DOE/NASA/0084-79/1
NASA CR-159746
WMB&A 131-4-R2

AN AUTOMATICALLY-SHIFTED TWO-SPEED TRANSAXLE SYSTEM FOR AN ELECTRIC VEHICLE

Hayden S. Gordon & Gregory V. Hassman
William M. Brobeck & Associates

January 1980

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN 3-84

for
U.S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
Office of Transportation Programs

NOTICE

DOE/NASA/0084-79/1
NASA CR-159746
WMB&A 131-4-R2

AN AUTOMATICALLY-SHIFTED TWO-SPEED TRANSAXLE SYSTEM FOR AN ELECTRIC VEHICLE

Hayden S. Gordon & Gregory V. Hassman
William M. Brobeck & Associates
1235 Tenth Street
Berkeley, California 94710

January 1980

Prepared for
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135
Under Contract DEN 3-84

for
U. S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
Office of Transportation Programs
Washington, D. C. 20545
Under Interagency Agreement EC-77-A-31-1044

TABLE OF CONTENTS

Title	Page
Summary	1
Introduction	3
Transaxle Description	5
Automatic Transaxle Shift Control System	9
Conclusions	17

APPENDICES

Appendix A - Vehicle and Motor Characteristics	19
Appendix B - Transaxle Operating Limitations	23

LIST OF FIGURES

Figure No.	Title	Page
1	Gear Train Schematic	6
2	Transaxle Envelope Dimensions	7
3	Controller/Shift Logic Schematic	10
4	Shift Logic Schematic	11
5	Shift Actuator—Linear	15
6	Shift Actuator—Rotary	16
7	Traction-Motor Speed/Torque Characteristics	21
8	Driving Cycle Torque Requirements and Transaxle Limitations	26
9	1814 N·m Torque Limiter	28
10	1689 N·m Torque Limiter	29

LIST OF TABLES

Table No.	Title	Page
1	Transaxle Shift Control Components	12
2	Vehicle Specifications & Requirements	19
3	Traction Motor Specifications	20
4	Breadboard Check-Out Operating Limits	24
5	Transaxle Motor Torque Limitations	25

SUMMARY

Under Department of Energy Interagency Agreement EC-77-A-31-1044, the NASA Lewis Research Center issued Contract DEN3-84 to William M. Brobeck and Associates for the design and fabrication of a state-of-the-art, ac vehicle propulsion system breadboard. A two-speed transaxle employing a Dana-Spicer IS-18 differential was built by McKee Engineering Corporation of Palatine, Illinois, for the breadboard. An overall reduction ratio of 9.667:1 or 19.333:1 is selected by the position of a shift rod linearly actuated by a shift mechanism.

In assessing the interrelationships of the ac motor-controller and the transaxle, it was determined that an automatic shifting system would prove advantageous in a vehicle. These advantages are a probable overall efficiency gain due to optimal shifting in both the powered and regenerative modes, smoother operation over the operating range and the fact that nearly all control information is already available in the ac controller. For these reasons it was decided to pursue a preliminary design of an automatic shifting system consisting of a control logic unit and shift-rod actuators.

INTRODUCTION

Under an interagency agreement with the Department of Energy, the NASA Lewis Research Center (NASA-LeRC) was authorized to issue contracts for the development of propulsion system and component technology for electric and hybrid vehicles to aid in implementing the Electric Vehicle Program initiated in 1976. In December, 1976, two contracts were issued to analyze and identify the best propulsion system that could be assembled from state-of-the-art components. One of these contracts resulted in a preliminary design of an ac propulsion system. The design is contained in Section 6 of the report, "Preliminary Power Train Design for a State-of-the-Art Electric Vehicle," Report No. DOE/NASA/0592-78/1.

In May, 1978, a Request for Proposal was issued to design, build and check-out test a state-of-the-art ac vehicle propulsion system resulting from the preliminary design. In December, 1978, a contract to undertake this work was awarded William M. Brobeck & Associates. The propulsion system design consists of a General Electric 29.8 kW (40 hp) ac motor flange-mounted to the integral two-speed transmission-differential housing. The ac motor was to be controlled by a Rohr Industries, Inc. three-phase, variable-voltage, variable-frequency converter. The system was to consist of these components provided with sufficient instrumentation to measure the independent component efficiencies. Inasmuch as a preliminary design existed for such a system, the implementation of the concept was expected to consist primarily of procurement of specified components and their incorporation in a static assembly--a breadboard--to be used at the NASA Lewis Research Center. It would serve as a benchmark system against which to measure future propulsion system developments. The vehicle and motor characteristics upon which the breadboard design was based are listed in Appendix A.

Coordination by NASA-LeRC of the various program elements dictated a fast-paced schedule for the breadboard project. Long-lead items required advance-procurement authorization before final design approval. It became clear that procurement of a critical component of the system--the ac-motor controller--required further development to reach even the prototype level with resulting unpredictable costs and schedule. After significant effort to establish an acceptable controller specification, the controller procurement fell behind the overall program schedule. Furthermore, other controller development contracts were underway as a part of the overall Electric and Hybrid Vehicle Program. Consequently, the major part of the work on the state-of-the-art project was terminated.

Design and fabrication of the two-speed transaxle was underway at the time of contract termination as a result of approval of advance procurement.

This report was authorized to make the information concerning an automatic shift concept available for use with the transaxle in other parts of the overall program. The automatic shifting system is projected to be as smooth as that experienced with the conventional torque converter automatic transmission while retaining the high efficiency of the manually-shifted transmission.

This report also includes the operating requirements assumed for the transaxle system, the results of stress calculations and the recommendations for lubrication of the transaxle system.

TRANSAXLE DESCRIPTION

The transaxle is a custom-fabricated unit comprising a modified Spicer differential, Model IS-18, on which is mounted a two-speed, synchromesh transmission built by McKee Engineering Corporation. All bearings are rolling-contact ball type with the exception of those for the differential bevel gears. All power train gears are helical-type including the differential ring and pinion gears. The latter change permitted replacement of the tapered roller bearing carrying the differential assembly with lower-loss ball bearings. Gears run continuously in mesh and speed selection is performed through synchromesh clutches. Figure 1 is a "roll out" which shows the transaxle elements as they would be displayed on a cutting plane passing through successive shaft centerlines. Figure 2 shows the external configuration of the transaxle.

The input shaft gears have 12 and 19 teeth meshing with countershaft gears of 48 and 38 teeth, respectively. The jackshaft output pinion of 18 teeth meshes with the differential ring gear of 87 teeth. These combinations give a high-gear ratio of 9.667 and a low-gear ratio of 19.333. The ratio is selected by the position of the shifting rod.

A question had arisen as to the torque handling capacity of the output splined shaft of the transaxle as designed when it was used with a locked-up differential as the sole output connection to the dynamometer. An inquiry made of Dana-Spicer elicited the information that the design endurance limit torque was 1193 N·m (880 lbs-ft) for each of the two shafts for the ASTM 1035 material used. The endurance limit for the other components of the differential was 1763 N·m (1300 lbs-ft). The maximum torque which could be exerted by the three-phase traction motor of 89.5 N·m (66 lbs-ft) corresponded to 89.5 x 19.333 = 1729 N·m (1276 lbs-ft) for the single output shaft. The McKee transaxle was, therefore, equipped with output spline shafts made of heat-treated and shot-peened ASTM 4340 alloy. The endurance limit of these shafts has been computed to be 2983 N·m (2200 lbs-ft). An analysis of the transaxle operating limitations when used with a locked-up differential is presented in Appendix B.

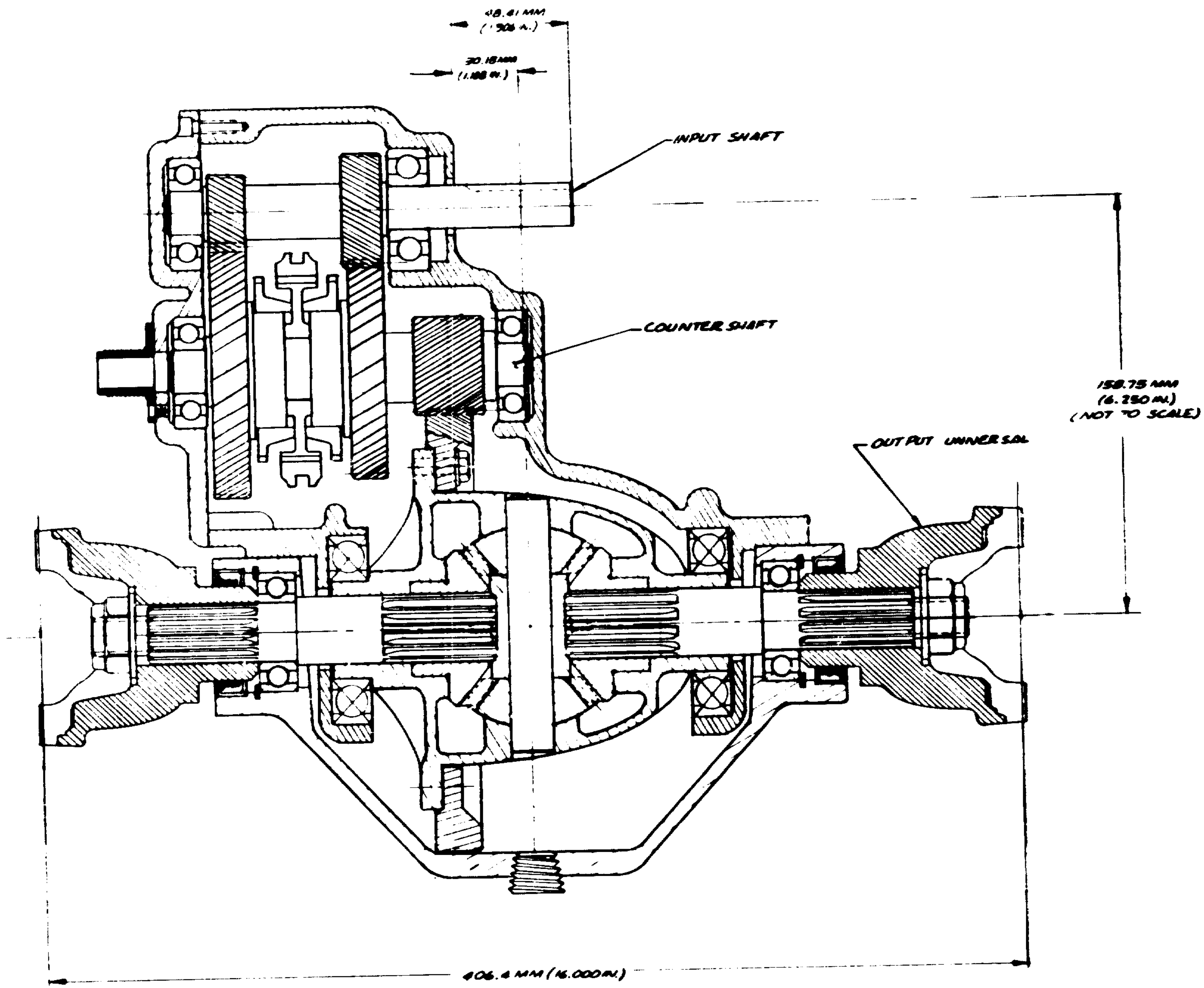


Figure 1. Gear Train Schematic

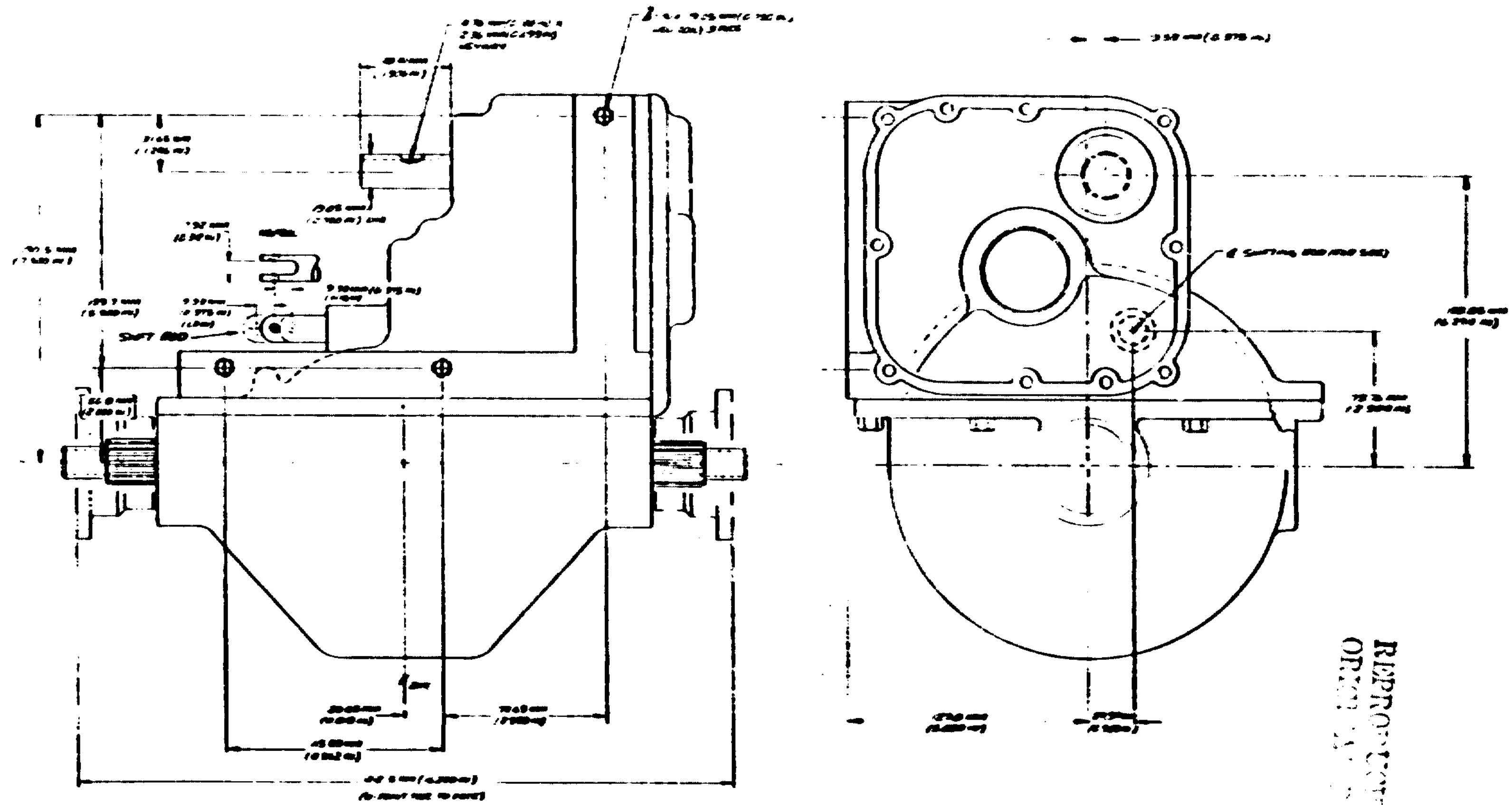


Figure 2. Transaxle Envelope Dimensions

AUTOMATIC TRANSAXLE SHIFT CONTROL SYSTEM

Automatic shifting of the transaxle of an electric vehicle may be more easily accomplished than that for an internal combustion powered vehicle because precise control of the torque and speed of the electric motor is inherently simpler. The motor torque can be easily reduced to a zero value for gear disengagement and the motor speed can be rapidly brought to its ideal synchronizing value for reengagement of the gears. The manner of speed and torque control varies with electric motor types. For the ac motor system studied in this contract, the motor speed and torque are controlled by the output frequency of the dc-ac controller making it fairly simple to create the ideal conditions for automatic shifting. For dc motors, the speed and torque are controlled by terminal voltage and field current.

For the ac system studied under this contract, changing the transaxle drive ratio while the vehicle is in motion is accomplished with the following steps:

First, the drive torque is reduced to zero by changing the controller output frequency to reduce the motor slip to zero. Second, the transmission is shifted to neutral disengaging the gears. Third, the motor speed is resynchronized for the new ratio by lowering the frequency for an upshift or raising it for a down shift. Fourth, the transmission is shifted to the new ratio. Fifth, a new value of slip is established to produce a rear-axle drive torque matching as closely as possible that which existed prior to initiation of the speed change ratio and, in the optimum case, the product of torque and speed, i.e. the power, would remain constant. The entire sequence can be executed in a fraction of a second, and will depend in part upon the rate of change of motor speed that can be forced. Smooth acceleration or deceleration will be interrupted only momentarily.

The shifting sequence is generated and controlled by the shifting logic. The shifting logic is interfaced between the controller and transaxle as shown in the block diagram Figure 3. A circuit schematic of the shifting logic is shown in Figure 4. The circuit description that follows and also the list of material, Table 1, both refer to the identification numbers that appear on the circuit schematic.

The instant of shifting is determined when a signal proportional to the difference of the vehicle velocity and a function of the developed torque reach the value set by the shift point adjust. In Figure 4, Shift Logic Schematic, amplifiers U1A and U1B modify the slip signal to develop a function of torque. Vehicle tachometer U12 produces a signal that represents the vehicle velocity. These signals are subtracted by U1C so its output represents a function of torque subtracted from the vehicle speed. This difference is then compared to the shift point voltage such that the output of U1D is either high or low corresponding to the required

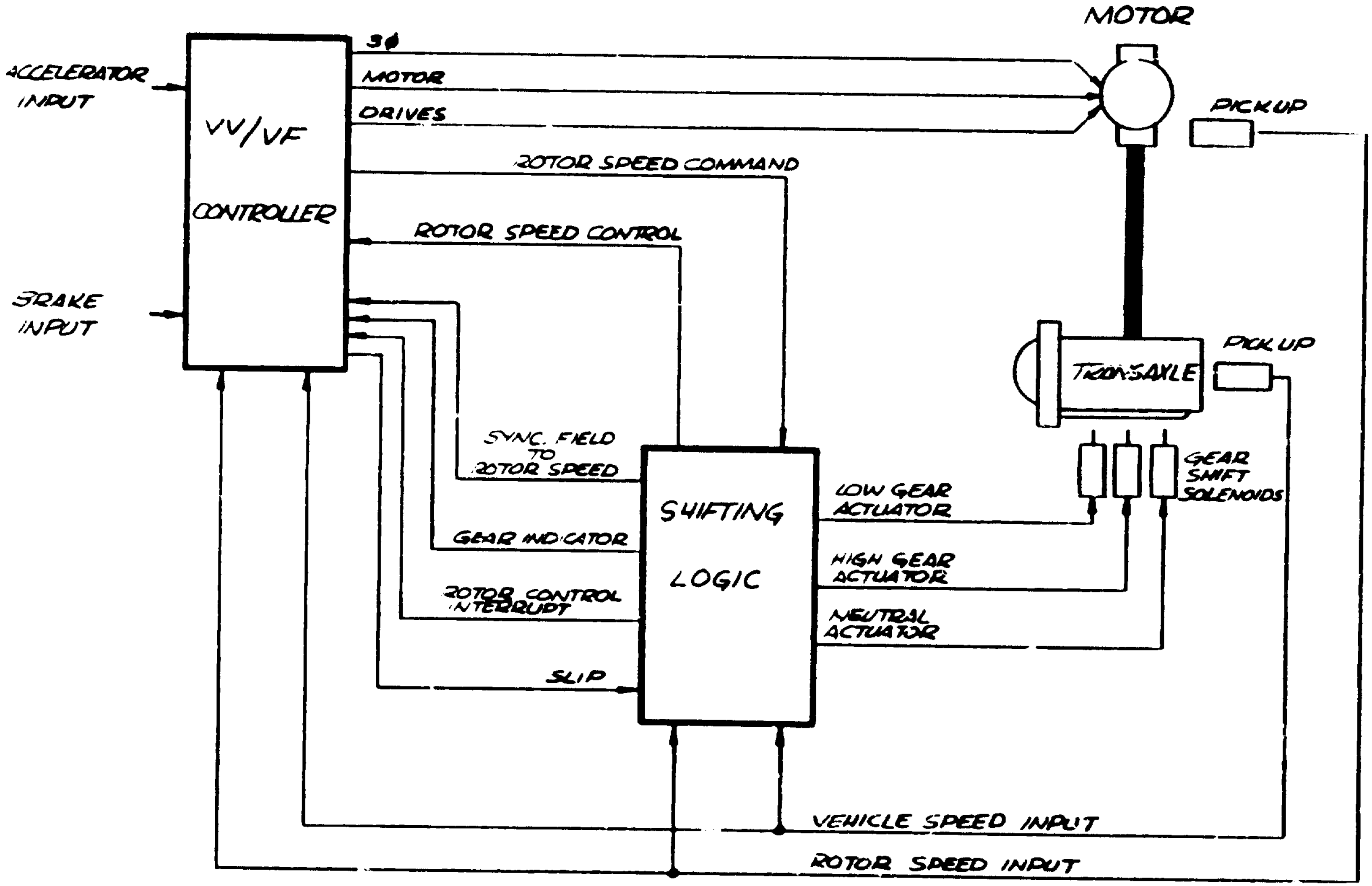


Figure 3. Controller/Shift Logic Schematic

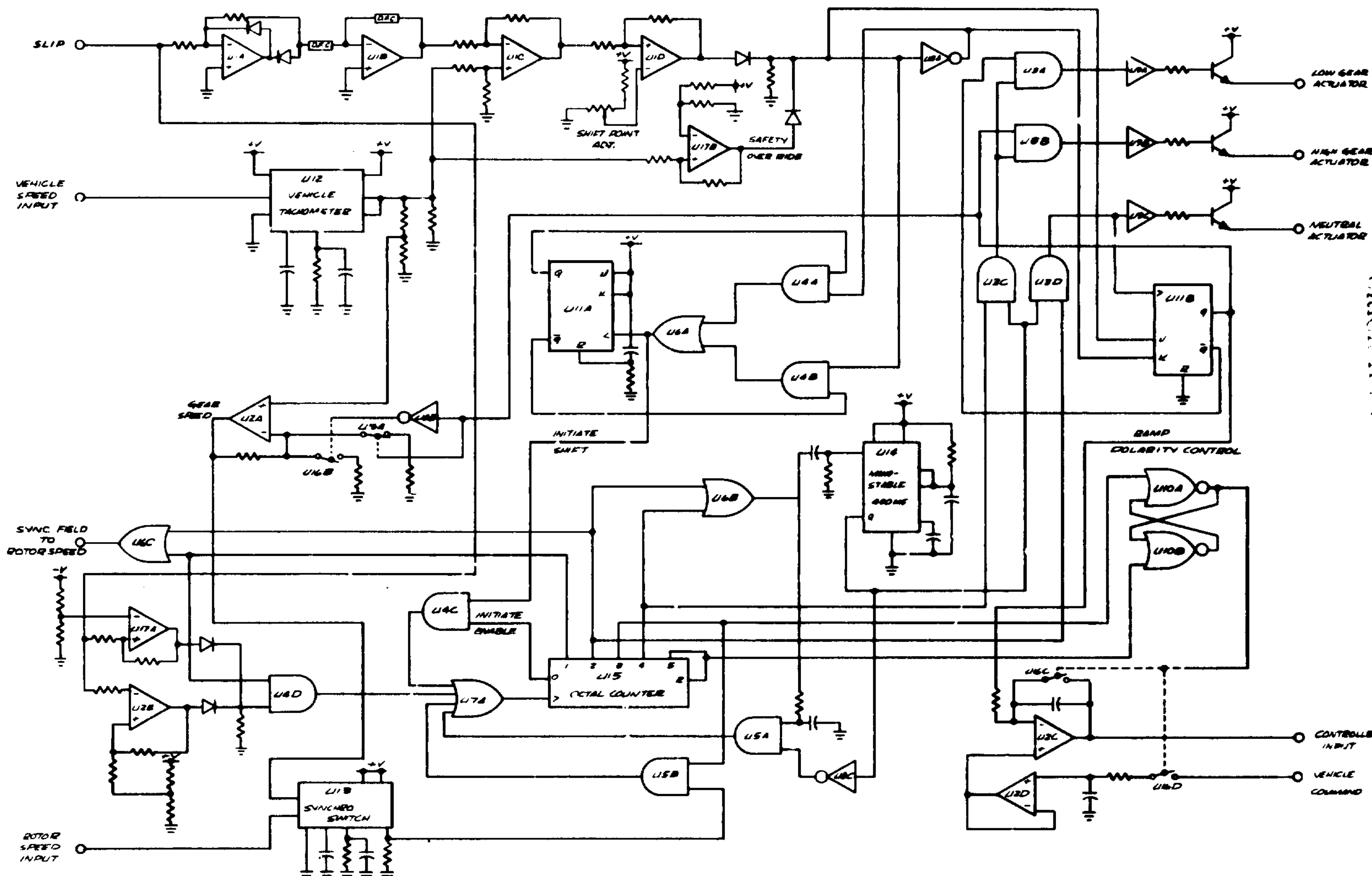


Figure 4. Shift Logic Schematic

Table 1. Transaxle Shift Control Components

Item No.	Symbol	Quantity	Part No.	Description
1	U1, U2, U17	3	LF347	Quad Bi-FET Operational Amplifiers
2	U3, U4, U5	3	CD4081B	C-MOS Quad Two Input AND Gate
3	U6	1	CD4071B	C-MOS Quad Two Input OR Gate
4	U7	1	CD4072B	C-MOS Dual Quad Input OR Gate
5	U8	1	CD4069B	C-MOS Hex Inverter
6	U9	1	CD4050B	C-MOS Hex Buffer
7	U10	1	CD4001B	C-MOS Dual Input NOR Gate
8	U11	1	CD4027B	C-MOS Dual Master-Slave Flip-Flop
9	U12, U13	2	LM2907	Frequency to Voltage Converter
10	U14	1	555	Timer
11	U15	1	CD4022B	C-MOS Octal Counter with Decoded Outputs
12	U16	1	CD4016B	Quad Bilateral Switch
13	R	42	N/A	1/4 Watt Resistors
14	Q_1, Q_2, Q_3	3	N/A	NPN Transistors
15	C	11	N/A	Capacitors
16	R	1	N/A	Trim Pot
16	CR	6	C-2	Signal Diodes

state of the transaxle. This output initiates three functions. First, it determines the state of the gear indicator U11B by setting its inputs prior to its being clocked by the neutral pulse. The gear indicator enables either the high or low actuator (through U3A or U3B), determines the polarity of the integrator U2C, and sends a gear indicator signal to the main controller. This gear indicator signal allows the main controller to scale the slip command and thereby maintain an even torque at the wheels within the limitations of the power train. This will eliminate jerks while shifting. The shifting can be smoother than that experienced with current automatic transmissions. Second, through U2A, U8B, U16A, and U16B, it changes the scale factor of the gear speed signal. This scaling produces, from the vehicle speed tachometer, a signal that represents the speed of the driven gear. The gear speed signal is used by the synchro switch to allow synchronization of the drive and driven gears prior to engaging. Finally, any change in the comparator output is decoded through U4A, U4B, U6A, and U11A to generate an initiate-shift pulse. This pulse is passed through U4C only when the counter is in position zero. In this condition the pulse is passed through U7A to increment the counter. The output of the comparator U1D is logically combined with a safety override U17B. This forces a shift when the rotor reaches a preset speed and prevents a down shift if the low gear will cause excessive rotor speed. U9A, U9B, and U9C are current drivers and perform no logic function.

Once the shifting sequence has been initiated the counter is incremented through five positions that performs the shift and then resets the counter. Position one sends a signal to the controller, through U6C, to set the slip to zero. It also enables U4D whose output further increments the counter when the slip nears zero. This low slip condition is determined by the comparators (U2B and U17A). The next state of the counter, position two, maintains the slip at zero and also initiates a 400 msec neutral pulse from U14 to disengage the gears through the now enabled U3D. Besides disengaging the gears the leading edge of the neutral pulse sets the gear indicator (U11B) to the correct state. The trailing edge of the pulse, through U8C, U5A, and U7A, increments the counter to its next position. Position three sets the S-R flip-flop (U10A and U10B) which implements synchronization of the drive gear to the driven gear. Prior to setting the S-R flip-flop, U16C and U16D are in the closed state. With the switches closed U2C and U2D are voltage followers and therefore pass the rotor speed command, unaltered, back to the main controller to control the rotor speed. When the S-R flip-flop is set, a signal is sent to the main controller which interrupts the rotor control, the two switches are also opened. With the switches open U2D and its components hold the instantaneous value of the rotor speed command while U2C ramps this value either up or down as determined by the ramp polarity control (output of the gear indicator). The result is that the drive gear speed is either increased or reduced toward the value of the driven gear's speed. At the point at which the two speeds are equal, the synchro switch (U13) generates a pulse that increments the counter through the now enabled U5B. Position four again initiates a 400-msec actuate pulse. This pulse is directed to the first or second gear actuator through U3C and either U3A or U3B, whichever is selected by the gear indicator (U11B). After the gear is engaged, the trailing edge of the pulse increments the counter as previously described. Position five then resets the S-R flip-flop and the counter back to position zero. Since the shifting mechanism and the actuators are unknown, the engage/disengage time is also unknown. Therefore, the 400 msec was assumed due to the lack of this information. The actual time would be calculated as a portion of the final design and may be shorter than 400 msec depending upon the controller/motor characteristics.

The use of a two-speed transaxle in an ac motor/controller vehicle propulsion system requires a shift-actuating mechanism if automatic speed changes are to be made. Two electric actuator concepts are shown in Figure 5 and Figure 6. Either concept uses a pulse from the shift control to actuate the electromagnet performing the desired function: shift to neutral, up-shift from neutral or down-shift from neutral. Common to both concepts is the need to develop sufficient force to overcome the shift rod detent when the electromagnet is in the minimum force position. The detent force should be adjusted to the minimum value necessary to maintain clutch engagement in either drive position. Final engagement is achieved by the electromagnet with the armature in the position of maximum attractive force. It is assumed that the magnet pulse lasts 400 milliseconds. After actuation the detent must provide the force necessary to maintain the shift rod in the selected position.

Of the two concepts illustrated, the first, Figure 5, is the simplest because all motions are linear and traction-type solenoids are used. Because the traction force is a minimum at the extended position of the armature, the force to overcome the detent is a minimum. Since detent and gear engagement forces are not known at this time, solenoid selection, force and power requirements cannot be calculated.

The second concept, Figure 6, uses a "rotary" solenoid. Due to the fact that the tractive force developed by the magnet elements is translated into rotary motion by means of balls rolling on properly contoured inclined ramps, the torque developed by the solenoid can be made substantially constant and the torque forces can be applied to the linear-motion-shaft rod in a calculable fashion. Although the system is more complex, it may prove more reliable.

It is likely that reliable shifting will require more force than can be developed by direct application of solenoids and a power-assisted system may be needed. An electric-motor-driven screw-jack or a hydraulic-actuated three-positon linear actuator could be used. Since it is probable that hydraulic power steering and power brakes would be used on the vehicle, a source of hydraulic power would be available. Power relays for the electric linear actuator or solenoid valves for the hydraulic servo-system could be designed to accept the input signals from the shift-control logic.

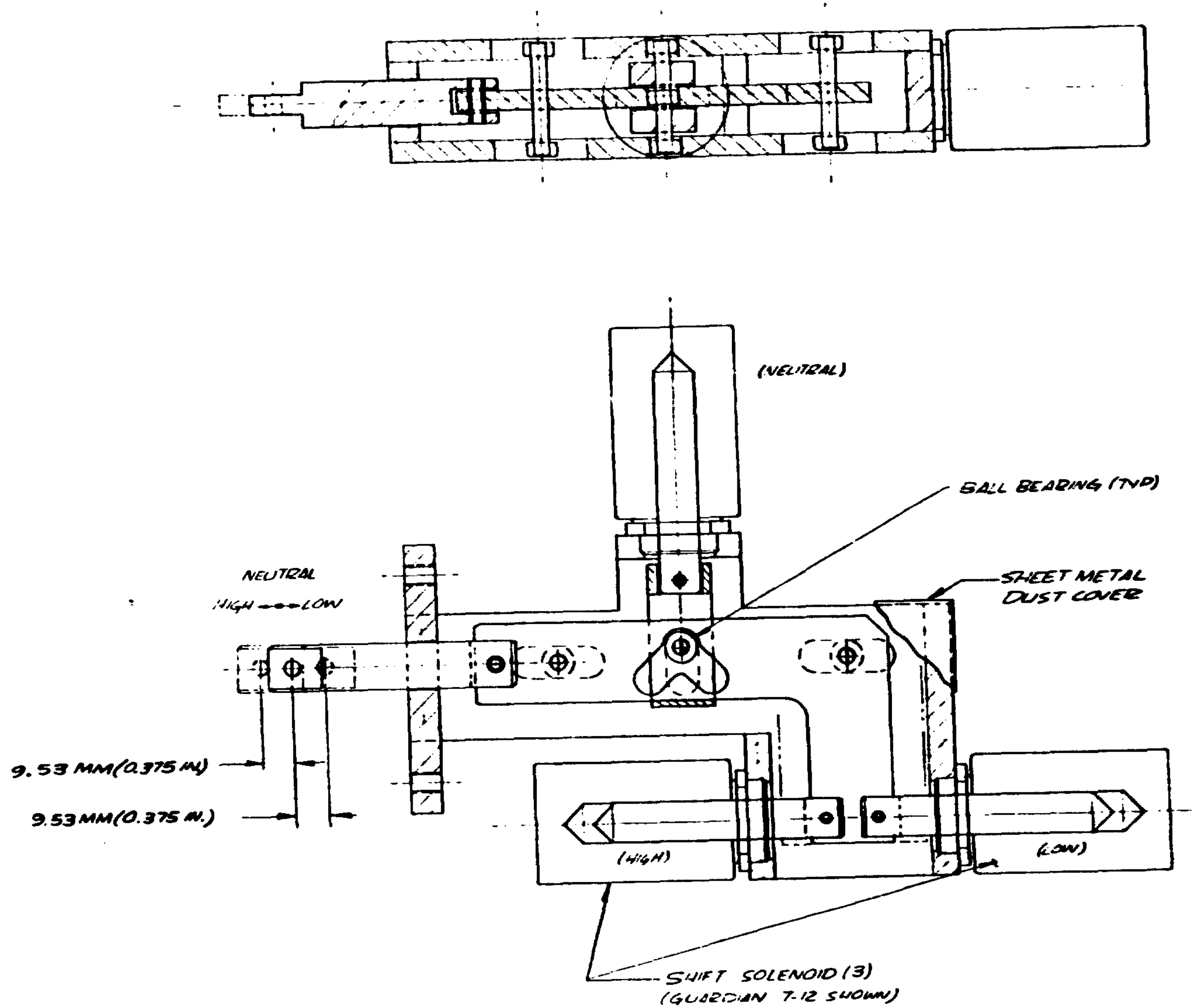


Figure 5. Shift Actuator--Linear

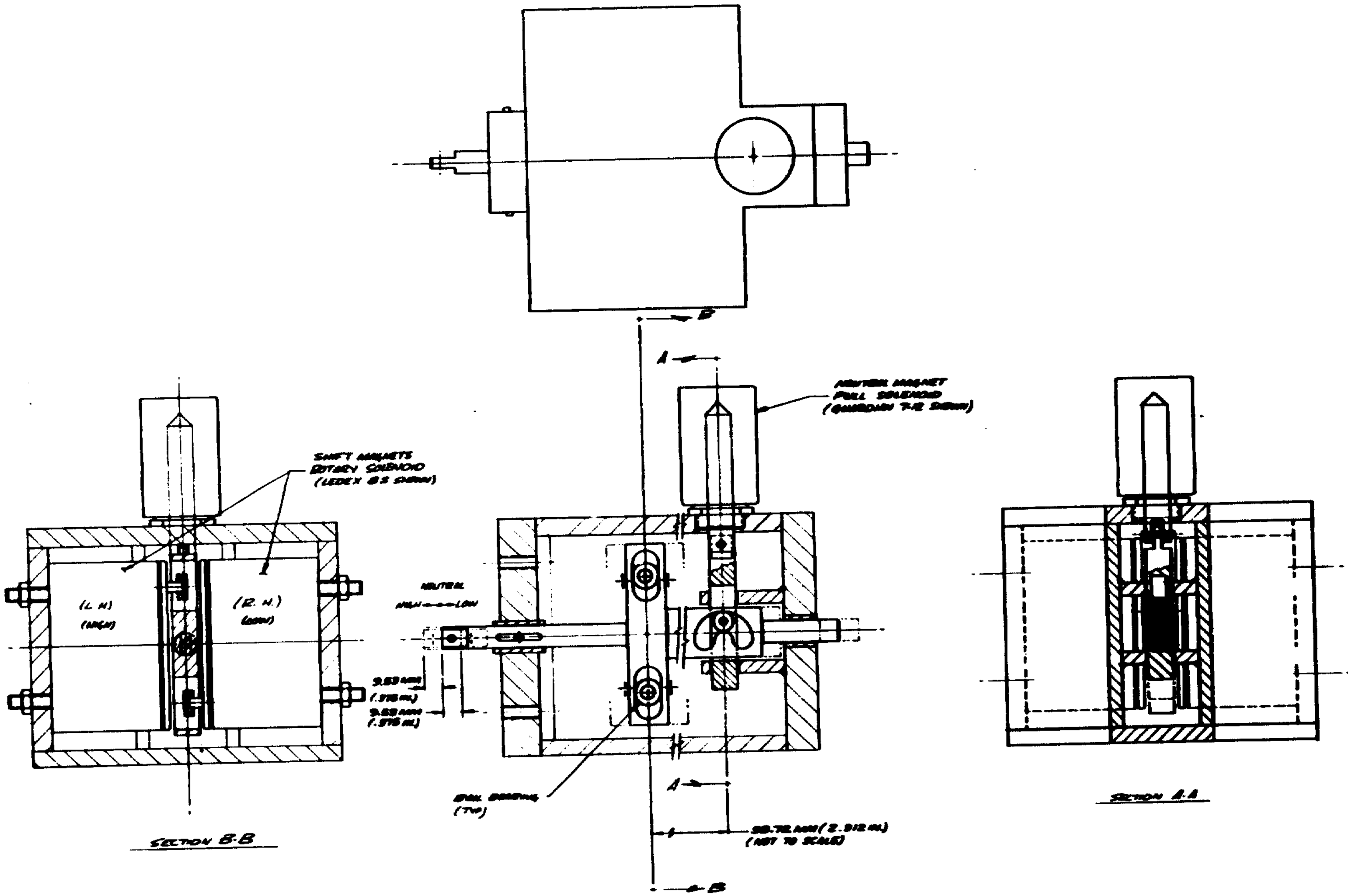


Figure 6. Shift Actuator--Rotary

CONCLUSIONS

A compact, light-weight, inexpensive and reliable shift-control logic system can be built for use in conjunction with a multi-speed transmission in electrically-powered vehicles. The logic system will permit vehicle performance optimization in terms of range, acceleration, cruising, hill climbing and dynamic braking requirements. Analog input parameters of vehicle speed and torque permit the logic system to be designed for either ac or dc motor drives.

APPENDIX A

VEHICLE AND MOTOR CHARACTERISTICS

The breadboard design was based upon an assumed vehicle having the specifications and requirements given in Table 2.

Table 2. Vehicle Specifications and Requirements

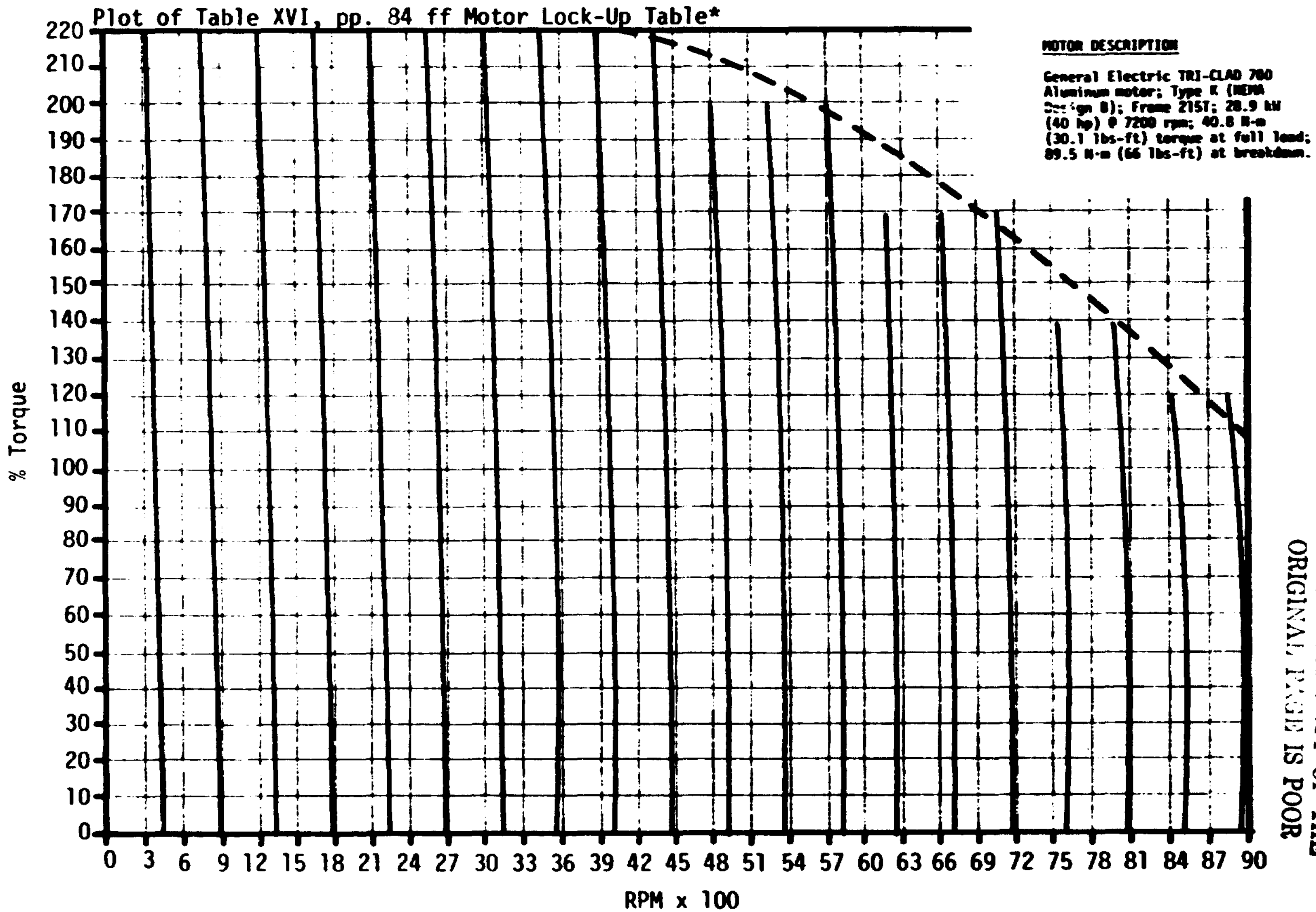
Gross vehicle weight	1456 kg
Aerodynamic drag coefficient, C_d	0.3
Frontal area	1.86 m^2
Rolling resistance	8.1 kg/1000 kg
Tire rolling radius	292 mm
Maximum braking rate from regenerative braking alone	3.05 m/sec^2
Maximum speed	96.5 km/hr
Minimum gradeability	10% @ 48.3 km/hr
Minimum acceleration in 15 seconds	0 to 72 km/hr

The corresponding traction motor specifications are given in Table 3.

Computed speed-torque data taken from Report No. DOE/NASA/0592-78/1 are plotted in Figure 7 to show the operating limits of the General Electric traction motor.

Table 3. Traction Motor Specifications

Manufacturer:	General Electric
Model Line:	TRI-CLAD 700 Aluminum Motor
Type:	K (NEMA Design B)
Frame:	215T
HP:	29.8 kW (40 hp) @ 7200 rpm
Torque:	40.8 Nm (30.1 ft lbs) at full load 89.5 Nm (66 ft lbs) at breakdown
Time Rating:	Continuous at rated full load torque
Synch. Speed:	7200 rpm
Volts:	69.2 (line to line)
Current:	364 amperes
Phases:	3
Frequency:	240 Hz
Enclosure:	Dripproof
Service Factor:	1.15
Ambient Temperature:	40° C
Mounting Position:	Horizontal
Bearings:	Ball
Special Provisions:	No internal fan (externally blower cooled) Dynamic balance to 9000 rpm



*Ross, James A. and Gerald A. Wooldrige, "Preliminary Power Train Design for a State-of-the-Art Electric Vehicle," NASA CR-135340

Figure 7. Traction-Motor Speed/Torque Characteristics

APPENDIX B

TRANSAXLE OPERATING LIMITATIONS

The external dimensions, input and output shaft and shift rod attachments have been shown in Figure 2. The operating limitations for the transaxle-motor-vehicle system specified in Appendix A are given in Table 4. Traction motor output and input power (propulsion and braking respectively) in excess of 28.9 kW at speeds above 7200 rpm represent short term conditions and may result in unacceptably high battery drain or charging current.

Table 4. Breadboard Check-Out Operating Limits

TRACTION MOTOR				TRANSAXLE OVERALL RATIO	SINGLE AXLE OUTPUT		VEHICLE SPEED	SYSTEM LIMIT
Operating Condition	Output (Input)							
	Speed rpm	Torque N·m	Power kW		Speed rpm	Torque N·m	km/hr	
Rated power @ max. speed	9000	31.6	29.8	9.667	931	305	102.5	Traction motor speed and rating
	9000	31.6	29.8	19.333	466	610	51.3	Traction motor speed and rating
Max. power @ max. speed	9000	45.6	43.0	9.667	931	441	102.5	Traction motor speed and torque
	9000	45.6	43.0	19.333	466	881	51.3	Traction motor speed and torque
Max. braking	9000	(30.0)	(28.3)	19.333	466	580	51.3	
	8915	(59.9)	(55.9)	9.667	922	579	101.5	
Rated output	72000	40.8	29.8	9.667	745	395	82.1	None
	7200	40.8	29.8	19.333	372	789	41.0	None
Max. power rated rpm	7200	66.6	50.2	9.667	745	644	82.1	Traction motor torque
	7200	66.6	50.2	19.333	372	1288	41.0	Traction motor torque
Max. torque	4200	89.5	39.4	9.667	434	865	47.8	Max. speed at 220% torque
	4200	81.6	35.9	19.333	217	1577	23.8	

Notes: 1. Transaxle and speed-increaser losses not included.

2. Traction motor speed-torque limits taken from Figure 7.

The transaxle gear train was analyzed by methods used for industrial gears which are rated for performance under continuous load. From the gear train analysis the capacity of the 18-tooth pinion was determined to be limiting. This gear is the most critical of the gears. This analysis implies that the pinion cannot carry the maximum motor torque in low gear. The allowable input torque is summarized in Table 5 which shows the maximum allowable input torque for both high and low gear.

Table 5. Transaxle Input Torque Limitations

Maximum Continuous Input Torque
Above these values fatigue life is reduced

Low Gear: 26 N•m

High Gear: 52 N•m

Maximum Peak Input Torque
Above these values gear damage will occur

Low Gear: 79 N•m

High Gear: 157 N•m

In Figure 8, these torque limitations are compared to the requirements of the SAE J227a Schedule D Driving Cycle for the assumed vehicle. It can be seen that the gear train will carry the required loading. The fatigue life of the gear will be reduced if full regenerative braking is used.

Two considerations should be noted in assessing the importance of the limits of peak torque and gear-tooth stress that have been computed. First, the limit is assumed to be set by stresses in the 18-tooth pinion engaging the differential ring gear. The pinion is 1.125 inches wide but the stresses are computed on the basis of the ring-gear-face width of 0.750 inches. Thus, no benefit has been assumed from the pinion-tooth overhang. However, the ring gear teeth, although subject to fewer stress cycles, have stresses comparable to those calculated for the pinion. Also, as previously mentioned, stresses calculated by the methods used are very conservative as they apply to industrial equipment design where long life under high, continuous loading is a requirement. Stresses calculated by these methods are substantially exceeded in automotive practice as a matter of course because of the low duty factor. It is therefore concluded that the transaxle can be satisfactorily used in a vehicle test program.

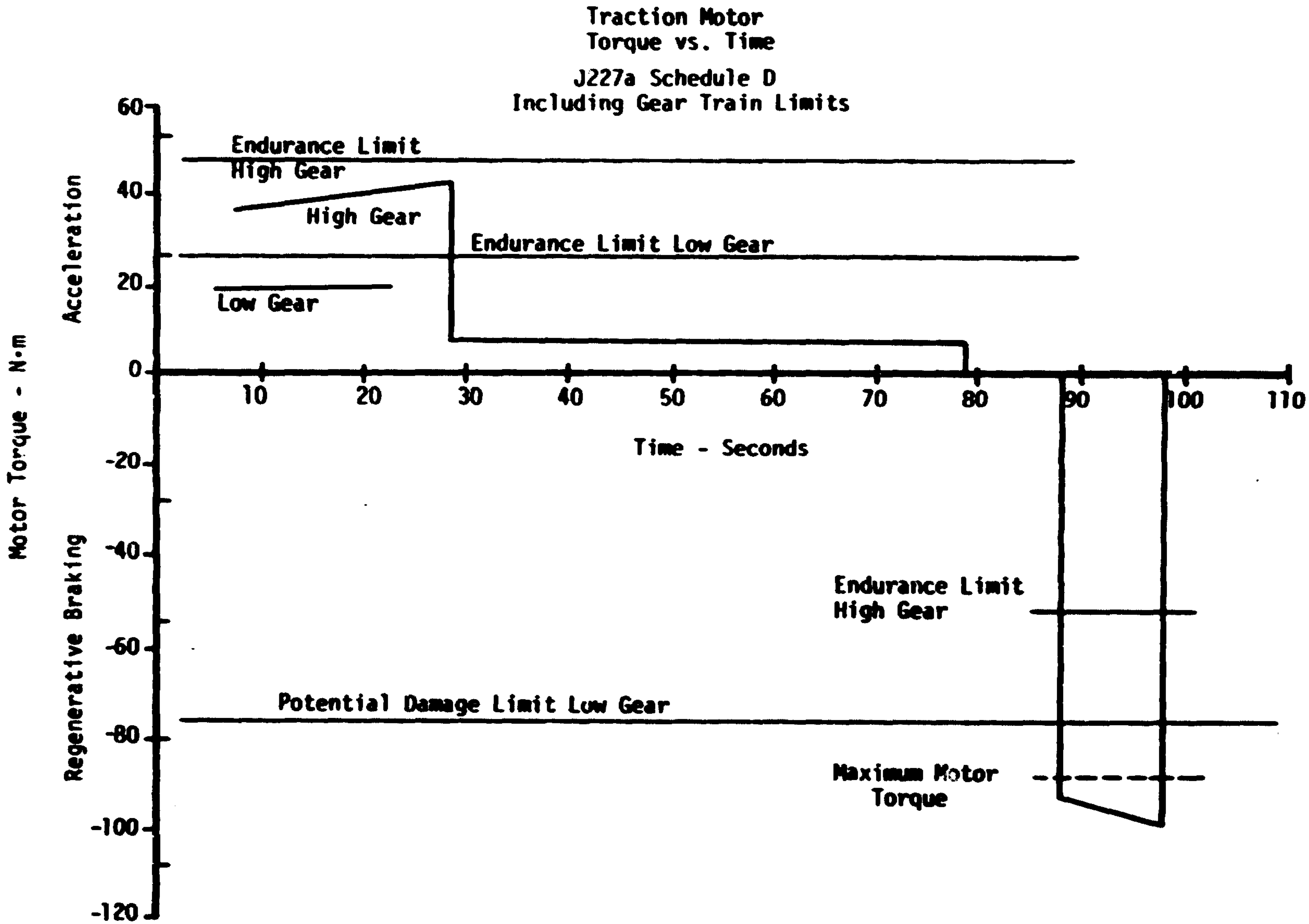


Figure 8. Driving Cycle Torque Requirements and Transaxle Limitations

<u>Yield Limit</u>

Due to the coarse pitch of the differential pinion and ring gear and the potential high torque loads at low speeds on these critical components, scoring of tooth faces is the most probable precursor of failure. Therefore, greater protection would be afforded by use of an EP lubricant such as multi-purpose hypoid-gear lubricant SAE Grade 90 (MIL-L-2015 B) (API Service GL-4) than obtained from a lower viscosity, automatic transmission fluid. The latter is compounded for service in systems which may reach high temperatures since it must cool clutches and brake bands, must have good oxidation resistance and anti-foaming properties, and must function as a hydraulic fluid in the shift servos. None of these requirements apply to the transaxle application although automatic transmission fluid has been the lubricant of choice in other electric vehicle transaxles due to its low viscosity and consequently reduced losses.

Due to the possibility of exceeding the strength limit of the transaxle in the NASA-LeRC dynamometer test facility, studies were made of the adaptation of the Browning Shear-Pin torque limiter to the transaxle output shaft. Two sizes are shown in Figures 9 and 10 with torque limits of 1814 N·m and 1689 N·m, respectively. These studies indicate that the breakdown torque of the traction motor of 89.5 N·m, corresponding to 1729 N·m torque on the single axle output torque of the transaxle, or the 1763 N·m limit of the differential system, can be safely protected from possible overload by the use of the shear pin limiter in the NASA-LeRC dynamometer test facility.

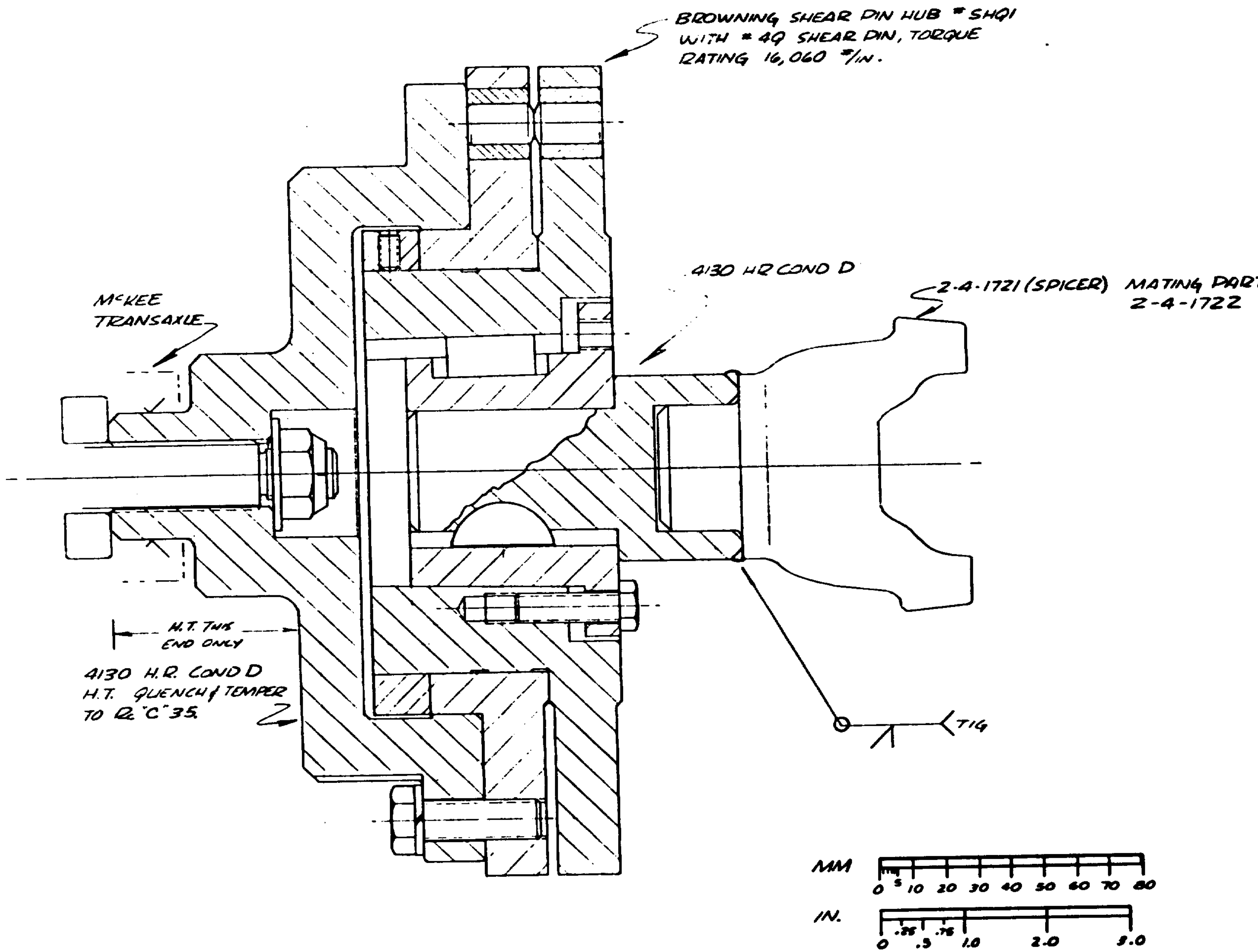


Figure 9. 1814 N.m Torque Limiter

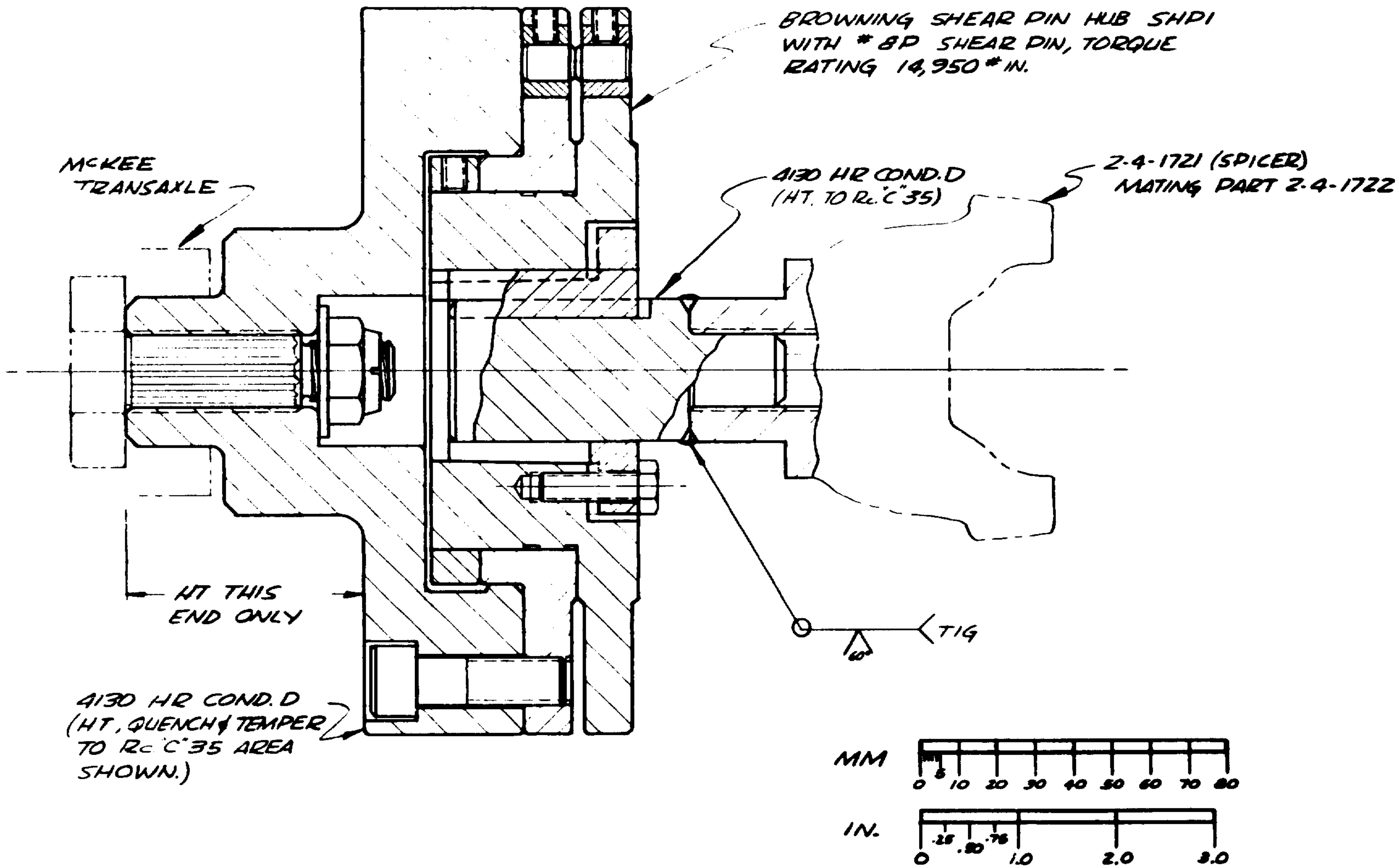


Figure 10. 1689 N·m Torque Limiter

1. Report No. NACA CR-159746	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle AN AUTOMATICALLY-SHIFTED TWO-SPEED TRANSAXLE SYSTEM FOR AN ELECTRIC VEHICLE		5. Report Date January, 1980
		6. Performing Organization Code
7. Author(s) Hayden S. Gordon & Gregory W. Hassman		8. Performing Organization Report No. WMB&A 131-4-R2
9. Performing Organization Name and Address William M. Brobeck & Associates 1235 Tenth Street Berkeley, California 94710		10. Work Unit No.
		11. Contract or Grant No. Den3-84
12. Sponsoring Agency Name and Address U. S. Department of Energy Office of Transportation Programs Washington, DC 20545		13. Type of Report and Period Covered Contractor Report 12/78 to 12/79
		14. Sponsoring Agency Code DOE/NASA/0084-79/1

15. Supplementary Notes

Final Report. Prepared under Interagency Agreement EC-77-A-31-1044. Project Manager, N. Sargent, Transportation Propulsion Division, NASA Lewis Research Center, Cleveland, Ohio 44135.

16. Abstract

This report describes an automatic-shifting scheme for a two-speed transaxle for use with an electric vehicle propulsion system. The transaxle system was to be installed in an instrumented laboratory propulsion system--a breadboard--of an ac electric vehicle drive train. The transaxle which had been fabricated is also described. The work was performed between December 28, 1978 and contract termination on April 26, 1979.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Electric Vehicle Transaxle Shift Control Logic	Unclassified--unlimited Star Category 44 and 85 DOE Category UC-96

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	38	

* For sale by the National Technical Information Service, Springfield, Virginia 22161

NASA-C-168 (Rev. 10-75)

PRECEDING PAGE BLANK NOT FILMED