 volts.

The simulated engine speed was measured using a Hewlett Packard optical encoder model HEDS-6000. A metal code wheel with a resolution of 1024 parts per revolution was fastened to the simulated engine shaft. An encoder interface was built which converted the signal from the encoder into a 16 bit binary code. The 16 bit binary code was a function of both the shaft rotational speed and the encoder interface sampling pulse rate. The resulting 16 bit binary code was input to the ISAAC 41 using all 16 of the available input binary bits. The maximum allowable ISAAC's data acquisition sampling rate was decreased when the encoder interface's binary output was used in combination with the analog inputs to the computer since the computer program had to be more complex. A comparison of the programs required for analog input and for both analog and binary input can be seen in Appendix A and Appendix B.

The flywheel speed was obtained directly from an Electro-Craft Corporation MCM moving coil tachometer generator model M100 which was coupled to the flywheel shaft. The tachometer generator's output voltage signal was a linear function with respect to speed. The gain was 326 revolutions per minute per 1 volt. The output signal did not require amplification.

The two thermocouples used in the accumulator tests were copper-constantan. The one in the accumulator was made of 30-gauge wire. It was connected to an Omega thermocouple amplifier, model Omni Amp I with a 100 Hz Frequency Response. The second thermocouple was located on the inlet to the accumulator. It was made with 24-gauge wire soldered directly onto the 1/2-inch (.0127 m) diameter steel inlet tube about 16 inches from the end of the accumulator. The reservoir oil temperature was measured using a thermometer.

3.1.1 Calibration

Several different methods were used to calibrate the various transducers used in the vehicle simulation. Since the results were to be used in efficiency calculations, great care was given in the equipment calibrations. The calibration of each transducer was checked at least once a day. The transducers were recalibrated if there was any noticeable drift in the output signals.

The pressure transducers were calibrated using an Ametek dead weight pressure tester model R-50. The amplifiers were first nulled at zero input. Then the Himmelstein amplifier gain was adjusted so that the highest expected pressure, normally 3000 psig (20.7 MPa), produced an output voltage near the 5 volt maximum allowed by the amplifier. Setting the gain high increased the pressure resolution. The pressure transducer's steady state output voltages were recorded at several pressure levels. The pressure was then calculated as a function of output voltage using a least square curve fit. Although the transducers were all rated as less than .24 percent nonlinear, a second order function was used in the data acquisition program since it was found be slightly more accurate than the first order curve fit.

Torque transducer calibration was done according to Himmelstein's procedure of shunt calibration resistance. This shunt calibration procedure was found to be accurate in a calibration test which consisted of placing known torques on the shaft and recording the output voltages. The torque was set as a linear function of the amplifier's output voltage based on the zero signal output voltage and the output voltage provided by Himmelstein during the shunt calibration procedure.

The LVDT's were calibrated by recording the voltage at several displacements as measured by a vernier caliper. The amplifier zero setting and gains were set to obtain the largest voltage resolution possible which increased the accuracy of the data. The resulting data were curve fit to obtain the relationship between the two variables. A first order curve fit was found to be a very good approximation.

The engine speed was recorded using an Hewlett Packard optical encoder model HEDS-6000. The output of the encoder was sent to an encoder interfacing counter circuit. This circuit counted the number of pulses occurring during a period of a square wave. Knowledge of the number of pulses, period of the square wave, and the number of pulses per revolution of the encoder shaft allowed for the determination of engine speed. The period of the square wave provided by the encoder interface was adjusted to provide maximum resolution within the limitations of the Isaac 41 16 bit binary input system. The 16 bit output from the optical encoder was compared for accuracy to a 60 pulse per revolution signal of the torquemeter on the engine shaft.

The tachometer generator which measured the flywheel rotational speed was calibrated by having the tachometer generator, a Hewlett Packard optical encoder, and a speed sensor from the torquemeter all mounted on the flywheel shaft. The voltage output from the tachometer generator, the speed produced using the optical encoder and its interface, and the 60 pulse per revolution signal from the torquemeter were all recorded for various flywheel speeds. The resulting data were curve fit to obtain a first order equation relating the output voltage of the tachometer generator and flywheel speed.

3.1.2 Accuracy

Table 4 lists the error band surrounding each of the recorded variables. These numbers represent our best estimate in quantifying the possible maximum error associated with each data point. Possible sources for the errors include drift, nonlinearities, hysteresis, and thermal sensitivity of the transducers and amplifiers, external noise, curve fit approximations, and resolution of the data acquisition system.

TABLE I: ERROR BANDS

Control Voltage	+/-	0.02	Volts
Flywheel Speed	+/-	7.5	RPM
Flywheel Torque	+/-	15.0	In-Lbf (1.694 N-m)
Pump/Motor Degrees	+/-	0.2	Degrees
Accumulator Pressure	+/-	15.0	PSIG (103.5 kPa)
Accumulator Position	+/-	0.025	Inch (.000635 m)
Engine Speed	+/-	5.0	RPM
Engine Torque	+/-	15.0	In-Lbf (1.694 N-m)
Pump Control Pressure	+/-	15.0	PSIG (103.5 kPa)

The error band ranges were determined by addition of the observed and rated maximum errors due to each component error source. Each transducer's drift was determined from the amount of recalibration required. The nonlinearities of the transducers were determined from the manufacturer's calibrations and from the curve fit approximation results. The external noise was determined from observing the output signal noise resulting from a nonvarying known input signal. This noise when analyzed often had a 60 Hz component due to all the electrical equipment in operation. Shielded cables were used to reduce the electrical noise. The Isaac 41 data acquisition resolution errors were a result of the 12 bit analog system capability. The Isaac 41 has a 2.44 millivolt analog input resolution. The pressure transducers which were calibrated to read 3000 psig at full scale had a 1.46 psig, .049 percent, resolution accuracy due to the 12 bit system resolution.

3.2.0 Data Acquisition Program

An example of a data acquisition program is given in Appendix A. The data acquisition program listed in the appendix is for multichannel analog input. The program was written in a very general code so that modifications such as sample rate, number of samples, number of channels, and scaling factors require only a change in the input file and not in the program code. The program outputs the acquired data in tabular form. Data from different channels is output in columns. Rows contain data recorded at the same time. The output is in a format compatible with Lotus 123, a spreadsheet and graphics program from Lotus Development Corporation. This compatibility allows for ease in manipulating and graphing the test results.

The program which was used to record the data during all of the complete system driving cycle tests is listed in Appendix B. The program which converted the data obtained by the program listed in Appendix B into engineering units is listed in Appendix C. The data acquisition rate for typical runs of the complete system was 6 samples per second. The program required both analog and binary inputs which lowered the maximum allowable data acquisition rate. The binary input recorded using the optical encoder and its interface was the engine speed. An additional subroutine was required which plotted the flywheel speed on the computer monitor at a rate equal to the sampling rate so that the driver could control the flywheel speed by adjusting the input command voltage. Although a continuous flywheel speed output was also sent to a digital oscilloscope, the oscilloscope's screen resolution was not fine enough to allowed the driver to control the flywheel speed as accurately as with the computer monitor.

The number of data points which could be collected was limited by the size of the computer's memory. Only 800 sampling points could be taken when the 9 variables were recorded.

4.0 CONTROL IN HYBRID ENERGY STORAGE VEHICLES

A system consisting of the required mechanical, hydraulic, data acquisition, and control components was assembled in order to test the effectiveness of proposed control strategies and techniques in hybrid energy storage vehicles. A microprocessor based control system is desirable because of the number and complexity of the factors required to determine the best control decisions at each instant in time. The speed of the control actions should be such that the driver does not perceive a lag in the vehicle response due to the controls. The ability to vary the test conditions was important since the effectiveness of the control could depend on many factors such as the specific system configuration, the type and properties of the components, the speed of the data acquisition and control microprocessor, the control program, the transducers, and the driver. The tests were run on a laboratory test bench rather than in a car to obtain the desired system flexibility.

4.1.0 Control of the Proposed System

The control system was evaluated on the passenger automobile dual pump/motor system shown in Fig. 3.1. In addition to the component arrangement shown, two limitations are made: the engine is large enough to provide all the desired power in the absence of the energy storage system, and the accumulator is as small as possible without noticeably reducing the fuel economy or vehicle operation and performance.

4.1.1 System Properties

The system acts as two subsystems: the engine-pump subsystem and the pump/motor subsystem. The two subsystems are linked by the high and low pressure lines. The high and low pressure accumulators in these lines reduce the dynamic interactions between the two subsystems. Since the two subsystems are virtually isolated from each other when the energy storage accumulators are in the system, the control of the two subsystems are generally independent of each other.

The engine-pump subsystem's primary function is to pump hydraulic fluid into the energy storage accumulator to maintain the pressure in the energy storage accumulator between high and low operating values which are determined by the controller. The required inputs to the engine-pump subsystem controller are the energy storage accumulator pressure and the vehicle speed. The state of the pump/motor subsystem is not a required input to control the engine-pump subsystem.

The pump/motor subsystem functions to provide the torque requested by the driver and to recover energy during regenerative braking. These functions can be accomplished by having the energy storage accumulator pressure and the driver's input as inputs to the pump/motor subsystem. The state of the engine-pump subsystem is not a required input to control the pump/motor subsystem.

The system components such as the engine and the hydraulic pump/motors normally do not have sharp efficiency peaks near the highest efficiency operating region. The component efficiencies are not significantly affected by reasonably small differences in the operating states. These efficiency profiles allow the control to be simple and have small errors without a noticeable loss in the system efficiency.

The control system's feedback is provided by the driver. The driver's commands are essentially a desired torque request. The driver continuously adjusts the position of the accelerator pedal to match the the car's performance to the desired performance. If the car is going too slow, the driver simply pushes down farther on the accelerator pedal. If the car is going too fast, the driver lets up on the accelerator pedal. Since the driver's input is based on his perception of the vehicle's velocity and acceleration, the relationship between the accelerator pedal position and the system output torque does not have to be a precisely defined and accurate function. The driver has little feel for absolute pedal position. The output torque should continuously and smoothly increase with an increase in the driver's input command. Discontinuities in the relationship would make the system difficult to control.

Since many of the system variables are related, variables needed for control can be calculated from other known variables. These relationships can be used in two ways: the number of transducers can be reduced or the values recorded by various transducers can be checked. The use of calculated variables will reduce the cost of the system. The system reliability could be increased by eliminating the least reliable transducers. One useful interrelationship between the variables is the torque of the pump/motor which can be calculated knowing the pressure, the displacement, and the mechanical efficiency at that operating point.

4.1.2 Proposed Control Policies

Even though an unstable system can be made to operate in a stable manner through compensation techniques, strategies that are inherently stable have some desirable operating characteristics. The possible danger using an inherently unstable system with compensation is that a variation or malfunction in a component can cause the system to go more unstable than a inherently stable system. The disadvantage of a very stable system is that the response is often not as fast as a less stable system.

The following control policies provide a basis for the control algorithms which provide control actions in a stable manner with adequate response and accuracy.

1) When the engine is off and the vehicle is operating, the engine is left off until the energy storage accumulator's pressure is reduced to a predetermined value. This value is determined from the amount of stored energy required to restart the engine. Depending on the other policies, the engine may be required to be turned on before the previously determined value if the accumulator can not provide sufficient power to meet the demand. This policy minimizes the engine on/off cycling rate and maximizes the engine off time.

2) When the engine is operating one of two strategies should be employed [13]. In the "ideal point" operating strategy, the engine is operated only at a torque and speed

which yields the most efficient system operation. In the "ideal line" operating strategy the engine is run most of the time at an established minimum allowable power output level which still provides an operating efficiency close to the peak efficiency. When more power is required than is produced at the minimum allowable power output level, the engine torque and speed are increased. These most efficient higher power level torques and speeds form a line on the engine's torque speed map. Following the "ideal line" operating policy postpones as long as possible the instant when the engine is shut off which reduces the engine cycling rate. The two strategies are illustrated in Fig 4.1.

3) When the engine is operating, it is kept on until the energy storage accumulator is charged to an energy level that is a function of the vehicle speed [14]. Fig. 4.1 illustrates the allowable accumulator energy levels as a function of the vehicle speed. When the vehicle is operating at low, urban driving speeds the energy capacity reserved for braking is essentially equal to the kinetic energy of the vehicle at the current operating speed. Actually the energy reserved is slightly less that the vehicle's kinetic energy since system losses will prevent all of the kinetic energy from being recovered. Since in normal driving conditions a hard stop from high highway

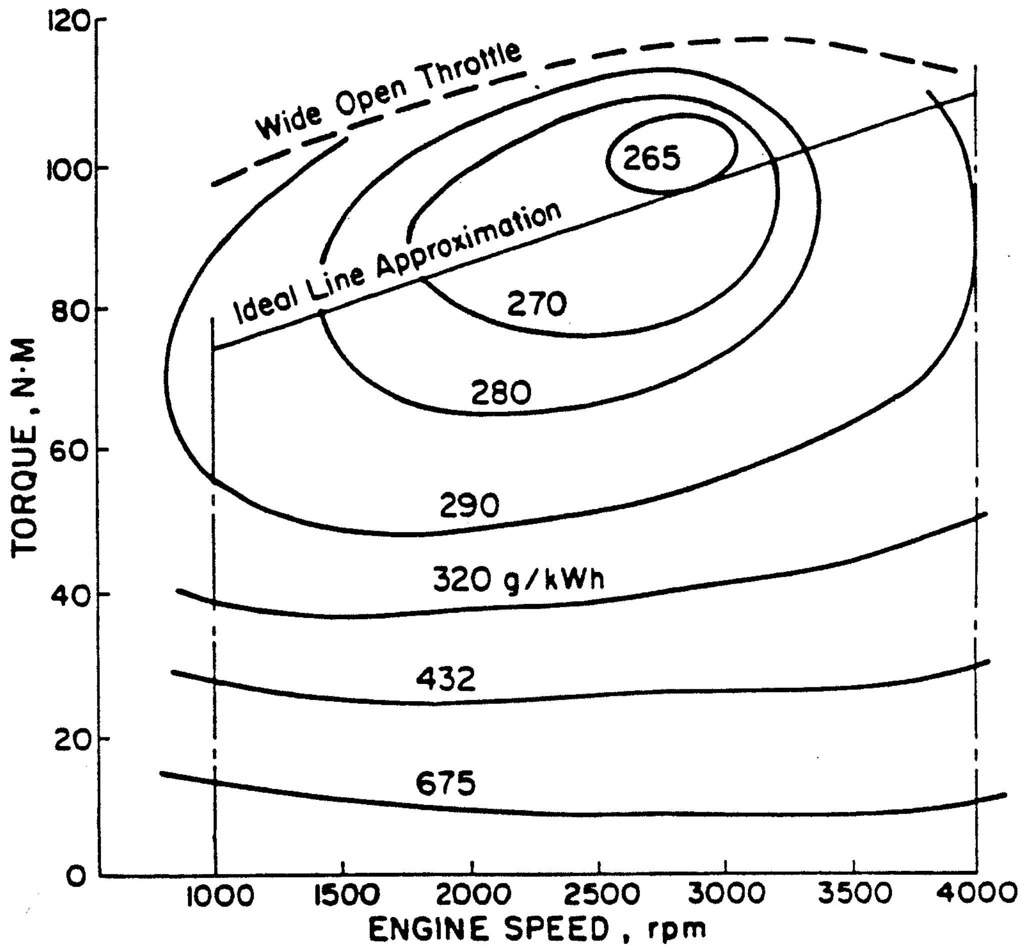

Fig. 4.1: BSFC Map for an IC Engine

driving speeds is rare, it would not be practical to reserve all the regenerative braking energy available as a result of hard stops from highway speed. Reserving such a large energy level would not allow the engine operation to be decoupled from the road load. During rare highway stops, some of the energy would be lost due to friction braking or by dumping some of the pump/motor's output over a relief valve.

4) The accumulator's energy level is such that under normal driving conditions most or all of the possible regenerative braking energy is recovered. The energy level must also provide sufficient ability to uncouple the road load from the power being provided by the engine. Fig. 4.2 shows the energy reserved for regenerative braking and the energy reserved for the power decoupling function. The allowable accumulator energy is the sum of the energies for these two functions. The most efficient accumulator size is a function of the driver, the road conditions, the vehicle size, and the driving cycles. The accumulator size is a somewhat empirical number based on numerous computer simulations [9,14].

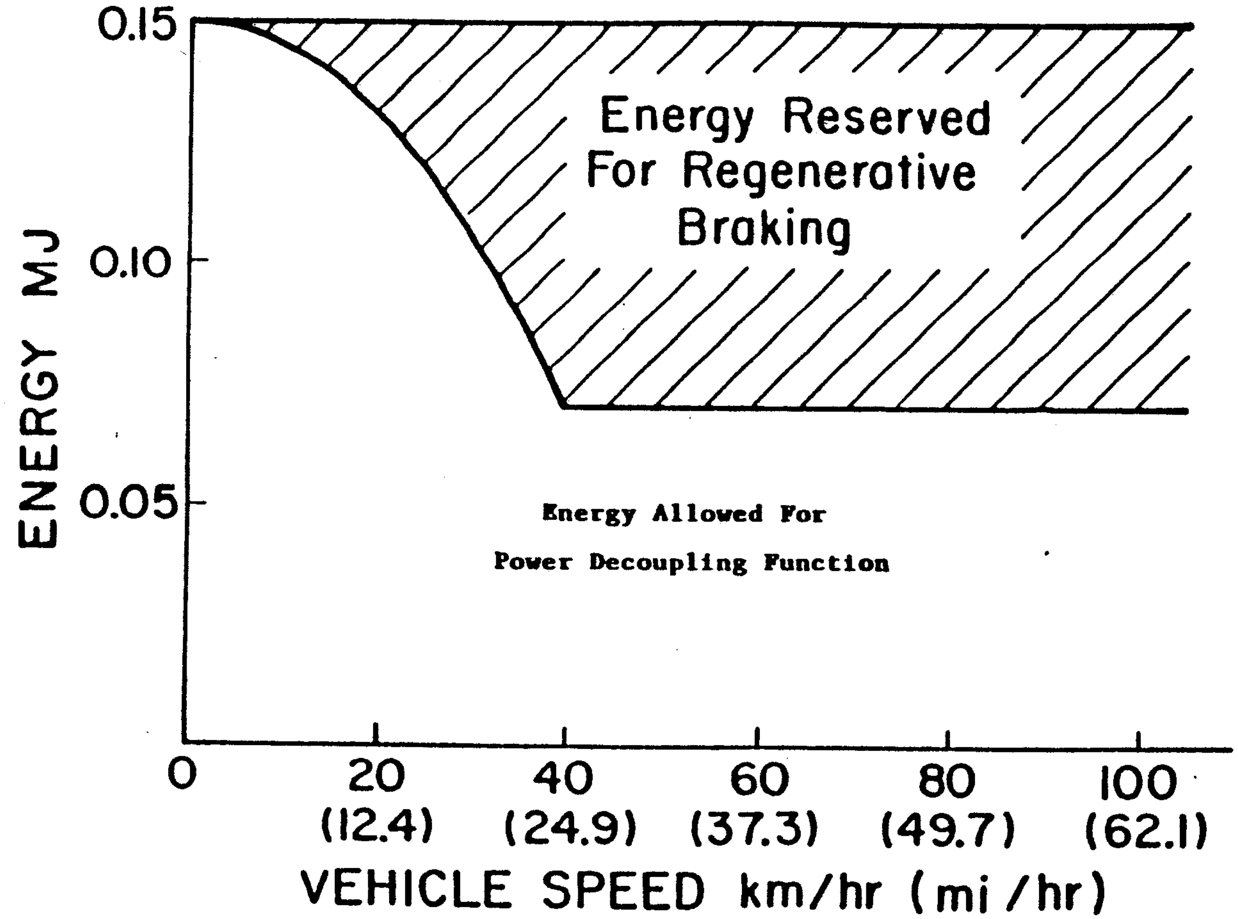

Fig. 4.2: Accumulator Energy Storage Control Policy

4.1.3 Implemented Control System

The microprocessor control system was implemented using an IBM computer model PC and a Data Translation control board model DT2801. The board has 12 bit analog to digital, A/D, and digital to analog, D/A, capability. The DT2801 board has 8 double ended A/D input ports, 2 double ended D/A output ports, and 16 binary input and output ports. The Data Translation board can be operated using a variety of languages, but it was designed to by controlled by commands written in IBM Personal Computer's BASIC language. All the control programs were operated using compiled BASIC.

The computer control flowchart for the program used to control the system during the driving simulation is shown in Fig. 4.3. A listing of the control program is given in Appendix D. The computer control system used during the driving simulation required 4 analog inputs, 2 binary outputs, and 2 analog outputs. The analog inputs represented the driver's input command, the accumulator pressure, the pump/motor displacement, and the flywheel speed. The binary outputs were inputs to the solenoid valves which controlled the pump/motor displacement. The analog outputs were for the pump/motor displacement servovalve controller and for the engine servovalve.

The control flowchart consists of two main loops. The

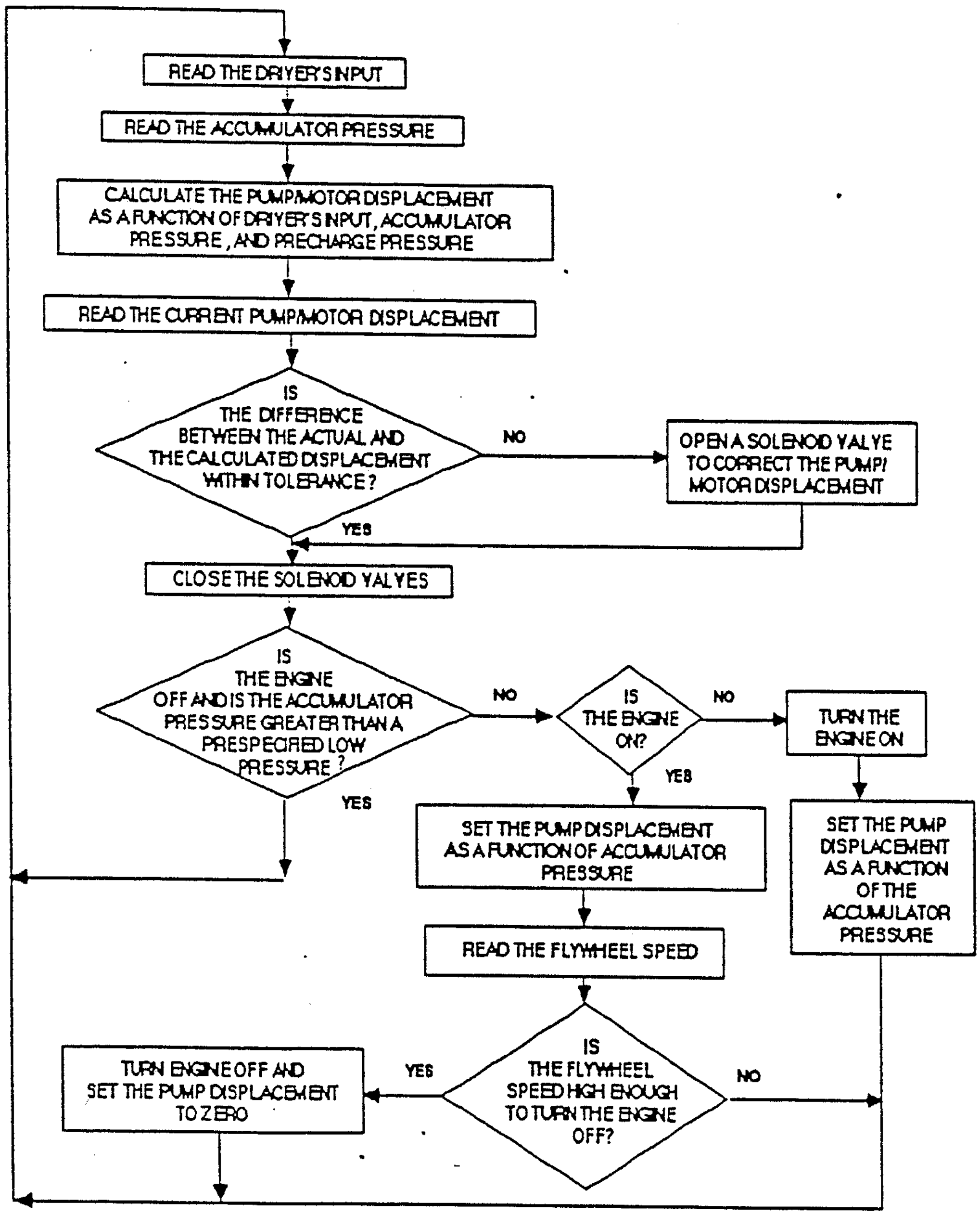

Fig. 4.3: Computer Control Flowchart

first loop is shown on the upper left of Fig. 4.3. The function of the first loop is to control the displacement of the hydraulic pump/motor. This loop is executed once during each controller cycle. The controller used executed the first loop alone at 31 Hz.

The execution steps in the first loop is as follows:

(1) The driver's input command is read. In our system the driver used a potentiometer to sent a voltage signal to the computer. The voltage ranged from -5 to +5 volts. Positive voltage values represented the driver's accelerator pedal position. Negative voltage values represented the driver's brake pedal position. The driver's input voltage value was equated to a torque value using a linear conversion equation.

(2) The energy storage accumulator pressure is read using a pressure transducer. In our tests the pressure transducers read the pressure in the nitrogen side of the accumulators. Mounting the pressure transducers on the nitrogen side of the accumulators allowed for ease in checking the precharge pressures in the accumulators. The pressure could have been read on the hydraulic fluid side of the accumulators. The pressure difference between the nitrogen and hydraulic fluid sides of an accumulator which

is operating above the precharge pressure is dependent on factors such as the type of accumulator, the charging and discharging rates, and the friction of the piston. During the normal operation of our system this pressure difference was very small; however, if the hydraulic fluid pressure was allowed to fall below the precharge pressure then the difference between the nitrogen and hydraulic fluid would be significant. If foam was not included in the nitrogen side of the accumulator to reduce the thermal effects during compression and expansion of the nitrogen then the pressure would not give as accurate an estimate of the energy stored.

(3) The desired pump/motor displacement is calculated. This desired displacement is a function of the driver's input command, the current accumulator pressure, and the accumulator's precharge pressure. The function used is discussed in detail in section 4.3.1.

The pump/motor displacement function is set so that the maximum torque is independent of the current accumulator pressure. This independence is desirable since it prevents the loss of available torque as the accumulator pressure varies. Losing available torque could be dangerous if it occurred during a driving situation such as passing a vehicle. The accumulator could be closed off in this case.

(4) The actual pump/motor displacement is read using a LVDT which in connected to the cylinder block of the pump/motor.

(5) The error between the actual displacement and the desired displacement is determined. The control program operates using an on-off control strategy. If the error is less than a tolerance voltage, then the displacement of the pump/motor is fixed at the current position by setting the binary output control port's voltages to close the solenoid valves which control the pump/motor's displacement. If the error is greater than the tolerance voltage, then one of the binary output control port's voltages is set to open the appropriate solenoid valve to control the pump/motor's displacement in the proper direction.

(6) If the engine is off and the accumulator pressure is greater than a previously established low pressure limit, then the control loop is sent to execute the first loop again. If the conditions are not satisfied, then the second loop is executed. The on/off state of the engine is maintained in the control program through the use of a flag.

The second control loop is shown on the lower right section of Fig. 4.3. The functions of the second loop are to control the displacement of the hydraulic pump and to turn the engine on and off. The controller used executed the first and second control loop at 21 Hz.

The execution steps in the second loop are as follows:

(1) The engine is turned on if the engine is off.

(2) The desired pump displacement is calculated. This desired displacement is a function of the current accumulator pressure. The function used is discussed in detail in section 4.2.2.

(3) The flywheel speed is read using a tachometer generator if the engine has not just been turned on.

(4) The flywheel speed is used to a calculate the allowable high accumulator pressure. The allowable accumulator pressure is calculated a function of the flywheel speed, the accumulator's maximum pressure, and the flywheel's inertia. Since the flywheel's inertia and the accumulator's maximum pressure are constants, the allowable accumulator pressure function has the flywheel speed as the only variable. The allowable accumulator pressure is calculated in the control program as the accumulator

pressure which has a potential energy equal to the difference of the maximum potential energy which could be stored in the accumulator and the current kinetic energy of the flywheel. This strategy allows all the kinetic energy to be regeneratively recovered during braking from any speed. Since it is rare for hard braking to occur from high speeds during the normal operation of an automobile, the energy storage level strategy used in this thesis, allowing all kinetic energy to be recovered, may decrease the overall cycle efficiency of the vehicle. In practice the allowable accumulator pressure could be raised at the higher vehicle speeds.

(5) If the calculated allowable accumulator pressure is less than the current accumulator pressure, then the engine is turned off and the pump displacement is set to zero. The engine is turned off by closing the simulated engine's servovalve in the current test setup.

(6) The control program is sent to the start of the first control loop.

4.1.4 Evaluation Criteria

The following criteria can be applied to evaluate the suitability of the control system: the overall fuel economy of the system should be maximized, the system should be responsive, stable, and have satisfactory driving characteristics, the required component's sizes and costs should be low, and the system reliability and life should be high.

Several tests were run to evaluate the system's controllability and stability. The evaluation was performed through the use of differing combinations of initial conditions. The initial conditions that were varied include the engine's initial on or off state, the energy or lack of energy stored in the accumulator, the driver's initial acceleration or braking command. Changing the initial conditions allows the transient and steady state responses to be observed.

4.2.0 Control of Components

The following sections discuss the control equations used and the theory behind those equations for several of the system's components. The hardware used in the control of the components was discussed in section 2.

Many combinations of hardware and software could be used in the control of the system's components. For example, the displacement of the hydraulic pump and pump/motor could be controlled using an electric or hydraulic stepping motor with rack and pinion, a piston and servovalve with an analog servovalve controller, a series of digitally controlled valves, or many other controllers. The choice of controller depends on factors such as the required displacement accuracy, the system stability, the required response speed, the controller and system efficiencies, the component sizes, and the cost.

4.2.1 Pump/Motor

The relationships between the pump/motor's operating variables used by the control program are explained in the following paragraphs.

The pump/motor displacement is a function of the current accumulator pressure and the driver's command. The driver commands a torque with both the accelerator and the brake pedals. The conversion factor between the driver's input command and the torque from the pump/motor is set such that the maximum torque allowed is equal to the ideal torque when the pump/motor is operating at near full displacement and at the precharge pressure of the accumulator. This strategy sets the maximum torque as high as allowed by the pump/motor displacement and the accumulator precharge pressure, but not so high as to reduce the independence of the current accumulator pressure and the available torque. The maximum torque is calculated using the following equation.

$$Tm = Dm * Pi \tag{4-1}$$

where Tm = the maximum torque

Dm = the maximum allowed displacement of the pump/motor

Pi = the precharge pressure

The mechanical efficiency of the pump/motor is not included in equation (4-1). The mechanical efficiency is not included in the any of the following equations. Not including the pump/motor's efficiency in the equations makes these equations approximations of the actual relationships. When the pump/motor is functioning as a motor the product of the motor's displacement and the pressure will be greater than the output torque of the unit by an amount that is dependent on the mechanical efficiency. When the pump/motor is functioning as a pump the product of the pump's displacement and the pressure will be less than the input torque to the unit by an amount that is dependent on the mechanical efficiency. The lower the mechanical efficiency the greater the approximation errors.

The efficiency of the pump/motor is not included in the equations for several reasons: the relationships are changed from simple linear equations to more complicated nonlinear equations by including the mechanical efficiencies, the correct mechanical efficiencies are usually not precisely known, the mechanical efficiencies will vary from model to model for a particular pump/motor design, the efficiencies of pump/motor's can vary significantly depending upon operating conditions such as the pump/motor's speed, displacement, pressure, oil temperature, and oil viscosity, and the driver will automatically correct for the approximations.

The driver will continuously adjusts the position of the accelerator pedal to match the the car's performance to the desired performance. In order for the driver to have the desired control over the vehicle's performance the relationship between the accelerator pedal position and the system's torque does not have to be a precisely defined and accurate function. The torque only has to be continuous and it should smoothly increase with an increase in the driver's input command. Any discontinuities in the relationship would make the system difficult to control. Since hydraulic pump/motors normally have smooth efficiency profiles, the driver has control over the vehicle's speed.

One disadvantage of not including the mechanical efficiency is that the actual maximum output torque of the motor or input torque of the pump will vary depending upon the operating conditions at that instant. The greater the variation of the efficiencies of the pump/motor, the greater the variations in the maximum available torque.

A linear relationship between the ideal maximum pump/motor torque and the driver's command voltage corresponding to the maximum pump/motor displacement was written in the following form.

$$Tm = K1 * Vdrm \tag{4-2}$$

where Tm = the maximum pump/motor torque

K1 = a constant

Vdrm = the driver command voltage at the maximum pump/motor displacement

Equation (4-2) allowed the constant K1 to be determined. The constant K1 determined the relationship between the pump/motor torque and the driver command voltage.

$$T = K1 * Vdr \tag{4-3}$$

where T = the pump/motor torque

K1 = a constant

Vdr = the driver command voltage

The constant K1 was then replaced by the appropriate variables to form equation (4-4).

$$T = ((Dm * Pi) / Vdrm) * Vdr \tag{4-4}$$

where T = the pump/motor torque

Dm = the maximum allowed displacement of the pump/motor

Pi = the precharge pressure

Vdrm = the driver command voltage at the maximum pump/motor displacement

Vdr = the driver command voltage

The relationship between the pump/motor displacement and the LVDT voltage was curve fit from test data using the following equation. The relationship was ideally a linear one. The curve fit was done only to find the value of the constant K2 which best accounted for the small nonlinearities of the LVDT.

$$D = K2 * Vlvdt \qquad (4\text{-}5)$$

where Vlvdt = the LVDT voltage

K2 = a constant

D = the pump/motor displacement

Equation (4-6) was then derived directly from equation (4-5).

$$Dm = K2 * Vlvdtm \qquad (4\text{-}6)$$

where Dm = the maximum allowed displacement of the pump/motor

K2 = a constant

Vlvdtm = the LVDT voltage at maximum pump/motor displacement

Equation (4-7) is the ideal relationship between the pump/motor's torque, displacement, and pressure.

$$T = D * P \qquad (4\text{-}7)$$

where T = the pump/motor torque

D = the pump/motor displacement

P = the current accumulator pressure

The pump/motor's displacement was replaced in equation (4-7) by the relationships shown in equation (4-5) and the constant K2 was replaced using equation (4-6) to form equation (4-8).

$$T = (Dm / Vlvdtm) * Vlvdt * P \qquad (4\text{-}8)$$

where T = the pump/motor torque

Vlvdt = the LVDT voltage

K2 = a constant

P = the current accumulator pressure

Equation (4-8) was solved for Vlvdt and the pump/motor torque was replaced using the relationship shown in equation (4-4) to form equation (4-9).

$$Vlvdt = (Pi * Vlvdtm * Vdr) / (P * Vdrm) \quad (4-9)$$

where Vlvdt = the desired LVDT voltage

Pi = the precharge pressure

Vlvdtm = the LVDT voltage at maximum pump/motor displacement

Vdr = the driver input voltage

P = the current accumulator pressure

Vdrm = the driver command voltage at the maximum pump/motor displacement

Equation (4-9) is seen in line 3130 of control program in Appendix D.

4.2.2 Pump

The control equations for the pump depend on the pump, the included pump displacement control piston, and the servovalve system used to control the pump's displacement. The schematic of the pump controller is shown in Fig. 2.8. The following relationships show the basic theory used to run the pump and the engine coupled to the pump at a constant torque and speed.

The theoretical relationship between the pump's displacement and the angle of the pump's cylinder block is:

$$D = K3 * \sin A \tag{4-10}$$

where D = the pump displacement

K3 = a constant

A = the pump angle

The angle of the pump's cylinder block was found through calibration tests to be accurately represented as a linear function of the control pressure at the displacement control piston.

$$A = K4 * Pc + K5 \tag{4-11}$$

where A = the pump angle

K4 = a constant

Pc = the pump angle control pressure

K5 = a constant

The proportional servovalve controller gains were adjusted until the desired linear relationship between the steady state command input voltage to the servovalve controller and the steady state control pressure was obtained. This relation is represented as:

$$Pc = K6 * Vc + K7 \tag{4-12}$$

where Pc = the pump angle control pressure

K6 = a constant

Vc = the servovalve control voltage

K7 = a constant

The previous three equations can be combined to show how the pump's displacement is a function of the control voltage.

$$D = K3 * \sin(K4 * (K6 * Vc + K7) + K5) \tag{4-13}$$

where D = the pump displacement

K3 = a constant

K4 = a constant

K6 = a constant

Vc = the servovalve control voltage

K7 = a constant

K5 = a constant

The ideal relationship for the pump torque is:

$$T = P * D \tag{4-14}$$

where T = the pump torque

P = the accumulator pressure

D = the pump displacement

Once the desired operating pump torque is chosen, the command voltage required to maintain the desired torque is calculated.

$$Vc = (\sin^{-1}(Tdes/K3*P)-K4*K7-K5)/(K4*K6) \tag{4-15}$$

where Vc = the servovalve control voltage

Tdes = the desired pump torque

K3 = a constant

P = the accumulator pressure

K4 = a constant

K7 = a constant

K5 = a constant

K4 = a constant

K6 = a constant

Since the test runs were made by operating at an ideal point engine torque and speed, the desired pump torque pump torque and speed were also constant. The relationship between the engine and pump speed and torque was dependent on the chain box gearing. Modifying the engine and pump to run at other points requires that the desired torque in equation (4-11) be changed from a constant to a variable.

Equation (4-11) was calculated assuming ideal pump efficiencies, linear relationships, and steady state operation. The accuracy of this equation was checked by varying the accumulator pressure and recording the change in the speed of the simulated engine. Since the simulated engine was operated at a single simulated throttle angle, a constant engine speed represented a constant torque.

The entire system was driven using equation (4-11) in the control program. The engine speeds were recorded in order to check for speed variation with accumulator pressure. The engine speed was found to vary +/- 4 percent from the desired operating speed. This small error was due to the idealized assumptions and the servovalve controller performance. Equation (4-11) was approximated using a third order equation. The third order approximation of the complex equation (4-11) increased the computer control rate. A trial and error curve fit adjustment process was used to decrease the engine speed variations. The final third order equation is contained in lines 3270 and 3295 in the control program listing in Appendix D.

4.2.3 Engine

The simulated engine was controlled to operate near a constant torque and speed. The value of the chosen torque and speed at which the engine operated is not as important as the ability of the controller to provide a stable and accurate control the engine's operation. The values of torque and speed at which the system is most efficient, the ideal point, is a function of the engine and the other hydraulic and mechanical components. In most systems the rate of change of efficiencies is very gradual near the ideal point. Operating in a ideal region about the ideal point will not cause a noticeable effect on the system efficiencies for systems using most of the currently available components. In our system the ideal engine torque was chosen as 725 in-lb (81.88 N-m). The ideal engine speed was chosen as 1100 rpm. This value was chosen on the basis of power supply limitations, engine cycling time, and similarity to actual internal combustion engine data. The entire torque versus speed curve for the simulated engine can be seen in Fig. 2.7. This figure also shows the power supply pressure Ps, represented by the higher line on the graph. It should be noted that our power supply did not have sufficient flow to maintain adequate pressure for faster engine speeds. This limitation caused the engine torque to drop off abruptly above 1,800 rpm. It did not affect the simulation since the engine was never operating in this range during the testing.

The engine was cycled on and off using a servovalve. The engine was turned on when the accumulator was almost fully discharged. The accumulator pressure at which the engine was turned on was set at a level such that cavitation did not occur in the hydraulic pump/motor even when the flywheel was accelerated at its maximum rate. If a real engine had been used, the accumulator pressure at which the engine was started would have been slightly higher than for the simulated engine since reserve energy from the accumulator would be used to start the engine.

In the control program the engine was turned off when the accumulator pressure reached a level which is a function of the vehicle's mass and speed. In the control program the engine was turned off when the sum of the current kinetic energy of the flywheel and the potential energy in the accumulator equaled the maximum allowable potential energy in the accumulator. This control policy allowed all the braking energy to be recovered during all possible driving conditions. As explained in section 4.1.2, at higher vehicle speeds it is not advantageous to allow all of the braking energy to be recovered since the engine cycling rate would be increased. In the actual automobile the friction brakes of the car would handle any additional braking requirements during emergency braking from high speeds.

The engine pump subsystem should naturally have stable operation when the slope of the engine fixed throttle line on the engine's torque speed map is negative at the intersection of the pump's torque speed map at the present pump displacement. If the engine torque temporarily rises above the level required to drive the pump at the ideal point, the excess engine torque will accelerate the engine pump subsystem to a slightly higher equilibrium speed. If the engine torque temporarily drops below the level required to drive the pump at the ideal point, the engine pump subsystem will decelerate to a slightly lower equilibrium speed. A more complete explanations of the stability of the subsystem in contained in Ref [13]. Since the slope of the engine's torque as a function of speed is naturally negative in a region around the ideal point for most engines, stable response should naturally result. For engine pump subsystems which do not have satisfactorily stable operation characteristics when controlled in the manner described above, other possible control policies are discussed in Ref [13].

Although a simulated engine can adequately model the characteristics of an actual internal combustion engine, a study with an internal combustion engine is necessary to answer some questions. There may be some problems associated with cycling the engine. Some of the topics which need to be examined are the effects on engine wear, the effect on emissions, and possible changes in the lubrication system, the cooling system, and the starting system.

4.3.0 Control for Safe Operation

Several safety measures were used during the system's operation. The keyboard of the computer was set to be a system shut down button once the control program was started. If any key was touched while the control program was running, the engine servovalve was closed which turned the engine off and the displacements of the hydraulic pump and pump/motor were set to zero. The flywheel could be stopped quickly by an emergency braking system. Emergency disk brakes were engaged by opening a valve which was connected to a small emergency braking accumulator.

4.4.0 Future Controls

There are several control modifications which can be implemented and evaluated with the test stand. A practical hydrostatic vehicle might require the provision of shutting off the accumulator and having direct hydrostatic drive. The direct hydrostatic drive would be accomplished by setting the throttle angle and pump displacement in an open loop manner. The system pressure would be controlled by varying the pump/motor displacement. In such a system the two subsystems are no longer uncoupled. A more complex and accurate control of the components would be required since the accumulator would no longer act as a buffer between the two subsystems.

The engine should be operated along an ideal line instead of at an ideal point. The ideal line is a range that minimizes brake specific fuel consumption while still allowing a wider range of engine power output. The additional control strategy is not expected to have any adverse effect on controllability since the engine and the pump/motor can be treated as two separate systems due to the buffering action of the accumulators. The operation along the ideal line is controlled just like the operation at the ideal point except that the engine operating point is continuously changing. Operating at lower engine speeds and torques causes the cycling rate to be decreased without much loss in efficiency. Operating at higher engine speeds and torques in ranges where the system efficiencies are slightly less than the peak system efficiency would allow the engine output power to be greater than at the ideal point.

Compatibility problems between the servovalve size and the servovalve controller caused stability problems in the simulated engine when we tried to operate the engine over an ideal line. This was not a system instability, but merely an instability of the simulated engine. A real engine is obviously stable in this respect. This problem, therefore, had no significance in terms of our system studies except for limiting what we could do.

A single-sided hydraulic pump/motor unit can replace the current pump/motor which has the capability of going overcenter. A single-sided pump/motor has an inherent efficiency advantage over the two-sided pump/motor since the compressibility losses are reduced. The single-sided pump/motor design would require additional valving which could produce a possible hydraulic shock caused by high velocity fluid being forced to reverse direction rapidly.

The control of the system could be modified to eliminate all but the accumulator pressure transducer and a vehicle speed sensor. The control of the hydraulic pump/motors may be accomplished though the used of simple limit switches and an adaptive control computer controller. Computing variables from other known variables would reduce the cost of the control system and could improve the system's reliability. The cost of computing is low and it should continue to go down while reliability increases.

5.0 TESTING AN EXPERIMENTAL HYBRID ENERGY STORAGE VEHICLE

Tests were performed to evaluate the controllability of the system, and to record significant system parameters. Tests were run using several different initial conditions and driving cycles. The initial condition variations were the starting accumulator pressure and engine operating state. A typical driving cycle consisted of an acceleration of the flywheel, a period of constant speed, and a deceleration. Several rates of accelerations and decelerations and several constant speed durations were tested. Data were also collected on random driving cycles. During all of the driving cycles, a data acquisition system recorded nine variables: driver's input command voltage, flywheel speed, flywheel torque, pump/motor stroke-control piston position, accumulator pressure, accumulator piston position, the simulated engine speed, the simulated engine torque, and the pump stroke control pressure. Fig. 5.1 through 5.36 show the results of some of the test runs.

5.1.0 Driving Cycles

The system was run with periods of constant acceleration, constant speed, and constant deceleration.

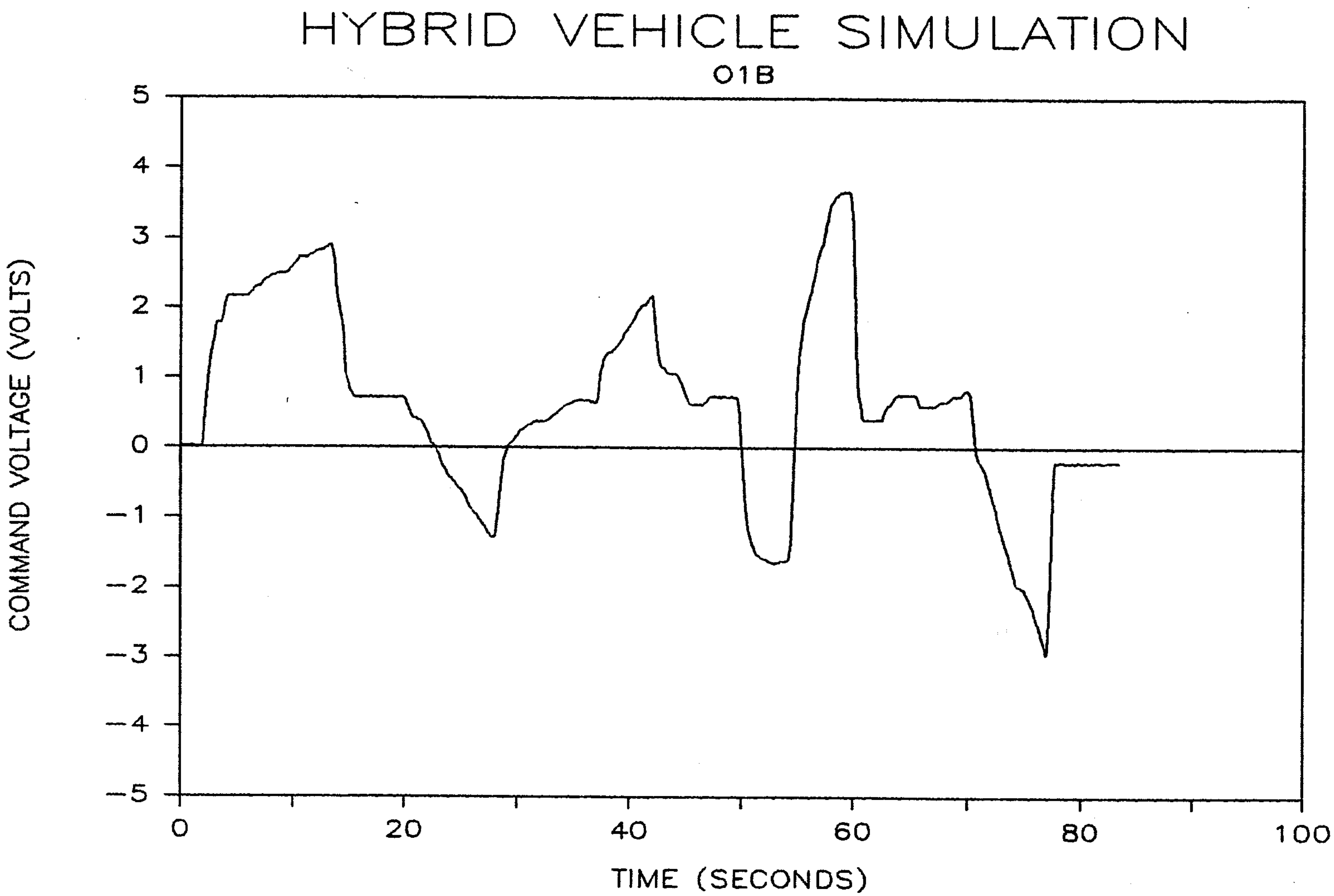

Fig. 5.1: Command Voltage (Test 1)

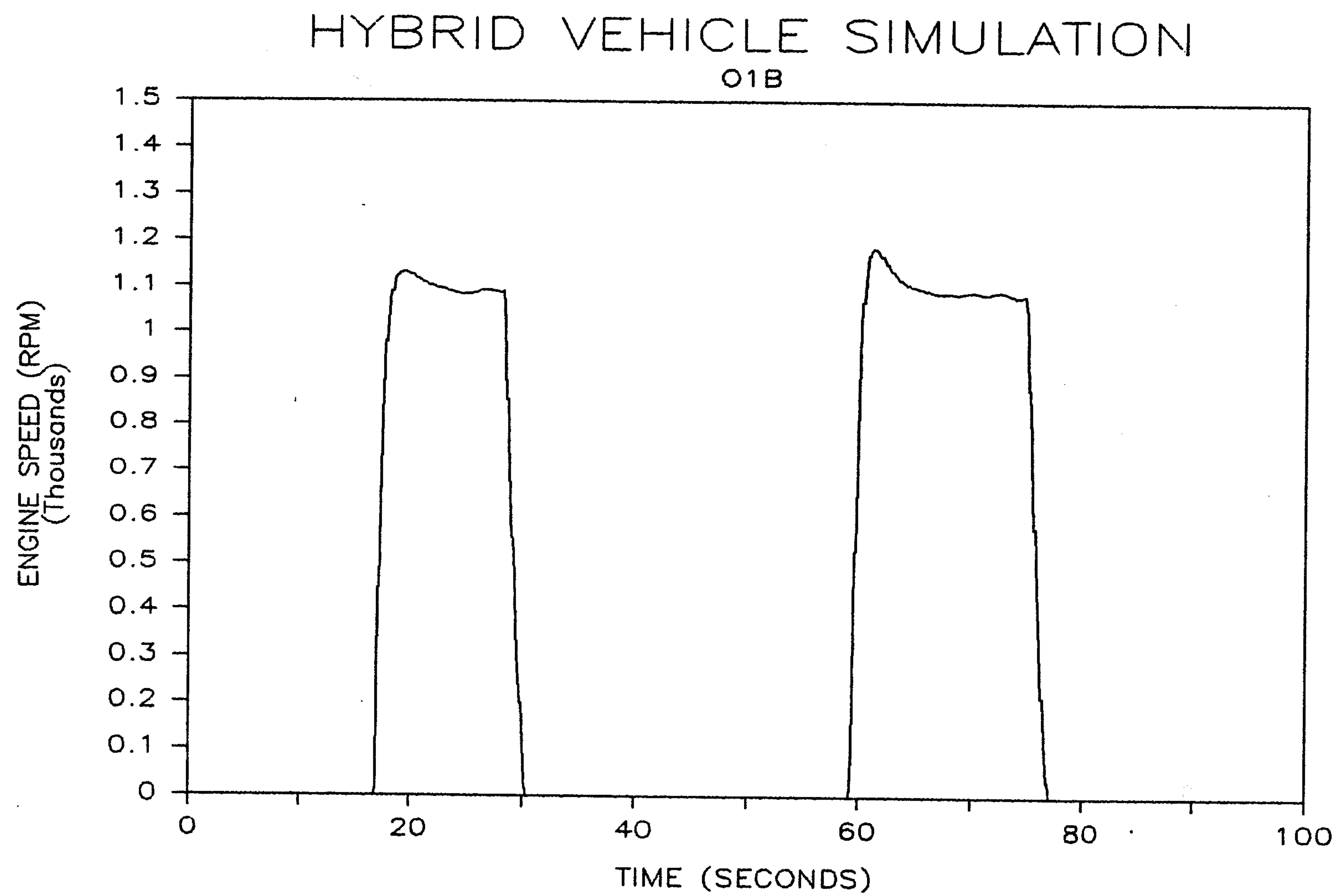

Fig. 5.2: Engine Speed (Test 1)

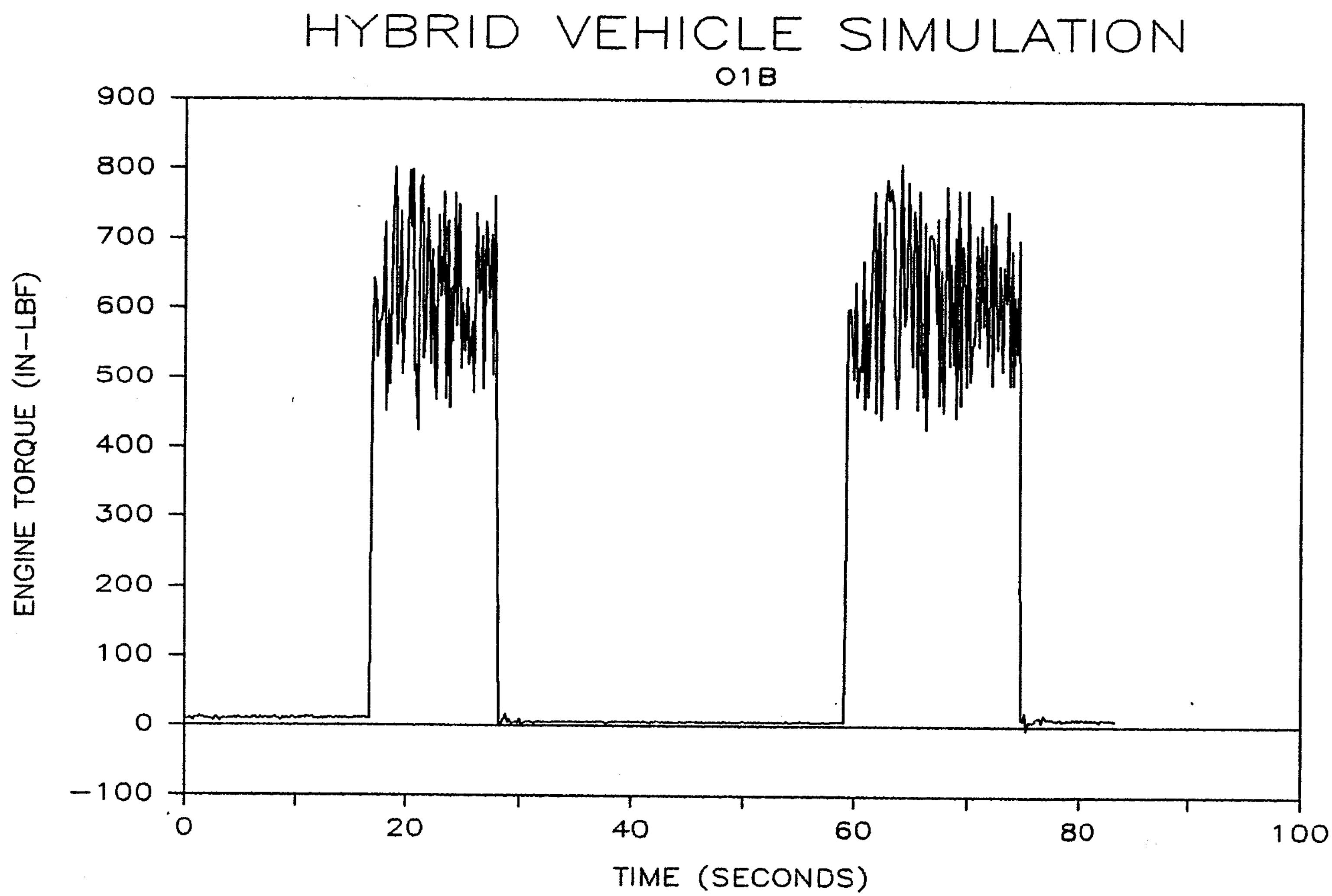

Fig. 5.3: Engine Torque (Test 1)

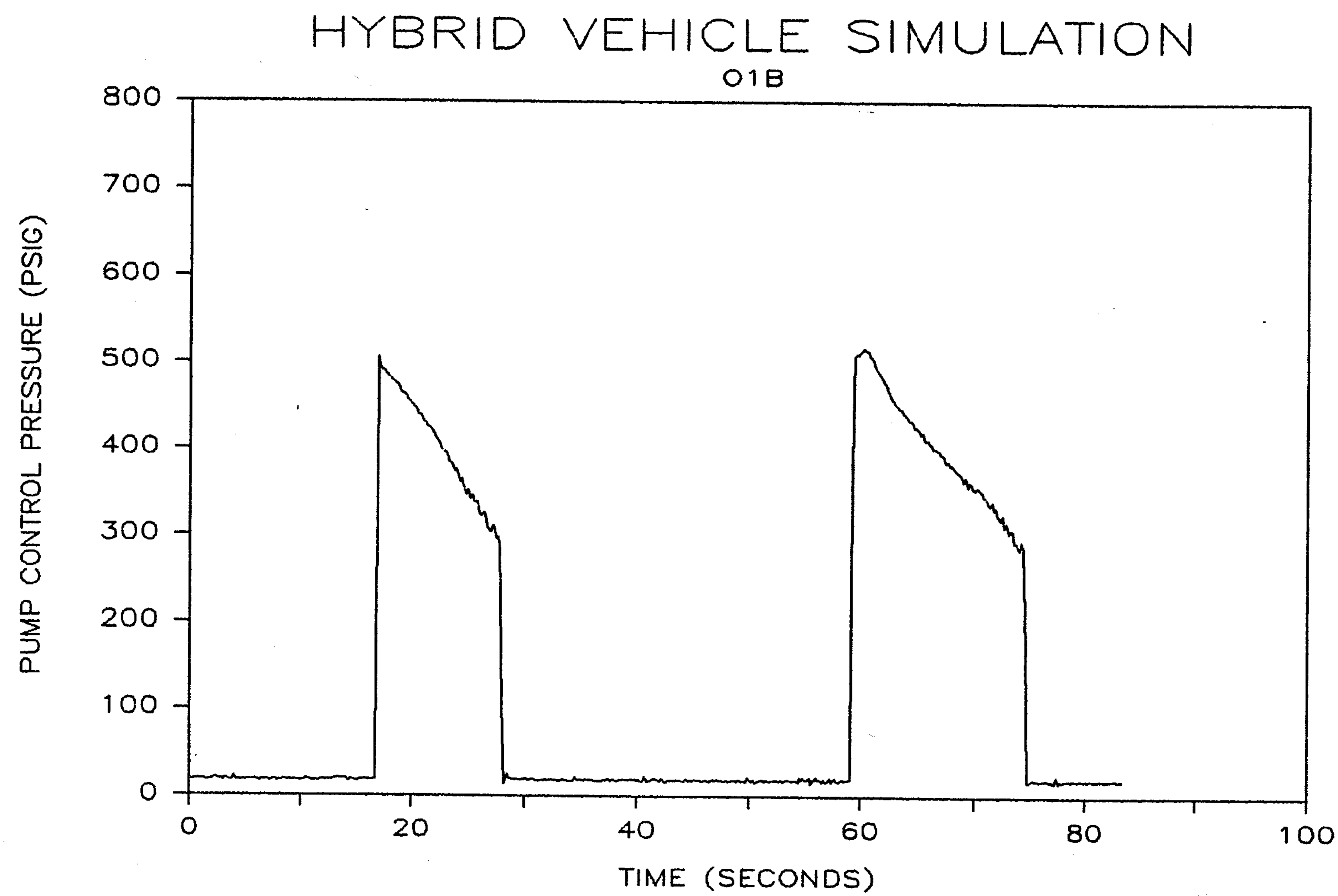

Fig. 5.4: Pump Control Pressure (Test 1)

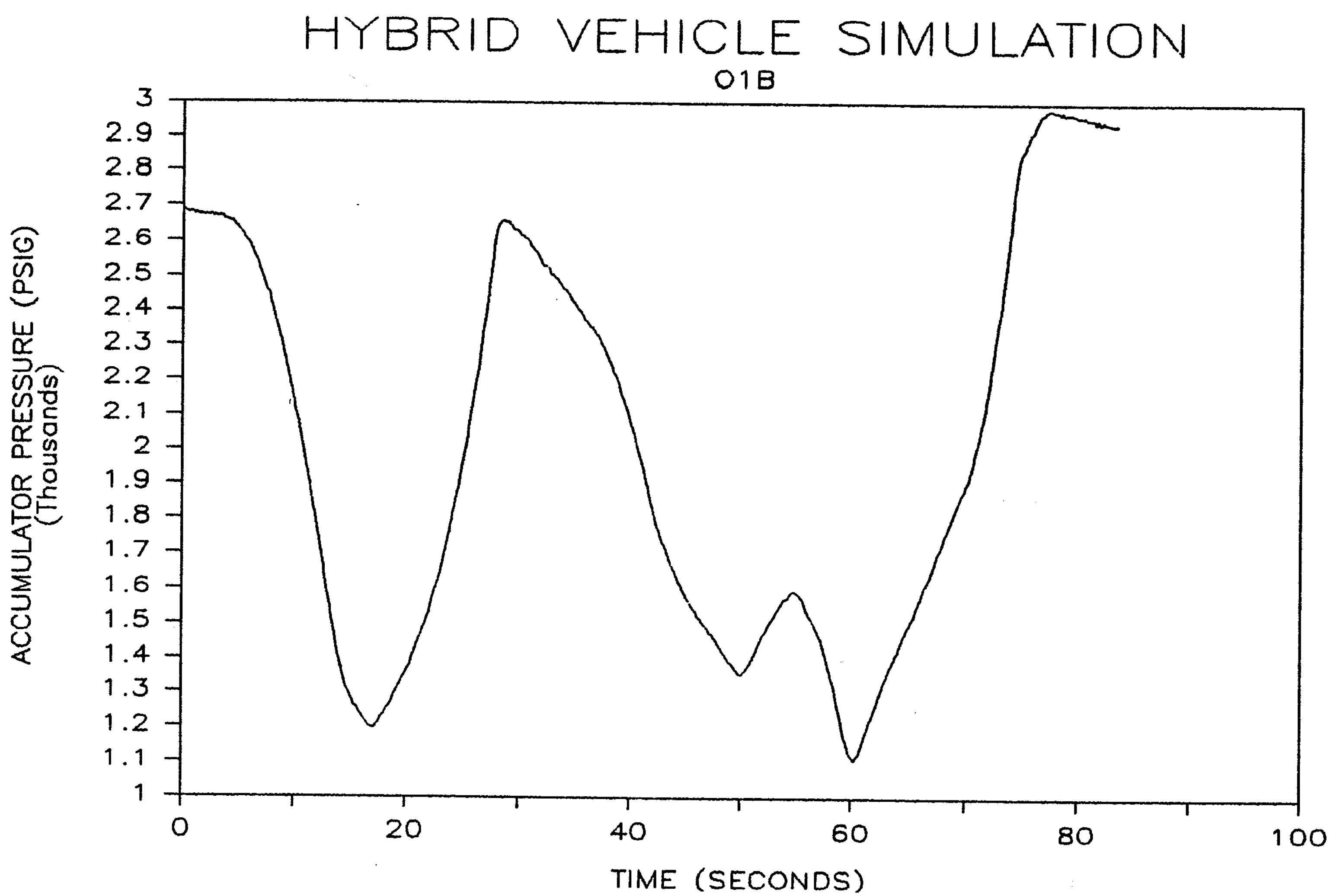

Fig. 5.5: Accumulator Pressure (Test 1)

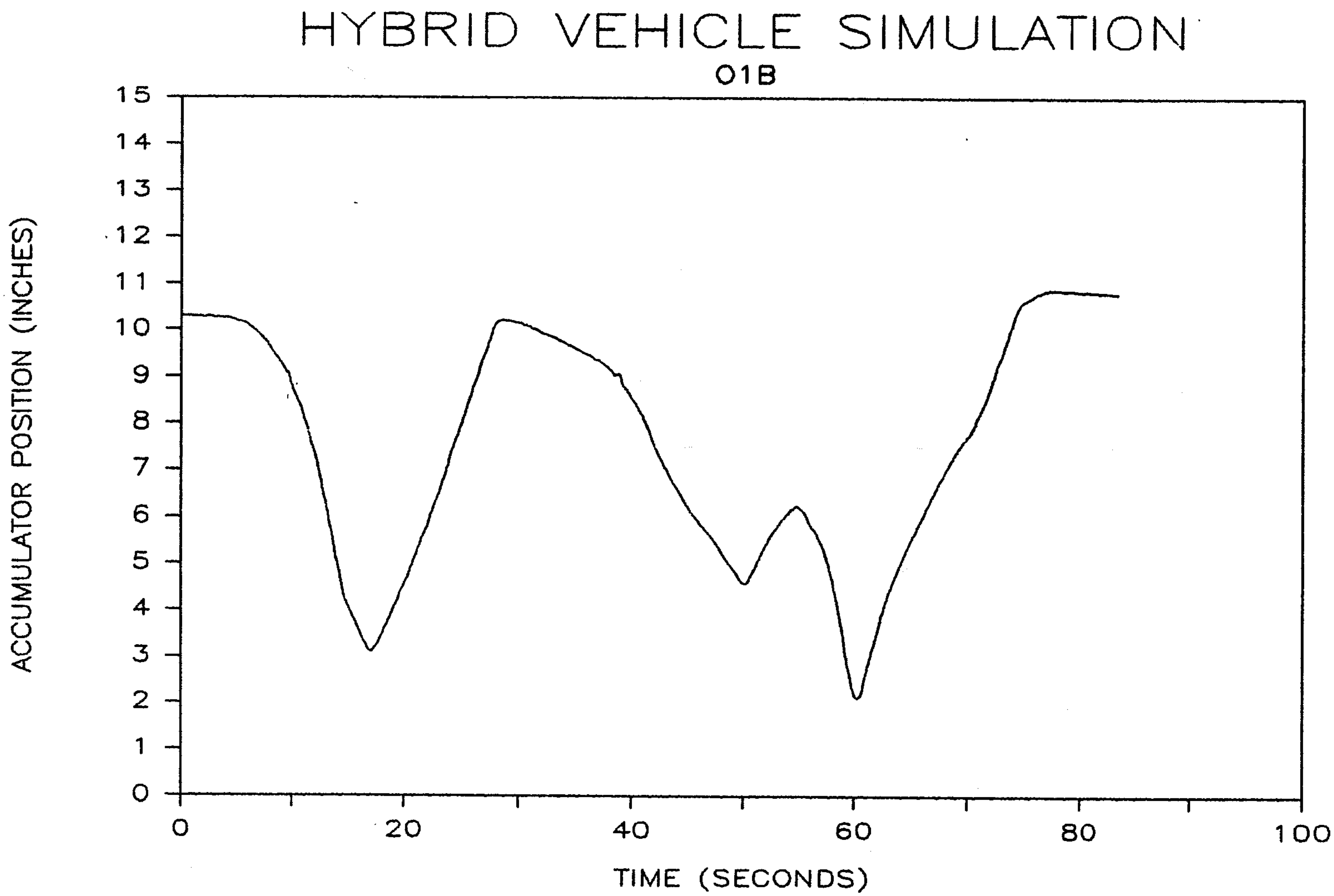

Fig. 5.6: Accumulator Position (Test 1)

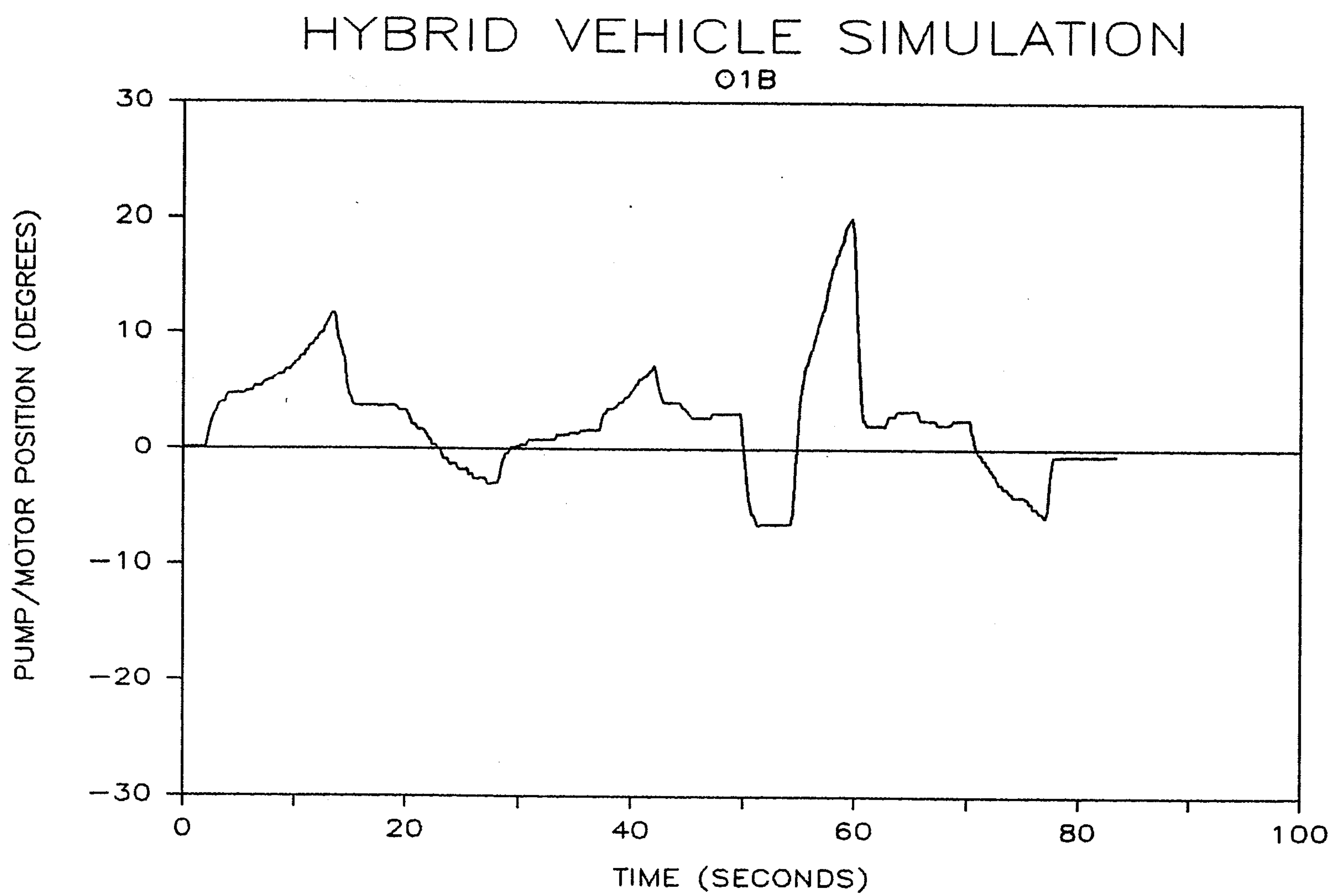

Fig. 5.7: Pump/Motor Position (Test 1)

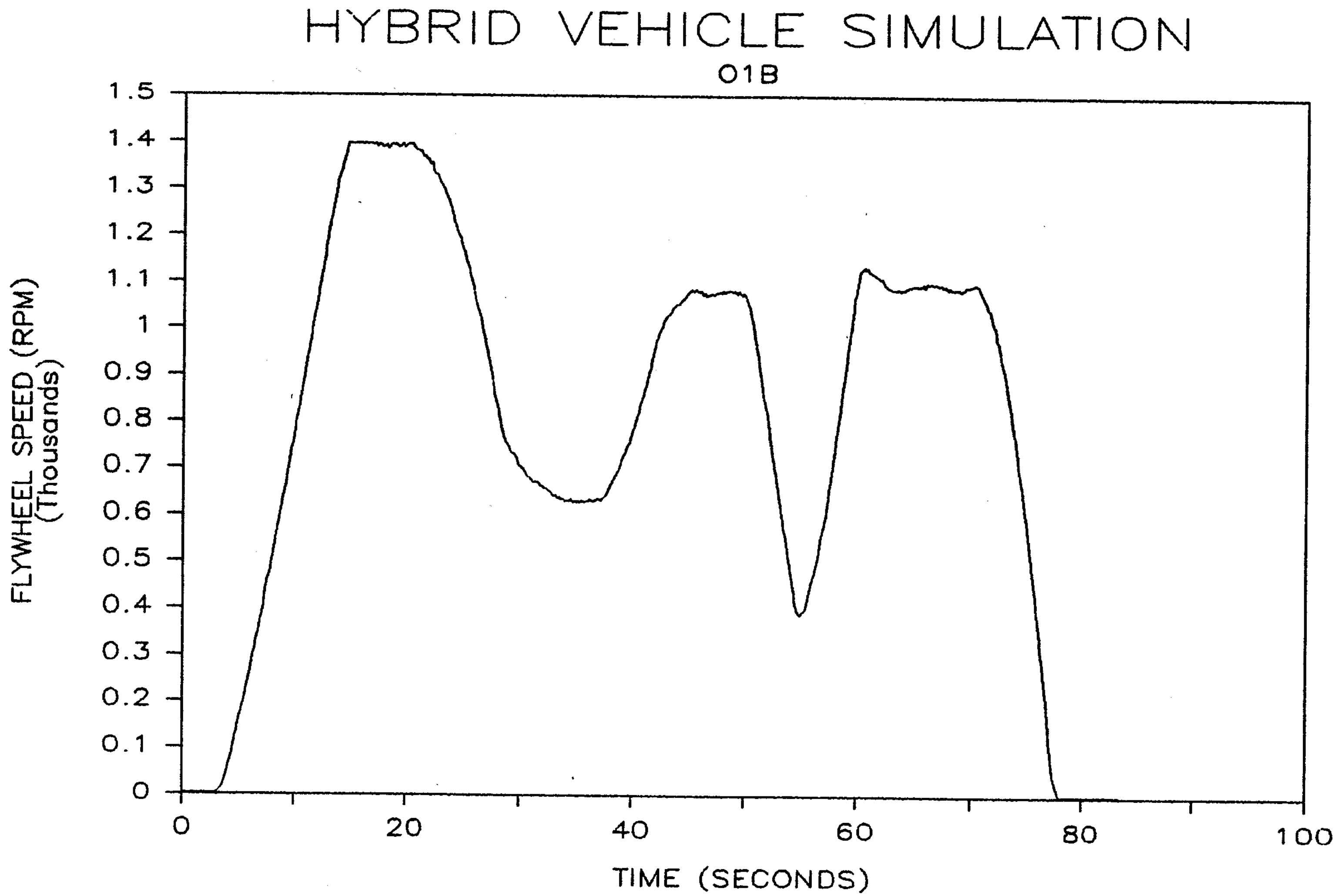

Fig. 5.8: Flywheel Speed (Test 1)

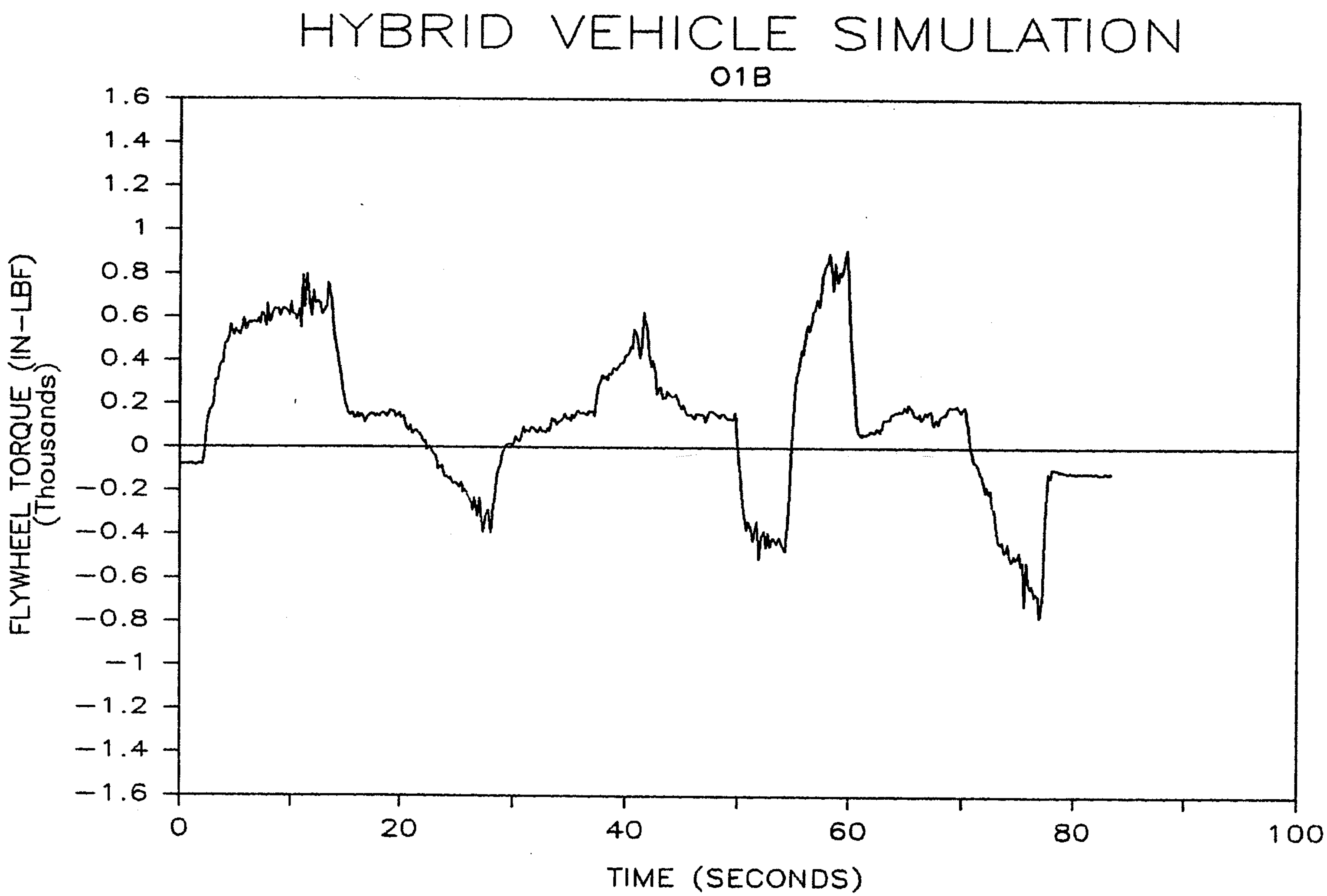

Fig. 5.9: Flywheel Torque (Test 1)

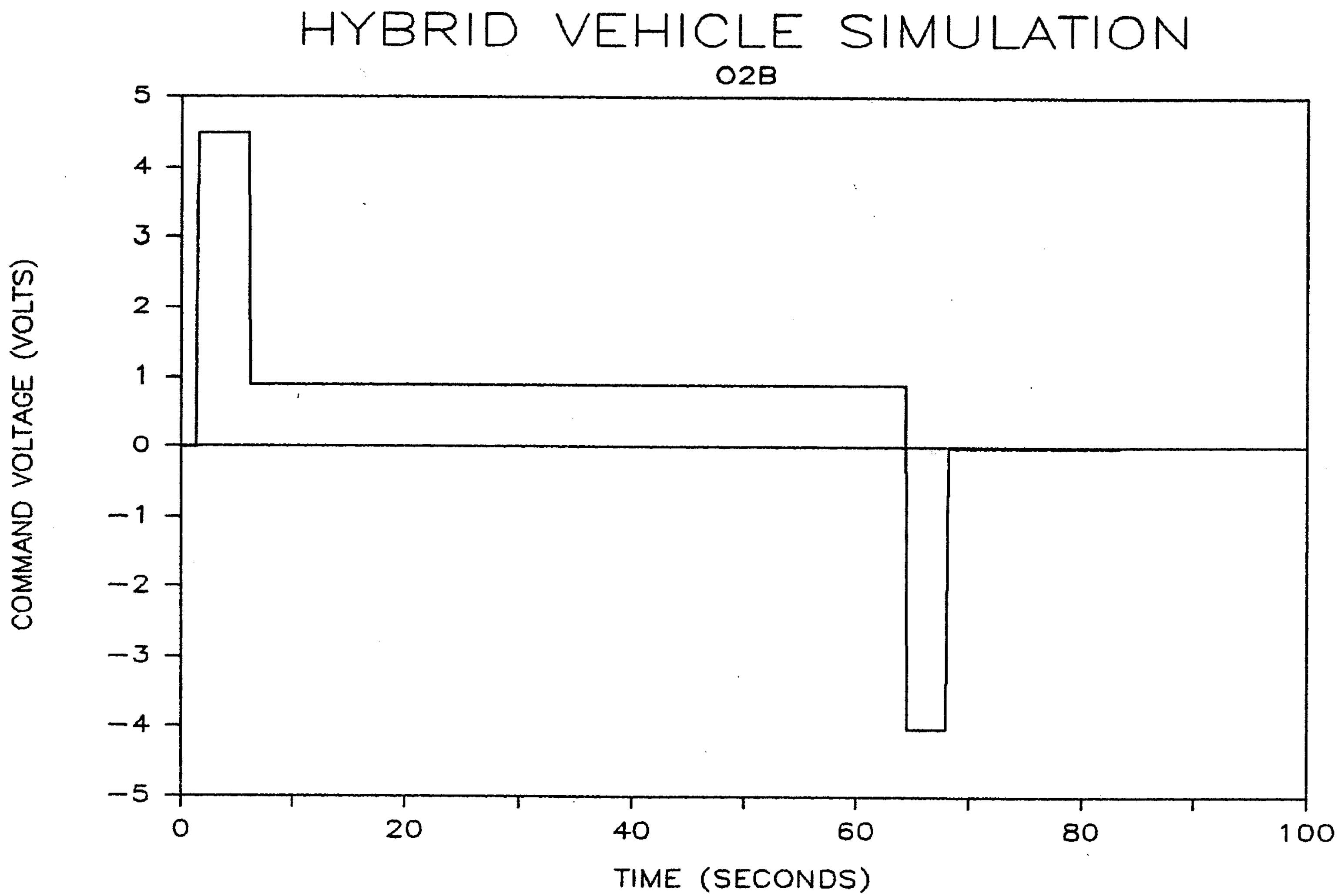

Fig. 5.10: Command Voltage (Test 2)

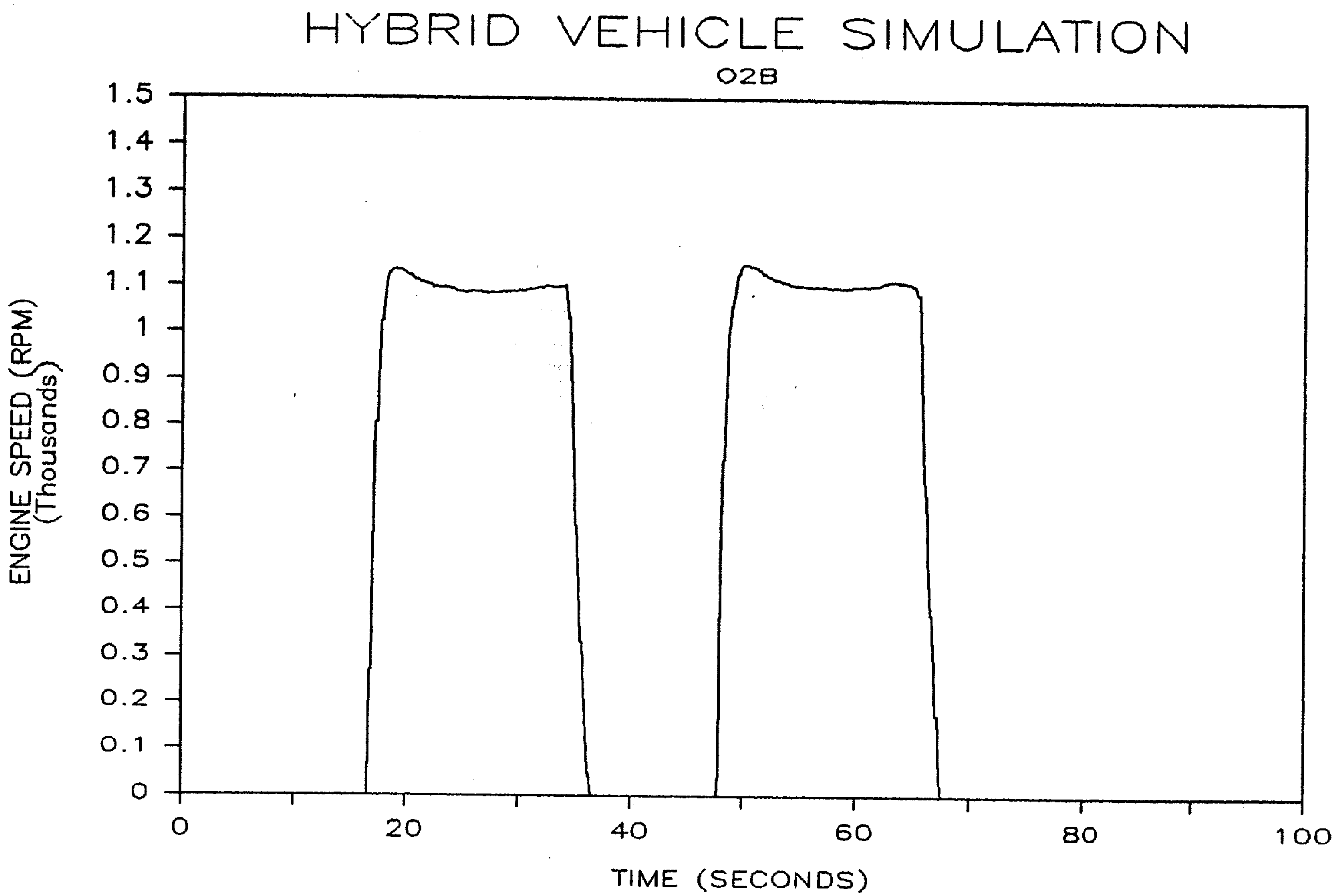

Fig. 5.11: Engine Speed (Test 2)

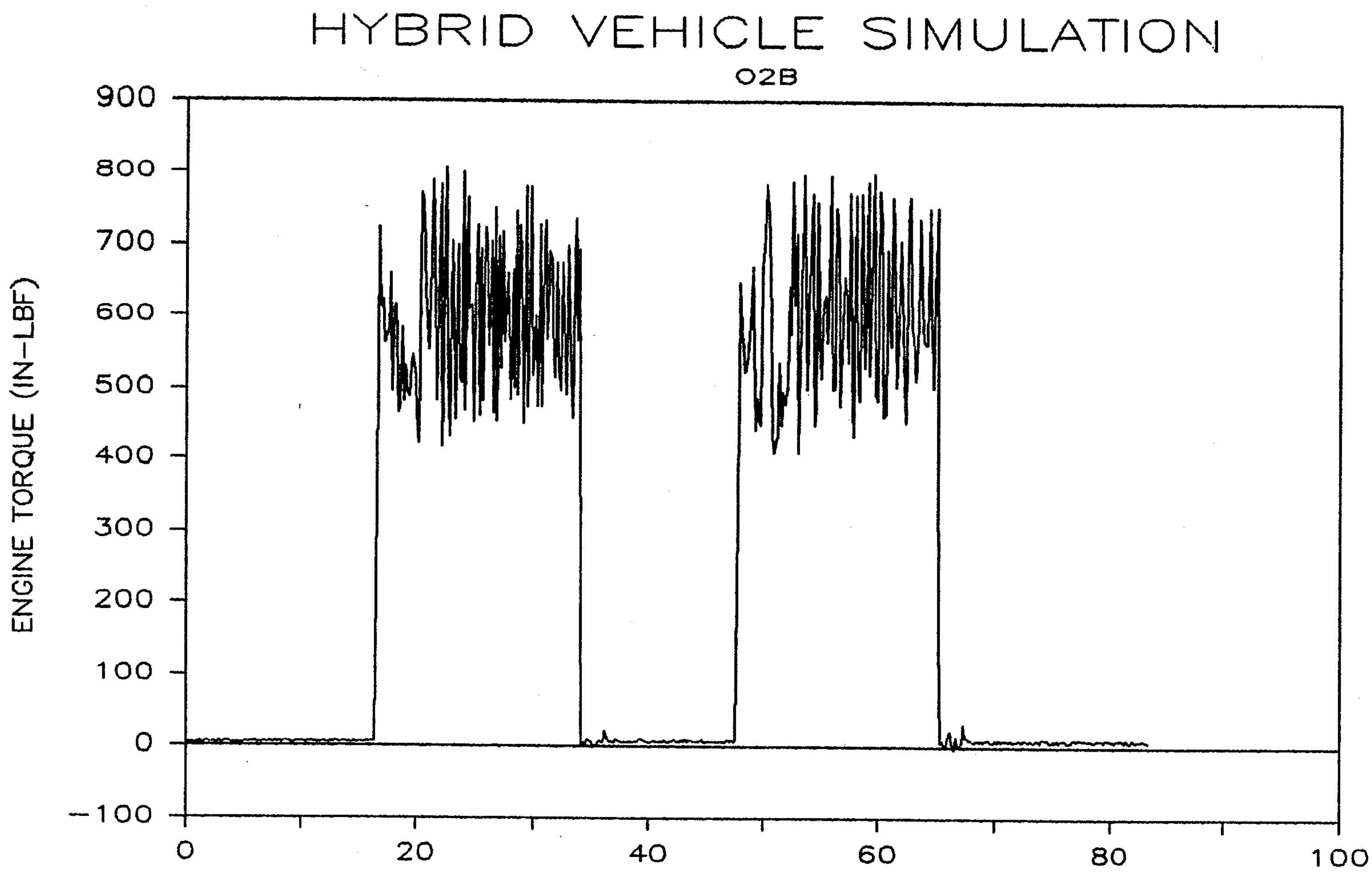

Fig. 5.12: Engine Torque (Test 2)

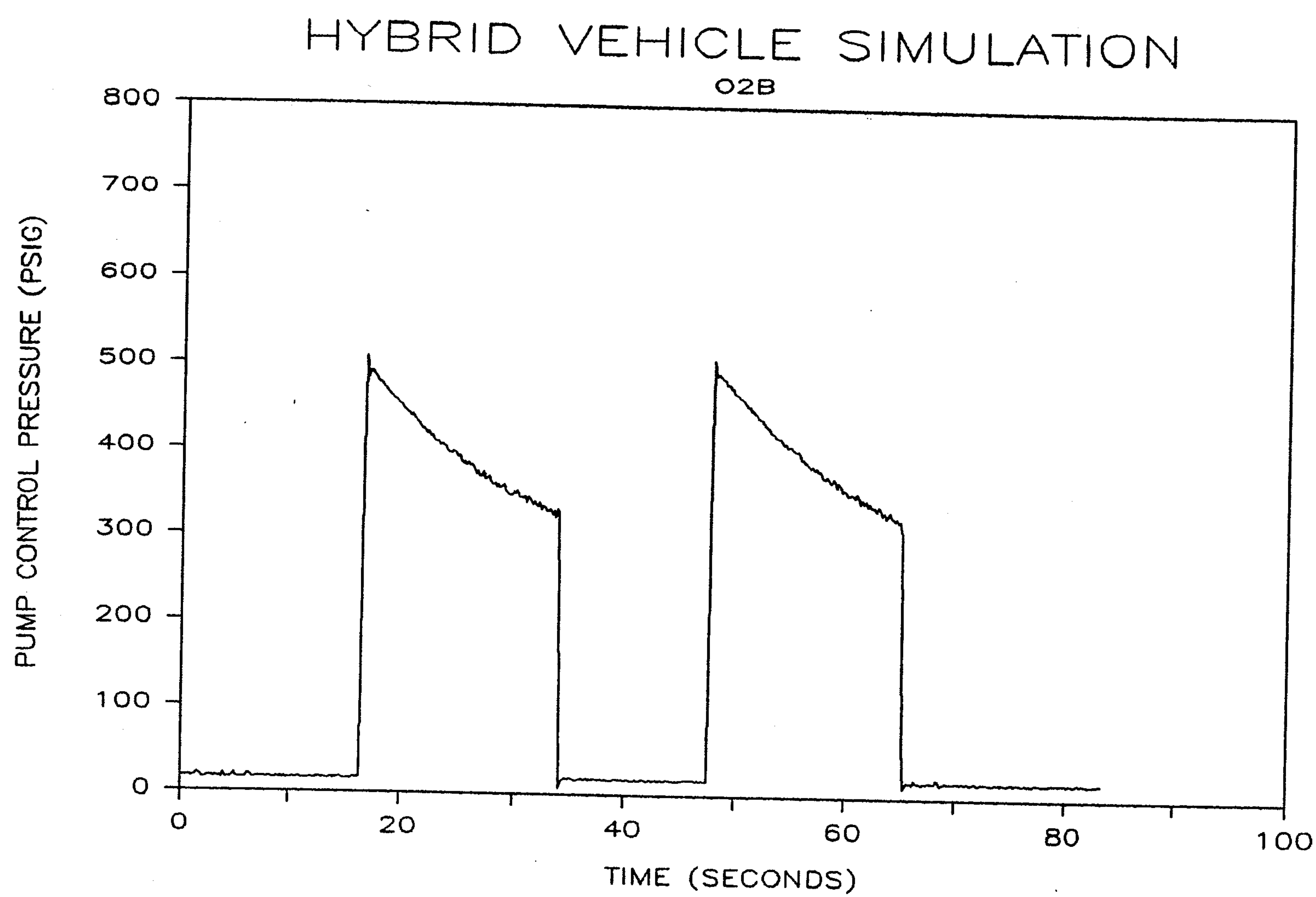

Fig. 5.13: Pump Control Pressure (Test 2)

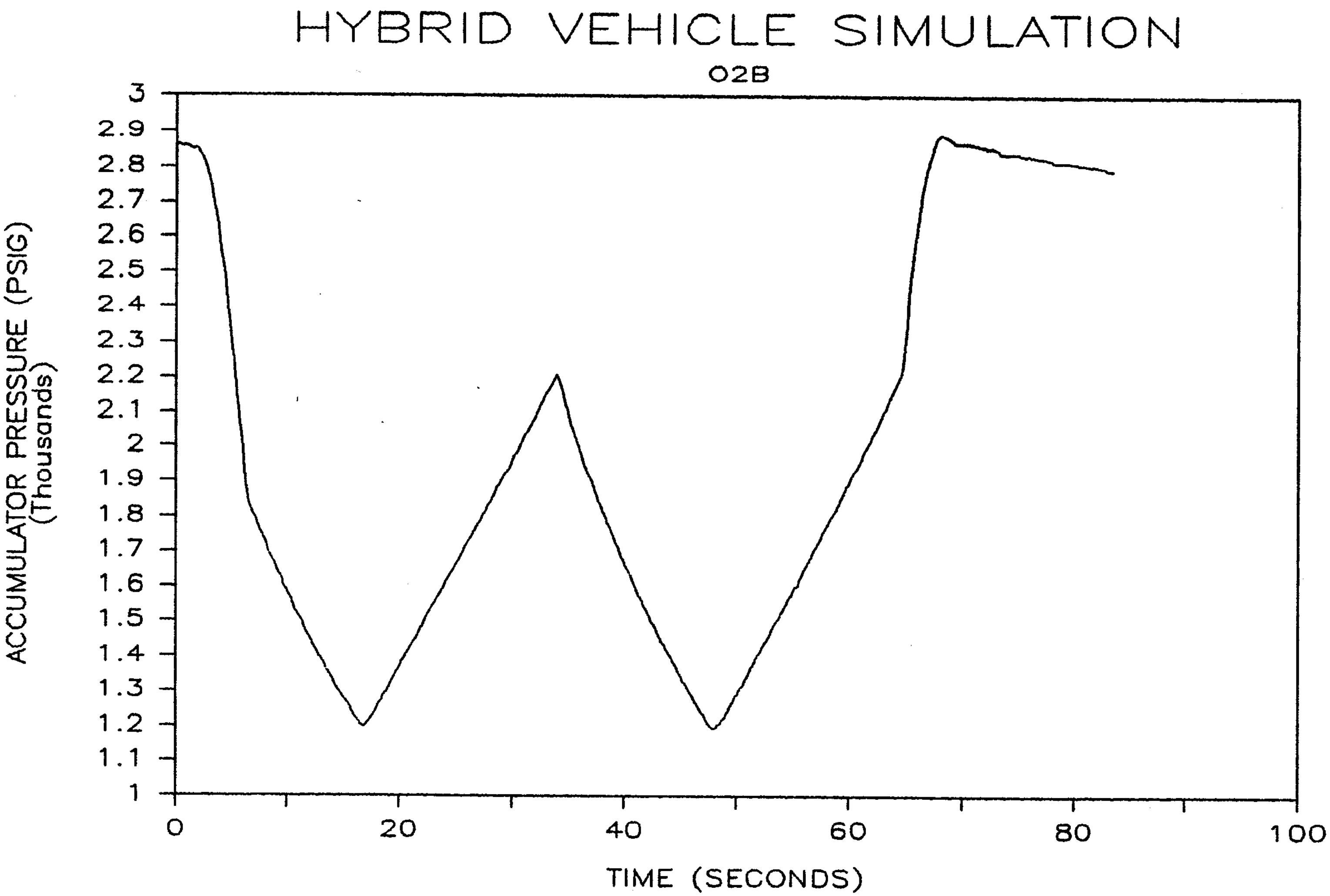

Fig. 5.14: Accumulator Pressure (Test 2)

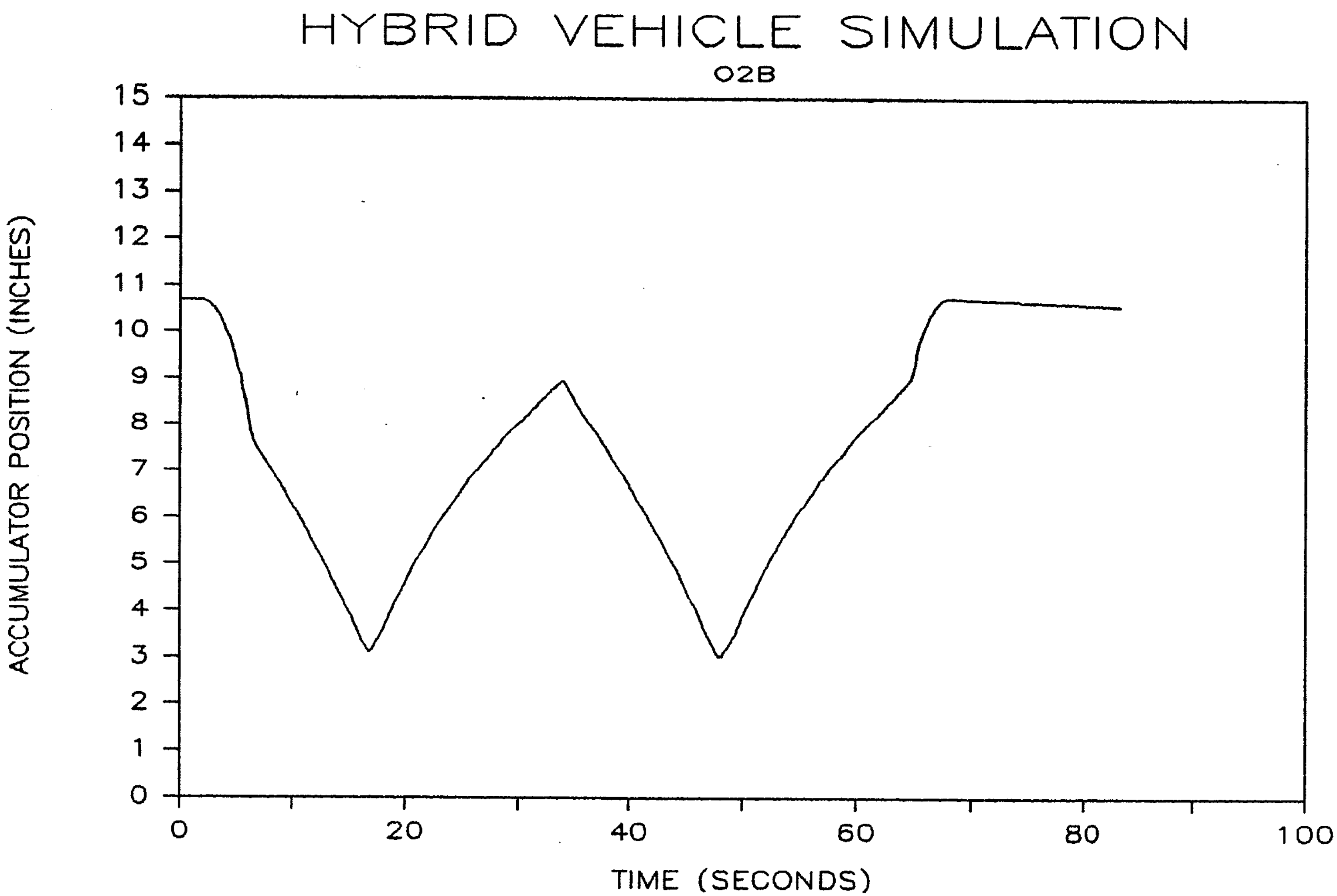

Fig. 5.15: Accumulator Position (Test 2)

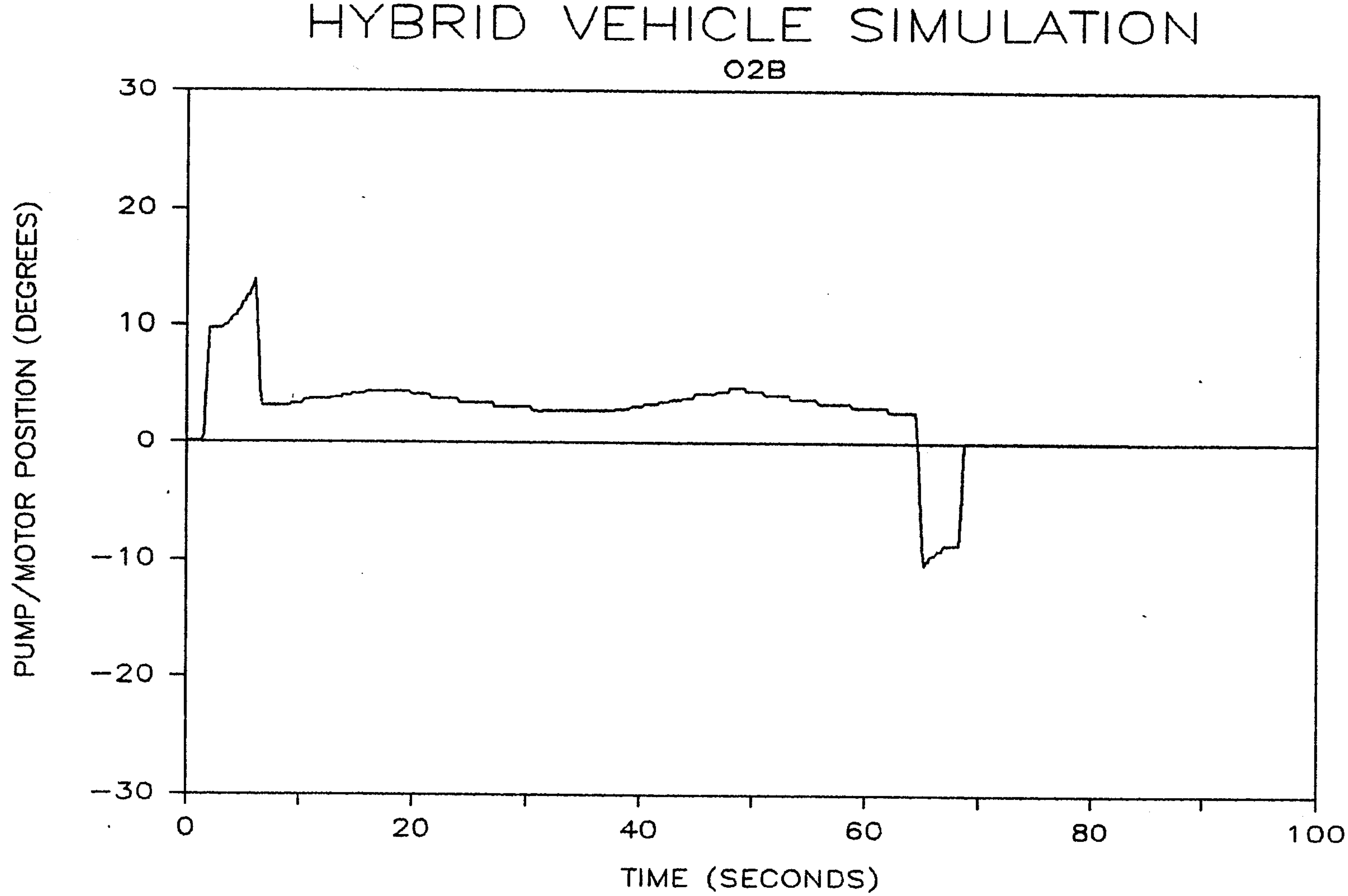

Fig. 5.16: Pump/Motor Position (Test 2)

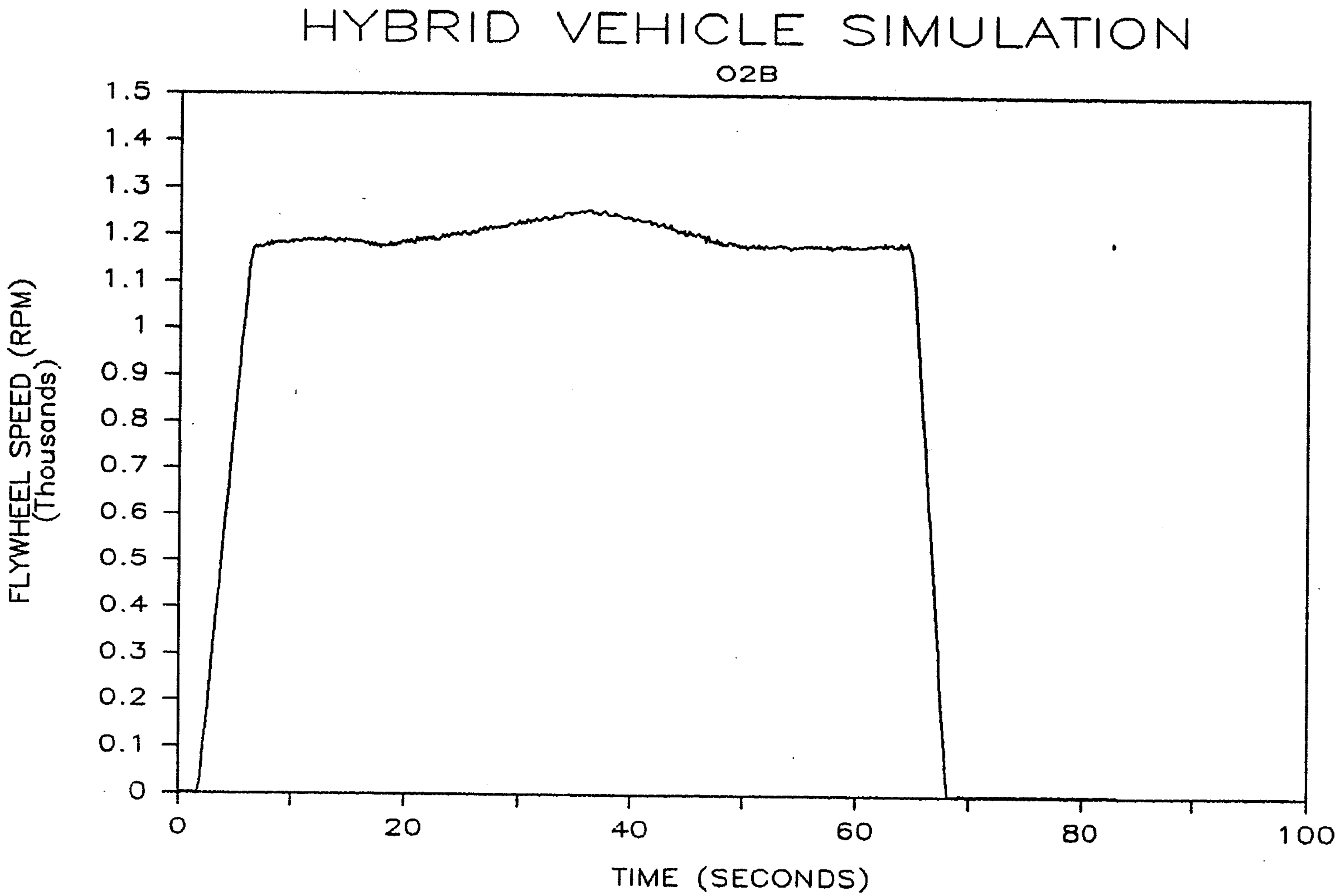

Fig. 5.17: Flywheel Speed (Test 2)

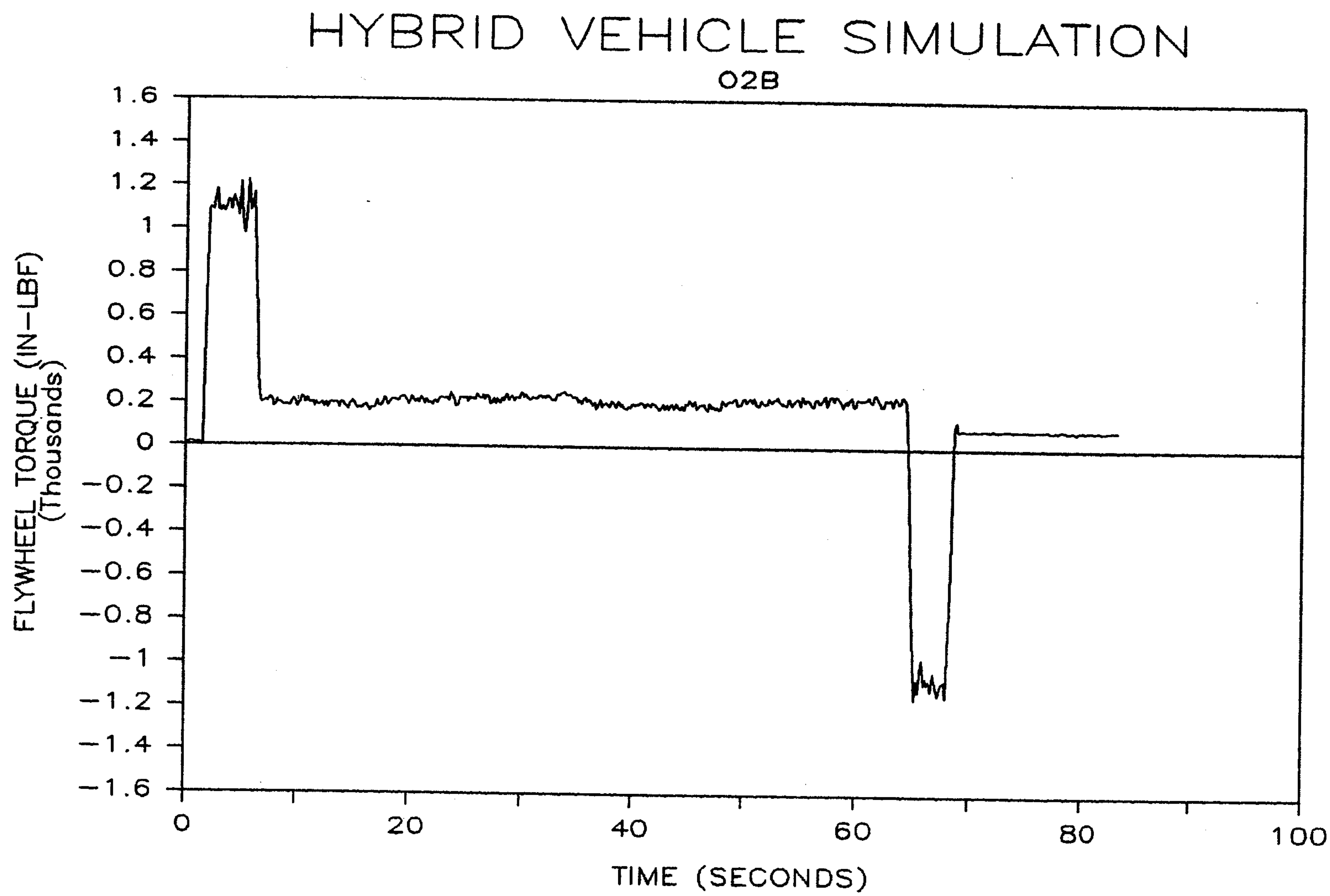

Fig. 5.18: Flywheel Torque (Test 2)

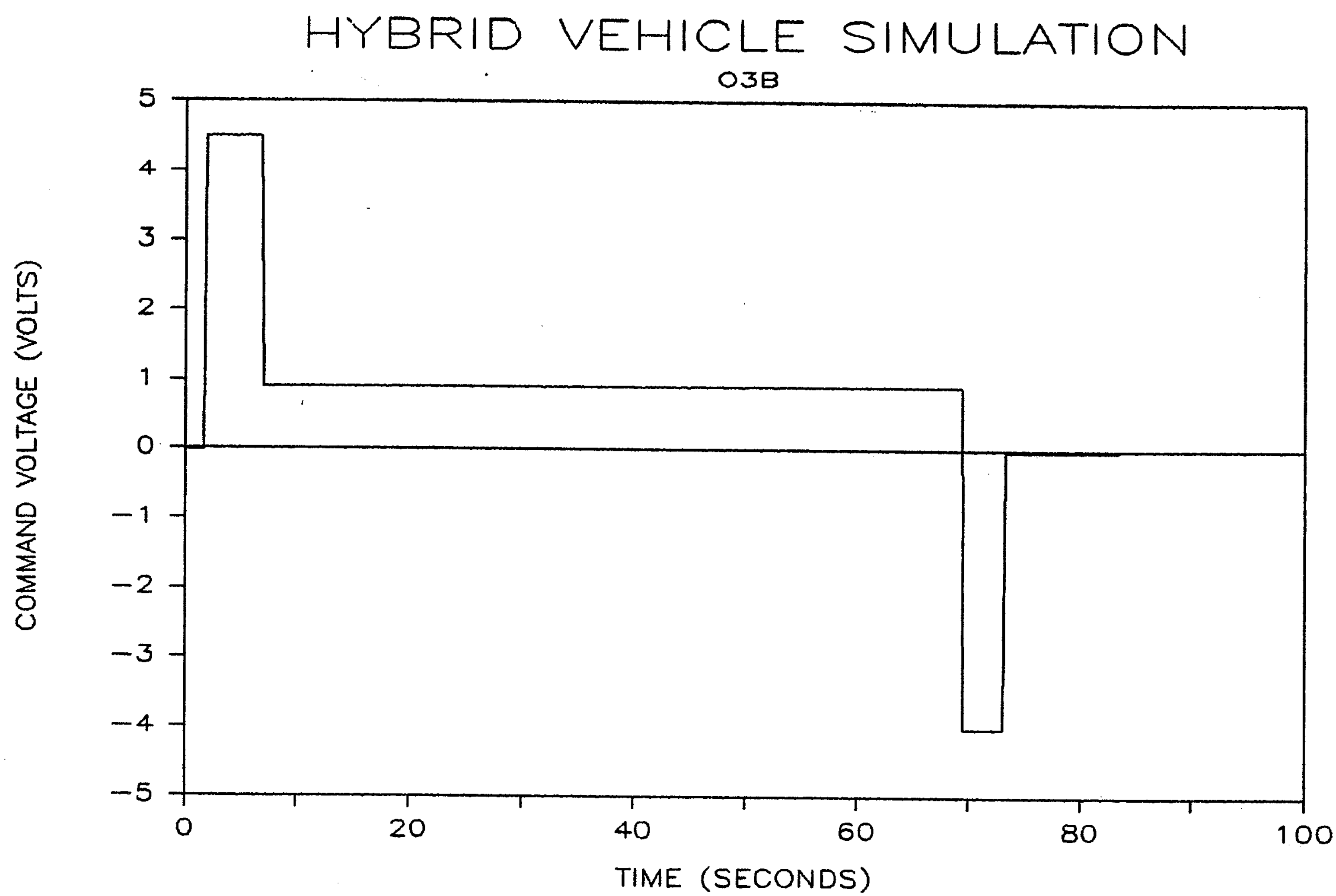

Fig. 5.19: Command Voltage (Test 3)

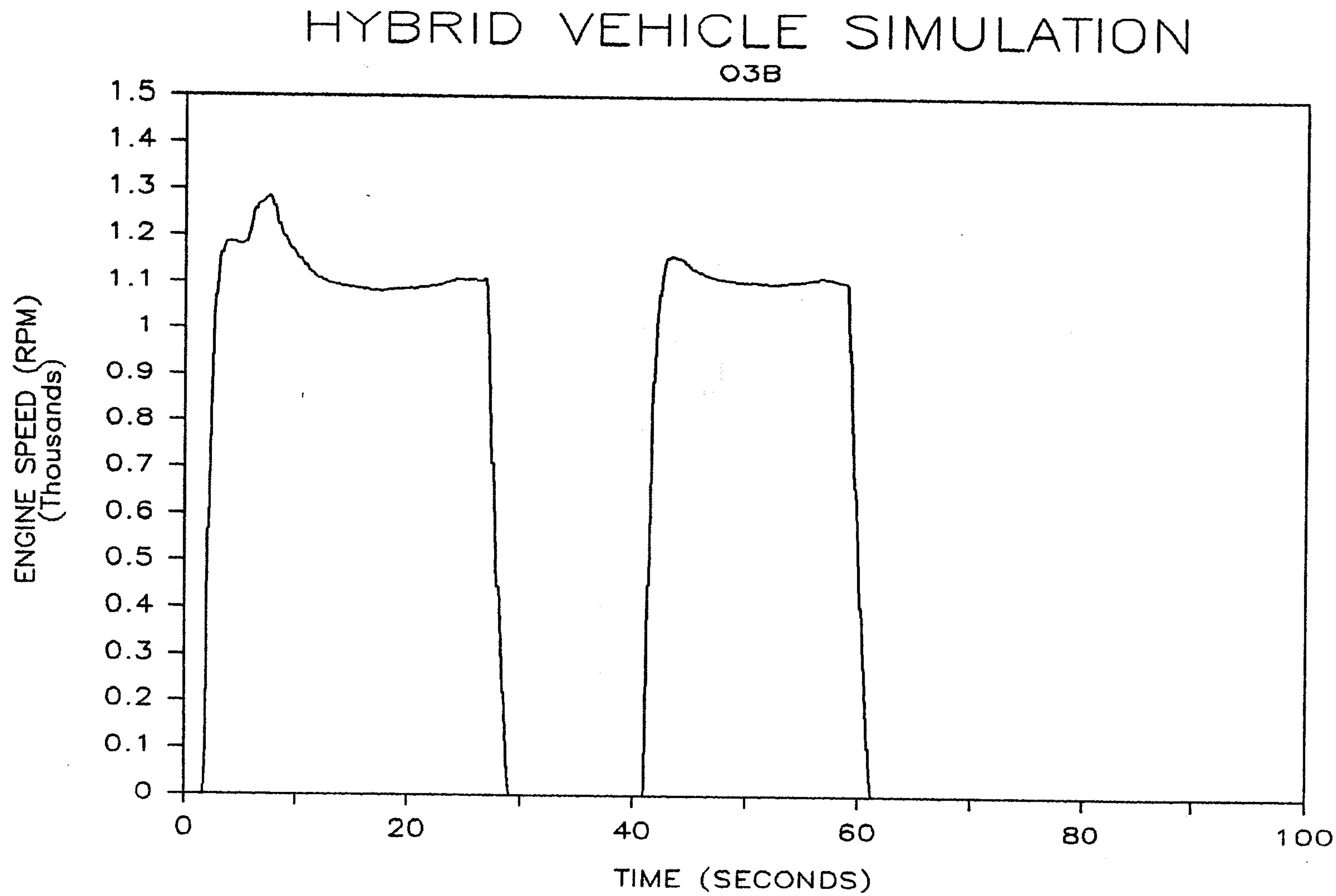

Fig. 5.20: Engine Speed (Test 3)

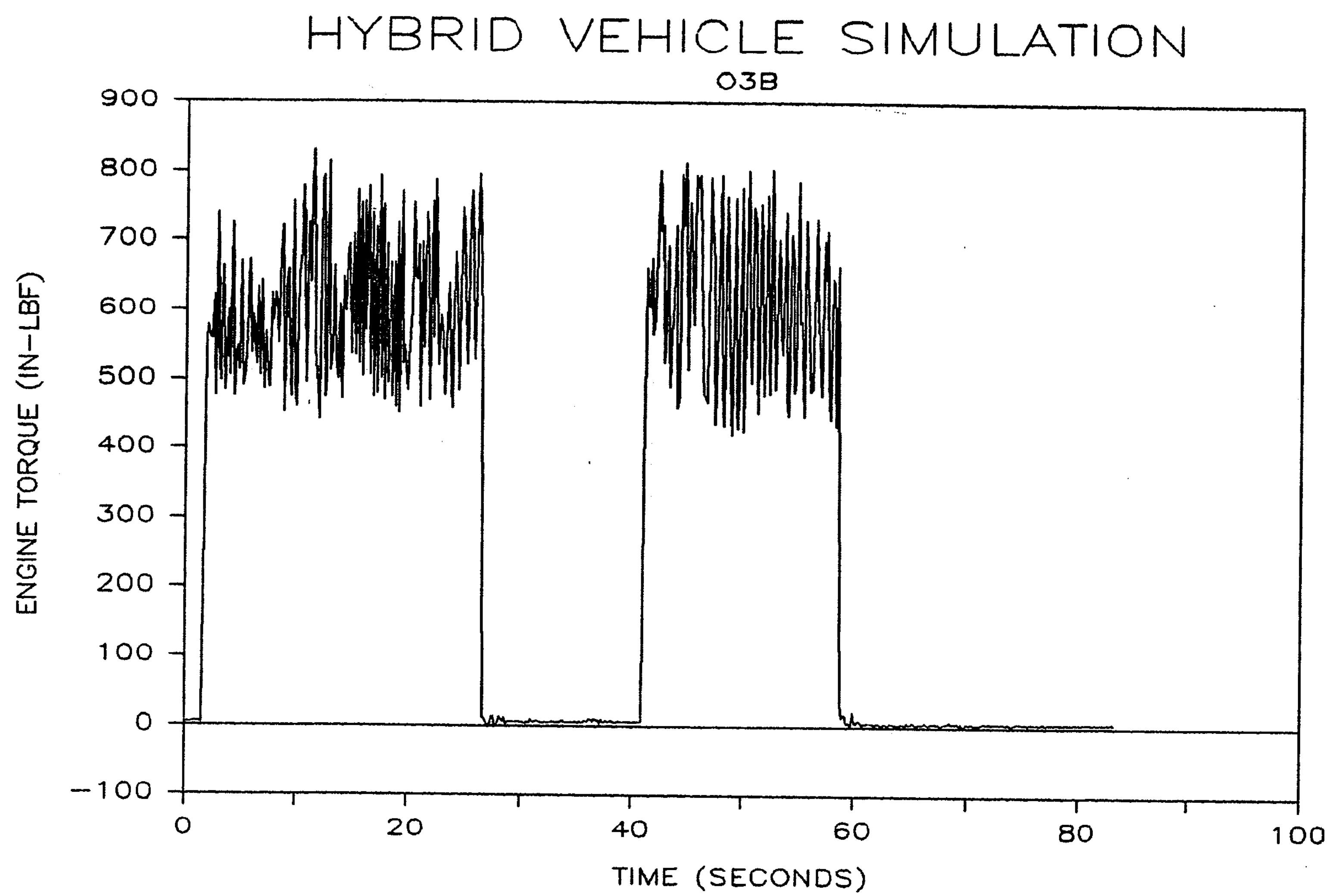

Fig. 5.21: Engine Torque (Test 3)

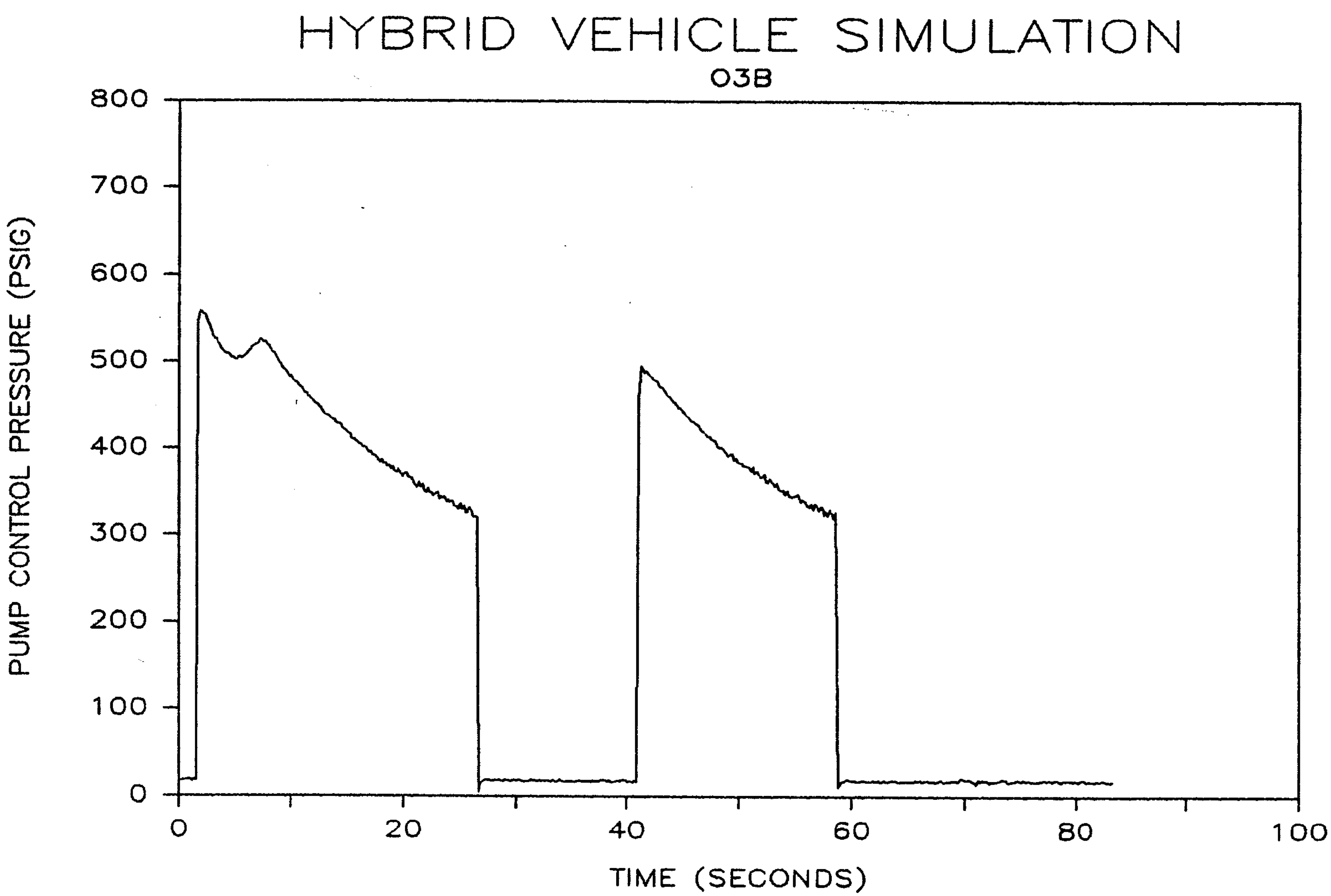

Fig. 5.22: Pump Control Pressure (Test 3)

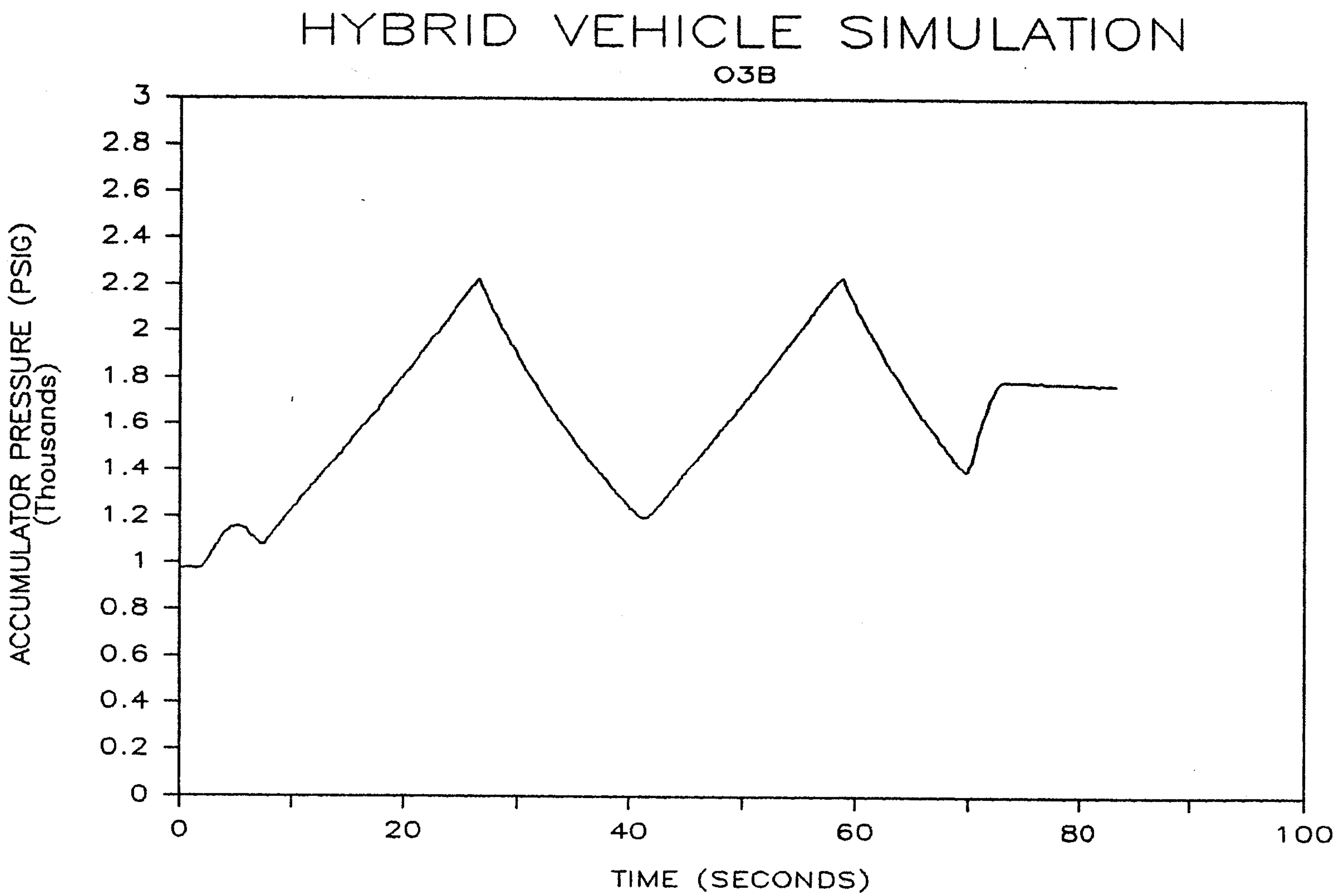

Fig. 5.23: Accumulator Pressure (Test 3)

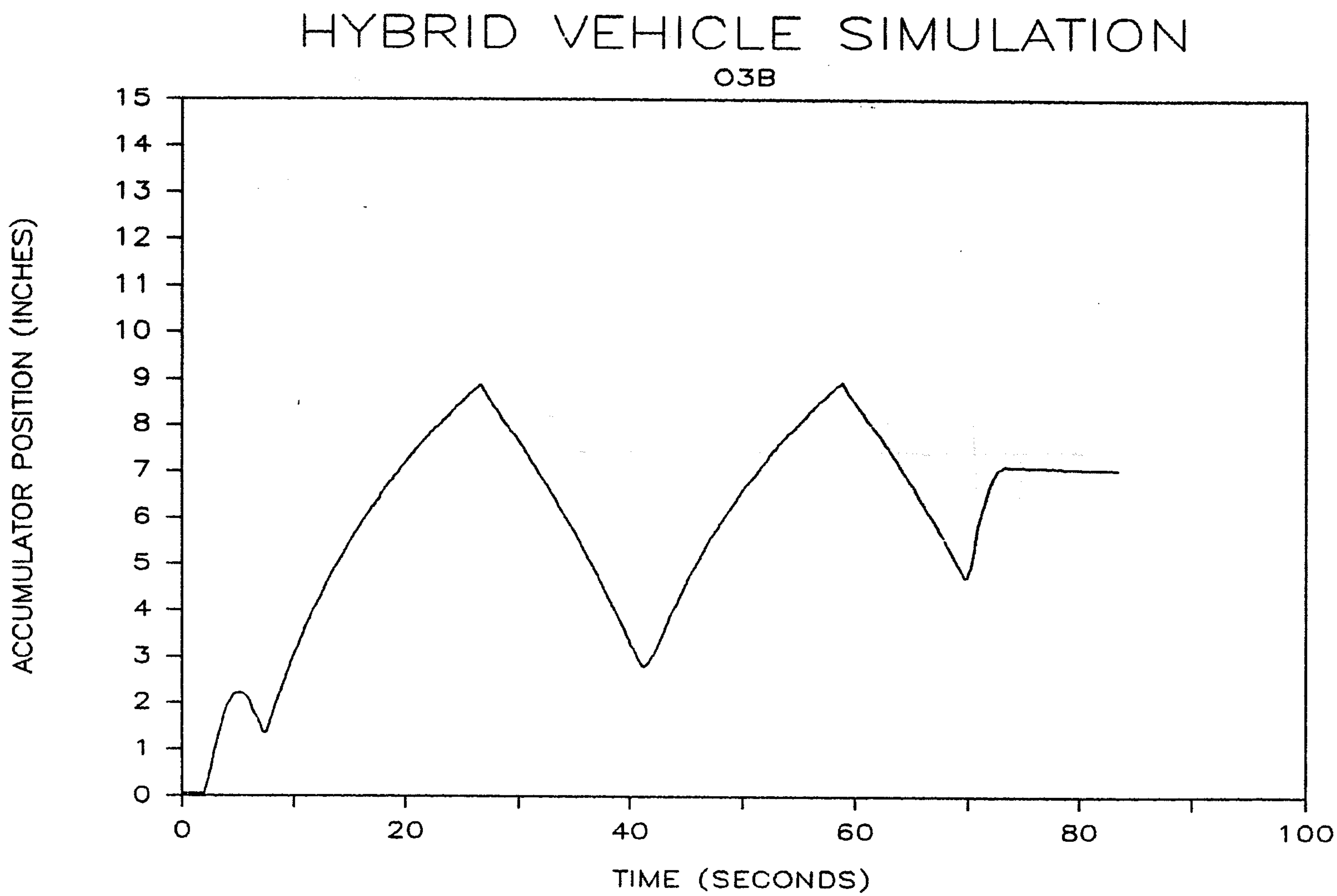

Fig. 5.24: Accumulator Position (Test 3)

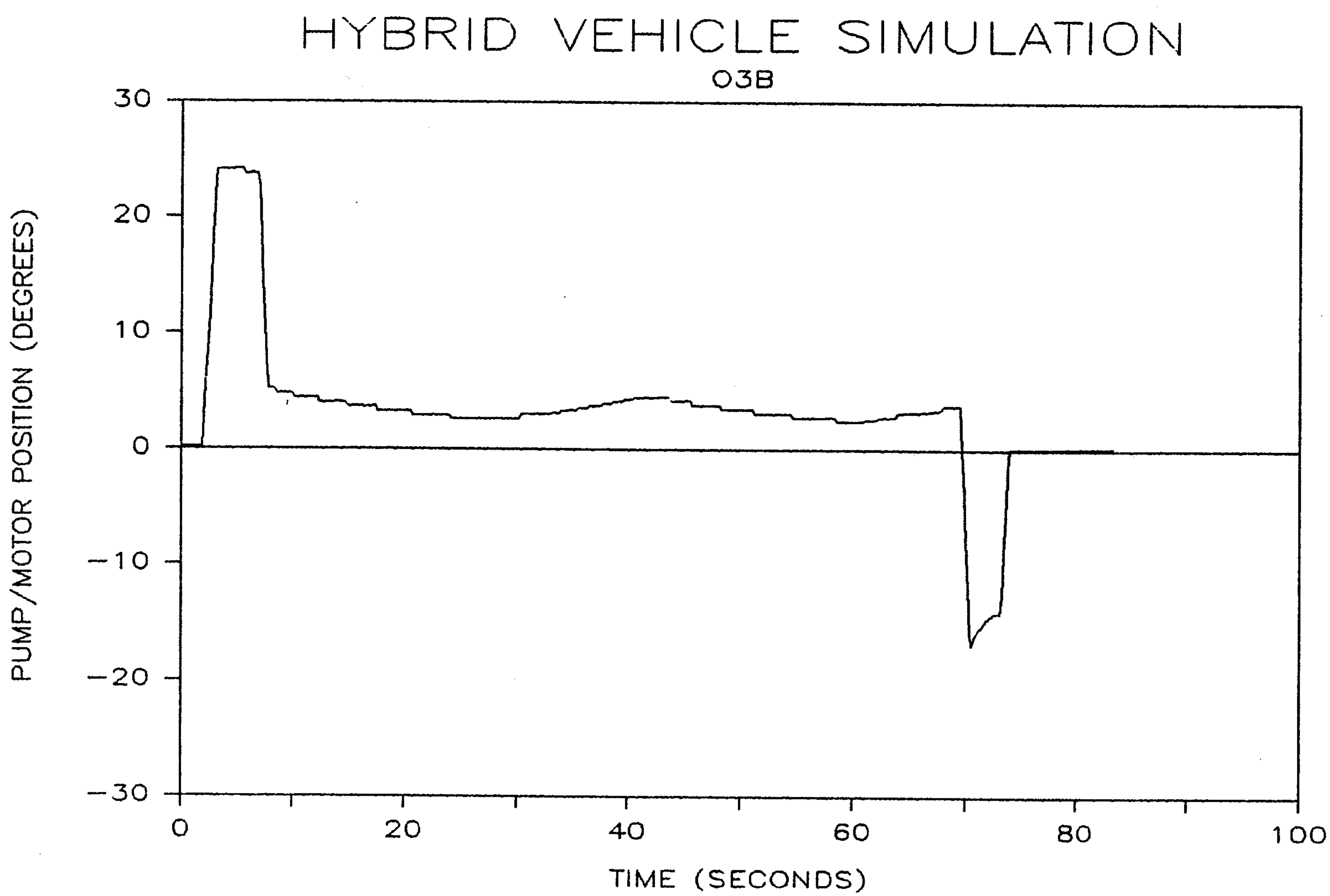

Fig. 5.25: Pump/Motor Position (Test 3)

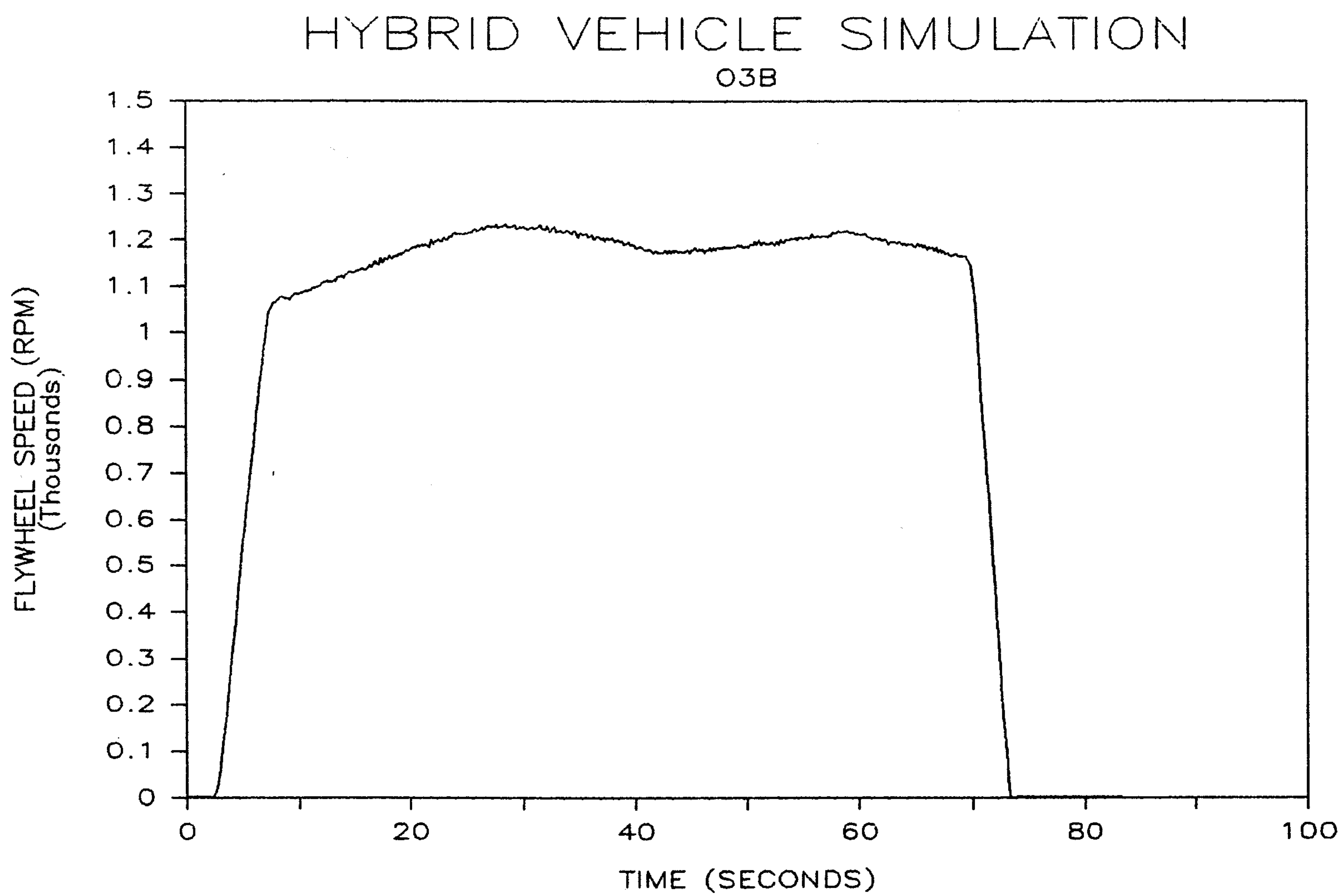

Fig. 5.26: Flywheel Speed (Test 3)

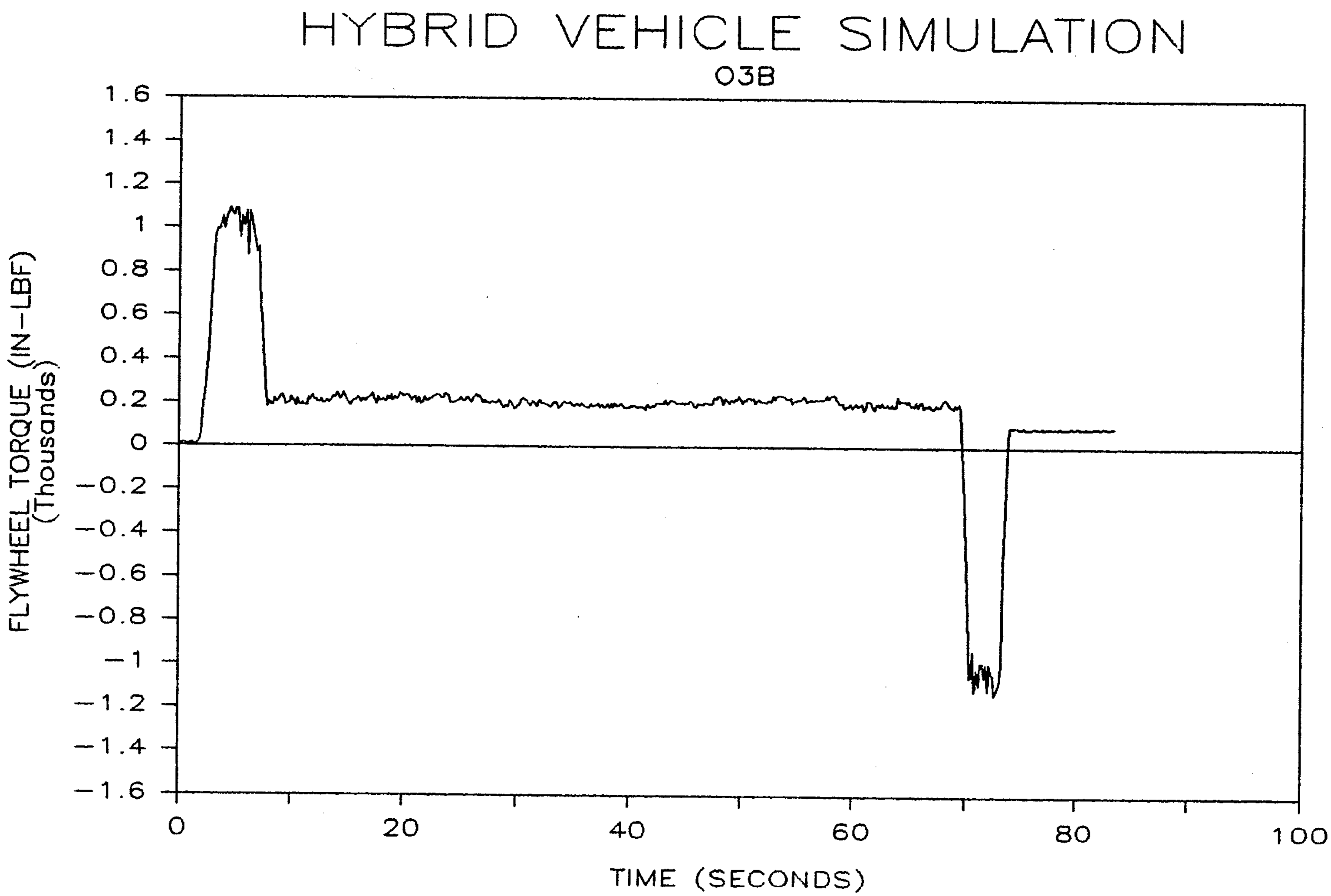

Fig. 5.27: Flywheel Torque (Test 3)

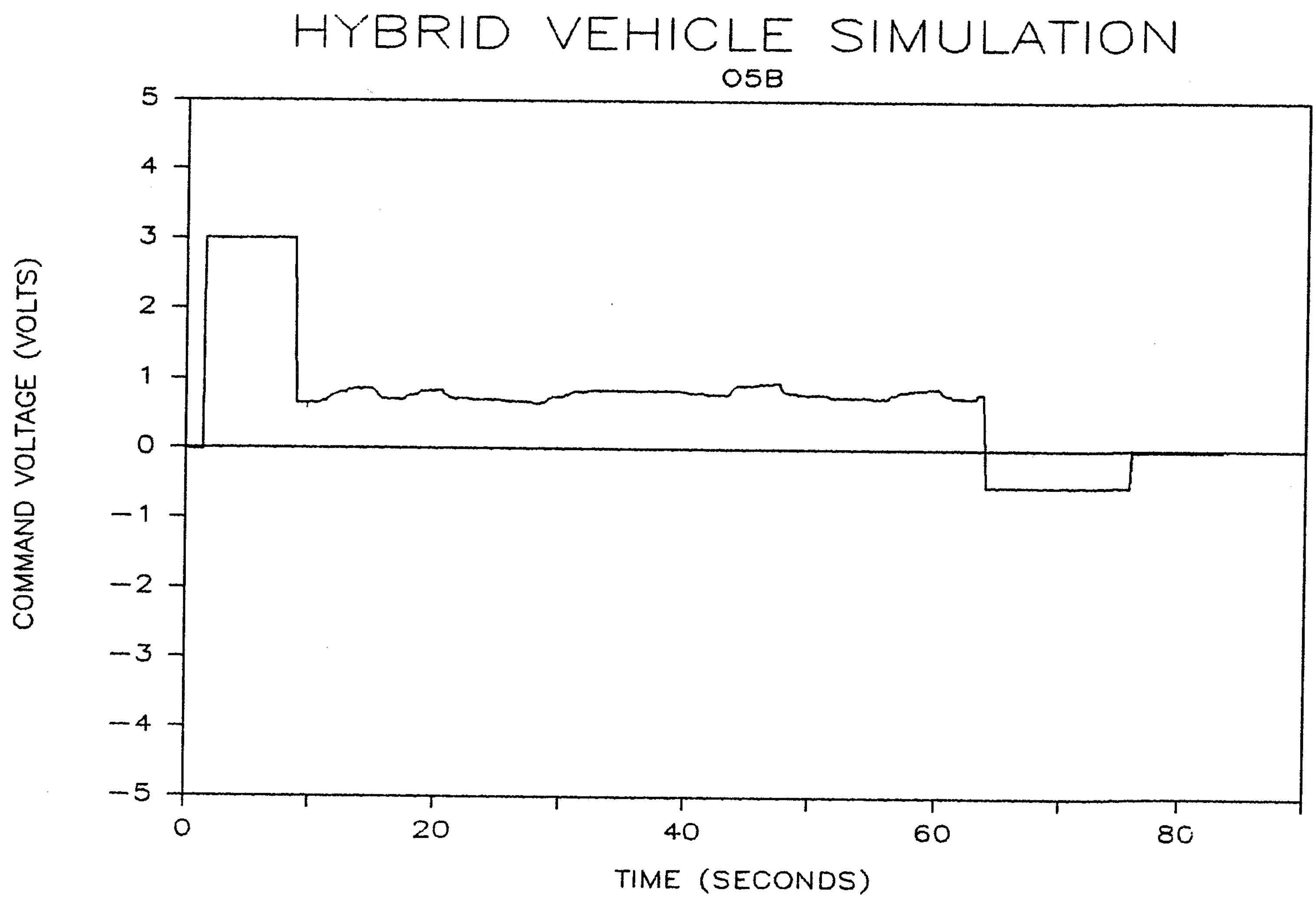

Fig. 5.28: Command Voltage (Test 5)

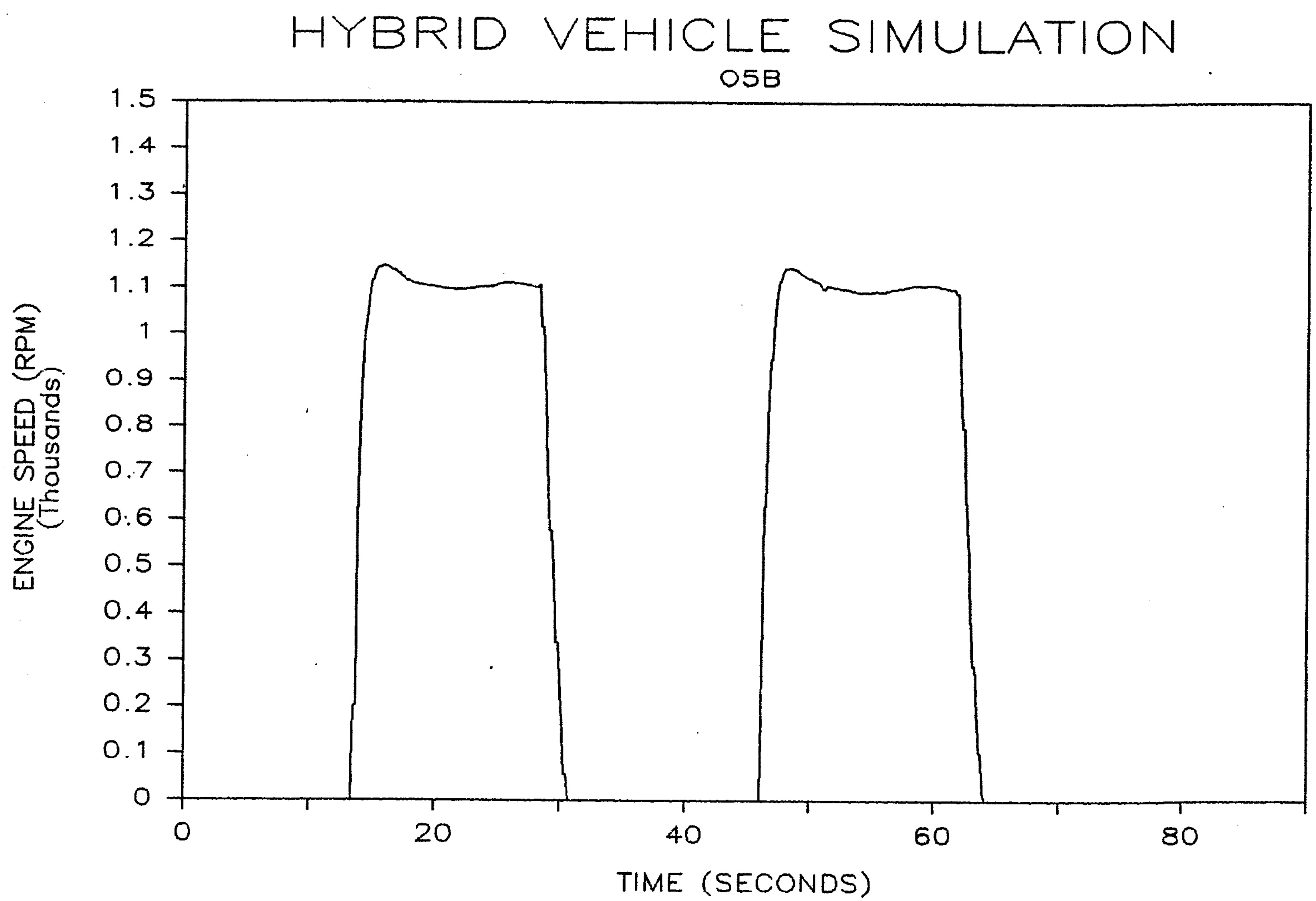

Fig. 5.29: Engine Speed (Test 5)

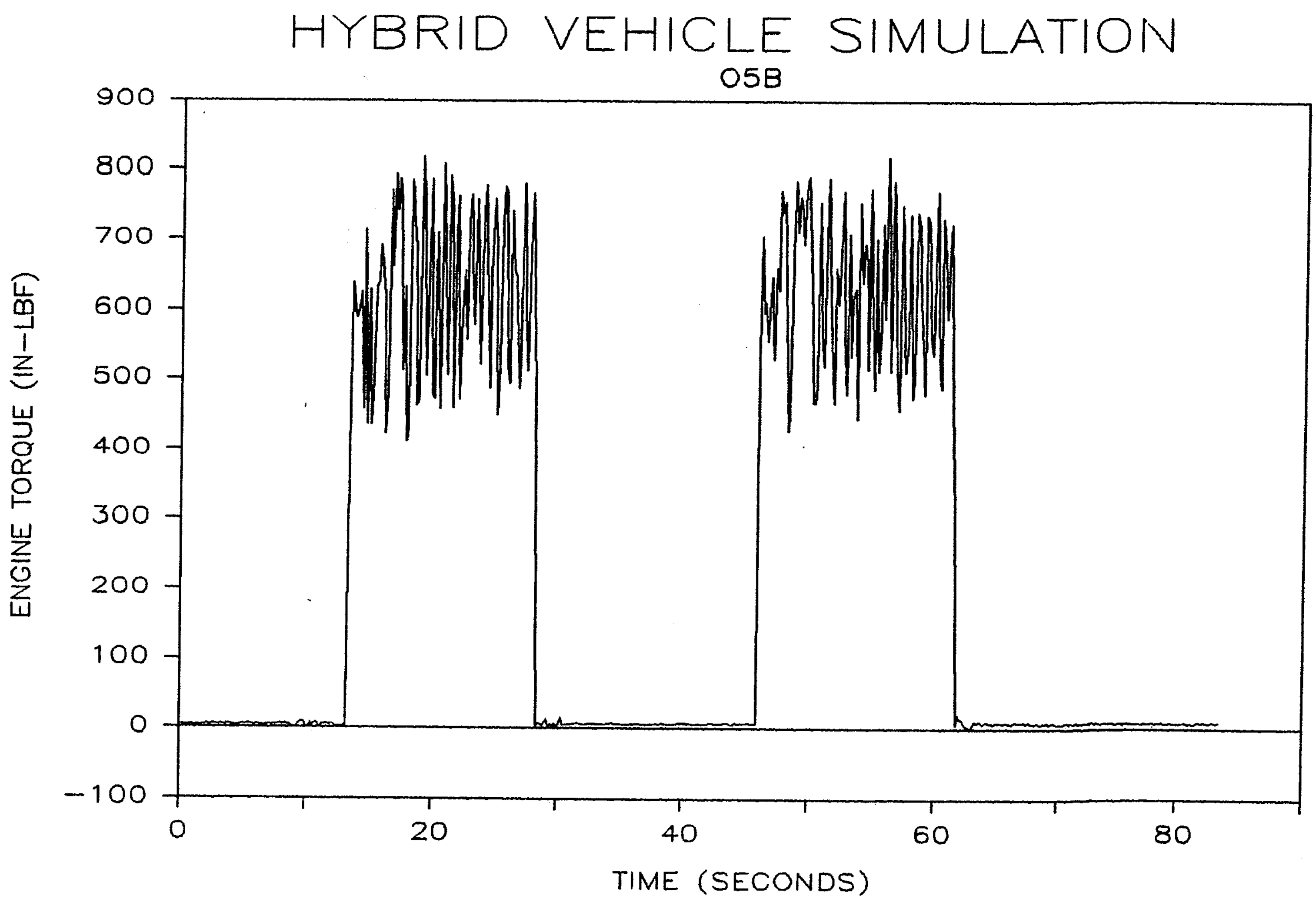

Fig. 5.30: Engine Torque (Test 5)

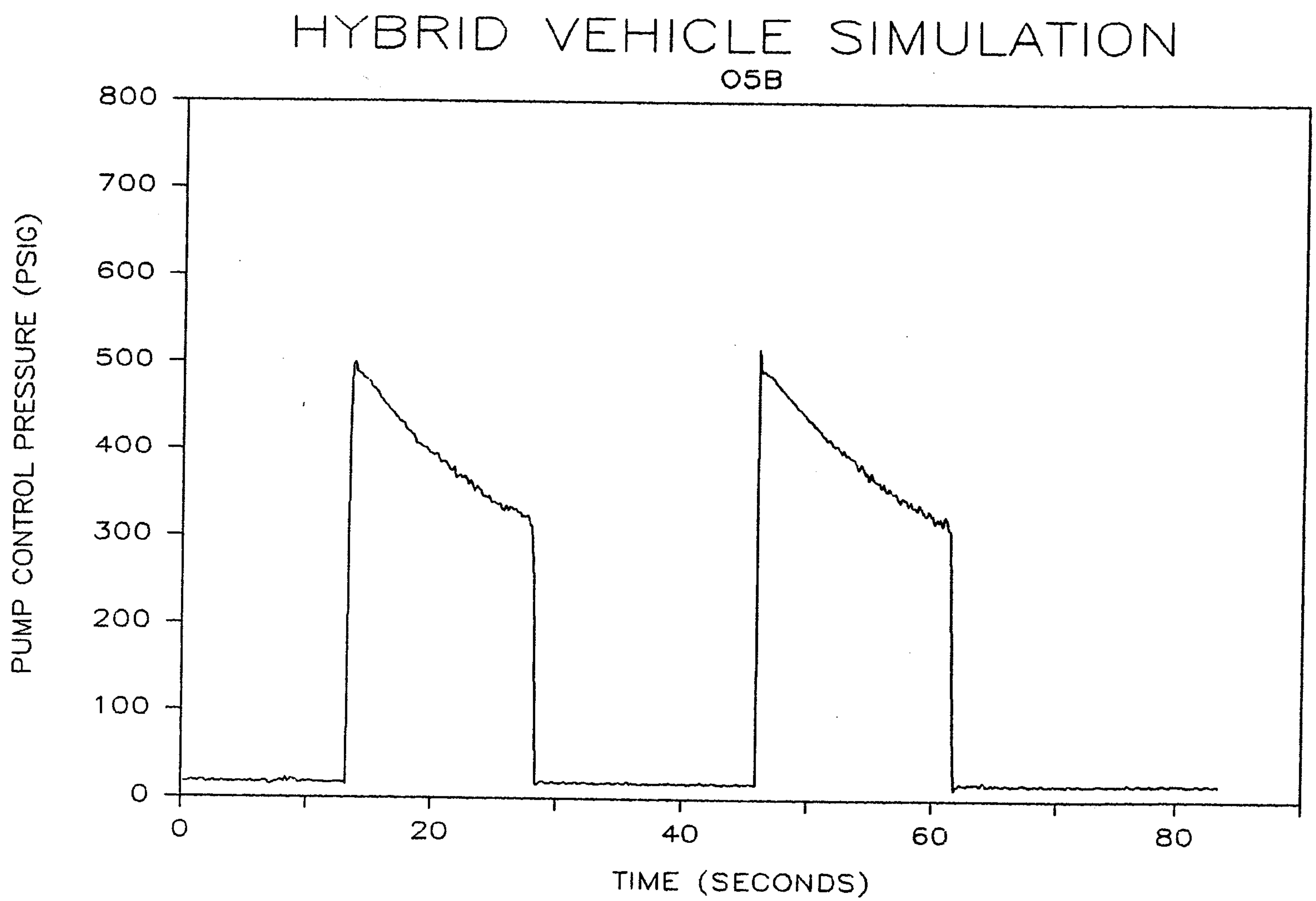

Fig. 5.31: Pump Control Pressure (Test 5)

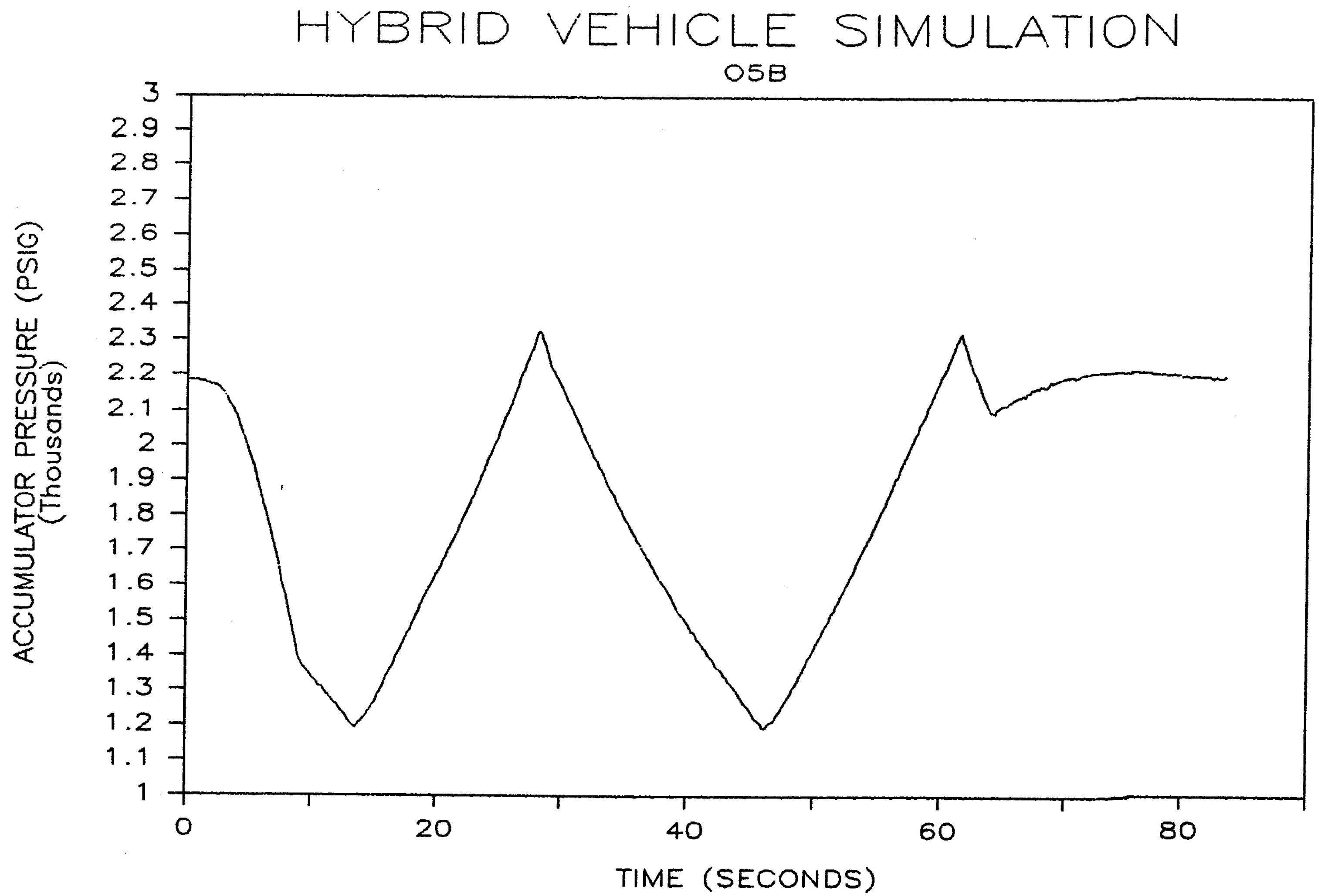

Fig. 5.32: Accumulator Pressure (Test 5)

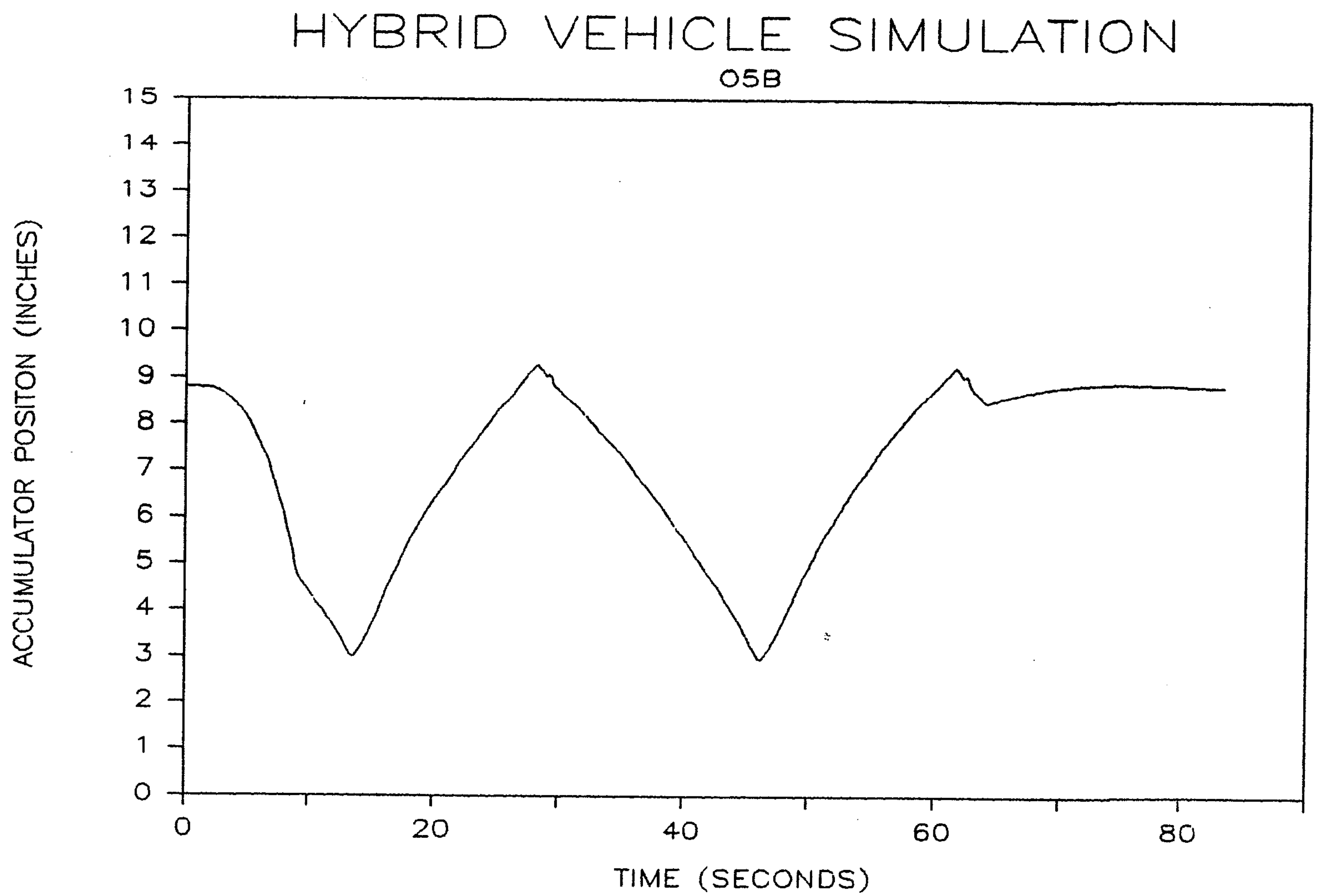

Fig. 5.33: Accumulator Position (Test 5)

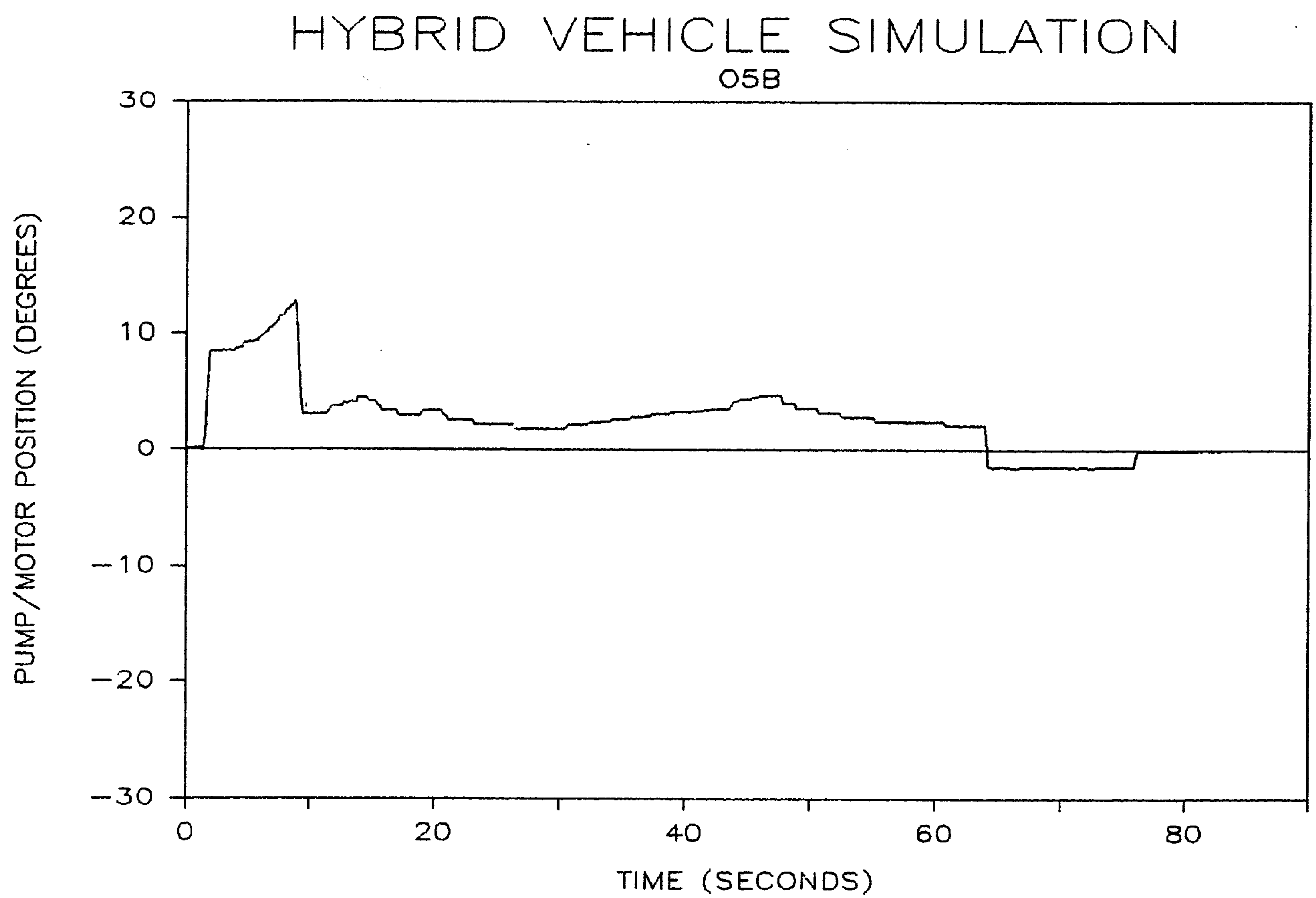

Fig. 5.34: Pump/Motor Position (Test 5)

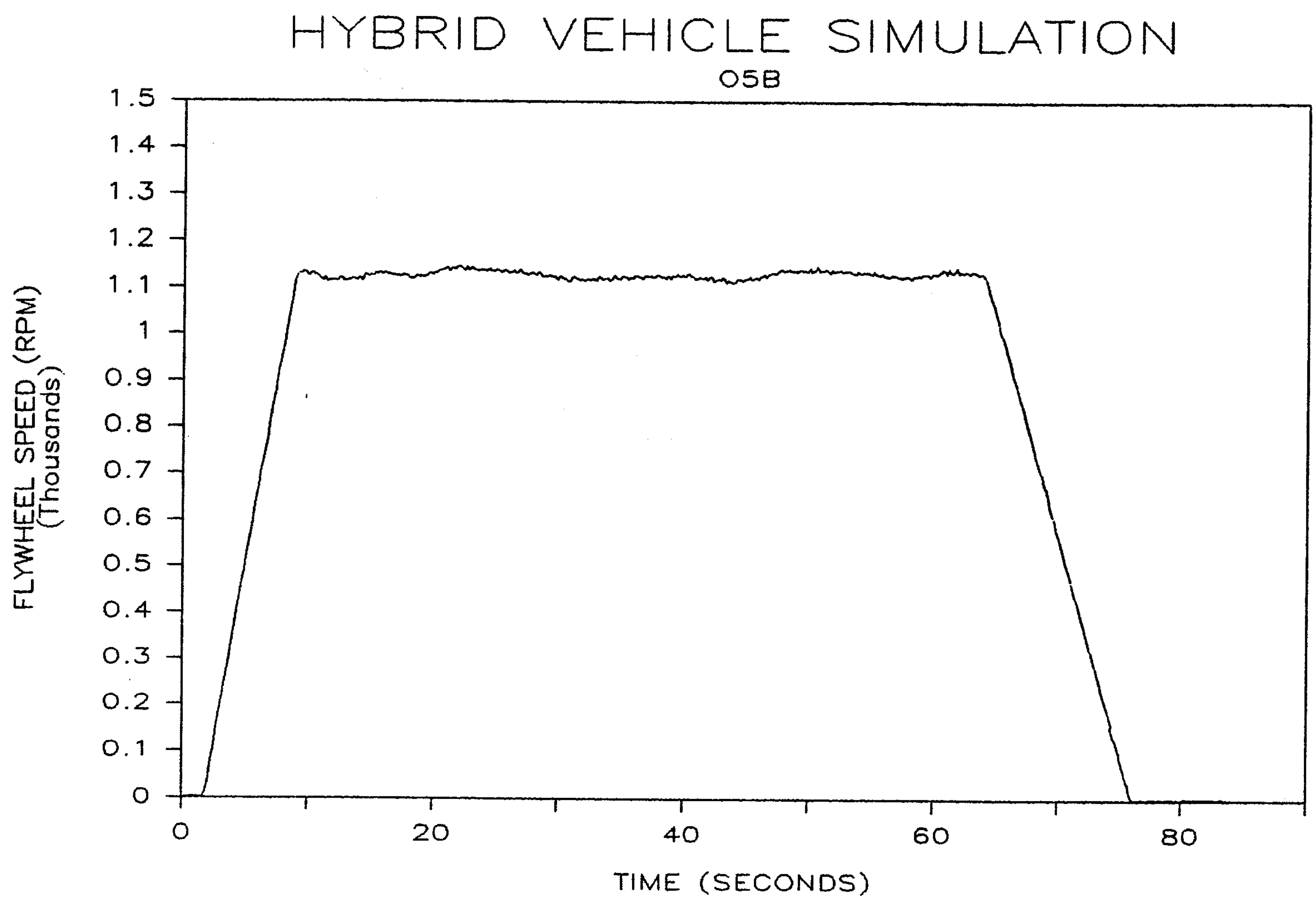

Fig. 5.35: Flywheel Speed (Test 5)

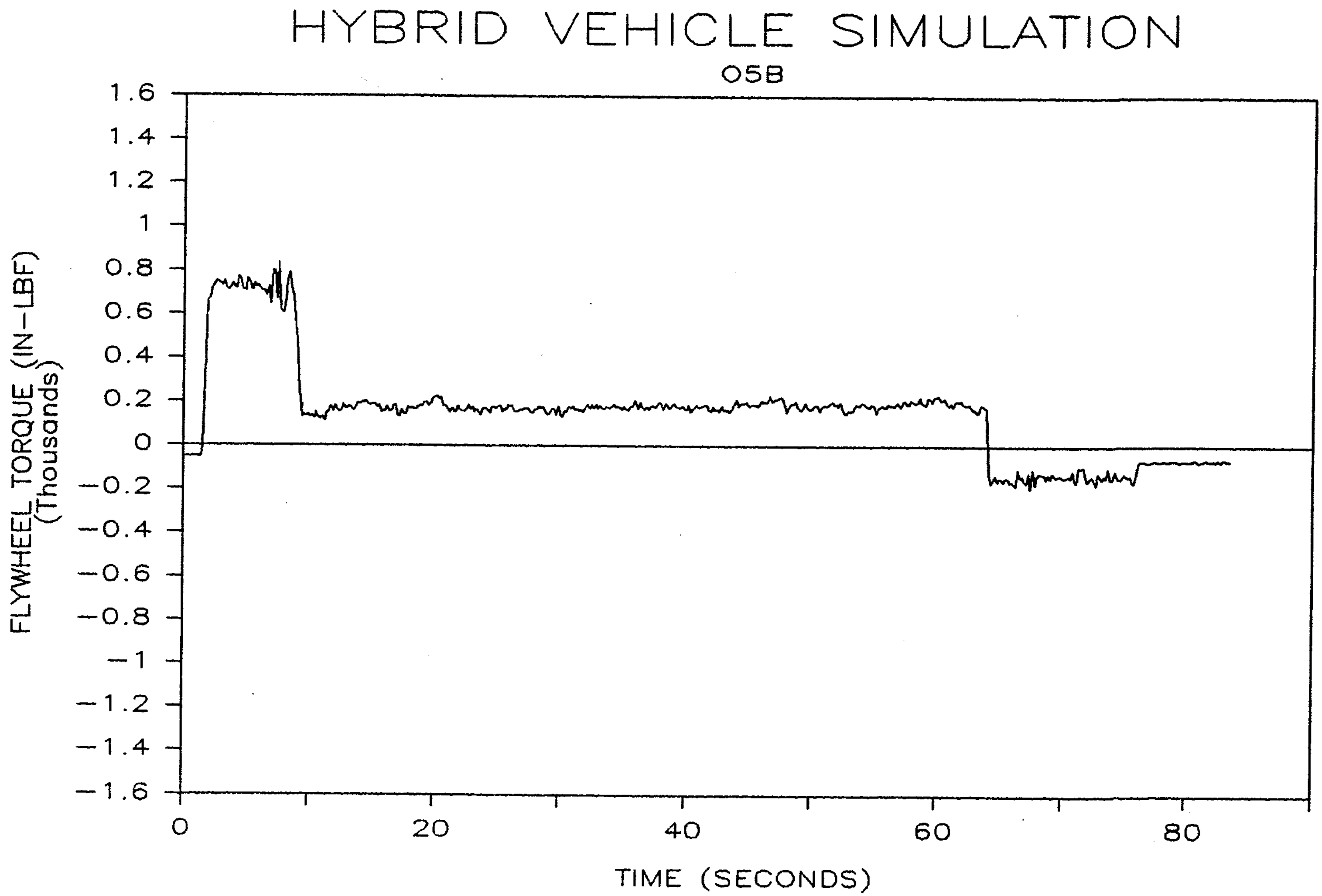

Fig. 5.36: Flywheel Torque (Test 5)

These simple driving cycles were performed to evaluate the ability of the system and the controller to follow a previously defined driving pattern. Two random driving cycles were also run to provide more data on system performance and response to a continuously varying driver signal. These were done using a driver who sent varying voltage signals to the controller, accelerating or decelerating the flywheel.

Seven tests were run with various starting conditions and driving cycles. Table II lists the various conditions for the tests.

TABLE II: TEST IDENTIFICATION

TEST NO.	STARTING ACCUMULATOR STATE	STARTING ENGINE STATE	TEST PATTERN AND COMMENTS
1	charged	off	random driving cycle
2	charged	off	fast acceleration, engine turns on and off during cruise, fast deceleration
3	discharged	on	start with engine on during fast acceleration; engine did not maintain constant speed

4	low	off	engine turns on during slow acceleration and off during cruise; fast deceleration
5	charged	off	medium acceleration, engine turns on and off during cruise, manual correction of input command by driver to maintain more constant speed during cruise, slow deceleration (only graphical data provided for this test)
6	low	off	engine turns on during slow acceleration, manual correction of input command by driver (as in test #5), medium deceleration
7	charged	off	random driving cycle; engine did not maintain constant speed
8	low	off	engine turns on during medium acceleration, manual correction of input command by driver (as in test #5), medium deceleration

5.2.0 Test Procedure

During each of the test runs a brake load was applied to the flywheel to simulate a road load. An accumulator was in the hydraulic circuit that actuated the brake. The accumulator was charged to the required pressure for the desired load and helped to maintain a constant brake pressure. The brake load however, was not always constant due to heating, cooling, and wear of the brake pads. The drag torque used for the simulation was approximately 200 in-lbf (22.59 N-m). That is, the brake was adjusted so that it took approximately 200 in-lbf (22.59 N-m) to drive the flywheel at a constant speed.

A typical run consisted of driving the "car" to adjust the constant speed voltage to compensate for brake wear and heating. Since it was desired to have the simulated engine turn on during different parts of the driving cycle, the accumulator pressure was adjusted so that the engine would either turn on immediately, during the acceleration period, during constant speed driving, or during the deceleration period. Once these variables were set, the data acquisition was started and the driving cycle was completed using one of the methods described above.

These driving cycles were accomplished using two different methods. The first method involved using "push button driving". In this method the control computer was fed a constant voltage from one of three potentiometers through a panel of pushbuttons used to switch the signal.

The first potentiometer was set to give a voltage signal which would produce the desired acceleration, the second was set for the constant speed voltage, and the last for the deceleration voltage. A driving cycle consisted of starting with the flywheel at rest and then switching the control voltage with the push buttons in order to change from acceleration to constant speed to braking. In the second method the acceleration and braking were run with the constant voltage from the potentiometers; however, the constant speed phase of the cycle was manually controlled by means of a driver monitoring flywheel speed and making small adjustments to maintain a more constant flywheel speed. In each of the methods the acceleration and deceleration rates were changed in order to provide more complete data on system performance.

5.3.0 Evaluation of the Test Results

The entire system was found to be very controllable and responded well to the driver's input command. Whether the engine was on or off, there was sufficient power to drive the flywheel. Transitions from engine-driving to accumulator-driving were smooth. This was largely due to the accumulators, which served as the link between two halves of the system. The accumulators acted as cushions and isolated much of each subsystem's dynamics.

5.3.1 Evaluation of the Engine-Pump Subsystem

The engine was operated near an "ideal point". The control program adjusted the pump's displacement in an attempt to maintain the operation at a constant engine torque and speed. Because the accumulator pressure changed relatively slowly no matter what the flywheel was doing, this subsystem was largely dynamically independent of the other. Several observations of the actual response of the engine-pump subsystem can been seen in Fig. 5.1 through 5.36.

The engine was turned on by the computer controller when the accumulator pressure was 1200 psia (8.28 MPa). The 1200 psia (8.28 MPa) value was chosen to prevent cavitation in the hydraulic pump/motor unit. Cavitation would occur if the accumulator pressure was allowed to drop below the 1000 psia (6.9 MPa) accumulator precharge pressure. The accumulator pressure dropped slightly below 1200 psia (8.28 MPa) as energy was being taking from the accumulator to drive the flywheel. The simulated engine rapidly accelerated to full power in the approximately 1 second. The accumulator pressure rose rapidly in most cases once the engine turned on since the output power from the engine was much greater than the power desired by the driver. The accumulator pressure rose more gradually if the flywheel was being accelerated when the engine turned on.

The speed of the engine-pump subsystem was not constant. The engine speed varied from 980 to 1030 rpm. The speed variations can be attributed to several sources. The accumulator pressure transducer was measuring the gas pressure inside one accumulator, not the oil pressure. There is a slight difference between these two due to the force needed to move the accumulator piston and to compress the foam. For this reason future implementations of this system should monitor the oil pressure instead of the gas pressure. The pump displacement for constant torque was probably not properly determined for low pressures. It should have been increased for this range. In all of the tests the engine tended to overshoot the desired speed when the accumulator pressure was low. Increasing pump displacement would increase the input torque, thus slowing the engine down. The problem of a non-constant engine speed is not critical to the whole system. Since an internal combustion engine has a wide area of high efficiency near the ideal line, the overall system efficiency will not be significantly reduced.

The torque of the engine-pump subsystem was not constant. The engine torque fluctuated from 480 to 720 in-lbs (54.21-81.31 N-m). The torque fluctuations can be attributed to the dynamics of the hydraulic units, the pump pulsations from the pistons. The torque signals were analyzed by a frequency analyzer. For the engine torque the predominant frequencies were 65 to 70 Hz. These frequencies roughly correspond to the frequency of pistons in the pump. In addition the output torque signal from the torque transducer contained a lot of noise. This noise can be seen on the engine torque curves even when the engine is off.

The time at which the engine turns off is shown to be a function of both the accumulator pressure and the flywheel speed. The accumulator pressures at which the engine shut off ranged from 2250 psia (15.525 MPa) to 2970 psia (20.49 MPa) for the tests results shown. The flywheel speed was 1200 rpm when the engine shut off at 2250 psia (15.525 MPa) and the flywheel speed was 100 rpm when the engine shut off at 2970 psia (20.49 MPa).

The pump control pressure which is proportional to the pump's displacement is seen as a function of the accumulator pressure. When the engine first turns on and the accumulator pressure is the lowest, the pump displacement is the largest. Then as the accumulator pressure increases, the pump displacement is decreased proportionately to provide a constant torque.

5.3.2 Evaluation of the Pump/Motor and Flywheel Subsystem

The pump/motor functions as a motor to accelerate the flywheel and functions as a pump to brake the flywheel. Because the accumulator pressure changed relatively slowly no matter what the flywheel was doing, this subsystem was largely dynamically independent of the other. Several observations of the actual response of the pump/motor and flywheel subsystem can been seen in Fig. 5.1 through 5.36.

The driver's input to the controller is interpreted by the control program as a torque command. The close similarities between the diver's command voltage and the flywheel torque reflects this relationship. The flywheel torque signal contained some noise. The torque fluctuated about 50 to 100 in-lbs (5.646 to 11.29 N-m). A frequency analysis was performed on the flywheel torque data. The most significant flywheel torque frequencies were 57 to 66 Hz, 81 Hz, and 114 Hz. The lowest ones also correspond to the frequency of piston pumping. The higher ones are probably due to the natural frequencies of the pump/motor-torquemeter-flywheel system. All of these frequencies were high enough that they were damped out by the flywheel inertia. The flywheel speed did not show any of these fluctuations.

The pump/motor position is seen to be positive, functioning as a motor, when the flywheel is accelerated. The pump/motor position is seen to be negative, functioning as a pump, when the flywheel is decelerated. The pump/motor position is slightly positive when the flywheel is held at a steady speed since some power is required to match the road load. The pump/motor displacement increased when the flywheel speed was constant in order to maintain a constant torque as the accumulator pressure decreases. In this way the subsystem compensates for the variation in accumulator pressure.

An exactly constant speed was not maintained for a constant driver command as the accumulator pressure varied. In general, as the accumulator pressure increased during the "constant speed" parts of the tests, the flywheel speed increased by as much as 120 rpm. The opposite occurred when the pressure decreased. The reason for this is due to the control algorithm. In the algorithm no correction in displacement is made unless the error is greater than about one third of a degree. If the displacement was set exactly right when the accumulator pressure started rising, the pump/motor should be destroked in order to maintain constant torque. This destroking of the pump/motor does not occur until the error is one third of a degree. In the intervening time the extra torque accelerates the flywheel.

This process repeats itself, and the flywheel continues to speed up. Another problem with the algorithm is that it calculates the proper displacement based on the equation that displacement equals torque divided by pressure. No correction was made for torque efficiency, which varies considerably at low displacements.

Maintaining an exactly constant speed is not critical to this system. In an automobile the driver provides the final feedback loop. He adjusts his foot to achieve the desired speed. In order to evaluate the accuracy with which the system could be controlled by a human driver, driving cycles consisting of accelerations, decelerations, and constant speed sections were run. The average percent deviations in flywheel speeds between the preassigned cycles speeds and the driven speeds were determined. The average percent deviations in speed were in the 2 to 4 percent range. Maintaining a constant flywheel speed was more difficult than maintaining a constant vehicle speed for a human driver since the driver in the moving vehicle has additional instantaneous visual and physical feedback while the driver in the simulation tried to match the flywheel speed curve to a curve drawn the computer monitor.

6.0 CONCLUSIONS

A versatile, well-instrumented test rig was designed and built in order to experimentally evaluate the efficiency, performance, and control aspects of a hydrostatic accumulator energy storage automobile. The results of previous analytical studies were used in determining preliminary hardware designs and tentative control policies. The tests conducted with the test rig have provided useful data on the system's efficiency and controllability. The components and system efficiency values were calculated and are contained in Ref. [15].

The results of the driving cycle tests show that the system is stable and is relatively easy to control. The two-stage pump/motor displacement controller in combination with a simple on/off computer control algorithm provided the driver with an adequate control over the simulated vehicle's velocity and acceleration. The two-stage pump/motor controller provides an inexpensive, low energy usage, and accurate replacement for servovalves in a wide range of applications. Research should be done on determining the effects of changes in the speed and accuracy of the controller on the system's controllability. Digital control techniques such as Pulse Width Modulation and Pulse Duration Modulation should provide a more precise and responsive control of the pump/motor's displacement than does the simple on/off algorithm. The poppet and solenoid valve characteristics should be determined in order to obtain a valid control system model.

The simulated engine should be modified to operate along an "ideal line" instead at an "ideal point" as was done in this study. Operating along an ideal line will reduce the engine's cycling rate and will allow the engine to run at its maximum output power when necessary.

The system should be tested with a hydraulic pump/motor that does not go "overcenter". The valving arrangement for such a pump/motor will be more complex than for the pump/motor currently used. Since a pump/motor which does not go "overcenter" can be made with less clearance volume than a pump/motor which does go "overcenter", the volumetric efficiency should be improved. The efficiency improvements should be most noticeable at low displacements.

An additional hydraulic pump/motor connected to the flywheel can be used to simulate the vehicle's road load. The vehicle's fuel economy can be predicted by operating the system over a driving cycle such as the Federal Urban Driving Cycle.

The results of the test rig studies can be used to design and build a hybrid hydraulic accumulator energy storage automobile.

REFERENCES

1. Nasvytis, A. L., "Flywheel Automotive Vehicles," U.S. Patent No. 3,672,244, June 1972.

2. Elder, F. T., and Otis, D. R., "Simulation of a Hydraulic Vehicle Power Train," ASME paper No. 73-ICT-50, 1973.

3. Schwarz, R., "Design Considerations for a Near-Term Hybrid Vehicle," 15th Intersociety Energy Conversion Engineering Conference, Washington, D. C., 1980, pp.1760-1765.

4. Honda, Shoji, "Developments and Future Prospects of Hybrid Vehicles," Electric Vehicle Developments, No. 12, Dec. 1981, pp.18-21.

5. Svoboda, J. V., "Analog and Digital Modelling in the Design of a Hydraulic Vehicular Transmission," Proc. I. Mech. E., Vol. 193, No. 12, 1979.

6. Frank, A. A., and Beachley, N. H., "Evaluation of the Flywheel Drive Concept for Passenger Vehicles," SAE Paper 790049, 1979.

7. Otis, D. R. and Pourmovahed, A.,"Improving Performance of Gas-Charged Accumulators Using Elastomeric Foams," Proceedings, International Symposium on Advanced and Hybrid Vehicles, U. of Strathclyde, Glasgow, Scotland, Sept. 1984.

8. Tollefson, S., Beachley, N. H., and Fronczak, F. J., "Studies of an Accumulator Energy-Storage Automobile Design with a Single Pump/Motor Unit," paper presented at the 1985 SAE Passenger Car Meeting.

9. Buchwald, P., et al., "Improvement of Citybus Fuel Economy Using a Hydraulic Hybrid Propulsion System - a Theoretical and Experimental Study," SAE paper 840056, 1984.

10. Lewandowski, E., "Design of a Test Facility for a Hydropneumatic Accumulator Energy Storage Vehicle Research," M. S. Thesis, University of Wisconsin - Madison, 1987.

11. Rappel, B., "Adaptive Control for Hydraulic Pump/Motors," , MSE Thesis report, University of Wisconsin - Madison, 1986.

12. Pourmovahed, A., Baum S. A., Fronczak, F. J., and

Beachley, N. H., "Experimental Evaluation of Hydraulic Accumulator Efficiency with and without Elastomeric Foam," Proceedings, 22nd Intersociety Energy Conversion Engineering Conference, Philadelphia, Pa., August 1986.

13. Beachley, N. H., "Control Strategies for an Internal Combustion Engine/ Accumulator Energy-Storage Automobile," Proceedings, Symposium on Advanced and Hybrid Vehicles, University of Strathclyde, Glasgow, Scotland, Sept. 1984, pp. 72-82.

14. Beachley, N. H., Anscomb, C., and Burrows, C. R., "Minimization of Energy Storage Requirements for Internal Combustion Engine Hybrid Vehicles," AMSE Journal of Dynamic Systems, Measurement and Control, June 1983, Vol. 105, pp. 113-119.

15. Baum, S. A., "Investigation of a Dual Pump/Motor Hydraulic Accumulator Energy Storage Automobile," M. S. Thesis, University of Wisconsin - Madison, 1987.

APPENDIX A

```
2000 '
2010 '*****************************************
2020 '* MAIN PROGRAM : THIS PROGRAM ALLOWS FOR  *
2030 '* MULTICHANNEL ANALOG INPUT.              *
2040 '*****************************************
2050 '
2060 CLS:SCREEN 2
2061 DIM VAR$(8)                     ' VARIABLE NAMES
2062 DIM CO(8,7)                     ' COEFFICIENTS OF EQUATIONS
2063 DIM RO%(8)                      ' ORDERS OF EQUATIONS
2064 DIM RAW%(4000)                  ' RAW UNCONVERTED DATA
2065 DIM DAT(5,1030)                 ' SCALED DATA
2066 DIM MINX(8) : DIM MAXX(8)       ' X-AXIS MINIMUM AND
                                       MAXIMUM VALUES
2067 DIM MINY(8) : DIM MAXY(8)       ' Y-AXIS MINIMUM AND
                                       MAXIMUM VALUES
2068 DIM PT(2,1030)                  ' PLOTTING XY VALUES
2070 GOSUB 3000 ' INPUT THE PARAMETERS
2080 INPUT "HIT RETURN TO START SAMPLING ",A$
2090 GOSUB 4000 ' START SAMPLING
2100 PRINT " SCALING DATA " : GOSUB 5000 ' CONVERT THE DATA
2120 CLS : SCREEN 2
2130 A$ = "0"
2140 PRINT "  PRESS A NUMBER (0-4) TO EXECUTE THE "
2145 PRINT "        FOLLOWING COMMANDS "
2150 PRINT
2160 PRINT "              1. VIEW A GRAPH
2170 PRINT "              2. DISPLAY X VS Y DATA
2180 PRINT "              3. WRITE TO A FILE
2185 PRINT "              4. EXIT
2190 PRINT
2200 A$ = INKEY$
2210 IF A$="1" OR  A$="2" OR  A$="3"  OR A$="4" THEN 2225
2220 GOTO 2200
2225 I = VAL(A$)
2230 ON I GOTO 2240,2260,2265,2310
2240 GOSUB 7000 : GOSUB 8000
2250 GOTO 2270
2260 GOSUB 7000 : GOSUB 9000 : GOTO 2270
2265 GOSUB 6000 : GOTO 2270
2270 A$ = " "
2280 A$ = INKEY$
2290 IF A$ = "" GOTO 2280
2300 GOTO 2120
2310 END
```

```
3000 '**************************************************
3010 '* SUBROUTINE : THIS SUBROUTINE LETS THE USER INPUT *
3020 '* THE PARAMETERS INTERACTIVELY OR AS A FILE.      *
3030 '**************************************************
3040 '
3050 PRINT "MAKE SURE THE CAPS LOCK KEY IS ACTIVE "
3060 PRINT
3070 INPUT "IS THERE AN INPUT FILE ? (Y/N) ",AN$
3080 IF AN$ = "Y" THEN 3330
3090 INPUT "ENTER THE NUMBER OF SAMPLES ",NSAM
3100 INPUT "ENTER THE NUMBER OF SAMPLES PER SECOND ",NSPS
3110 INPUT "ENTER THE NUMBER OF CHANNELS ",NCHAN%
3200 FOR A% =  1 TO NCHAN%
3210 PRINT "CHANNEL ";A%
3220 INPUT "ENTER THE VARIABLE NAME ",VAR$(A%)
3230 INPUT "ENTER THE ORDER ",RO%(A%)
3240 FOR G% = 0 TO 7
3250 CO(A%,G%) = 0
3260 NEXT G%
3270 FOR F% = 0 TO RO%(A%)
3280 PRINT "COEFFICIENT ";F%
3290 INPUT "ENTER THE COEFFICIENT ",CO(A%,F%)
3300 NEXT F%
3310 NEXT A%
3320 GOTO 3500
3330 INPUT "ENTER THE INPUT FILENAME ",FILE1$
3340 B$ = "B:"
3350 FILE2$ = B$ + FILE1$
3360 OPEN FILE2$ FOR INPUT AS 1
3370 INPUT #1,NSAM
3380 INPUT #1,NSPS
3390 INPUT #1,NCHAN%
3400 FOR A% =  1 TO NCHAN%
3410 FOR G% = 0 TO 7
3420 CO(A%,G%) = 0
3430 NEXT G%
3440 INPUT #1,VAR$(A%)
3450 INPUT #1,RO%(A%)
3460 FOR F% = 0 TO RO%(A%)
3470 INPUT #1,CO(A%,F%)
3480 NEXT F%
3490 NEXT A%
3500 CLOSE #1
3510 RETURN
```

```
4000 '
4010 '***************************************************
4020 '* SUBROUTINE : THIS SUBROUTINE SAMPLES THE DATA   *
4030 '***************************************************
4040 '
4050 LOW% = 1 : HIGH% = LOW% +NCHAN% - 1 : OPT$ = " "
4060 PRINT "START SAMPLING"
4070 CALL AINSC(LOW%,HIGH%,NSAM,NSPS,RAW%(1),OPT$)
4080 PRINT "END SAMPLING"
4090 RETURN
5000 '
5010 '***************************************************
5020 '* SUBROUTINE : THIS SUBROUTINE CONVERTS THE RAW   *
5030 '* SAMPLED DATA TO SCALED DATA.                    *
5040 '***************************************************
5050 '
5060 COUNT% = 1
5070 FOR C% = 1 TO NSAM
5080 DAT(0,C%) = (C% - 1)/NSPS
5090 FOR B% = 1 TO NCHAN%
5100 DAT(B%,C%) = 0
5110 FOR F% = 0 TO RO%(B%)
5120 IF F%>0 THEN 5150
5130 DAT(B%,C%) = CO(B%,F%)
5140 GOTO 5160
5150 DAT(B%,C%) = DAT(B%,C%) + (RAW%(COUNT%)^F%)*CO(B%,F%)
5160 NEXT F%
5170 IF C% > 1 THEN 5210
5180 MINY(B%) = INT(DAT(B%,C%)-1)
5190 MAXY(B%) = INT(DAT(B%,C%)+1)
5200 GOTO 5230
5210 IF DAT(B%,C%)>MAXY(B%) THEN MAXY(B%)=INT(DAT(B%,C%)+1)
5220 IF DAT(B%,C%)<MINY(B%) THEN MINY(B%)=INT(DAT(B%,C%)-1)
5230 COUNT% = COUNT% +1
5240 NEXT B%
5250 NEXT C%
5260 RETURN
6000 '
6010 '***************************************************
6020 '*SUBROUTINE : THIS SUBROUTINE OUTPUTS THE SCALED *
6030 '*DATA TO A FILE WHICH IS COMPATIBLE TO LOTUS 123.*
6040 '***************************************************
6050 '
6060 INPUT "ENTER THE OUTPUT FILE NAME ",FILE1$
6070 B$ = "B:"
6080 P$ = ".PRN"
6090 FILE2$ = B$ + FILE1$ + P$
6100 OPEN FILE2$ FOR OUTPUT AS 1
```

```
6110 NR% = NSAM
6120 FOR E% = 1 TO NR%
6130 ON NCHAN% GOTO 6140,6160,6180,6200,6220,6240,6250,
     6260,6280
6140 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
6150 GOTO 6300
6160 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%)
6170 GOTO 6300
6180 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%)
6190 GOTO 6300
6200 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%);DAT(4,E%)
6210 GOTO 6300
6220 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%);DAT(4,E%);DAT(5,E%)
6230 GOTO 6300
6240 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%);DAT(4,E%);DAT(5,E%);DAT(6,E%)
6250 GOTO 6300
6260 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%);DAT(4,E%);DAT(5,E%);DAT(6,E%)
     ;DAT(7,E%)
6270 GOTO 6300
6280 PRINT #1,USING "##.#########^^^^ ";DAT(0,E%);DAT(1,E%)
     ;DAT(2,E%);DAT(3,E%);DAT(4,E%);DAT(5,E%);DAT(6,E%)
     ;DAT(7,E%);DAT(8,E%)
6290 GOTO 6300
6300 NEXT E%
6310 CLOSE #1
6320 RETURN
7000 '
7010 '*********************************************
7020 '* SUBROUTINE : THIS SUBROUTINE SETS UP THE    *
7030 '* INPUT PARAMETERS FOR THE PLOTTING.          *
7040 '*********************************************
7050 '
7060 CLS:SCREEN 2
7070 INPUT "ENTER THE CHANNEL ",PCH%
7075 IF PCH% > NCHAN% GOTO 7070
7080 NP = NSAM
7090 XMIN = 0
7100 XMAX = (NSAM-1)/NSPS
7110 YMIN = MINY(PCH%)
7120 YMAX = MAXY(PCH%)
7130 XTITLE$ = "TIME, SECONDS"
7140 YTITLE$ = VAR$(PCH%)
```

```
7160 FOR D% = 1 TO NSAM
7170 PT(2,D%) = DAT(PCH%,D%)
7180 PT(1,D%) = (D% - 1)/NSPS
7190 NEXT D%
7200 RETURN
8000 '
8010 '***************************************************
8020 '* SUBROUTINE : THIS SUBROUTINE PLOTS THE XY GRAPHS *
8030 '***************************************************
8040 '
8050 'NP = NUMBER OF POINTS
8060 'TITLE$ XTITLE$ YTITLE$
8070 'XMIN XMAX YMIN YMAX
8080 'X=PT(1,A) Y=PT(2,A)
8090 KEY OFF:SCREEN 1:CLS
8100 PSET(30,155)
8110 LINE-STEP(260,0):LINE-STEP(0,-145):LINE-STEP(-260,0)
     :LINE-STEP(0,145)
8120 LOCATE 1,20-LEN(TITLE$)/2
8130 PRINT TITLE$
8140 LOCATE 22,20-LEN(XTITLE$)/2 :PRINT XTITLE$
8150 FOR A =0 TO LEN(YTITLE$)
8160 LOCATE (11-(LEN(YTITLE$)/2)+A),1
8170 PRINT MID$(YTITLE$,(A+1),1)
8180 NEXT A
8190 A$ = STR$(XMIN):LOCATE 21,4 :PRINT A$
8200 A$ = STR$(XMAX):LOCATE 21,38-LEN(A$):PRINT A$
8210 A$ = STR$(YMIN):LOCATE 20,1 :PRINT A$
8220 A$ = STR$(YMAX):LOCATE 2,1  :PRINT A$
8230 FOR A = 0 TO 10
8240 PSET(30+A*26,155)
8250 LINE-STEP(0,3)
8260 NEXT A
8270 FOR A= 0 TO 10
8280 PSET (30,10+A*14.5)
8290 LINE-STEP(-3,0)
8300 NEXT A
8310 XP% = INT(30+260*((PT(1,1)-XMIN)/(XMAX-XMIN)))
8320 YP% = INT(155-145*((PT(2,1)-YMIN)/(YMAX-YMIN)))
8330 PSET(XP%,YP%)
8340 FOR A = 2 TO NP
8350 XP% = INT(30+260*((PT(1,A)-XMIN)/(XMAX-XMIN)))
8360 YP% = INT(155-145*((PT(2,A)-YMIN)/(YMAX-YMIN)))
8370 LINE -(XP%,YP%)
8380 NEXT A
8390 RETURN
```

```
9000 '
9010 '***************************************************
9020 '* SUBROUTINE : THIS SUBROUTINE ALLOWS THE XY      *
9030 '* VALUES TO BE PRINTED ON THE SCREEN.             *
9040 '***************************************************
9050 '
9060 CLS
9070 FOR D% = 1 TO NSAM
9080 PRINT PT(1,D%) TAB(20) PT(2,D%)
9090 NEXT D%
9100 RETURN
```

APPENDIX B

```
801 DIM DAT(1,801)
803 DIM RAW%(4801)
804 DIM BRAW%(801)
810 CLS:SCREEN 2
820 INPUT "IS THERE AN INPUT FILE ? (Y/N) ",AN$
830 IF AN$ = "Y" THEN 1080
840 INPUT "ENTER THE NUMBER OF SAMPLES ",NSAM
850 INPUT "ENTER THE NUMBER OF SAMPLES PER SECOND ",NSPS
860 INPUT "ENTER THE NUMBER OF CHANNELS ",NCHAN%
870 GOTO 1331
1080 INPUT "ENTER THE INPUT FILENAME ",FILE1$
1090 B$ = "B:"
1100 FILE2$ = B$ + FILE1$
1110 OPEN FILE2$ FOR INPUT AS 1
1120 INPUT #1,NSAM
1130 INPUT #1,NSPS
1140 INPUT #1,NCHAN%
1330 CLOSE #1
1331 OPT$ ="INTSOFF"
1332 LOW% = 1 : HIGH% = LOW% + NCHAN% - 1
1333 ONE = 1 : FAST = 10000
1334 I% = 1
1350 PRINT "START SAMPLING"
1351 STARTTIME = TIMER
1352 FOR A% = 1 TO NSAM
1354 ENDTIME = STARTTIME + A%/NSPS
1356 ACTTIME = TIMER
1358 IF ACTTIME >= ENDTIME THEN 1360 ELSE 1356
1360 CALL AINSC(LOW%,HIGH%,ONE,FAST,RAW%(I%),OPT$)
1361 I% = I% + NCHAN%
1362 CALL BINS(BRAW%(A%),OPT$)
1363 DIFTIM = ACTTIME - STARTTIME
1364 DAT(0,A%) = DIFTIM
1365 NEXT A%
1370 PRINT "END SAMPLING"
1380 INPUT "ENTER THE RAW DATA OUTPUT FILE ",FR$
1385 FR$ = B$ + FR$
1390 OPEN FR$ FOR OUTPUT AS 1
1400 FOR A% = 1 TO NSAM*NCHAN%
1410 PRINT #1,RAW%(A%)
1420 NEXT A%
1430 FOR A% = 1 TO NSAM
1440 PRINT #1,BRAW%(A%)
1450 NEXT A%
1451 FOR A% = 1 TO NSAM
1452 PRINT #1,DAT(0,A%)
```

```
1453 NEXT A%
1460 CLOSE #1
1470 END
```

APPENDIX C

```
800 DIM VAR$(10)
801 DIM CO(10,7)
802 DIM RO%(10)
803 DIM RAW%(4301)
804 DIM SPD(801)
805 DIM DAT(9,801)
806 DIM MINX(9)       : DIM MAXX(9)
807 DIM MINY(9)       : DIM MAXY(9)
808 DIM BCO(10)
809 DIM PT(2,801)
810 CLS:SCREEN 2
811 INPUT "ENTER THE ENCODER TIME BASE ",SR
820 INPUT "IS THERE AN INPUT FILE ? (Y/N) ",AN$
830 IF AN$ = "Y" THEN 1080
840 INPUT "ENTER THE NUMBER OF SAMPLES ",NSAM
850 INPUT "ENTER THE NUMBER OF SAMPLES PER SECOND ",NSPS
860 INPUT "ENTER THE NUMBER OF CHANNELS ",NCHAN%
940 AP1% = NCHAN% + 1
941 INPUT "ENTER THE BINARY VARIABLE NAME ",VAR$(AP1%)
950 FOR A% =  1 TO NCHAN%
960 PRINT "CHANNEL ";A%
970 INPUT "ENTER THE VARIABLE NAME ",VAR$(A%)
980 INPUT "ENTER THE ORDER ",RO%(A%)
990 FOR G% = 0 TO 7
1000 CO(A%,G%) = 0
1010 NEXT G%
1020 FOR F% = 0 TO RO%(A%)
1030 PRINT "COEFFICIENT ";F%
1040 INPUT "ENTER THE COEFFICIENT ",CO(A%,F%)
1050 NEXT F%
1060 NEXT A%
1070 GOTO 2020
1080 INPUT "ENTER THE INPUT FILENAME ",FILE1$
1090 B$ = "B:"
1100 FILE2$ = B$ + FILE1$
1110 OPEN FILE2$ FOR INPUT AS 1
1120 INPUT #1,NSAM
1130 INPUT #1,NSPS
1140 INPUT #1,NCHAN%
1221 INPUT #1,BV$
1230 FOR A% =  1 TO NCHAN%
1240 FOR G% = 0 TO 7
1250 CO(A%,G%) = 0
1260 NEXT G%
1270 INPUT #1,VAR$(A%)
1280 INPUT #1,RO%(A%)
1290 FOR F% = 0 TO RO%(A%)
1300 INPUT #1,CO(A%,F%)
```

```
1310 NEXT F%
1320 NEXT A%
1330 CLOSE #1
2020 INPUT "ENTER THE RAW DATA INPUT FILE ",FR$
2025 FR$ = B$ + FR$
2030 OPEN FR$ FOR INPUT AS 1
2040 FOR A% = 1 TO NSAM*NCHAN%
2050 INPUT #1,RAW%(A%)
2060 NEXT A%
2070 FOR A% = 1 TO NSAM
2080 INPUT #1,SPD(A%)
2090 NEXT A%
2091 FOR A% = 1 TO NSAM
2092 INPUT #1,DAT(0,A%)
2093 NEXT A%
2100 CLOSE #1
2110 COUNT% = 1
2120 FOR C% = 1 TO NSAM
2130 FOR B% = 1 TO NCHAN%
2140 DAT(B%,C%) = 0
2150 FOR F% = 0 TO RO%(B%)
2160 IF F%>0 THEN 2190
2170 DAT(B%,C%) = CO(B%,F%)
2180 GOTO 2240
2190 IF B% = 1 THEN 2220 ELSE 2200
2200 DAT(B%,C%) = DAT(B%,C%) + (RAW%(COUNT%)^F%)*CO(B%,F%)
2210 GOTO 2240
2220 TNU = (((RAW%(COUNT%)/409.6)-5)/9.403273669#)
2230 DAT(B%,C%)  = 57.29577951#*ATN(TNU/SQR(1-TNU^2))
2240 NEXT F%
2250 IF C% > 1 THEN 2290
2260 MINY(B%) = INT(DAT(B%,C%)-1)
2270 MAXY(B%) = INT(DAT(B%,C%)+1)
2280 GOTO 2310
2290 IF DAT(B%,C%)>MAXY(B%) THEN MAXY(B%)=INT(DAT(B%,C%)+1)
2300 IF DAT(B%,C%)<MINY(B%) THEN MINY(B%)=INT(DAT(B%,C%)-1)
2310 COUNT% = COUNT% +1
2320 NEXT B%
2330 NEXT C%
2340 FOR A% = 1 TO NSAM
2350 BBB% = NCHAN% + 1
2360 DAT(BBB%,A%) = SPD(A%)*60/ (2048*SR)
2370 NEXT A%
2380 B% = NCHAN% + 1
2390 FOR C% = 1 TO NSAM
2400 IF C% > 1 THEN 2440
2410 MINY(B%) = INT(DAT(B%,C%)-1)
2420 MAXY(B%) = INT(DAT(B%,C%)+1)
2430 GOTO 2460
2240 IF DAT(B%,C%)>MAXY(B%) THEN MAXY(B%)=INT(DAT(B%,C%)+1)
```

```
2450 IF DAT(B%,C%)<MINY(B%) THEN MINY(B%)=INT(DAT(B%,C%)-1)
2460 NEXT C%
2470 INPUT "ENTER THE OUTPUT FILE NAME ",FILE1$
2480 B$ = "B:"
2490 FILE3$ = B$ + FILE1$
2500 OPEN FILE3$ FOR OUTPUT AS 1
2510 FOR A% = 1 TO NSAM
2520 PRINT #1,DAT(0,A%);" ";DAT(1,A%);" ";DAT(2,A%);" "
     ;DAT(3,A%);" ";DAT(4,A%);" ";DAT(5,A%);" ";DAT(6,A%)
     ;" ";DAT(7,A%)
2530 NEXT A%
2540 CLOSE #1
2970 INPUT "ENTER THE .DIF OUTPUT FILE NAME ",FILE1$
2980 B$ = "B:"
3000 DD$ = ".DIF"
3010 FILE2$ = B$ + FILE1$ + DD$
3020 OPEN FILE2$ FOR OUTPUT AS 1
3030 NR% = NSAM + 1
3040 NC% = NCHAN% + 2
3050 C$ = STR$(NC%)
3060 R$ = STR$(NR%)
3070 O$ = "0,"
3080 PRINT #1,"TABLE"
3090 PRINT #1,"0,1"
3100 PRINT #1,CHR$(34);CHR$(34)
3110 PRINT #1,"VECTORS"
3120 NPC$ = O$ + C$
3130 PRINT #1,NPC$
3140 PRINT #1,CHR$(34);CHR$(34)
3150 PRINT #1,"TUPLES"
3160 NPR$ = O$ + R$
3170 PRINT #1,NPR$
3180 PRINT #1,CHR$(34);CHR$(34)
3190 PRINT #1,"DATA"
3200 PRINT #1,"0,0"
3210 PRINT #1,CHR$(34);CHR$(34)
3220 PRINT #1,"-1,0"
3230 PRINT #1,"BOT"
3240 FOR F% = 1 TO NC%
3250 PRINT #1,"1,0"
3260 IF F% = 1 THEN 3290
3270 PRINT #1,CHR$(34);VAR$(F%-1);CHR$(34)
3280 GOTO 3300
3290 PRINT #1,CHR$(34);"TIME";CHR$(34)
3300 NEXT F%
3310 PRINT #1,"-1,0"
3320 PRINT #1,"BOT"
3330 FOR E% = 1 TO NR%-1
3340 FOR F% = 1 TO NC%
3350 PRINT #1,"0,";
```

```
3360 PRINT #1,USING "##.##############^^^^";DAT(F%-1,E%)
3370 PRINT #1,"V"
3380 NEXT F%
3390 PRINT #1,"-1,0"
3400 IF E% = NR%-1 THEN 3420
3410 PRINT #1,"BOT" : GOTO 3430
3420 PRINT #1,"EOD"
3430 NEXT E%
3440 PRINT #1," "
3450 CLOSE #1
3460 CLS:SCREEN 2
3470 INPUT "ENTER THE CHANNEL TO PLOT ",PCH%
3480 NP = NSAM
3490 XMIN = 0
3500 XMAX = DAT(0,NSAM)
3510 YMIN = MINY(PCH%)
3520 YMAX = MAXY(PCH%)
3530 XTITLE$ = "TIME, SECONDS"
3540 YTITLE$ = VAR$(PCH%)
3550 INPUT "ENTER THE GRAPH TITLE ",TITLE$
3560 FOR D% = 1 TO NSAM
3570 PT(2,D%) = DAT(PCH%,D%)
3580 PT(1,D%) = DAT(0,D%)
3590 NEXT D%
3600 GOSUB 3700
3610 KK$ = INKEY$
3620 IF (KK$ = "L") THEN 3640 ELSE 3630
3630 IF (KK$ = "C") THEN 3460 ELSE 3610
3640 CLS
3650 FOR D% = 1 TO NSAM
3660 PRINT PT(1,D%) TAB(20) PT(2,D%)
3670 NEXT D%
3680 GOTO 3610
3690 END
3700 'NP = NUMBER OF POINTS
3710 'TITLE$ XTITLE$ YTITLE$
3720 'XMIN XMAX YMIN YMAX
3730 'X=PT(1,A) Y=PT(2,A)
3740 KEY OFF:SCREEN 1:CLS
3750 PSET(30,155)
3760 LINE-STEP(260,0):LINE-STEP(0,-145):LINE-STEP(-260,0)
     :LINE-STEP(0,145)
3770 LOCATE 1,20-LEN(TITLE$)/2
3780 PRINT TITLE$
3790 LOCATE 22,20-LEN(XTITLE$)/2 :PRINT XTITLE$
3800 FOR A =0 TO LEN(YTITLE$)
3810 LOCATE (11-(LEN(YTITLE$)/2)+A),1
3820 PRINT MID$(YTITLE$,(A+1),1)
3830 NEXT A
3840 A$ = STR$(XMIN):LOCATE 21,4 :PRINT A$
```

```
3850 A$ = STR$(XMAX):LOCATE 21,38-LEN(A$):PRINT A$
3860 A$ = STR$(YMIN):LOCATE 20,1 :PRINT A$
3870 A$ = STR$(YMAX):LOCATE 2,1  :PRINT A$
3880 FOR A = 0 TO 10
3890 PSET(30+A*26,155)
3900 LINE-STEP(0,3)
3910 NEXT A
3920 FOR A= 0 TO 10
3930 PSET (30,10+A*14.5)
3940 LINE-STEP(-3,0)
3950 NEXT A
3960 XP% = INT(30+260*((PT(1,1)-XMIN)/(XMAX-XMIN)))
3970 YP% = INT(155-145*((PT(2,1)-YMIN)/(YMAX-YMIN)))
3980 PSET(XP%,YP%)
3990 FOR A = 2 TO NP
4000 XP% = INT(30+260*((PT(1,A)-XMIN)/(XMAX-XMIN)))
4010 YP% = INT(155-145*((PT(2,A)-YMIN)/(YMAX-YMIN)))
4020 LINE -(XP%,YP%)
4030 NEXT A
4040 RETURN
```

APPENDIX D

```
100 '******************** HEADING ************************
110 REM $LIST+
120 ''
130 ''
140 '' define constants
150 ''
160 ''
170 DEFINT A-Z
171 OSS = 0
174 EFLAG=0
180 BASE.ADDRESS     = &H2EC
190 COMMAND.REGISTER = BASE.ADDRESS + 1
200 STATUS.REGISTER  = BASE.ADDRESS + 1
210 DATA.REGISTER    = BASE.ADDRESS
220 COMMAND.WAIT     = &H4
230 WRITE.WAIT       = &H2
240 READ.WAIT        = &H5
250 ''
260 CCLEAR  = &H1
262 CDAOUT  = &H8
270 CADIN   = &HC
280 CSTOP   = &HF
290 CSOUT         =&H5
300 CDIOOUT =&H7
310  ''
312 TOP.RANGE# = 10
314 BOTTOM.RANGE# = -10
320 ADGAIN    = 0
330 BASE.FACTOR#=4096
350 ''
360 '  stop and clear DT2801 board
370 ''
390 OUT COMMAND.REGISTER, CSTOP
400 TEMP = INP(DATA.REGISTER)
410 WAIT STATUS.REGISTER, COMMAND.WAIT
420 OUT  COMMAND.REGISTER, CCLEAR
430 ''
440 ' wait until DT2801 is ready
450 WAIT STATUS.REGISTER, COMMAND.WAIT
460 OUT COMMAND.REGISTER, CSOUT
470 WAIT STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
475 DIOPORT = 1
480 OUT DATA.REGISTER, DIOPORT
```

```
1000 ' ****************   MAIN PROGRAM   *****************
1005 ' *                                                 *
1010 ' ********   SUBROUTINES WHICH CAN BE CALLED ********
1020 ' *                                                 *
1025 ' * SUBROUTINE     LINE  INPUTS            OUTPUTS   *
1026 ' * ----------     ----  ------            -------   *
1030 ' * ANALOG INPUT   5000  CHANNEL           VOLTS#    *
1040 ' * BINARY OUTPUT  6000  DIOPORT,DATA.VALUE          *
1050 ' * ANALOG OUTPUT  7000  CHANNEL,VOLTS#              *
1060 ' *                                                 *
1070 ' ***************************************************
1075 '
1080 '  EXAMPLES
1085 '
1086 ' CHANNEL = 7 : GOSUB 5000 :INVOLTS# = VOLTS# :
     PRINT INVOLTS#
1087 ' DIOPORT=1 : DATA.VALUE = 2: GOSUB 6000
1088 ' CHANNEL = 1 : VOLTS# = -5.1 :GOSUB 7000
1090 '
2000 '' get a/d and d/a channels
2001 DRCHANNEL = 7
2002 PRCHANNEL = 3
2003 PMCHANNEL = 1
2004 VMCHANNEL = 1
2005 EPCHANNEL = 0
2006 SPCHANNEL = 5
2007 DIOPORT   = 1
2010 INPUT "ENTER THE DRIVER CHANNEL (0-7) ",DRCHANNEL
2020 INPUT "ENTER THE PRESSURE CHANNEL (0-7) ",PRCHANNEL
2030 INPUT "ENTER THE PUMP/MOTOR DISPLACEMENT CHANNEL
     (0-7)",  PMCHANNEL
2040 INPUT "ENTER THE VOLVO MOTOR CHANNEL ";VMCHANNEL
2050 INPUT "ENTER THE REXROTH ENGINE-PUMP CHANNEL ";
     EPCHANNEL
2060 INPUT "ENTER THE FLYWHEEL SPEED CHANNEL ";SPCHANNEL
2070 INPUT"DIGITAL PORT 0,1";DIOPORT
2080 INPUT "POSITION ACCURACY (DEAD BAND/2), VOLTS";ACC#
2090 INPUT "ENTER THE PRECHARGE PRESSURE ",PCPR#
2100 INPUT "ENTER THE LOW ACCUMULATOR PRESSURE ";LP#
2120 EFLAG = 0
3000 A$ = ""
3010 STP = 0
3020 CHANNEL = EPCHANNEL
3030 VOLTS# = 0
3040 GOSUB 7000
3041 CHANNEL = VMCHANNEL
3042 VOLTS# = 0
3043 GOSUB 7000
3050 WHILE A$=""
3051  ' IF OSS = 1 THEN 3052 ELSE 3053
```

```
3052  ' OSS = 0 : VOLTS# = 0 : CHANNEL = 0 :GOSUB 7000 :
      GOTO 3060
3053  ' OSS = 1 :VOLTS# =  2 : CHANNEL = 0 :GOSUB 7000
3060  CHANNEL = DRCHANNEL : GOSUB 5000 : DRVOLTS# = VOLTS#
3090  CHANNEL = PRCHANNEL : GOSUB 5000 : PRVOLTS# = VOLTS#
3120  PRESSURE# = PRVOLTS# * 666.67
3122  IF PRESSURE# > 10 THEN 3130 ELSE 3124
3124  OVOLTS# = 0 ; GOTO 3140
3130  OVOLTS# = PCPR#*DRVOLTS#/PRESSURE#
3140  IF OVOLTS# > 3.8 THEN OVOLTS# = 3.8
3150  IF OVOLTS# <-3.8 THEN OVOLTS# =-3.8
3160  CHANNEL = PMCHANNEL : GOSUB 5000 : PMVOLTS# = VOLTS#
3190  ER# = OVOLTS# - PMVOLTS#
3200  IF ABS(ER#) > ACC# THEN 3220
3210  DATA.VALUE = 0 : GOTO 3230
3220  IF ER#<0 THEN DATA.VALUE=1 ELSE DATA.VALUE=2 :
      GOSUB 6000
3231  IF STP = 1 THEN 3232 ELSE 3240
3232  CHANNEL = EPCHANNEL : VOLTS# = 0 : GOSUB 7000
3233  CHANNEL = VMCHANNEL : VOLTS# = 0 : GOSUB 7000 :
      GOTO 3350
3240  IF (EFLAG = 0 AND PRESSURE# > LP#) THEN 3350
3242  IF (PRESSURE# > LP# OR EFLAG = 1) THEN 3294
3250  EFLAG = 1
3251  CHANNEL = VMCHANNEL : VOLTS# = 2.55 : GOSUB 7000
3260  CHANNEL = EPCHANNEL
3270  VOLTS# =(PRESSURE#* (PRESSURE#* (PRESSURE# *
      -1.957699E-10 + 1.558302E-06) - 4.523E-03)
      + 7.101959
3280  GOSUB 7000
3290  GOTO 3350
3294  CHANNEL = EPCHANNEL
3295  VOLTS# =(PRESSURE#* (PRESSURE#* (PRESSURE# *
      -1.957699E-10 + 1.558302E-06) - 4.523E-03)
      + 7.101959
3296  GOSUB 7000
3300  CHANNEL = SPCHANNEL : GOSUB 5000
3302  IF VOLTS#>2 THEN HP#=3515 - 342.5*VOLTS#
      ELSE HP#=3000 - 116.65*VOLTS#
3303  IF HP# > 3000 THEN HP# = 3000
3309  IF(PRESSURE# > HP# AND EFLAG = 1 AND
      PRESSURE# >LP#) THEN 3310  ELSE 3350
3310  CHANNEL = EPCHANNEL : VOLTS# = 0 : GOSUB 7000
3331  CHANNEL = VMCHANNEL : VOLTS# = 0 : GOSUB 7000
3340  EFLAG = 0
3350  IF STP<>0 THEN 3401
3360  A$=INKEY$
3370  IF A$="" THEN GOTO 3420 ELSE GOTO 3380
3380  STP = 1
3390  PCPR# = 0
```

```
3400  PRINT "KEYBOARD SET DISPLACEMENT TO 0 " : GOTO 3410
3401  A$ = INKEY$
3403  IF A$ = "R" THEN END
3410  A$ = ""
3420 WEND
3430 END
5000 ' ************* ANALOG INPUT SUBROUTINE ***********
5010 WAIT STATUS.REGISTER, COMMAND.WAIT
5020 OUT  COMMAND.REGISTER, CADIN
5030 ''
5040 '' wait until the DT2801 DATA IN FULL flag is clear
5050 '' ,then write a/d gain to the data in register
5060 ''
5070 WAIT STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
5080 OUT  DATA.REGISTER, ADGAIN
5090 '' wait until the DT2801 board DATA IN FULL flag is
        clear,
5100 '' then write the a/d channel to data in register
5110 ''
5120 WAIT STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
5130 OUT  DATA.REGISTER, CHANNEL
5140 ''
5150 '' read two bytes from data out register after
     DATA OUT ready
5160 ''
5170 WAIT  STATUS.REGISTER, READ.WAIT
5180 LOW = INP(DATA.REGISTER)
5190 WAIT  STATUS.REGISTER, READ.WAIT
5200 HIGH = INP(DATA.REGISTER)
5210 RAWVALUE#=HIGH*256+LOW
5220 VOLTS#=RAWVALUE#*(20/BASE.FACTOR#)-10
5230 RETURN
6000 ' ************** BINARY OUTPUT ********************
6010 '' wait until DT2801  ready, then write the WRITE
6020 ''  DAC IMMEDIATE command to command register
6030 WAIT  STATUS.REGISTER, COMMAND.WAIT
6040 OUT  COMMAND.REGISTER, CDIOOUT
6050 ''
6060 '' wait until DATA IN FULL flag is clear, then
6070 '' write dachannel to DATA IN REGISTER
6080 WAIT  STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
6090 OUT DATA.REGISTER, DIOPORT
6100 '' write low and high to DATA IN REGISTER after a
6110 '' clear of DATA IN FULL flag
6120 ''
6130 WAIT STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
6140 OUT DATA.REGISTER, DATA.VALUE
6150 RETURN
```

```
7000 '******** ANALOG OUTPUT SUBROUTINE ***********
7010 'INPUTS CHANNEL, VOLTS#, TOP.RANGE#, BOTTOM.RANGE#
7020 IF VOLTS# > TOP.RANGE# THEN VOLTS# = TOP.RANGE#
7030 IF VOLTS# < BOTTOM.RANGE# THEN VOLTS# = BOTTOM.RANGE#
7040 RANGE# = TOP.RANGE# - BOTTOM.RANGE#
7050 DATA.VALUE# = (VOLTS# - BOTTOM.RANGE#) * 4096/RANGE#
7060 DATA.VALUE# = CINT(DATA.VALUE#)
7070 IF DATA.VALUE# > 4095 THEN DATA.VALUE# = 4095
7080 ''
7090 '' wait until the DT2801 series board READY flag set,
7100 '' then write the WRITE DAC IMMEDIATE command
7110 '' byte to Command Register.
7120 WAIT STATUS.REGISTER, COMMAND.WAIT
7130 OUT  COMMAND.REGISTER, CDAOUT
7140 ''
7150 ''  Wait until the DT2801 series board DATA IN FULL
7160 ''  flag is clear, then write the DAC SELECT byte
7170 ''  to the Data In Register.
7180 WAIT STATUS.REGISTER, WRITE.WAIT, WRITE.WAIT
7190 OUT  DATA.REGISTER, CHANNEL  ' channel 0 or 1
7200 ''
7210 '' Divide DATA.VALUE# into high and low bytes and
7220 '' write both bytes to the Data In Register, waiting
7230 '' for a clear DATA IN FULL flag before each write.
7240 ''
7250 HIGH = INT(DATA.VALUE#/256)
7260 LOW = DATA.VALUE# - HIGH*256
7270 WAIT STATUS.REGISTER,  WRITE.WAIT,  WRITE.WAIT
7280 OUT DATA.REGISTER,  LOW
7290 WAIT STATUS.REGISTER,  WRITE.WAIT,  WRITE.WAIT
7300 OUT  DATA.REGISTER,  HIGH
7310 ''
7320 '' Wait until the DT2801 series board READY flag
7330 '' sets, indicating command completion, then check
7340 '' the Status Register ERROR flag.
7350 WAIT STATUS.REGISTER,  COMMAND.WAIT
7360 STATUS = INP(STATUS.REGISTER)
7370 IF (STATUS AND &H80) THEN GOTO 7390
7380 GOTO 7420
7390 ''
7400 '' ERROR
7410 ''
7420 RETURN
```

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DATE

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