VEHICLE LUNARIZATION STUDY

U.S. ARMY M-274 'MULE' VEHICLE

VOLUME - II

PART 1, TECHNICAL DISCUSSION

BY D MACCHIA

DEPARTMENT of THE ARMY

OFFICE, CHIEF of ENGINEERS

ARMY MATERIEL COMMAND

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U.S. ARMY M-274 "MULE" VEHICLE

FINAL REPORT

VOLUME II

PART 1 - TECHNICAL DISCUSSION

Prepared By

DEPARTMENT OF THE ARMY
Office, Chief of Engineers

and

Army Materiel Command

Prepared For

National Aeronautics and Space Administration
Office of Manned Space Flight
Washington, D.C.

April 1966

PREFACE

This is the final report of a study conducted for the purpose of determining if it would be feasible to modify a current military cargo type vehicle to operate in the lunar environment with a reasonable degree of confidence.

The study was requested by the Office of Manned Space Flight, National Aeronautics and Space Administration and was conducted by personnel from the Office of the Chief of Engineers; Headquarters, Army Materiel Command; Army Automotive Tank Center; and Engineer Research and Development Laboratories.

The military vehicle selected for modification analysis was the Army M-274 vehicle which is a 4 x 4, 1/2 ton, weapons carrier. This versatile vehicle is known more commonly as the "Mule". This vehicle was subjected to three orders of modification. The first modification was the minimum changes necessary to provide a modified vehicle having a capability consistent with the study criteria and at minimum cost. The second and third modifications were made to provide for comparison modified vehicles with increased capability for lunar surface mobility.

The Report is presented in two volumes, Volumes I and II. Volume II consists of two parts. Part I consists of the technical discussions concerned with vehicle modifications, development program schedules and cost estimates. Part 2 consists of study criteria and supplementary material developed to assist in the details of vehicle modification. Volume I is a separate summary which presents in less detail the material contained in Volume II - Part 1.

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1.0 INTRODUCTION

1.1 General

One of the objectives of the Apollo Program is scientific exploration on the lunar surface. However, the basic Apollo system has only limited stay time and payload capabilities. The Apollo Applications Program (AAP) has been proposed as the next step toward increasing the nation's capabilities to explore the moon. The space vehicle configuration and mission profile for an AAP mission are outlined in Vol II, Part 2, Appendix, Section 1.0.

The Apollo Applications Program concerns itself with extension of man's stay time and capabilities on the linar surface. A key element is the addition of some degree of surface mobility for the astronaut. To date, studies of lunar surface transportation systems have emphasized optimum performance under a very wide range of operating conditions. This approach has produced concepts for cross-country vehicles of unconventional design and requiring development of many new components and subsystems. In order to properly evaluate the worth of this approach, NASA has found it desirable to compare the cost and expected performance of such unconventional vehicles with vehicles of somewhat less sophistication and which use proven design principles and components. To support this comparative evaluation, NASA established the requirement for a study to determine the feasibility and cost of development for the modification of a proven terrestrial vehicle to perform in the lunar environment with a reasonable degree of confidence.

The Office of Manned Space Flight, Headquarters, NASA, requested the Department of the Army to conduct this study. Responsibility for conduct of the study was assigned to the Chief of Engineers and the Commanding General, Army Materiel Command.

1.2 Study Objectives

1.2.1 General.

Preliminary discussion between representatives of OMSF, NASA, and the U. S. Army identified the study guidelines and restraints. Consideration of payload limitations and volume restrictions supported a tentative selection of the Army's M-274 "Mule" vehicle. It was also determined that should further study show this vehicle to be unsatisfactory for the intended use, consideration of other vehicles would be limited to vehicles in production for which design and supporting data are available.

1.2.2 Specific Objectives

1.2.2.1 Vehicle Survey. A precursory survey of U. S. Army vehicles indicated the M-274 (Mule) vehicle as the most logical vehicle to use as a basis for this study. Actual vehicle selection was to be based upon trade-offs and comparisons of mass and volume, adaptability of the vehicle to the transportation system payload, and terrestrial vehicle performance.

1.2.2.2 Conceptual Design/Modification Study. Upon final selection of the vehicle, the vehicle system was to be examined in greater depth to determine which of the subsystems and components require "lunarization" and the extent to which such modification was required. In developing the results of such an analysis, it was possible that several degrees of modification could be established, each resulting in slight increases in one or more of the vehicle's overall operational capability or performance characteristics. The prime consideration in selecting a minimum level of modification was to be cost. The effect of this task was to establish a "baseline" or minimum modification vehicle; one which, due to cost considerations, would be capable of satisfying in a minimum manner the performance requirements. The conceptual design/modification was to define the following elements:

- Composite System
- Power System
- Mobility System
- Crew Station

1.2.2.3 Test Program Definition. This task was to define the system, subsystems and component engineering and test program required to demonstrate the capability of the vehicle system.

1.2.2.4 Vehicle Performance Data Definition. This task was to evaluate the modified vehicles performance characteristics and operational capability and limitations.

1.2.2.5 Cost Estimates. This task was to develop a cost breakdown for the development project. In addition to development costs, the procurement cost was to be estimated for a total of three operational units.

1.3 Study Team Organization

The study team was staffed by personnel from the Extraterrestrial Research Agency, Office of the Chief of Engineers; Directorates of Research and Development and Quality Assurance, Headquarters, Army Material Command; Research and Engineering Directorate, Army Tank Automotive Center; and Electrical Department, Engineer Research and Development Laboratories.

The study team functioned as a unit in the Extraterrestrial Research Agency under the general direction of Mr. H. N. Lowe, Jr., Chief of the Agency.

1.4 Study Participants

D. Butler	OCE, Project Manager
W. Eason	OCE, Project Engineer
Dr. G. Frysinger	AMC, ERDL, Asst Project Engineer
A. Jokl	AMC, ERDL

P. Spanski	AMC, ATAC
J. Eilers	AMC, ATAC
R. Liston	AMC, ATAC, Consultant
J. Hausman	AMC, HQ
D. Butcher	AMC, HQ, Consultant
J. Malcolm	OCE
B. Hall	OCE
A. Allen	OCE
D. Gimber	OCE
M. Cooper	OCE

1.5 Study Approach

The guidelines and constraints for this study were provided by NASA. These are given in Vol II, Part 2, Appendix, Section 1.0.

The study flow plan is given in Figure 1-1. A review of terrestrial vehicles was conducted as one of the initial steps. It was followed by final selection of the study vehicle.

The orders of vehicle modification and their designation are given in Figure 1-2.

The first effort was devoted to defining a minimum modification program consistent with the guidelines and constraints and minimum cost. The resulting vehicle was defined as the first order modification.

Two additional and successive orders of modification were then developed to define opportunities for improvement in vehicle performance. The third-order modification defined the maximum degree of modification possible and still retain the original vehicle structure.

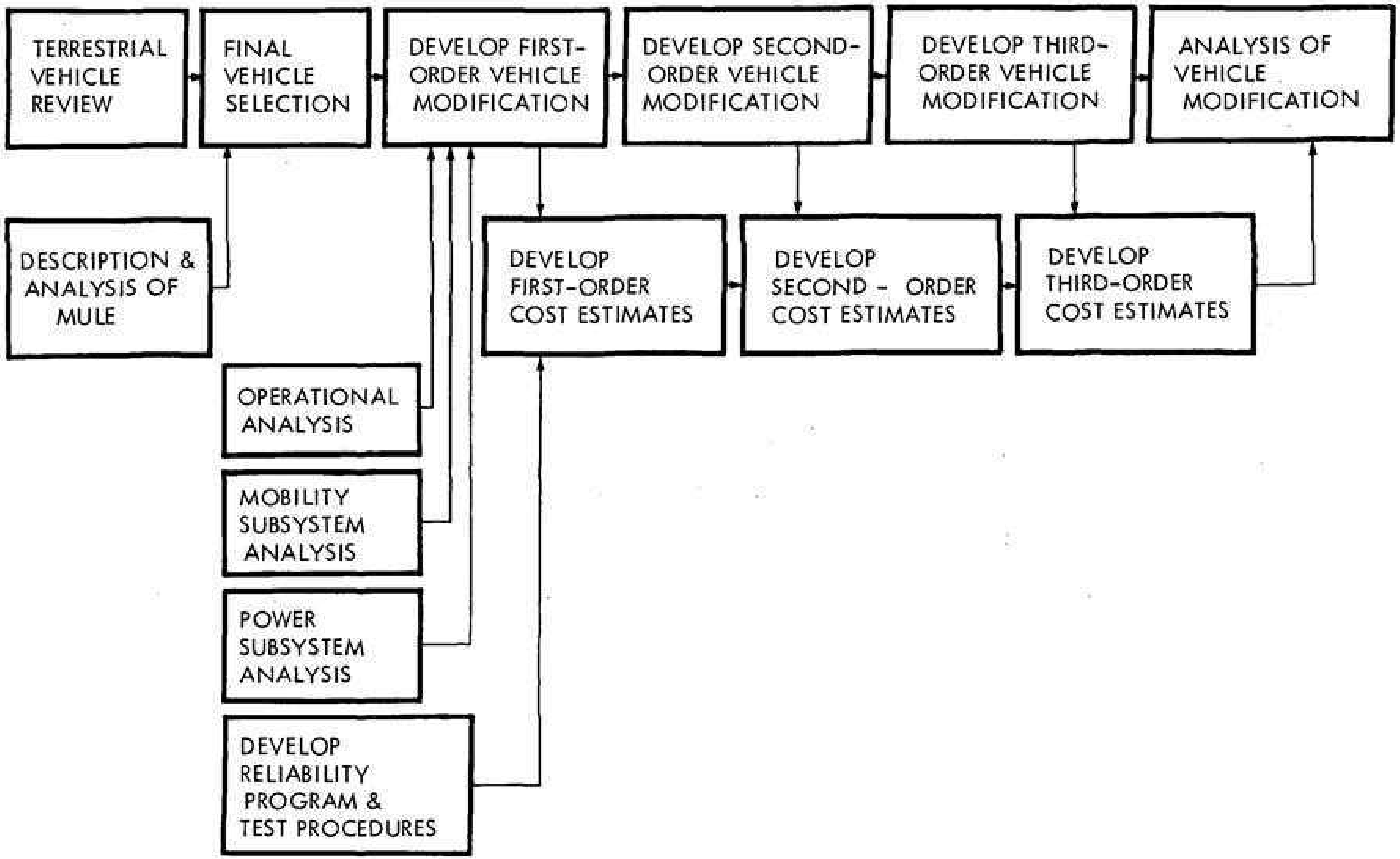


Figure 1-1. Study Flow Plan

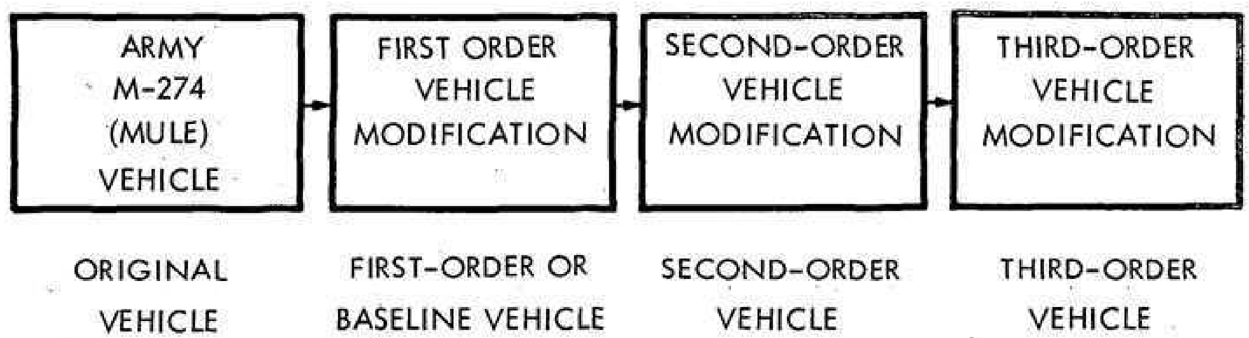


Figure 1-2. Orders of Vehicle Modification

The study philosophy in the first-order modification was to make as few changes as possible to achieve desired performance. In all cases the functioning of each subsystem and component in the lunar environment was examined in detail. Where modifications were adjudged as necessary, new components and subsystems were selected with primary emphasis being upon simplicity of operation and high reliability. Maximun use was made of new components which already exist or are in some stage of a scheduled development program. Preference was given to components already developed and space qualified.

2.0 STUDY GUIDELINES AND CONSTRAINTS

The Statement of Work for this study with five (5) Appendices was furnished by OMSF.

This statement of work contains considerable background information as well as study guidelines and constraints. The detailed listing of the study guidelines and constraints is contained in Vol II, Part 2, Appendix, Section 1.0.

The following is a summary of some of the primary guidelines and constraints:

- The vehicle must be carried and packaged above the LEM/S descent stage between the shelter and adapter.
- The vehicle must be capable of withstanding an initial three months of storage and fourteen days of operational time on the lunar surface.
- The vehicle will be operated only during one lunar day (14 days earth time).
- The total operational range per 14 day mission should be 150 miles (240 km).
- The total operational range per sortie should be 15 miles (24 km) (10 sorties total).
- The vehicle must carry the operator and, as an alternate payload, one other astronaut. This sets the minimum cargo (excluding operator) as 145.5 Kg or 320 lbm (space suit assembly 125 lbm plus 195 lbm astronaut).
- The vehicle must carry a maximum cargo of 704 lbm plus a spare PLSS of 63 lbm for operator.
- The vehicle should be capable of speeds of at least 3 mph (statute) on level soft soils ($K\phi = 0.5$, $n = 0.5$) and at least 5 mph on level compacted soils ($K\phi = 3.0$, $n = 1.00$).
- Operational sorties will be up to 6 hours in length (PLSS life - 2 units - estimated at 1200 btu/hr expenditure) plus 2 hours for stops (8 hours maximum sortie duration).

- Operational radius of vehicle will be limited to 6 km line-of-sight.
- The vehicle must be capable of traversing lunar surface terrain with slopes up to 15 degrees with the soil characteristics and slope frequency distribution as specified in the Engineering Lunar Model Surface (ELMS).
- The mass of the vehicle without operator or cargo, but fully fueled (for 150 mile (240 km) range) will not exceed 683 Kg or 1500 lbm.
- The vehicle mass allowance will be charged for all LEM/S power (fuel for shelter fuel-cell) furnished to the vehicle.
- The vehicle development test program should be formulated to demonstrate a minimum modified-vehicle reliability of 0.90 at an 80% confidence level to accomplish the operational mission outlined.

3.0 VEHICLE REVIEW AND SELECTION

3.1 General

A review of terrestrial vehicles was conducted. This review has covered the class of vehicles designated as remote area vehicles. This terminology covers a class of small, mechanized personnel and cargo carriers designed to operate in a harsh environment in the absence of prepared surfaces. It is felt that the lunar surface qualifies as a truly harsh environment and certainly is without prepared surfaces.

This vehicle review was conducted primarily to assist in the selection of possible subsystems and components in the vehicle modification process. An initial review indicated that an Army vehicle, the M-274, 1/2-ton cargo vehicle, (the "Mule"), was the most logical vehicle for use in this study. However, the final selection was made after a review of many other vehicles. The basic vehicle has to be small and compact, rugged, have an adequate cargo platform, and a history of performance. The first order vehicle modification was to be a minimum departure from the basic vehicle consistent with minimum cost and meeting the criteria. Consequently, the M-274 vehicle was the best choice of production class Army vehicles for use as the basic vehicle.

3.2 Vehicle Review and Analysis

The largest and most complete body of information on remote area vehicles has been compiled by the U.S. Army, and in particular by the U.S. Army Tank and Automotive Center in Warren, Michigan. References 1, 4, 5, 6, 7, 8 and 10 were researched for potential vehicles which could conceivably meet the criteria established. Over 750 remote area vehicles have been evaluated in these reports.

Figure 3-1 indicates several of the interesting vehicle concepts reviewed. The vehicles shown in Figure 3-1 are:

XM561, 6 x 6, Articulated, Wheeled
RAM I, 4 x 4, Wheeled
Marsh Screw
PATA, Inflatible Air-Bag Tracked
Swamp Spryte-Model 1301, Tracked
M116, Tracked
Swamp Skipper, 4 x 4, Wheeled
Gemini Test Bed, 4 x 4, Articulated, Wheeled

A division of vehicles reviewed was made between wheeled and tracked vehicles. While tracked vehicles are generally chosen for the most difficult terrain situations, consideration of other lunar environment parameters resulted in the selection of a wheeled vehicle. Primarily, the exposure of a mechanism as complex as a track and suspension system to the very low lunar atmospheric pressure was the factor responsible for wheel selection. Army experience indicates that tracks are traditionally less durable than wheels and thus have a lower reliability.

After deciding upon the wheeled vehicle, many interesting concepts were reviewed. In addition to references mentioned previously, references 2, 3 and 9 were consulted. The wheeled vehicles were divided between rigid-frame and non-rigid frame vehicles. The non-rigid frame vehicles were further divided into articulated frames and flexible frames.

The rigid-frame wheeled vehicle offers the maximum simplicity and is therefore potentially the most reliable. However, a properly designed, high-wheel-travel suspension is essential to maintain good ground contact and hence vehicle control. The quality of the ride is primarily affected by the suspension design. Certain small remote area vehicles are designed without a suspension system for additional simplicity. While potentially very reliable, this lack of suspension drastically limits the vehicle speed over rough terrain and even affects the vehicle's soft soil mobility.

XM 561, 6X6 ARTICULATED, WHEELED

SWAMP SPRYTE, MODEL 1301, TRACKED

RAM I, 4x4, 1¼ TON, WHEELED

M116 TRACKED CARRIER

MARSH SCREW

SWAMP SKIPPER, 4x4, WHEELED

INFLATABLE AIR BAG, TRACKED

GEMINI TEST BED, 4x4, ARTICULATED

Figure 3-1. M-274 Vehicle, Characteristics Sheet

The mobility degradation is caused primarily by the inability to maintain constant wheel loadings. In fact, a four wheel vehicle of this type may be considered to have only three wheels for mobility purposes except in the very ideal situation of smooth level surfaces.

The articulated non-rigid frame vehicle offers several advantages over the basic four-wheel, rigid-frame vehicle. These advantages stem from the use of the articulation joint for steering and the roll freedom allowance of the vehicle sections relative to each other. Steering by articulation imposes less stresses on the soil than either Ackerman type or skid type steering systems. Thus soil strength can be utilized more fully for vehicle mobility during steering maneuvers. Also, since the wheels of the vehicle do not turn relative to the vehicle, wheel wells can be smaller and fit more closely; larger wheels also may be used.

When the articulated joint includes yaw freedom as well as roll freedom, the vehicle attains an order of magnitude improvement in the ability to conform to the terrain profile. Optimum wheel loadings can thus be maintained. This is true even when the individual wheels are not suspended. The rigid suspension still, in this case, limits the vehicle speed over rough terrain.

The disadvantage of articulated vehicles lie in their more complex design and hence higher cost. Demonstration of a specified reliability would require more testing time than for a similar class rigid-frame vehicle.

A step beyond the articulated non-rigid frame vehicle is the flexible frame vehicle concept. While no practical production vehicles of this type have been constructed, several proposals have been advanced, references 2 and 9. The flexible-frame is combined generally with articulated steering or wagon steering. The flexible-frame vehicle has the greatest potential ability to conform to the terrain profile. The flexible frame (having a definite spring rate) acts as an auxiliary suspension device and improves the speed of crossing rough terrain.

Vertical obstacle negotiation capability is remarkable with the flexible-frame concept. A model of a long wheel base six-wheel, flexible-frame vehicle has demonstrated an ability to climb a vertical obstacle somewhat higher than the diameter of the vehicle wheels, (see reference 7). In climbing a vertical obstacle, the flexible frame transfers the horizontal force encountered by the front wheels against the obstacle to the rear wheels. This transfer of force generates additional vertical loads on the rear wheels and thus increases the available tractive effort from the soil-wheel interaction. This, in turn, increases the force of the front wheels upon the obstacle. This self-energizing cycle is most effective with a multi-wheel (more than 4) configuration and a frame which is highly flexible in the pitch plane. This multi-wheel, flexible-frame, articulated vehicle presents a magnitude of mobility in wheeled vehicles previously not developed. A vehicle concept of this type would be a primary choice for an optimum remote-area vehicle.

3.3 Relative Vehicle Development Costs

An example of the relative costs of developing such types of terrestrial military vehicles can be given. The Army M-274 and the XM-561 vehicles are utilized for this purpose. The M-274 is a 4 x 4, rigid-frame, suspensionless, 450 kg (1/2 ton) capacity cargo transporter. The XM-561 is a 6 x 6, articulated, independent suspension, 1150 kg (1-1/4 ton) capacity cargo transporter. During 1952-1955, a total of $1,700,000 excluding prototypes costs and pre-production engineering was spent to engineer and develop the M-274 vehicle. Since 1961, approximately $8,500,000 has been spent to engineer and develop the XM-561 vehicle. Some of the cost difference between the XM-561 and the M-274 vehicles can be attributed to the larger size of the XM-561 and the time increase in cost of equivalent engineering work. The cost increase was due primarily to the increase in vehicle complexity.

3.4 Vehicle Selection

The criteria for this study emphasizes a minimum vehicle development cost consistent with the study vehicle performance and program reliability requirements. No definitive obstacle size and distribution was given. The development costs of vehicles climb rather rapidly with increasing vehicle capability. The development of a flexible frame vehicle to a specified reliability would entail the most expensive development program of the vehicle types discussed.

Consideration of the available vehicles, against all the given criteria, resulted in the selection of the standard U.S. Army M-274 1/2 ton cargo carrier, the "Mule", as the basic vehicle for modification concepting. The Mule vehicle is not only dimensionally suitable but is extremely rugged and has a demonstrated reliability history. When fitted with terra tires, the Mule also has significant soft soil crossing capability. Preliminary investigations indicated that it could be lunarized to meet the criteria with a minimum of modification.

3.5 REFERENCES

3-1 Aberdeen Proving Ground, Development & Proof Services, "Test Report—Military Potential of the Dyna-Mite Articulated Vehicle" Report No. DPS 1530, December 1964.

3-2 Bekker, M. G., "6 x 6 Flexible Frame Lunar Vehicle Model", General Motors Defense Systems.

3-3 Bekker, M. G., "Off the Road Locomotion—Research and Development in Terramechanics", University of Michigan Press, Ann Arbor, Michigan, 1961.

3-4 General Motors, Chevrolet Motor Division and General Motors Defense Research Laboratory, "Sidewinder", Four Wheel, Articulated Vehicle Concept.

3-5 U.S. Army Tank Automotive Center, Research & Engineering Directorate, "RAVE I, Remote Area Vehicle Evaluation", 29 October 1962.

3-6 U.S. Army Tank Automotive Center, Research & Engineering Directorate, "RAVE II, Remote Area Vehicle Evaluation", 13 June 1963.

3-7 U.S. Army Tank Automotive Center, Research & Engineering Directorate, "Wheel-Track, Comparative Operational Mobility Evaluation", 21 June 1962.

3-8 U.S. Army Tank Automotive Center, Research & Engineering Directorate, "Special Vehicles", 22 January 1962.

3-9 U.S. Army Tank Automotive Center, Land Locomotion Laboratory, "Flexible Frame Test Rig".

3-10 U.S. Army Tank Automotive Center, Research & Engineering Directorate, Economic Engineering Branch, "Evaluation of Past and Present Vehicles Capable of Performing in Adverse Terrain".

4.0 DESCRIPTION AND ANALYSIS OF THE M-274 VEHICLE

4.1 History of Development

There was a recognized need for a light, cross-country, logistic military carrier for several decades. Such a vehicle was needed to transport weapons and supplies over difficult terrain in the battlefield area. During World War I, the mule (animal) and the motorcycle performed this function, in addition to the soldier himself. A report issued after World War I established an Army requirement for a self-propelled, four wheeled vehicle to replace the mule and the motorcycle and also to relieve the soldier from this duty. Funding was severely curtailed at that time and no development was accomplished.

In World War II, a vehicle of this class was developed. This was the "Jeep" 1/4 ton carrier which was a versatile, cross-country, logistic vehicle. The Jeep had several deficiencies, however; notably, the vehicle height and width and poor soft-soil mobility. A report was issued in 1946 re-establishing the need for a platform-type cargo carrier. Again funds were not available for development.

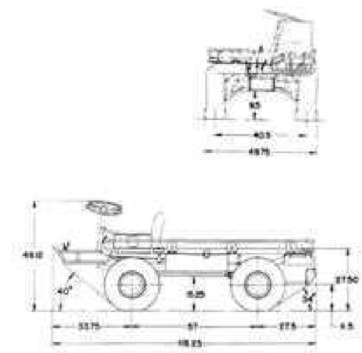
In April, 1952, the mechanical mule vehicle development project was started. Development was completed and the vehicle was type-classified in April 1956. The U.S. Marine Corps became interested in this vehicle and began purchases in 1960. To date, procurement by the Army and Marine Corps of this vehicle total approximately 5,700 units.

4.2 Vehicle Characteristics

Figure 4-1 is the characteristic sheet on the standard M-274 vehicle.

CHARACTERISTIC SHEET **ORDNANCE TANK AUTOMOTIVE COMMAND**

VEHICLE: ~~CARRIER, LIGHT WEAPONS, INFANTRY~~, 1/2 TON, 4X4, M274 SECTION: SWV-1 STATUS: STANDARD
WHEELBASE: 57 **IN.** **TYPE** WEAPONS CARRIER CODE: SWV-6 DATE: 1 SEPT. 1958



CURB WEIGHT, FULLY EQUIPPED, LESS PAYLOAD & PERSONNEL — W/W: FRONT AXLE N/A LBS.; REAR AXLE OR BOGIE N/A LBS.; TOTAL N/A LBS. — WO/W: FRONT AXLE 338 LBS.; REAR AXLE OR BOGIE 587 LBS.; TOTAL 925 LBS.

GROSS LOAD, CROSS COUNTRY N/A LBS. 1000 LBS.
HIGHWAY N/A LBS. 1000 LBS.

TOWED LOAD ALLOWANCE, CROSS COUNTRY N/A LBS. HIGHWAY N/A LBS.

SHIPPING DIMENSIONS. CU. FT. 81.64
SQ. FT. 40.5 REDUCIBLE TO 32 SQ. FT.
LENGTH 118.25 IN. REDUCIBLE TO 103.12 IN.
WIDTH 49.75 IN. REDUCIBLE TO IN.
HEIGHT. N/A 49.12 IN. REDUCIBLE TO 27.50 IN.

TREAD. FRONT 40-1/2 IN. REAR 40-1/2 IN.

CAB TO ℄ REAR AXLE OR BOGIE IN. ℄ REAR AXLE TO END OF PLATFORM 27.5 IN.

MINIMUM GROUND CLEARANCE. REAR AXLE OR BOGIE 8.5 IN. UNDER CHASSIS 8.5 IN.

ANGLE OF APPROACH: N/A ° 40 °
ANGLE OF DEPARTURE: 34 °
MIN. TURNING RADII (OUTER CORNER): LEFT N/A FT. IN. 9 FT. 10 IN.
RIGHT N/A FT. IN. 9 FT. 2 IN.

FORDING DEPTH. N/A WITH KIT N/A IN. LESS KIT 18 IN.

ENGINE:
MAKE WILLYS MODEL AO-53 TYPE HORIZ. OPPOSED, AIR COOLED
DISPLACEMENT 53.5 CU. IN. BORE & STROKE 2.75 × 2.25 COMPRESSION RATIO 6.5:1
FUEL 80 MIN. OCTANE RATING. CAPACITY 8 GALS. GOV. SPEED 4200 RPM.
HORSEPOWER: MAX. GROSS 15 @ 3200 RPM MAX. NET @ RPM.
TORQUE: MAX. GROSS 30 LB.FT. @ 2100 RPM MAX. NET LB.FT. @ RPM.
LUBRICATION PRESSURE CRANK CASE CAPACITY 2 QTS. REFILL
GOVERNOR: MAKE HOOF MODEL SO5N-1 TYPE VELOCITY 2½ QTS. DRY
COOLING SYSTEM CAPACITY N/A QTS. PUMP TYPE DRIVE

FUEL SYSTEM: CARBURETOR, MAKE TILLOTSON MODEL HD107A TYPE HORIZONTAL
FUEL PUMP: MAKE AC MODEL FO

CLUTCH: MODEL AUBURN #100163 TYPE SINGLE-DISC DIAM. 6½ IN.
EFFECTIVE 17.27 SQ. IN. TORQUE RATING 36-40 LB.FT.

TRANSMISSION MODEL WILLYS (INTEGRAL W/REAR AXLE) NO. OF SPEEDS 3 FWD. 1 REV.
RATIOS: 1ST 2.93:1 2ND 1.71:1 3RD 1.00:1 4TH 5TH REV. 2.93:1
PTO OPENINGS NONE INPUT TORQUE RATING LB.FT.
LUBRICANT CAPACITY W/PTO QTS. WO/PTO 2 QTS.

TRANSFER CASE: MODEL WILLYS (INTEGRAL W/TRANS.) NO. OF SPEEDS 2 FWD. REV.
RATIOS: HIGH 2.84:1 LOW 5.31:1 REVERSE HIGH LOW
CONTROL OF FRONT AXLE DRIVE NONE LUB. CAPACITY QTS.

AXLES: FRONT: MODEL WILLYS TYPE SPIRAL BEVEL
RATIO 1.87 : 1 TORQUE RATING 235 LB.FT. LUB. CAP. 6 OZ. PROV. FOR DUAL N/A
DROP GEAR RATIO 2.20 : 1 TORQUE RATING TYPE
REAR: MODEL QTS.
RATIO 1.87 : 1 TORQUE RATING INTEGRAL W/TRANS.
DROP GEAR RATIO 2.20 : 1 TORQUE RATING
FRAME: TYPE TUBULAR DIMENSIONS: SIDEBAR 3.355 I.D. X 3.500 O.D. X 14 GAGE
SECTION MODULUS .679 IN. NO. OF CROSS-MEMBERS 3 REINFORCED
EXTENSION TUBES - 12 GAGE

SUSPENSION: FRONT SPRINGS: TYPE NONE SIZE WIDE LONG
NO. LEAVES RATE LB/IN. SHOCK ABSORBERS
REAR SPRINGS: TYPE SIZE WIDE LONG
NO. LEAVES RATE LB/IN. SHOCK ABSORBERS
BOGIE: MODEL W.B. IN. SPRINGS WIDE LONG
NO. LEAVES RATE LB./IN. SHOCK ABSORBERS

WHEELS: ORDNANCE NO. D7966747 TYPE CAST MAGNESIUM ALLOY DIAM. 10 X 5.50 IN
BEAD CLIPS: ORD. NO. NONE (DROP CENTER TYPE) SINGLE REAR SINGLE REAR
TIRES: TYPE FLOTATION SIZE 7.50 X 10 PLY 4
TREAD DESIGN: ALL SERVICE-CROSS COUNTRY-MILITARY NO. SPARES NONE

STEERING: MODEL ROSS S-12 TYPE CAM & LEVER
RATIO: 14:1 POWER ASSIST MODEL NONE

BRAKES: SERVICE: MAKE TYPE INTERNAL EXPANDING
DRUM SIZE: FRONT: 5-1/2" DIA. X 1-3/8" REAR
EFFECTIVE AREA: FRONT 17.4 SQ. IN. REAR/BOGIE SQ. IN.
ACTUATION. CABLE & CAM W/FOOT PEDAL

BRAKES: PARKING: MAKE CABLE & CAM W/HAND RATCHET LEVER TYPE PROPELLER SHAFT-INTERNAL EXPANDING
DRUM SIZE EFFECTIVE AREA 17.4 SQ. IN.

ELECTRICAL SYSTEM: WATERPROOF N/A RADIO SUPPRESSED YES
POTENTIAL STARTING SYSTEM: MANUAL PULL W/RECOIL CABLE
GENERATOR ORD. NO. NONE RATING AMP. DRIVE
BATTERY (IES) NO. NONE TYPE ORD. NO.
MAGNETO REGULATOR: ORD. NO.
LIGHTING: HEADLAMPS: NO. NONE BO DRIVING LAMP NO.
B.O. MARKER LAMPS: NO. NONE-REFLECTIVE TAPE-LOC. OF PLATFORM TRAILER COUPLING SOCKET NO.

PERFORMANCE: MILES PER GALLON 17.6 @ 5 M.P.H. CRUISING RANGE HIGHWAY 100 MI.
GRADEABILITY: DIRECT 44.5 % FORWARD, TOTAL OF 1830 LBS.
SIDE SLOPE-40% LOW 60 % FOR GVW, TOTAL OF 1830 LBS.
FORWARD-60% MAXIMUM SPEED 25 MPH AT ENGINE GOVERNED SPEED 4200 RPM.

CAB: TYPE NONE CAPACITY 1 MEN

WINCH: MAKE NONE MODEL
CAPACITY LBS. 1ST LAYER ROPE DIAM. IN.
MAX. ROPE LENGTH PTO SPEEDS FWD. REV.

PINTLE HOOKS: ORD. NO. NONE
TIE DOWNS: NO. LOCATION HAND RAILS ON PLATFORM
PLATFORM: FLAT BED FABRICATED OF EXTRUDED MAGNESIUM ALLOYS
SIZE: 96" LONG X 46" WIDE
SPECIAL EQUIP. AVAIL. - 106 MM RIFLE INF. WEAPON KIT
BRACKETS - AXLE MOUNTED AIRBORNE SLINGING PROVISION
- HELICOPTER LIFTS - TWO (2) REAR CORNERS AND ONE (1) CENTER OF FRONT EDGE OF PLATFORM

FOR PURPOSE, REFERENCE (TCTS), HISTORY AND CURRENT STATUS, SEE REVERSE SIDE.

PROJECT ENGINEER Paul Emelion
UNIT CHIEF Ronald G. Patak
SECTION CHIEF L. F. Muttinger
BRANCH CHIEF W. L. Dunne
CHIEF ENGINEER J. B. Hayes
DATE APPROVED 8 Sept 1958

OAP FORM 1028 6 JAN 57 Previous editions of this form are obsolete.

Figure 4-1. M-274 Vehicle, Characteristics Sheet

The current vehicle has a flat bed platform of extruded magnesium alloys 121.9 cm wide x 259 cm long (48" x 102"). The axles are solidly mounted to the platform with no suspension. Airborne and helicopter lift points are provided. The vehicle has a shock load rating of 30 g for airborne drops.

The engine is mounted under the platform in a protective basket directly behind the transmission-transfer case which is integral with the rear axle. There are no differentials and the wheels are mounted to the axles through constant-velocity joints and drop-gears. A drive shaft extends forward to the front axle and a shoe brake is mounted on this shaft just behind the front axle. The current vehicle has four wheel Ackerman steering; but the rear steer has been eliminated on the newest models.

The vehicle can be turned on its side by two men for repairs and, with the sealed fuel and lubrication systems, can be driven immediately when righted. It can be driven three-wheeled (one wheel off), because of the rigid structure, by redistribution of the cargo.

The current vehicle has a 10.44 kilowatt (14 horsepower) air-cooled, gasoline-fueled, horizontal-opposed, internal combustion engine with 41 newton-meters (30 foot-pounds) gross torque. The transmission has three forward gears (2.93:1, 1.71:1, 1.00:1) and one reverse gear (2.93:1). There are two transfer ranges (2.84:1, 5.31:1). The axles are spiral bevel type with a 1.87:1 ratio and the drop gears have a ratio of 2.20:1.

4.3 Vehicle Mass

The current vehicle has a mass total of 433.2 kilograms (955 pounds) fueled, which is distributed as follows by materials:

Magnesium	35.4 kg	(78 lb)	(platform and wheels)
Aluminum	61.6 kg	(136 lb)	(engine, axles & gear case)
Copper	2.7 kg	(6 lb)	(electrical)

Ferrous Metals	235.4 kg (519 lb)
Rubber	29.94 kg (66 lb)
Assembly, Paint and Cotton	40.8 kg (90 lb)
Fuel and Oils	27.2 kg (60 lb)

4.4 Vehicle Performance

The vehicle will carry a cargo mass of 453.6 kg (1,000 pounds) including the driver and has a range of 160.9 km (100 miles) at 7.57 km/liter (17.8 miles/gal) of gasoline fuel. It will negotiate a 60% forward slope and a 40% side slope and can negotiate obstacles up to 25.4 cm (10") at low speed. The angle of approach is 40 degrees and the angle of departure is 34 degrees. The vehicle has a ground clearance of 21.6 cm (8.5").

The vehicle reliability has been estimated as about 0.92 for 1609.3 km (1,000 miles) of operation. However, in tactical use under combat conditions, the Mule rarely lasts for more than 805 km (500 miles) of operation.

4.5 Development Program

The "Mule" vehicle development program is summarized in Figures 4-2, 4-3 and 4-4.

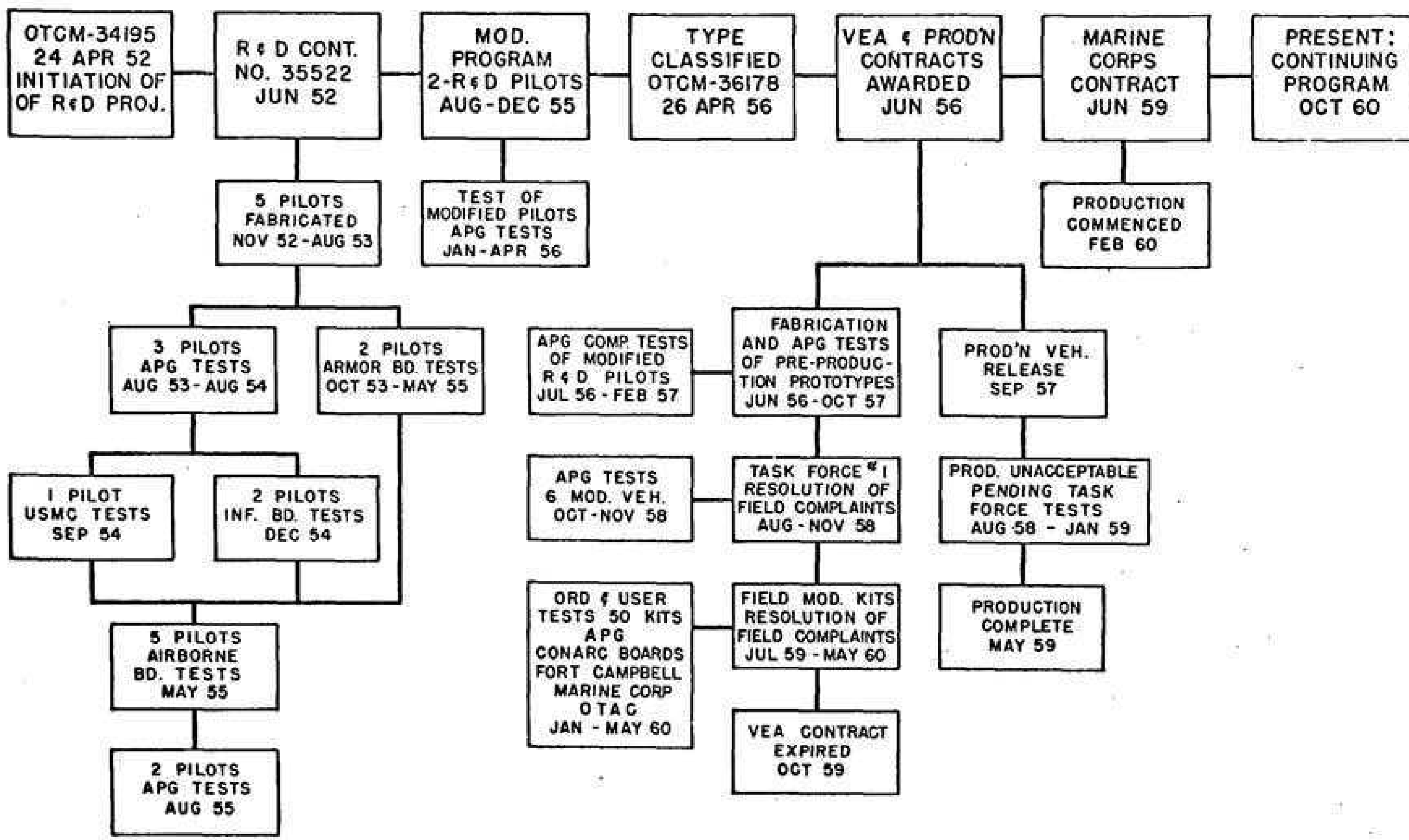


Figure 4-2. M-274 Vehicle, R&D Program Cycle

TEST AND LOCATION	TYPE	DATE	NO. OF VEH.	TOTAL MILEAGE	COMMENTS
R & D • APG • CONARC • USMC	DURABILITY AND ENGINEERING USER EVALUATION	SEP 53 TO NOV 54	5	15,000 APPROX.	EXCELLENT PERFORMANCE NEGOTIATING TEST COURSES - POOR ENGINE PERFORMANCE - HIGH OIL CONSUMPTION.
R & D APG	PROOF TESTS FOR TYPE CLASS	JAN 56 TO APR 56	2	3,632	MODIFICATIONS IMPROVED VEHICLE PERFORMANCE AND DURABILITY - MOBILITY GOOD OVER TEST COURSES.
PRE-PRODUCTION (COMPONENTS) • APG	SUITABILITY AND DURABILITY OF MODIFIED COMPONENTS FOR PRODUCTION.	JUL 56 TO FEB 57	1	2,573	MODIFICATIONS IMPROVED VEHICLE PERFORMANCE AND DURABILITY: DEFICIENCIES REPORTED: ENGINE, MECHANICAL STARTER, WHEEL HUBS, STEERING SYSTEM, PLATFORM AND CHASSIS.
PRE-PRODUCTION (VEHICLE) • APG	DURABILITY AND ENGINEERING	FEB 57 TO OCT 57	2	9,082	VEHICLE DESIGN GENERALLY SATISFACTORY - HOWEVER, DEFICENCIES REPORTED IN ENGINE, MECHANICAL STARTER, STEERING SYSTEM AND CHASSIS.
PRODUCTION (MODIFIED COMP.) • APG	ACCELERATED DURABILITY AND ENGR. TESTING	SEP 58 TO NOV 58	6	24,069	STARTING SYSTEM MODIFICATIONS ENHANCED RELIABILITY - THE MAGNETO AND INTAKE MANIFOLD WERE ACCEPTABLE.
PRODUCTION (VEH.) • CONARC	USER EVALUATION	OCT 58 TO MAR 59	1	6,227	ARMOR BOARD CONCLUDED THAT THE M274 CARRIER IS UNSUITABLE FOR ARMY USE.
PRODUCTION (VEH.) • CONARC	USER EVALUATION	OCT 58 TO FEB 59	3-6	NOT AVAIL.	THE VEHICLE SUITABLE FOR THE MARINE CORPS, HOWEVER, CERTAIN MECHANICAL DEFICIENCIES EXIST WHICH REQUIRE EARLY CORRECTION.
PRODUCTION (MODIFIED COMP. AND KIT MATERIAL) • APG	DURABILITY AND ENGINEERING	SEP 59 TO MAY 60	1	4,725	GENERAL PERFORMANCE SATISFACTORY - COLD START DOWN TO -25°F SATISFACTORY - KIT ITEMS PERFORMED SATISFACTORY, HOWEVER, REQUIRES MINOR REDESIGN.
PRODUCTION (KIT MATERIAL) • ARMOR BOARD • FORT CAMPBELL • INFANTRY BOARD • AIRBORNE BOARD	USER EVALUATION	JAN 60 TO MAY 60	5 BOARDS 28 FIELD	10,732 NOT AVAIL.	THE MODIFICATIONS HAVE SIGNIFICANTLY IMPROVED THE DURABILITY AND RELIABILITY. CORRECTION OF DISCREPANCIES AND TEST VERIFICATION OF SAME SHOULD MAKE THE VEHICLE SUITABLE FOR ARMY USE.
PRODUCTION (KIT MATERIAL) • USMC	USER EVALUATION	OCT 59 TO MAY 60	6	11,047	THE KIT IMPROVES ENGINE STARTING, PROLONGS ENGINE LIFE AND IMPROVES VEH. PERFORMANCE AND IS ACCEPTABLE FOR MARINE CORPS VEH.
PRODUCTION (VEH.) • APG	ICT	OCT 57 TO MAY 59	6	6,000	TO CHECK QUALITY LEVEL OF PRODUCTION. NOTE: THESE WERE (6) SEPARATE TESTS. NONE ARE SCHEDULED FOR CURRENT USMC PROD. BECAUSE OF LACK OF FUNDING.

Figure 4-3. M-274 Vehicle, Ordnance Test Summary

TYPE	NUMBER	DATE	COST	REQUIREMENTS
R & D	35522	JUN 1952	$392,506	CONCEPT STUDY AND DRAWINGS NOTES ON DEVELOPMENT TYPE MATERIEL 5 PILOT CARRIERS
R & D	37165	JUN 1954	$28,085	DEVELOPMENT COMPONENTS SPARE PARTS ENGR. TEST FOLLOW-UP
R & D	37213	JUN 1954	$35,600	ENGINEERING LIAISON DRAWING REVISIONS SPARE PARTS ENGR. TEST FOLLOW-UP
R & D	1915	JUN 1955	$85,747	ENGINEERING LIAISON DRAWING REVISIONS SPARE PARTS FURNISH REVISED COMPONENTS ENGR. TEST FOLLOW-UP
PROD.	2217	JUN 1956	$6,420,968	FURNISH 1,747 VEHICLES SPECIAL TOOLING FOR PRODUCTION KITS AND CONCURRENT SPARE PARTS
VEA	2216	JUN 1956	$895,929	ENGINEERING SERVICES FURNISH ORD. DRAWINGS FURNISH FOUR (4) PRE-PRODUCTION VEHICLES FURNISH TWO (2) SPARE ENGINES ONE (1) SET OF SPARE PARTS FOR TEST SUPPORT
VEA	2536	JUN 1957	$610,890	PROVIDE ENGINEERING SERVICES GENERAL VEHICLE PRODUCT IMPROVEMENT ENGR. OBSERVER SERVICE MAINTAIN ENGR. RECORDS MAINTAIN ORD. DRAWINGS
PROD. (USMC)	3125	JUN 1959	$2,565,481	FURNISH 705 VEHICLES PILOT LOT TEST ITEMS CONCURRENT SPARE PARTS

Figure 4-4. M-274 Vehicle, Contract Summary

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5.0 MATERIALS ANALYSIS

5.1 General

An investigation was made of the effects of the lunar environment upon the materials of which the current vehicle is composed and those materials involved in the vehicle modifications. This materials information is presented in detail in Vol II - Part 2, APPENDIX, Section 2.0, MATERIALS, of this study report. The general effect of each parameter of lunar environment was considered.

5.2 Elimination of Existing Vehicle Components

Use of the material presented in the Appendix has been made in discussing elimination of certain existing vehicle components from the modification orders. In particular, reference is made to the discussions in Section 7.0, MULE VEHICLE MODIFICATIONS, in which the existing internal combustion engine and pneumatic rubber tire were considered.

5.3 Component Modification

The materials discussions in the Appendix consider materials use in components selected for vehicle modifications. Lubrication materials, organic materials, seals, wire and insulation, optical materials, insulation and heat transport fluids are among those discussed.

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6.0 OPERATIONAL TERRAIN ANALYSIS

6.1 General

The operation of a vehicle on the lunar surface is dependent upon the terrain and the vehicle response reaction to the terrain. The essential terrain information includes topography and trafficability. Topographic data needed are the slopes that have to be climbed or descended in traverses; the walls, crevasses, craters, and large obstacles that have to be detoured; the sizes and frequency of distribution of obstacles that must be negotiated; and surface roughness that affects wheel passage. Data needed for trafficability estimates are the engineering properties of soil at the wheel-soil interface that affect the mobility of the vehicle.

6.2 Topographic Data

6.2.1 Slopes

The work statement instructs that the study would utilize the topographic data contained in the ELMS report with the additional instruction that slopes greater than 15 degrees would not be utilized. The ELMS report presents a profile of a representative traverse. The slopes, both positive and negative, and the percent of total traverse distance of each slope is given. We have modified this typical traverse profile to eliminate slopes greater than 15 degrees. Table 6-1 contains the slopes, percentage of total traverse distance of each slope, and the total distance (of a 24 km traverse) of each slope in the traverse, respectively.

TABLE 6-1

Slopes, Lunar Model (Modified from ELMS Table 4-1)

TERRAIN PROFILES FOR A REPRESENTATIVE
24.14 km (15 mile) TRAVERSE
ON THE LUNAR MARIA SURFACE

Slope (degrees)	% of Traverse	Total Distance	K_ϕ	n	ϕ
0	11.0	2.655 km (1.6 mi.)	0.5	0.5	32°
-1°	13.0	3.138 km (2.0 mi.)	0.5	0.5	32°
+1°	9.5	2.293 km (1.4 mi.)	0.5	0.5	32°
-2°	12.8	3.089 km (1.9 mi.)	0.5	0.5	32°
+2°	11.7	2.824 km (1.7 mi.)	0.5	0.5	32°
-3°	5.5	1.328 km (0.8 mi.)	0.5	0.5	32°
+3°	10.5	2.534 km (1.6 mi.)	0.5	0.5	32°
-4°	3.0	0.724 km (0.5 mi.)	0.5	0.5	32°
+4°	7.0	1.690 km (1.0 mi.)	0.5	0.5	32°
-5°	3.0	0.724 km (.5 mi.)	1.0	0.75	32°
+5°	4.5	1.086 km (.7 mi.)	1.0	0.75	32°
-7.5°	1.8	0.434 km (.3 mi.)	1.0	0.75	32°
+7.5°	1.2	0.269 km (.2 mi.)	1.0	0.75	32°
-10°	.6	0.145 km (.1 mi.)	3.0	1.0	32°
+10°	1.2	0.270 km (.2 mi.)	3.0	1.0	32°
-12.5°	0.72	0.182 km (.1 mi.)	3.0	1.0	32°
+12.5°	0.48	0.126 km (.1 mi.)	3.0	1.0	32°
-15.0°	1.55	0.384 km (.2 mi.)	3.0	1.0	32°
+15°+	0.95	0.241 km (.1 mi.)	3.0	1.0	32°
	100.00	24.14 km (15.0 mi.)			

K_ϕ = Modulus of soil deformation due to frictional ingredients of soil (lb/inch n+2)

n = A dimensionless factor reflecting stratification of soil

ϕ = Angle of friction (between soil grains), degrees.

6.2.2 Walls, Crevasses

Though not directly affecting vehicle operation on the surface, as they would be detoured, these terrain features are not defined in the criteria. However, a discussion of these terrain features and how they probably will affect traverse average distances, which must be travelled to go from point-to-point, is discussed in Vol II - Part 2, APPENDIX, Section 3.0, TERRAIN ANALYSIS, of this report.

6.2.3 Obstacles

The work statement defines no obstacle sizes or their frequencies of distribution for the lunar surface. It required only an increase of the total power, required for the vehicle to negotiate the representative traverse, by 20 percent as an allowance for obstacle negotiation.

6.2.4 Surface Roughness

Surface roughness is a factor which affects vehicle operation. It is a combination of obstacle representation and slopes with distance or separate factors of each. A distance in this sense would be considered small, in units which are orders of vehicle length and wheel diameter. This specific information is not available for the lunar surface.

6.3 Trafficability

Locomotion of any vehicle in any medium is directly dependent on vertical forces called lift and horizontal forces called thrust or drag, if it is assumed that, in uniform motion, thrust is balanced by drag. These two force vectors have to be defined whenever a general expression of trafficability is sought. Characteristics of the lunar soil have been obtained from the ELMS report. These values are given in Table 6-1. The use of these soil characteristics values and the selection of the relevant vehicle values will enable a definition of a functional relationship between lift and thrust or drag, which definition is per se, the relationship between soil trafficability and vehicle mobility, (Reference 1).

6.4 Vehicle Speed

The work statement provides that the vehicle should be capable of speeds of at least 4.83 km/hr (3 mph) in the soft soil group ($K\phi = 0.5$, $n = 0.5$) and 5.44 km/hr (5 mph) in the hard soil group ($K\phi = 3$, $n = 1.0$) on level ground (zero slope). The vehicle prime power requirement has been determined for these two conditions. The prime power requirement has been found to be that necessary to maintain the required vehicle speed in the soft soil group. Utilizing this maximum power availability, the speed over the ELMS model traverse slopes has been derived.

6.5 Terrain Analysis

A study was conducted to investigate the feasibility of using a particular terrestrial area for modified vehicle testing. The analogies between known and suspected aspects of the lunar terrain and the Yuma Arizona Proving Ground are discussed in Vol II, Part 2, APPENDIX, Sect. 3.0, TERRAIN ANALYSIS, of this report.

6.6 REFERENCES

6-1 Bekker, M. G., "Off-the-Road Locomotion", Research and Development in Terramechanics, University of Michigan Press, Ann Arbor, Michigan, 1960.

7.0 MULE VEHICLE MODIFICATIONS

7.1 GENERAL

The M-274 "Mule" vehicle has been subjected to three orders of modifications: first order, second order and third order.

The first order, Mod 1, is the minimum cost modifications necessary to the basic vehicle, the Mule, to make it capable of operation on the lunar surface within the study guidelines and constraints.

The second order, Mod 2, is modifications to the basic order vehicle which provides somewhat better mobility characteristics to the modified vehicle than the first order modifications and at a modest increase in cost over the first order.

The third order, Mod 3, of modifications represents the maximum extent of modification which can logically be made and still retain any portion of the basic vehicle structure.

7.2 MAJOR COMPONENTS REVIEW

A review is presented of the basic vehicle's existing major components. Technical Manual references are cited in the subsection headings.

7.2.1 Engine, Fuel, Electric, and Controls TM9-2805-211.35P

The engine is a 4-cyl. 4 stroke cycle, air-cooled, internal combustion engine, 53 cu in displ, 16 hp @ 3200 rpm, and has 30 lbs torque @ 2100 rpm. The drive train is sized to this power input.

7.2.2 Front Axle Assembly - Ch 4, TM9-2320-213-34

The multiple piece cast housing containing a right angle spiral bevel gear drive, two axle shafts, two 3-gear drop cases, two constant velocity universal joints, and bearings. The unit from hub to hub is water proof. The front axle has an externally-mounted, unsealed internal expanding brake. Spiral bevel gearing incurs heavy unit pressure loadings and sliding contact.

7.2.3 Rear Axle Assembly & Transmission - Ch 5, TM9-2320-213-34

The rear axle housing contains the same components as the front axle; however, the center section is greatly enlarged and contains a three-speed manual transmission and two-speed transfer case. The rear axle assembly is also sealed against water. The assembly has one output shaft to the front axle and three linear motion input shafts for gear changing purposes.

7.2.4 Steering Gear & Linkage - Ch 3, TM9-2320-213-34

The steering gear is a manually-operated cam and lever type. The gear is sufficiently sealed to prevent loss of a heavy SHE 90 gear lube. Unit loadings between cam faces and lever is high.

7.2.5 Frame & Body - Ch 6, TM9-2320-213-34

Frame components bolted to the axle assemblies combine to form the vehicle frame. The flat bed cargo deck is then bolted to the frame. Materials used are magnesium and aluminum extrusions and steel.

7.2.6 Miscellaneous Components - Ch 7, TM9-2320-213-34

7.2.6.1 Engine Guard. Welded assembly of steel tubing and expanded metal grate.

7.2.6.2 Driver's Seat. Steel Frame, canvas-covered cotton and rubberized hair cushions.

7.2.6.3 Footrest Assembly. Welded steel tubing assembly complete with gas, brake and clutch pedal assembly. Several unsealed plain bearings are used.

7.2.6.4 Propeller Shaft. Drawn steel tube with forgings welded to each end to carry conventional cross-type universal joints. The joints are sealed against water.

7.2.6.5 Brake & Shift Lever Support Assembly. Exposed forged and stamped metal components, unsealed plain bearings. Mechanism is to facilitate control of vehicle.

7.2.6.6 Engine Starter Assembly. Manual recoil starter, automatic rewind.

7.2.6.7 Fuel Tank. Fuel tank is of a welded and soldered steel construction.

7.2.6.8 Wheels & Tires. Wheels are cast magnesium, drop center type 10 x 5.50 F. Tires are of conventional construction 7.50 x 10 size, overall diameter 25.3. The tires are used with a tube.

7.3 MAJOR COMPONENTS NOT SUSCEPTIBLE TO PRACTICAL LUNARIZATION

The existing subsystems and components were evaluated in detail as to the practicality of their lunarization. The constraints and guidelines stress simplicity and cost. Any of the existing subsystems and components could be modified to operate on the lunar surface. It is a matter of whether it is practical to do so. It was concluded that the prime power unit (the internal combustion engine) and the pneumatic tire were the two major components on the existing vehicle not susceptible to practical lunarization. Substitute components were selected on the basis of cost and efficiency.

7.3.1 Pneumatic Tire and M-274 Wheel

The M-274 vehicle is normally furnished with wheel and standard pneumatic rubber tire as shown in Figure 7.3-1. This is a 4 ply type, 7.50 x 10 size with the all-service cross-country tread. The characteristics of this tire are set forth in the military specification Mil-T-467529. The specification requirements do not include the elastomeric materials to be used, but only state the minimum composition of 75% natural rubber hydrocarbon for natural rubber base tires, and 87% total synthetic elastomer hydrocarbon for synthetic elastomer tires. The specification requirements are established to assure operational suitability over the environmental temperature range from 220°K (-65°F) to 322°K (120°F). Internal operating temperatures should in any event be less than the maximum noted in racing tires, that is 405°K (270°F).

An ultra-wide flotation tire is commercially available for the M-274 vehicle, pressurized normally at 0.276 bars gauge (4 psi), and used in very soft soils at internal pressures one half this value. This tire produced by the Goodyear Tire and Rubber Company is known as the "Terra-Tire".

The military tire is fabricated from natural rubber, or synthetic elastomers, such as a butadiene-styrene copolymer or isobutylene diolefin, see Para. 2.1.8, sect. 2.0, Vol. II - Part 2, APPENDIX for a complete discussion.

Properties of selected elastomeric materials are presented for comparison in Table 7.3-1.

To produce a tire the elastomers are compounded with carbon black, vulcanizing agents (sulfur and its compounds) and vulcanizing accelerators and antioxidants.

Natural rubber is a preferred material for the treads because of its excellent resistance to abrasion and tear. Butadiene-styrene copolymers are also used in tread formulations, but the tear resistance is not as good as that of the natural rubber; however, these elastomers are more generally used in tire sidewalls.

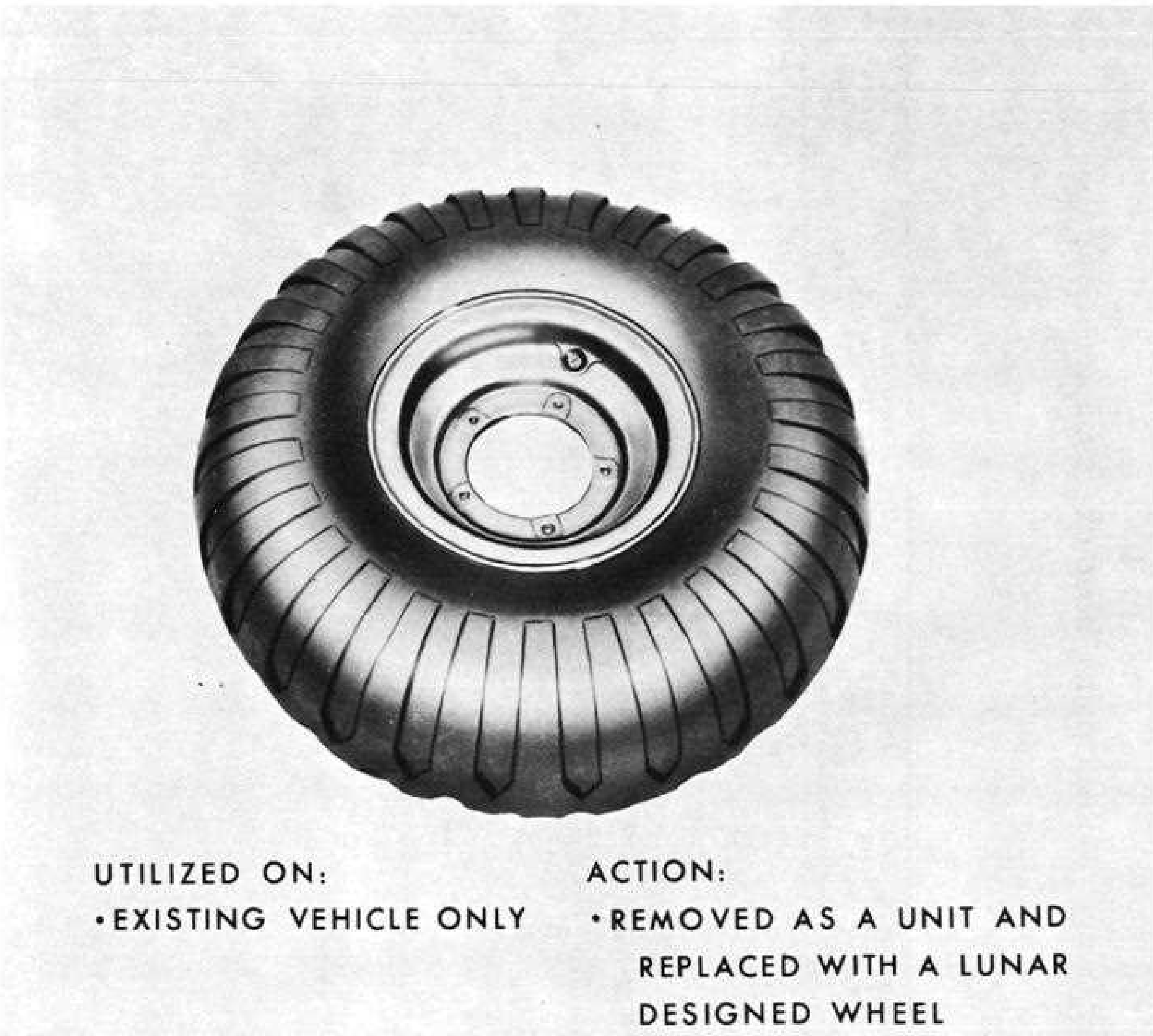


Figure 7.3-1. Existing Wheel and Tire Assembly

Table 7.5-1

ELASTOMER CHARACTERISTICS

Elastomeric Material	Cold Brittle Point °K	Cold Brittle Point °C	Cold Brittle Point °F	Tensile Strength At 298°K (76°F) Kilonewtons/cm²	Crescent Test Tear Resistance Newtons/cm.	Radiation Dose For Threshold Damage - Rads	Temperature For 10% Mass Loss in 1 yr* °K	Temperature For 10% Mass Loss in 1 yr* °F	Max. Rated Use Temp. °K	Max. Rated Use Temp. °F	Gas Impermeability	Abrasive Wear Resistance
Cis-isomeric Natural Rubber (Hevea)	219	-54	-65	3.08	1,220	2×10^6	465	380	380	225	Fair	Excellent
Butadiene-Styrene (GR-S)	213	-60	-76	2.74	635	2×10^6	510	460	394	250	Fair	Good
Polychloroprene (Neoprene)	233	-40	-40	2.30	735	2×10^6	366	200	394	250	Good	Very Good
Isobutylene Diolefin (Butyl)	233	-40	-40	2.05	735	2×10^6	394	250	421	300	Very Good	Good
Dimethyl Siloxane (Silicone)	193	-80	-112	0.55	78	1.3×10^6	477	400	590	600		
Butadiene Acrylonitrile (Nitrile)	235	-38	-36.4	2.74	490	2×10^6	421	300	421	300	Fair	Good
Polyurethanes (Adiprene)**	185	-88	-125	3.5 to 5.5	165 to 270	(Possibly greater than above)	***	***	366 to 394	200 to 250		(Presumed Good)

* At 1.3 nanobars pressure

** Trade Mark of the "DuPont" Polyurethane

*** Above 365°K (205°F) considered upper limit

Polyurethane polymers show promise for meeting the requirements for use in the lunar daytime environment as noted in Table 7.3-1. The polyurethane elastomers have low outgassing rates, and are resistant to x-ray radiation. Tires have been fabricated using the polyurethane elastomers, see para. 2.1.8, sect. 2.0, Vol II, Part 2, APPENDIX for a complete discussion. However, the performance at temperatures above 370°K (207°F) have not been noted as entirely satisfactory and difficulty has been experienced in making a fabric reinforced tire body with these elastomers.

The study constraints do not explicitly specify the range of temperatures anticipated in the proposed one (1) day-time lunar mission (14 earth days). However, in this 14-day (336 hour) period, the critical environments with respect to low temperature will occur in the times closely following sunrise, and closely preceding sunset. At these periods, the surface temperatures will either rise through, or fall below the 220°K minimum operational temperature for the military tire. But more important, at these hours of twilight, vertical surfaces facing away from the sun will act as radiators to space. An elastomeric tire then would, in time, equilibrate to the "effective sink temperature". If it faced away from the sun, this effective sink temperature, as has been calculated in reference 1 for a particular radiator surface, would be considerably lower than the 220°K lower limit for the military tire.

Figure 7.3-2 shows the results of effective sink temperature calculations for a vertical, cylindrical radiator, having a surface α/ϵ ratio of about 0.21. In this figure the angular positions of the sun of interest, insofar as the minimum equilibrated surface temperature of a tire exposed in the lunar environment is concerned, are from 90° to 270° of arc, that is facing away from the sun. In this range of orientation, the α/ϵ ratio is no longer controlling, since the value of α is for solar radiation rather than the long wave radiation reflected from the lunar surface.

From Figure 7.3-2, it is shown that the equilibrated surface temperature (i.e., effective sink temperature) would be exemplified by the range, as lunar sunset progresses into night, from 180°K (-135°F) down to 80°K (-315°F). For the sunrise conditions, for

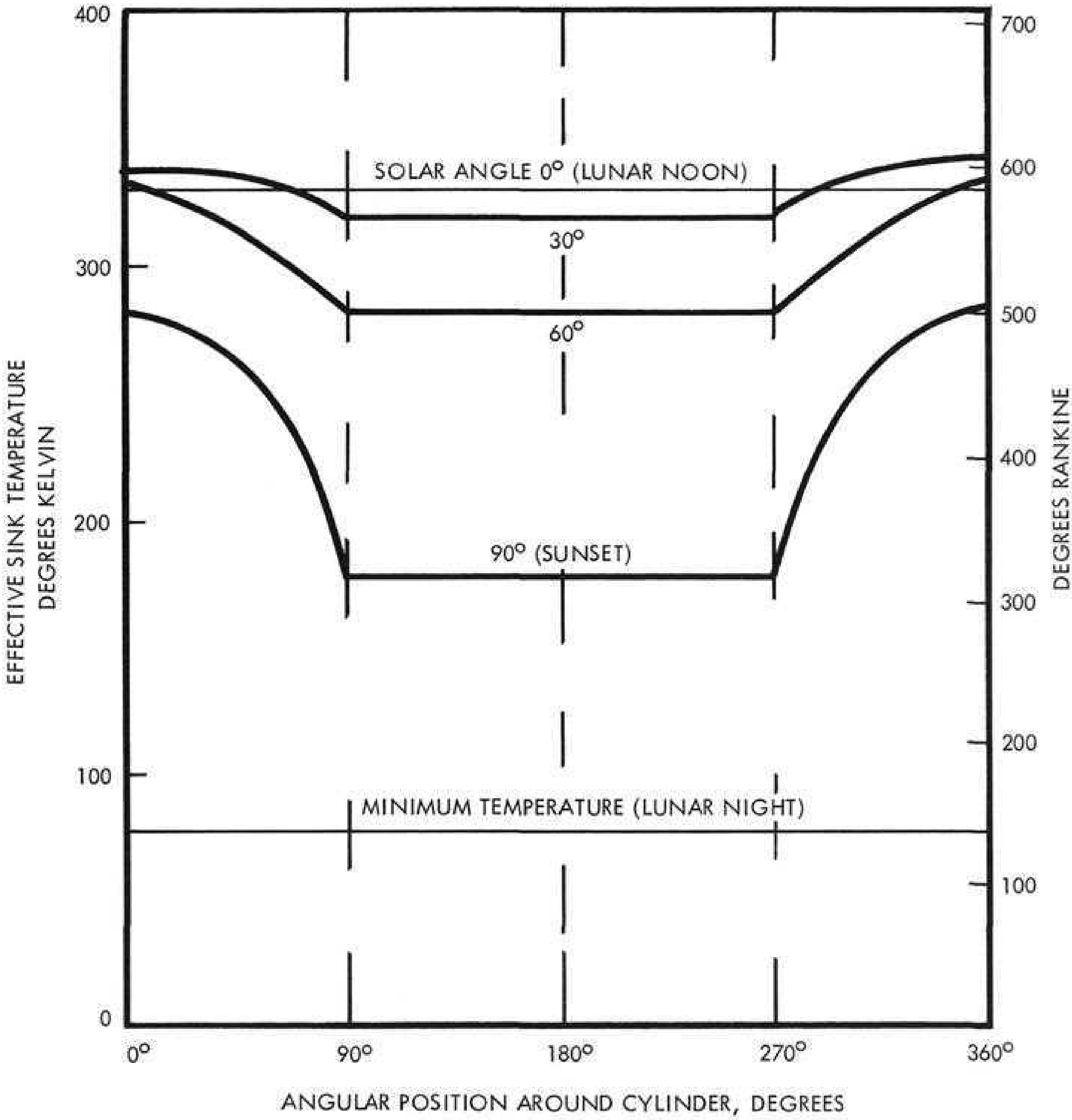


Figure 7.3-2. Effective Sink Temperature Distribution

the same solar angle over the horizon, the effective sink temperature would be somewhat lower than that at sunset, since the ground temperature will be lower. It is evident that the number of days available for vehicle use would be limited with a military type vehicle tire because: One, the surface operating temperature can be expected to be below the minimum for suitable tire flexibility; and Two, the only elastomer tire compound showing promise of suitable low temperature flexibility has not been satisfactorily used in a fabric reinforced tire. Therefore, development would be required to produce a satisfactory pneumatic tire, meeting all requirements.

Other problems with the polyurethane elastomers could concern the abrasion resistance and tear resistance of the compounded formulations. Since the ELMS criteria, used in this study, did not specify characteristic surface roughness, no attempt has been made to evaluate the feasibility of developing a compounded polyurethane elastomer from the standpoint of these lunar surface characteristics.

Additional environmental characteristics must also be considered. During the daylight portion of the lunation cycle, the sunlight will impinge on the lunar surface for the most part at oblique angles, thus causing shadows on sides facing away from the sun. At times near sunrise and sunset these shadows will be long, and will engender lower sink temperatures than the surrounding obliquely illuminated lunar surface and, if sufficiently extensive, would deny operations with a conventional military tire in the shadowed area.

In view of the operational limitations anticipated with attempted use of a conventional type military tire and the uncertainties with respect to undertaking a program to develop a tire, suitable over the spectrum of anticipated temperatures, the pneumatic tire was not considered suitable for the lunarized M-274 vehicle concepts.

7.3.2 Internal Combustion Engine - Dynamic Power Sources

The purpose of this discussion is to review the possibility of adapting the existing basic vehicle engine for use on a lunarized Mule vehicle and to consider other possible dynamic engine power sources.

7.3.2.1 Existing Engine. The existing engine utilized in the terrestrial version of the basic vehicle is the military model A042, which is an air-cooled, 4 cycle, 2 cylinder gasoline engine of 44 cubic inch displacement with a rating of 15.2 horsepower at 3200 rpm. A general view of the engine is shown in Figure 7.3-3. This engine is a result of a six years development program and an expenditure in excess of $4,000,000. Preliminary calculations of the lunarized vehicle power requirements for the first order modification indicated a maximum need of 2.5 horsepower. Use of a wide tread rigid wheel reduced this to 1.75 horsepower. The existing engine develops over 15 horsepower. Even if it were lunarized, it would be considerably overpowered for use on the lunar vehicle and its retention would, therefore, result in an unnecessary weight and space penalty on the vehicle. The oversized engine would also be operating at a low power level which results in poor efficiency. Therefore, a higher fuel consumption would result than with a smaller properly sized engine. For these reasons, the A042 engine is not considered compatible with the lunar vehicle requirements.

7.3.2.2 Otto Cycle Engines. After elimination of the existing engine because of the overpower situation, an Otto cycle engine of the proper power can be selected for analysis. The selected engine is the military model 2A016 gasoline engine, 4 cycle, 2 cylinder, 16 cubic inch displacement, air-cooled with a rating of 2.8 continuous horsepower and 5 horsepower maximum at 3200 rpm. This engine horsepower rating would be more appropriate to the modified vehicle's power requirements. A photograph of this engine is shown in Figure 7.3-4. Even with a properly sized Otto cycle engine the adaptation of the engine for lunar use presents a number of major problems:

- Cooling System - The 2A016 engine is designed for air-cooling. An air-cooled engine cannot be utilized in the lunar environment because of the absence of an atmosphere. Redesign of the cooling fins to allow heat rejection by radiation transfer only is considered unfeasible since there is a widely varying temperature gradient between the heat and the crank case of the engine. The attempt at direct radiation cooling would involve a major redesign of the entire engine assembly. The practicality and effectiveness of such modifications are highly questionable. A more direct solution to the cooling

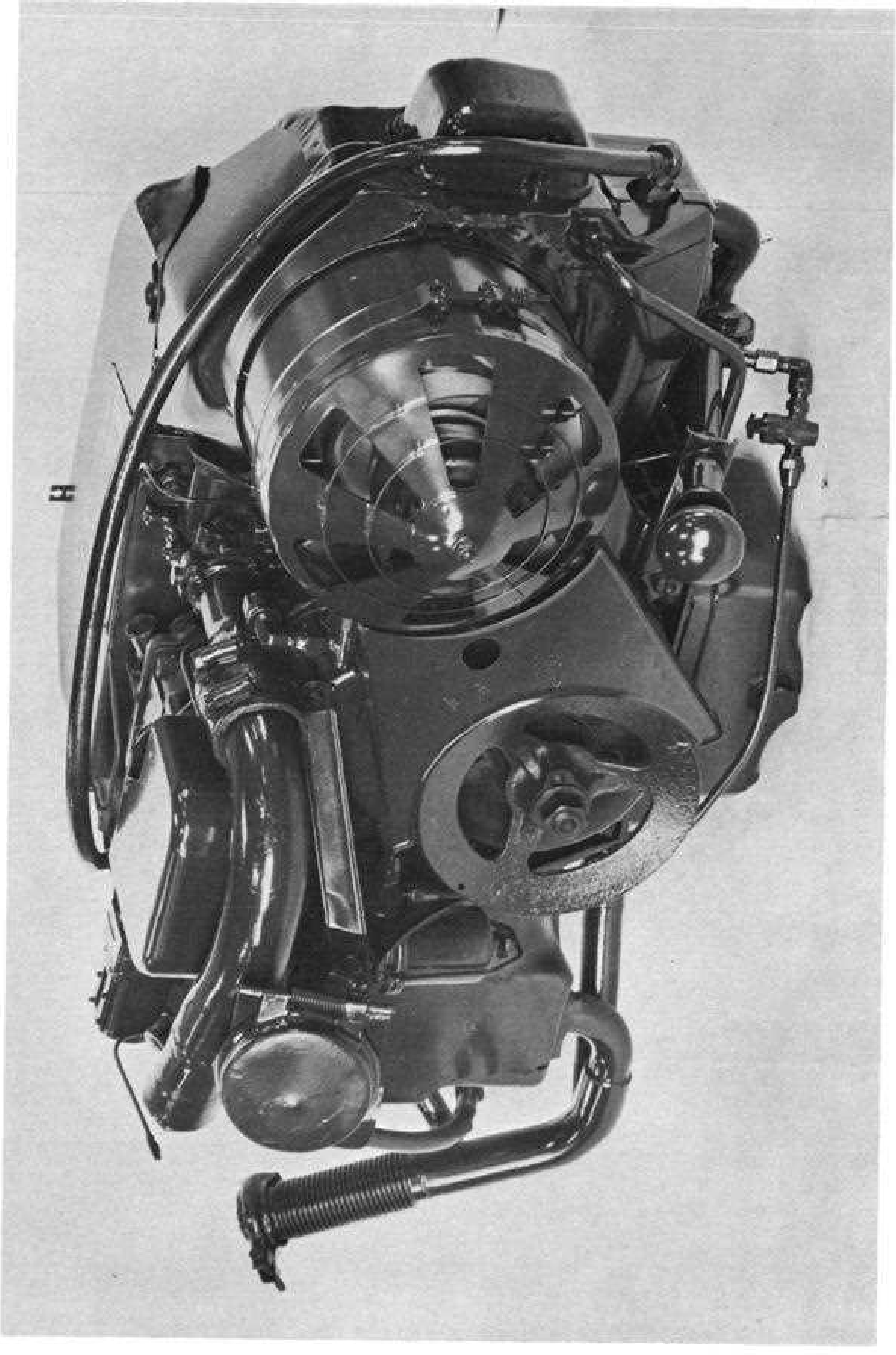


Figure 7.3-3. A042 Military Standard Gasoline Engine

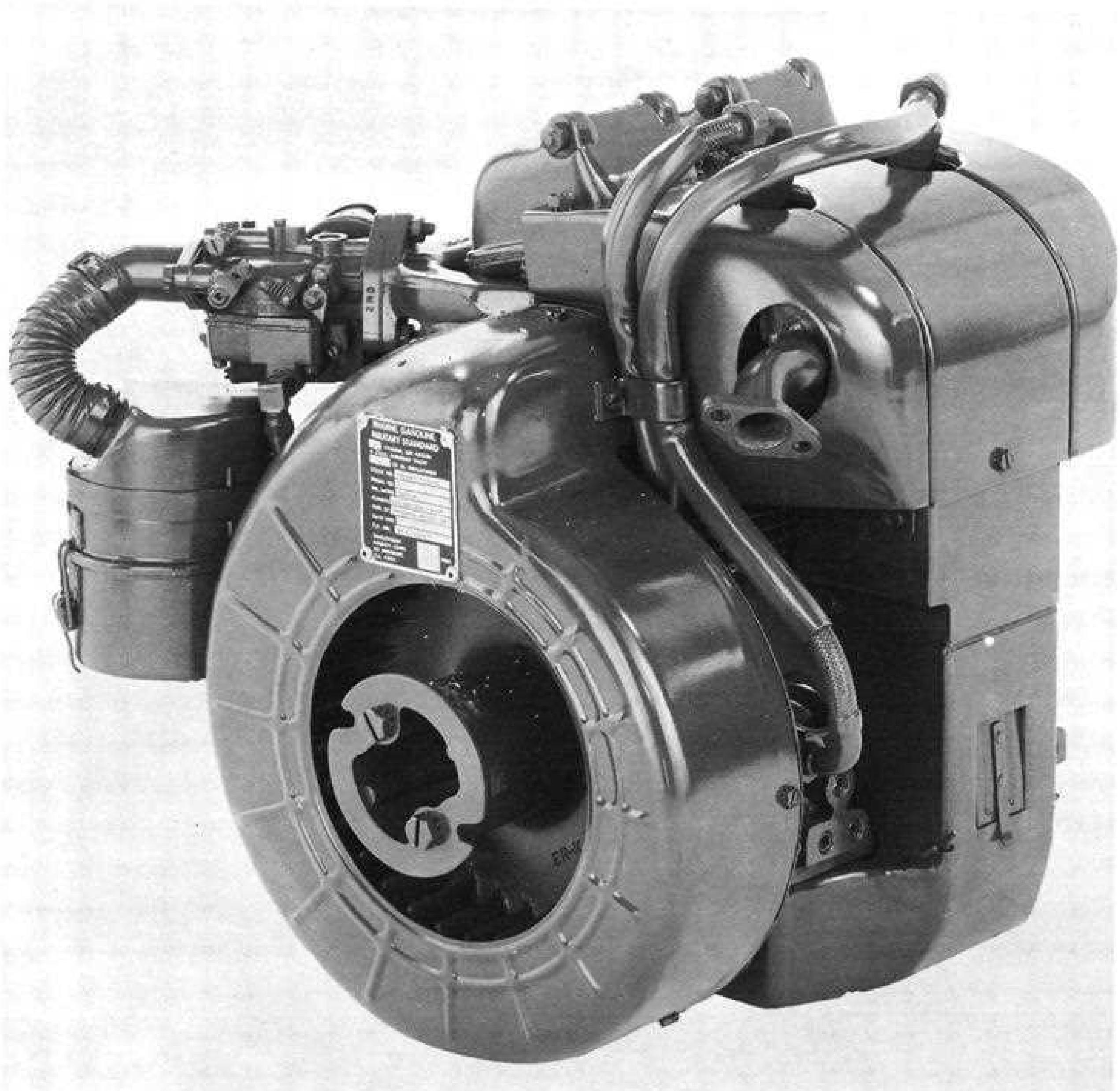


Figure 7.3-4. 2A016 Military Standard Gasoline Engine

problem would be to use a small liquid cooled engine which could be directly coupled to an external radiator system. Although small horsepower engines have been recently manufactured with liquid cooling, the production has been very limited and background test information is not readily available. By utilizing a liquid cooled engine, the cooling problem associated with engine modifications would not be too extensive.

- Lubrication System - Conventional petroleum lubricants could be utilized in a lunarized dynamic engine only if the oil could be maintained within the required limits of 233°K to 394°K (-40°F to +250°F). Methods would have to be provided to supply heat to the crank case and all other lubricated parts in the engine to maintain this temperature while in a running or static condition. Coolant coils would be required to prevent excessive temperature rises. New types of synthetic oils increase the temperature performance range but would still not equal the temperature range expected on the Moon. In addition, the crank case of the engine would require a dynamic sealing system in order to prevent evaporation of the lubricants to the lunar vacuum. Dry film lubricants such as graphite-moly or iodine based lubricants may, in the future, prove to be usable on short-life engines. Development of a reliable design or dry lubed engine would be costly and time consuming with no guarantee of success. Although the lubrication system presents a problem, it could be solved without major engine redesign.
- Fuel System - The normal Otto cycle engine utilizes gasoline as a basic fuel. Lunarization of the engine would require that the standard carburetor be replaced by a fuel injection system or other means of maintaining pressure on the gasoline. The vacuum prevents the effective use of the carburetor. Heating systems would be required to keep the gasoline fluid at the low lunar temperatures, as its' freezing point is 213°K (-76°F). In addition, protection must be provided for the fuel tank and lines against excessively high temperatures resulting from solar radiation or other heat inputs as the fuel will boil and vapor lock the fuel system at temperatures around 367°K (+200°F). Utilization of gasoline is not considered as effective for this

application. The energy release rate being considerably lower than some of the more exotic fuels, results in high fuel consumption and the necessity of transporting excessive amounts of gasoline. A substitution of some of the more high energy fuels could be contemplated; however, this results in a completely new engine design.

- Induction System - The induction system for the conventional engine is composed of a simple air filter in combination with an intake manifold and carburetor assembly. Because of the absence of an atmosphere, an oxygen supply would be required for a lunarized engine. This oxygen could be supplied as a cryogen with necessary vaporization systems. The supply of pure oxygen to an internal combustion engine is not feasible, however, since the resultant combustion temperatures become excessively high and result in melting of the engine components with only a few minutes of operation. On earth, the nitrogen in the atmosphere acts as a diluent to the oxygen, lowering actual combustion temperatures. A pressurized diluent can be included in the fuel system; however, the use of a nonrecovered diluent-oxygen mixture for an engine of this size results in excessive tankage requirements. A more logical system would be to provide an initial diluent supply and recirculate the exhaust gas to allow reuse of the diluent. This requires a very large radiator system to handle the gas flow and produce sufficient cooling to arrive at essentially the same volume as that prior to combustion. The exhaust recirculation system also requires a filtering system to eliminate diluent contaminants produced during combustion. The exhaust recirculation with its requirements of a large radiator and filtering system is considered unfeasible and is the major reason for the elimination of the Otto cycle engine.
- Exhaust System - The exhaust system of the conventional Otto cycle engine is normally left open to the environment and would have to be sealed for lunar use. If left open to the vacuum, internal pressure within the engine (crank case, combustion chamber, etc.) would be lost and the resultant vacuum could produce cold welding of the valve train components, piston rings, cylinder walls and all other major parts subject to high loading. The

exhaust recirculation system discussed above would eliminate this problem but contains its own development problems.

- Seals - Static and dynamic seals will be required on any dynamic engine or electric motor system; this is a general problem of sealing moving parts and is no greater a problem for the existing engine than for any dynamic system. It is anticipated that adequate seals could be developed for the required short mission.
- Ignition System - Present ignition system components external to the cylinder interior are designed dependent upon the electric breakdown characteristics of atmospheric air. When placed in a vacuum situation, there are many unknown factors regarding the operation of a conventional magneto of a battery ignition system. The voltage requirement to jump a spark in a vacuum is approximately 20 million volts per inch of gap. Since present ignition distributors utilize a jump gap system for distribution of high voltage, they would be inoperative in the vacuum high voltage gradient. Redesign of the distribution rotor for zero clearance would be required. The breaker points cannot be expected to operate in a vacuum because of thermal problems, and some type of inert gas pressurization would be required. It is, therefore, expected that the standard ignition furnished on the engine would require major modification and development before any degree of reliability could be assured for lunar operation.

The previous discussions have been related to the four cycle engine concept. The two cycle engine would have essentially all of the same problems and required development as for the four cycle engine.

The pressurization and crankcase lubrication would be simpler problems, since the two cycle engine utilizes a fuel lubricant mixture drawn through the crank case. In exchange for this slight gain an additional problem is introduced. The two cycle engine is subject to carbon or combustion product deposition on the intake and exhaust gas ports. The operating time between port cleaning for terrestrial applications is approximately 75 to 100 hours. When this is combined with the exhaust diluent recirculation system modification with a high contamination content, the port cleaning problem may be expected to increase.

The development cost for only the general Otto cycle modifications indicated above for lunar use is estimated at 3 million dollars. The end product would probably work, but the reliability would be questionable, the fuel consumption excessive, and the weight and size larger than other power concepts basically more aligned to operation in a lunar environment. The Otto cycle engine was dropped from consideration at this time. Consideration was given to other dynamic engine concepts.

7.3.2.3 Non-Conventional Dynamic Power Sources. These are many non-conventional dynamic engines which can be considered for the modified vehicle. The term non-conventional, as applied in this case, refers to engines which have been developed and run primarily as prototypes or test units without extensive manufacturing and field use experience. They include reciprocating external and internal combustion engines operating on hydrogen-oxygen or hypergolic fuels, and engine concepts operating on the Sterling, Rankine or Brayton cycles. Generally, these types of engines may be considered to have the major disadvantage of high fuel consumption and low reliability as compared to the static power sources. Their chief advantage is high specific power per unit volume of package size which may be particularly useful when applied to such intermittent power requirements as a lunar roving vehicle. The trade-off requirements to determine the best power system for development depends on the exact nature of the vehicle mission, the power profile and the size and weight of the end item. The excess weight of fuel consumed may be compensated for on high power, short run applications because of the reduction in the initial weight of the power system, which can be accomplished by utilizing a dynamic engine. The lunarized mule vehicle power requirements are too low to provide a competitive advantage to most dynamic sources, while the mission time requirement could allow effective use of either the static or dynamic sources. Another major disadvantage is that the dynamic engine normally represents a single power source lacking in the redundancy which creates high system reliability. Static sources, on the other hand, are predominantly modular in nature and provide this redundancy.

7.3.2.3.1 Internal Combustion Reciprocating Engines Operating on Special Fuels— The feasibility of operating small reciprocating engines on hypergolic or hydrogen-oxygen fuels has been demonstrated by several companies. A single cylinder engine in the 6-7 horsepower range, using nitrogen tetroxide as the oxidizer and a 50-50 blend of hydrazine and unsymmetrical dimethyl hydrazine as a fuel has been demonstrated and run for something less than 100 hours. The prime problem with this type of engine has been excessive fuel consumption and lack of reliability. A 3 horsepower hydrogen-oxygen fueled engine has been under development for several years and performance and limited endurance tests have been conducted. These two types of engines suffer from most of the inadequacies that were listed for the Otto cycle engine; a prime improvement being the elimination of the fuel and oxidizer induction system. The same problems, crankcase lubrication, cold welding of internal parts and dynamic seals would still exist and require development. The state of the art in these engines has not sufficiently progressed at this time to allow their consideration for use in the modified vehicles.

7.3.2.3.2 Open and Closed Cycles—The open cycle engine system in which the exhaust products are released directly to the lunar vacuum is not the most economical system with respect to fuel consumption. For long mission durations (several times the specified study mission), the exhaust products could be condensed and methods derived to reconstitute the products to a usable fuel. In short range missions (as specified), the minimum treatment for a more economical operation would be to condense the exhaust product of a hydrogen-oxygen fueled engine (for use of the resultant product, water). Engines utilizing fuels that do not directly produce water or steam as the exhaust product, such as 50-50 hydrazine and unsymmetrical dimethyl hydrazine, would not lend themselves toward a closed loop system and would have to operate on an open cycle. The question of whether an engine system should operate on a closed or open cycle is dependent on the nature of the fuel; the complexity of reconstituting the exhaust products; the length and nature of the mission; and the weight and size limitations of the condensate and/or fuel reconstitution systems. On vehicles of the size-weight range needed for the modified vehicle, a fuel reconstitution system is not feasible. A fuel condensation system is feasible if the original fuel is hydrogen-oxygen; if contamination is not excessive; and if there is a valid use for the resultant product, water. If given water credit, the dynamic H_2-O_2 engine compares more favorably with the static power sources. On the basis of the limited weight and volume available or the lunarized Mule vehicle, the closed cycle concept of engine operation will no longer be considered and only an open cycle will be discussed. On some engine designs, the working fluid may be in a closed loop system but the fuel burning is still an open loop system.

7.3.2.3.3 Sterling Cycle Engine—The Sterling cycle engine basic concept actually predates the present day Otto cycle. It was originally manufactured as a stationary prime power source but was replaced by the compact, more reliable Otto cycle engine. Presently there is considerable interest and money (in excess of 5 million dollars) being devoted toward development of this engine both by the U.S. Army and by industrial firms. The primary U.S. Army interest lies in the engine utilization in a silent power system. Commercial objectives are both engine utilization in silent power systems and in development of an efficient low temperature refrigeration system. Preliminary tests have been completed by the U.S. Army on an experimental engine. These tests included a successful 500 hour endurance test. Figure 7.3-5 is a cross-sectional diagrammatic view of the construction and major components of the engine. A single cylinder houses the power piston and displacer piston combination. The piston rods are concentric and drive the geared crank shafts through a diamond crank mechanism which restricts the rods to a linear motion. A more complete explanation of the operation of this engine may be found in Reference 2. The engine could be supplied in many configurations: as an internal combustion cycle; as an external combustion cycle; in the normal configuration; or as a solar power system. The engine is liquid cooled and, in this case, has an external combustor and uses hydrogen as the working fluid in a closed inner loop system. The Sterling cycle engine would appear to be one of the better competitors to the static power sources. This engine efficiency can be as high as 21%. The improvement of the present engine reliability would be primarily a matter of redesign of mechanical components. The engine could utilize any available fuel in a properly designed combustor, exhausting directly

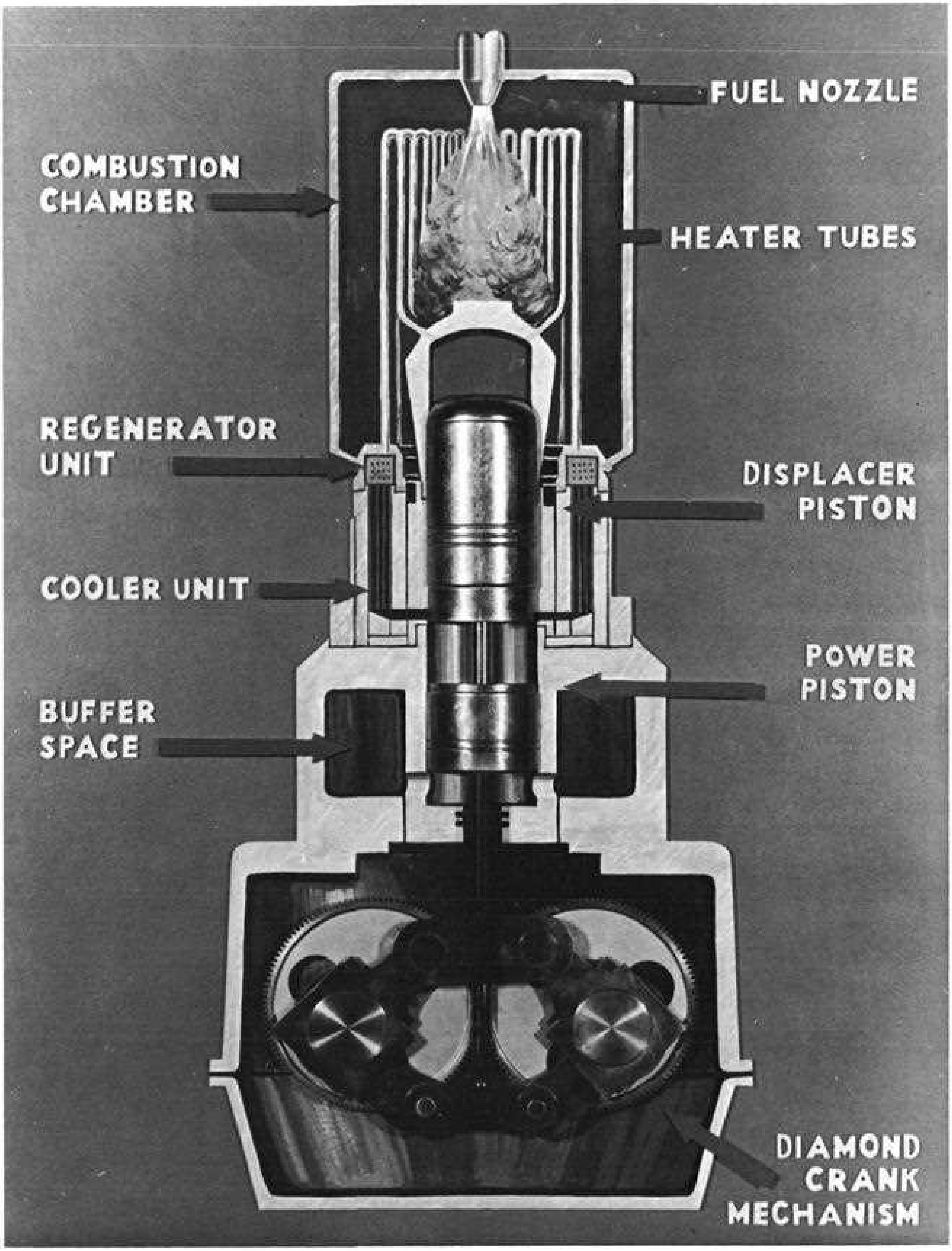


Figure 7.3-5. Sterling Cycle Engine

to space. Crank case lubrication and seal problems are essentially the same as in the Otto cycle engine. A three horsepower engine, of the same basic design illustrated, would weigh less than 100 pounds with all accessories; and would require about two years to develop at a cost of approximately 1.5 to 2 million dollars.

7.3.2.3.4 Rankine Cycle Engine—The Rankine cycle engine usually operates on the basis of externally heating some working fluid to the boiling point, then causing the vapors to pass through some mechanical energy converter such as a turbine wheel. The working fluid is then condensed, pumped back to the heat source for reheating and repeat of the cycle. As implied above, the Rankine has one or more closed heat loop systems within the engine system. The combustion loop is normally open, as would be the case for lunar vehicle application. Since heat energy is the only direct engine requirement, almost any type of fuel and oxidizer combination could theoretically be used. Gasoline or hydrocarbons and oxygen are again a poor choice because of the low energy release rate and the complexity of storing gasoline. The more ideal fuel, as with most dynamic engines, is hydrogen-oxygen; however, the stoichiometric mixture temperature of 2800°K (5500°F) is higher than combustor designs or materials can withstand. A diluent and diluent recovery system would be necessary for efficient operation. The other recourse, to lower combustion temperature, requires the use of excessive quantities of hydrogen. This procedure produces excessively high fuel consumption. A diagrammatic arrangement of a typical Rankine Cycle power unit is shown in Figure 7.3-6. The system illustrated uses mercury as the working fluid and has an electrical output rating of 3 KW. A 4.5 mechanical KW (6 horsepower), 24,000 rpm, single-stage turbine operated by the expanding mercury vapor drives a 3 KW alternator and mercury feed pump on a common shaft. The assembly is mounted in a hermetically sealed housing with no protruding shafts. Other principal components in the cycle are the condenser-subcooler module, the mercury preheater and the mercury vapor generator or boiler. The entire mercury loop is hermetically sealed after conditioning to prevent contamination from the environment or loss of working fluid. An external heat source, a liquid petroleum fuel burner in the system illustrated, furnishes the necessary heat for boiling the mercury. Several industrial firms have been working on development of Rankine cycle engines utilizing several

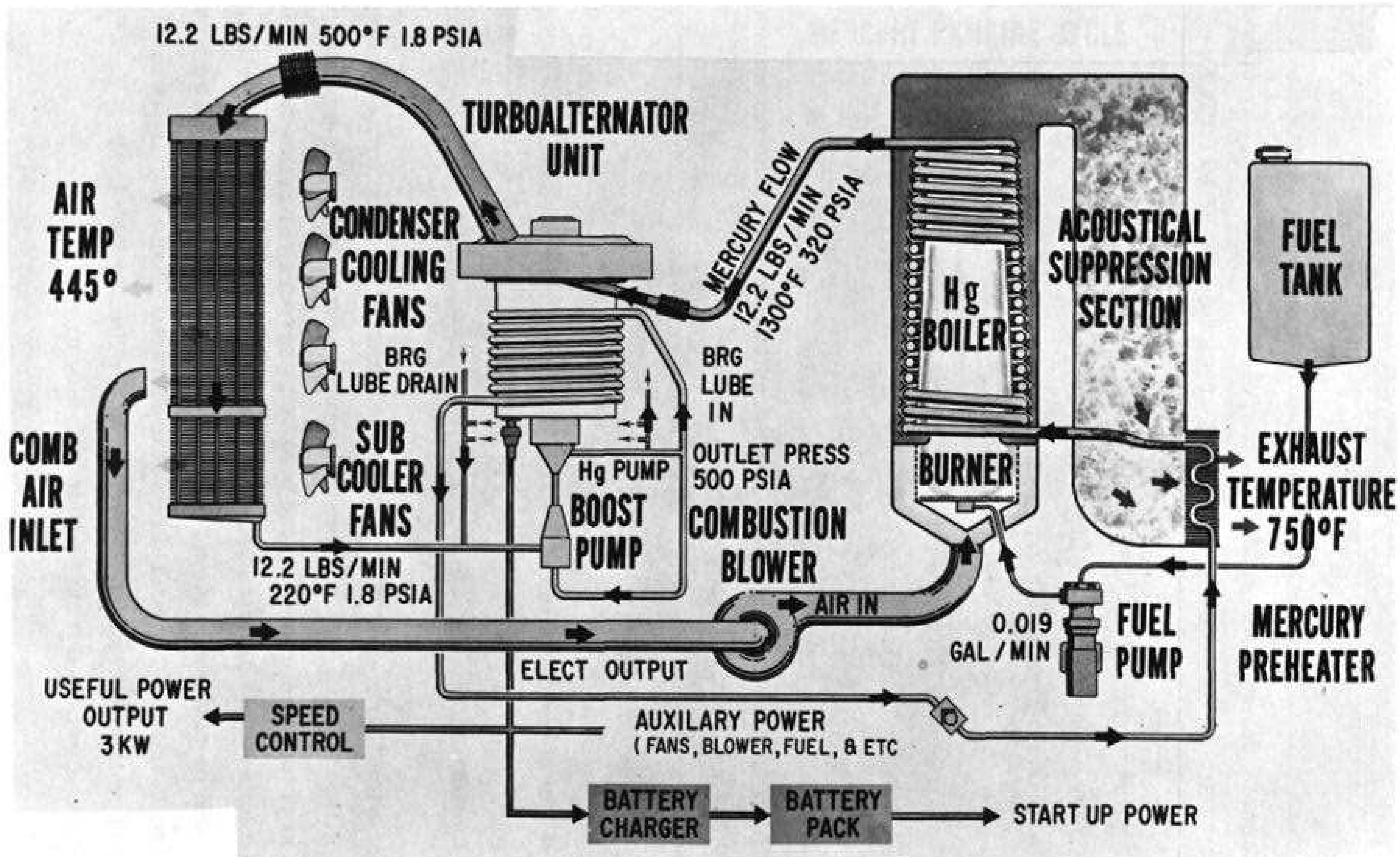


Figure 7.3-6. Mercury Rankine Cycle Engine

types of working fluids. The U.S. Army has been conducting evaluation tests on a 3 KW mercury Rankine cycle engine to determine its potential as a silent, ground power unit. NASA has substantially funded the development of a space qualified Rankine cycle, in the SNAP 8 power plant.

The Rankine cycle engine has a considerable development background and offers possibilities as a lunar roving vehicle power plant. Several problem areas caused this cycle to be eliminated as the selected power plant. These are:

- In comparison to the more efficient static sources, it would have a high fuel consumption which would become even higher when the engine is designed with a low horsepower (3 hp or less) rating.
- Non-modular construction lowers the system reliability.
- The high speed turbine (24,000 rpm) precludes the use of a transmission and requires an electric motor driven vehicle.
- Dynamic sealing of the output shaft to prevent leaking of high temperature working fluid would be difficult.

It is estimated that a 1.5 electrical KW (3 HP) Rankine cycle power supply would have a mass of 68 kg (150 lbs) and would cost 1.5 to 2 million dollars for the adaptation of a SNAP-8 type converter.

7.3.2.3.5 Brayton Cycle—The better known terrestrial Brayton cycle engines are the gas turbines utilized on aircraft and as stationary power plants. They operate on the basis of compression of air, fuel injection and constant pressure expansion through a turbine. These engines suffer from almost every disadvantage that the Rankine cycle has and, in addition, would be physically larger in size and of greater weight. The lower operating temperatures and/or pressures of the Brayton cycle usually result in considerably larger radiators than are required for the Rankine cycle. The Brayton cycle is therefore eliminated from further consideration.

7.3.2.4 Conclusions. Based on the above discussion it is concluded that:

- A lunarized version of the existing basic vehicle engine, or any other conventional internal combustion engine, is neither practical nor desirable due to the lack of previous space use modifications or development and the unknown factors of material compatibility, cooling requirements, lubrication, and fuel requirements with operation in the lunar environment.
- A Sterling Cycle engine appears to be potentially feasible as a prime mover for the modified vehicle in conjunction with mechanical transmission.
- A Rankine Cycle engine appears to be potentially feasible as a power source if an electric drive is considered for the modified vehicle.
- Because of high development costs, high fuel consumption and reliability considerations, the dynamic engines are not considered as the best possible selection for this particular vehicle application, where the vehicle operating power requirements are low.

Changes in the basic criteria, under which this study was conducted and to which the modified vehicle will perform, could affect a dynamic engine power versus static power system choice. Longer mission operating durations and higher power density requirements are examples of such changes.

7.4 SUMMARY OF VEHICLE MODIFICATION

7.4.1 General

To facilitate an understanding of the three orders of vehicle modification made in this study, a summary of each order of modification is presented initially. Later in Section 7.0, detailed discussions on each order of modification are presented. The information presented on each order of vehicle modification is divided into the following three major subsystem areas, according to the NASA format:

- Mobility and structures

- Power system, and
- Crew station

7.4.2 Existing Vehicle Configuration

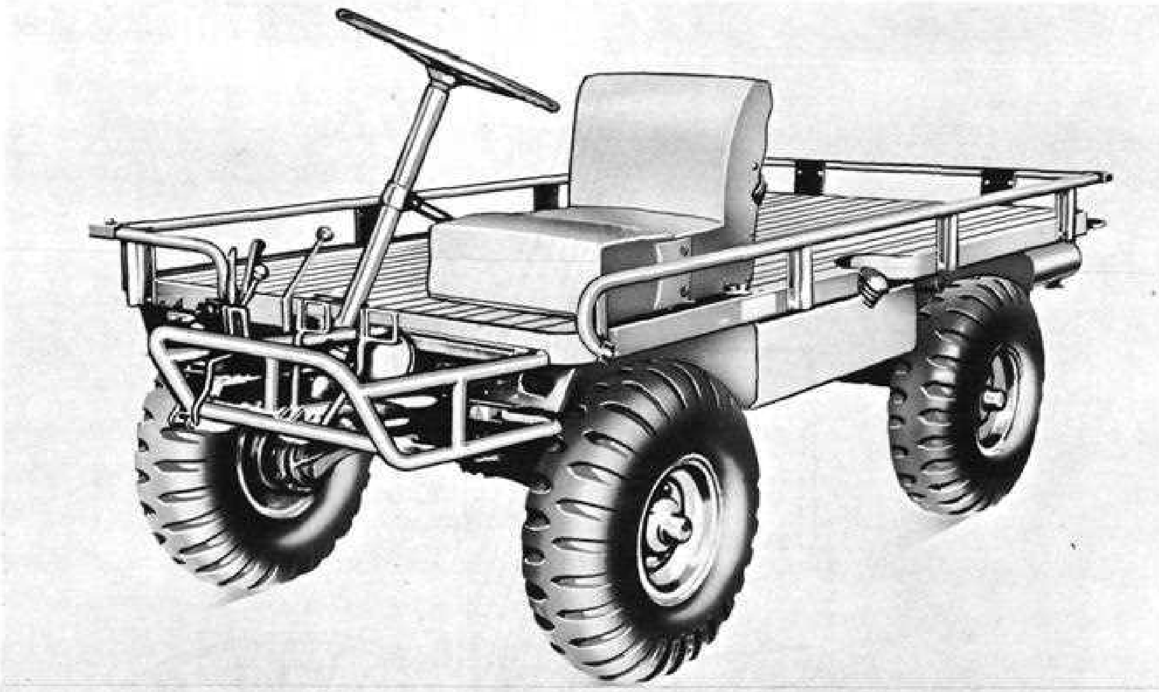
The configuration of the basic M-274 vehicle is given in Figure 7.4-1.

7.4.3 First Order Modified Vehicle

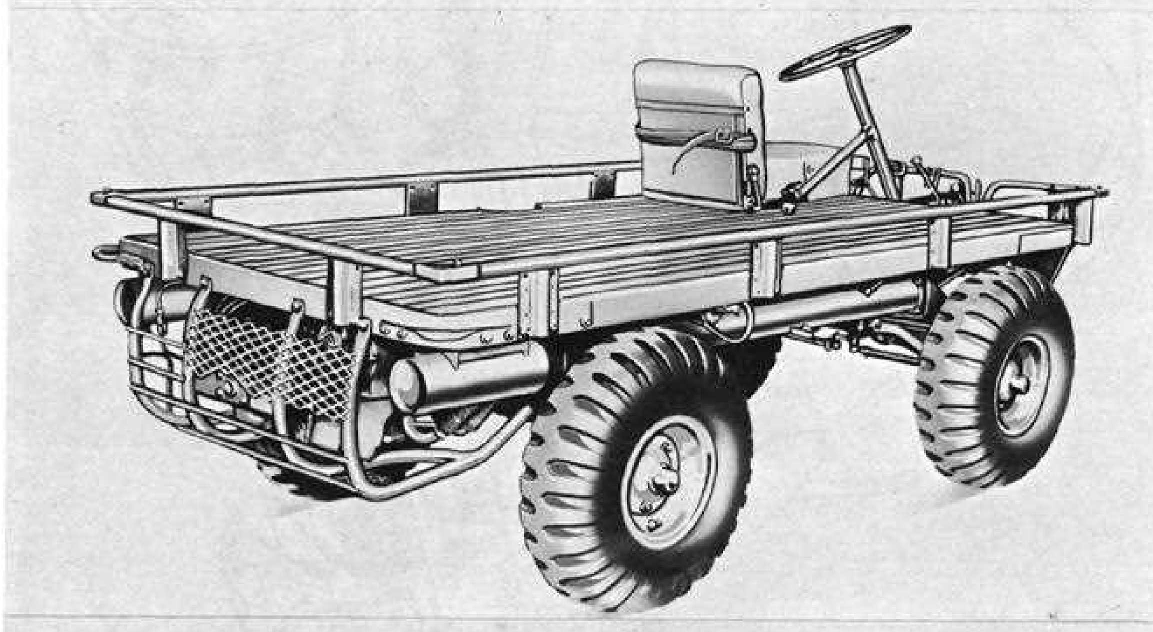
7.4.3.1 Vehicle Configuration. The configuration concept of the first order modified vehicle is given in Figure 7.4-2.

7.4.3.2 Modification Actions Taken. The modification actions taken in the first order are tabulated in Figure 7.4.3. The actions taken to the major components in the three major subsystem areas are explained by action code designations, as described in footnotes on this figure. The P code action, or replaced with new component action, refers to the two main items previously discussed, i.e., the internal combustion engine and the pneumatic-tired wheels. Referring to Figures 7.4-1 and 7.4-2, the following main changes to the basic vehicle are readily apparent:

- The pneumatic-tired wheels have been replaced with solid-disc metal wheels.
- The cargo platform rails have been eliminated. They are replaced by tie-down points.
- Typical cargo packages are shown on the platform. These packages are to scale with cargo package dimensions furnished in the criteria.
- Roll bars have been provided to protect the operator. Proper plate extensions are provided on these roll bars to prevent sinkage in soft soil.
- The seat has been redesigned to accommodate the suited astronaut.
- The foot control basket has been redesigned to provide easier operator ingress and egress.
- The gear shift handle has been extended.



CARRIER, LIGHT WEAPONS: INFANTRY, 1/2 TON, 4×4 M274-LEFT FRONT VIEW



CARRIER, LIGHT WEAPONS: INFANTRY, 1/2 TON, 4×4 M274-RIGHT REAR VIEW

Figure 7.4-1. Existing Mule Vehicle

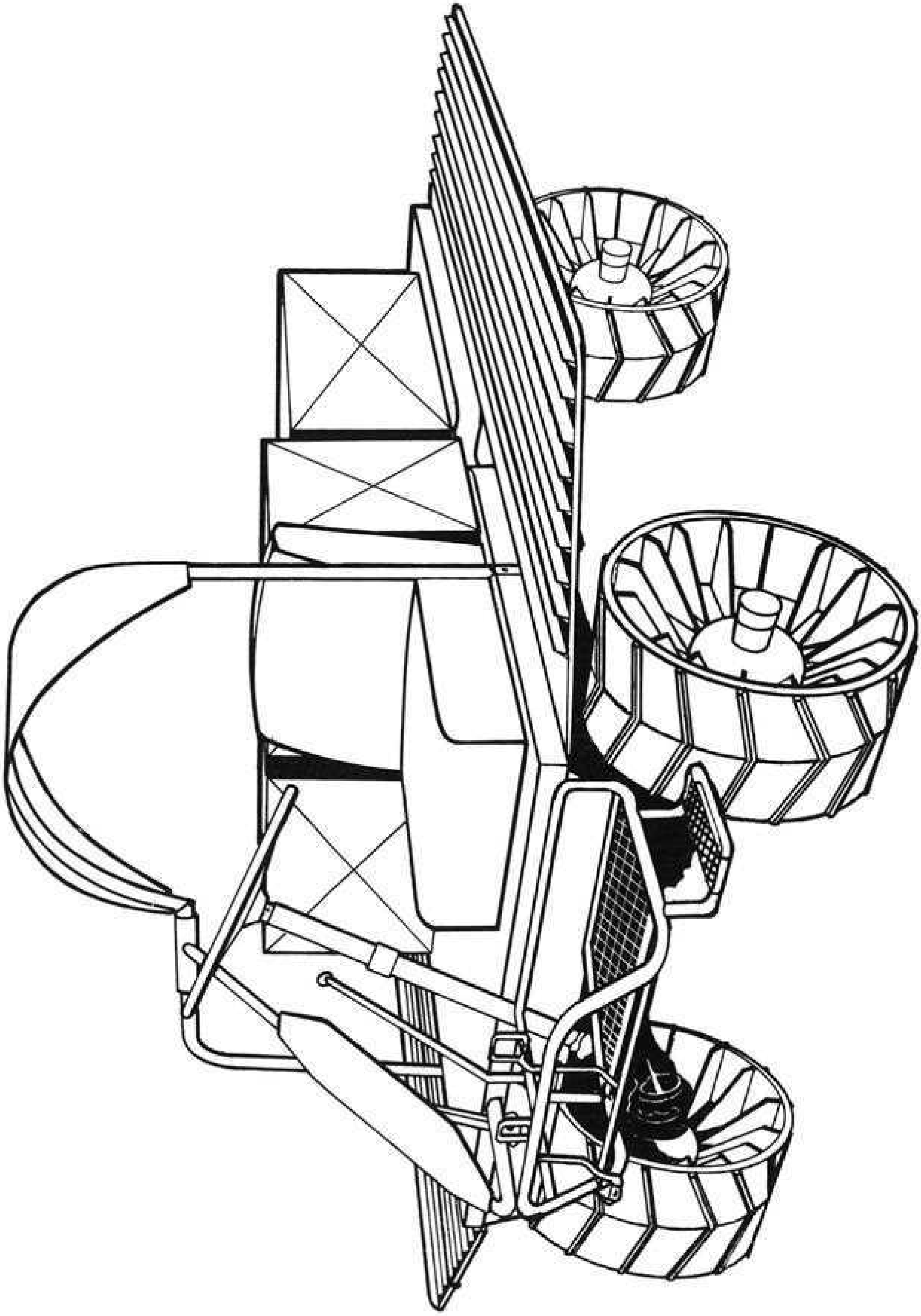


Figure 7.4-2. General Concept - First Order Vehicle

	EXIST	NEW	COMPONENT ASSEMBLY	ACTIONS	FIRST ORDER
POWER	X		I.C.E. ENGINE	P	UNSATISFACTORY - REPLACED AS UNIT
		X	BATTERY PACK	A	NEW POWER SOURCE-4 BATT. PACKS
		X	BATTERY CHARGE & CONTROL	A	D.C. PULSE CONTROL
		X	RADIATOR	A	SPACE RADIATOR DESIGN
	X		FUEL TANK	R	NOT REQUIRED
MOBILITY & STRUCTURES	X		WHEEL ASSEMBLY	P	SOLID DISC WHEEL
	X		ACKERMAN STEERING (4 WHEELS)	L	LUBRICATION & SEALING
	X		.STEERING GEAR ASSEMBLY	L	" " "
	X		TRANSMISSION	L	LUBE.,SEAL,COOL
	X		TRANSFER CASE	L	" " " CHANGE HIGH GEAR
	X		CLUTCH	M	MANUAL TO CENTRIFUGAL
	X		PROPELLER SHAFT	L	LUBRICATION
	X		PARKING BRAKE	L	LINING MATERIAL CHANGE
	X		FRONT AND REAR AXLE	L	LUBE, SEAL
	X		WHEEL UNIVERSALS & BEARINGS	L	LUBRICATION
	X		PAYLOAD PLATFORM	U	46" X 94"
	X		FRAME ASSEMBLY	M	A/E SURFACE TREATMENT ONLY
	X		FRAME TUBES	M	A/E SURFACE TREATMENT ONLY
	X		BODY RAIL	P	REPLACED BY TIE DOWN POINTS
				-	---
		X	ELECTRIC DRIVE MOTOR	A	SINGLE MOTOR COUPLED TO TRANS.
				-	---
CREW STATION	X		SEAT	M	REDESIGN FOR SUITED MAN
	X		STEERING WHEEL	M	" " " "
	X		ACCELERATOR (FOOT)	M	" " " "
	X		CLUTCH PEDAL	R	NO PEDAL REQUIRED - CENTRI. CLUTCH
	X		THROTTLE, CHOKE, HAND STARTER	R	NOT REQUIRED
	X		BRAKE PEDAL	L	REDESIGN FOR SUITED MAN
	X		TRANSMISSION SHIFT LEVER	L	EXTENDED LENGTH
	X		TRANSFER SHIFT LEVER	R	NO SHIFTING-TRANSFER GEAR LOCKED
		X	INSTRUMENTATION	A	SPEEDOMETER, BATT. PRESS. CURRENT
		X	ROLL BAR	A	REQUIRED TO PROTECT MAN
	X		FOOT CONTROL BASKET	M	REDESIGN FOR SUITED MAN

ACTION CODE: A= ADDED COMPONENT
R= REMOVED COMPONENT
L= LUNARIZED COMPONENT
M= MODIFIED COMPONENT
U= UTLIZED WITHOUT MODIFICATION
P= REPLACED WITH NEW COMPONENT OR SYSTEM

Figure 7.4-3. First Order Modifications Chart

- Radiators have been provided along each side of the cargo platform. These radiators are a part of the modified vehicle's thermal control system. They also serve as fenders to provide deflection of debris thrown up by the wheels (as per the criteria).

This first order modified vehicle consists generally of a vehicle in which the internal combustion engine of the basic vehicle has been replaced by a D.C. electric motor, which is directly connected to the basic vehicle's transmission from which the transfer case is eliminated. The primary electric power source will be four battery packs placed under the cargo platform. Each battery pack consists of two (2) silver-zinc battery units. The basic vehicle's mechanical drive train is retained with special seals. The basic vehicle's steering system is retained. The operator utilizes a foot pedal type drive motor accelerator to control vehicle speed. A brake pedal operates the basic vehicle's propeller shaft brake. No clutch pedal is required as the manual clutch has been replaced with a centrifugal clutch which engages the drive when a specified minimum motor speed is reached. All grease lubrication points have been sealed.

7.4.4 Second Order Modified Vehicle

7.4.4.1 Vehicle Configuration. The configuration concept of the second order modified vehicle is given in Figure 7.4-4.

7.4.4.2 Modification Actions Taken. The modification actions taken in the second order are tabulated in Figure 7.4-5. The only modification change to the basic vehicle made in the second order which is different from that made in the first order is the wheel assembly. All other changes are the same. In the second order, a metalastic wheel is used. The wheel shown is a wheel type now under development by an industrial firm. No preference is intended of the several elastic metal wheel types now in development.

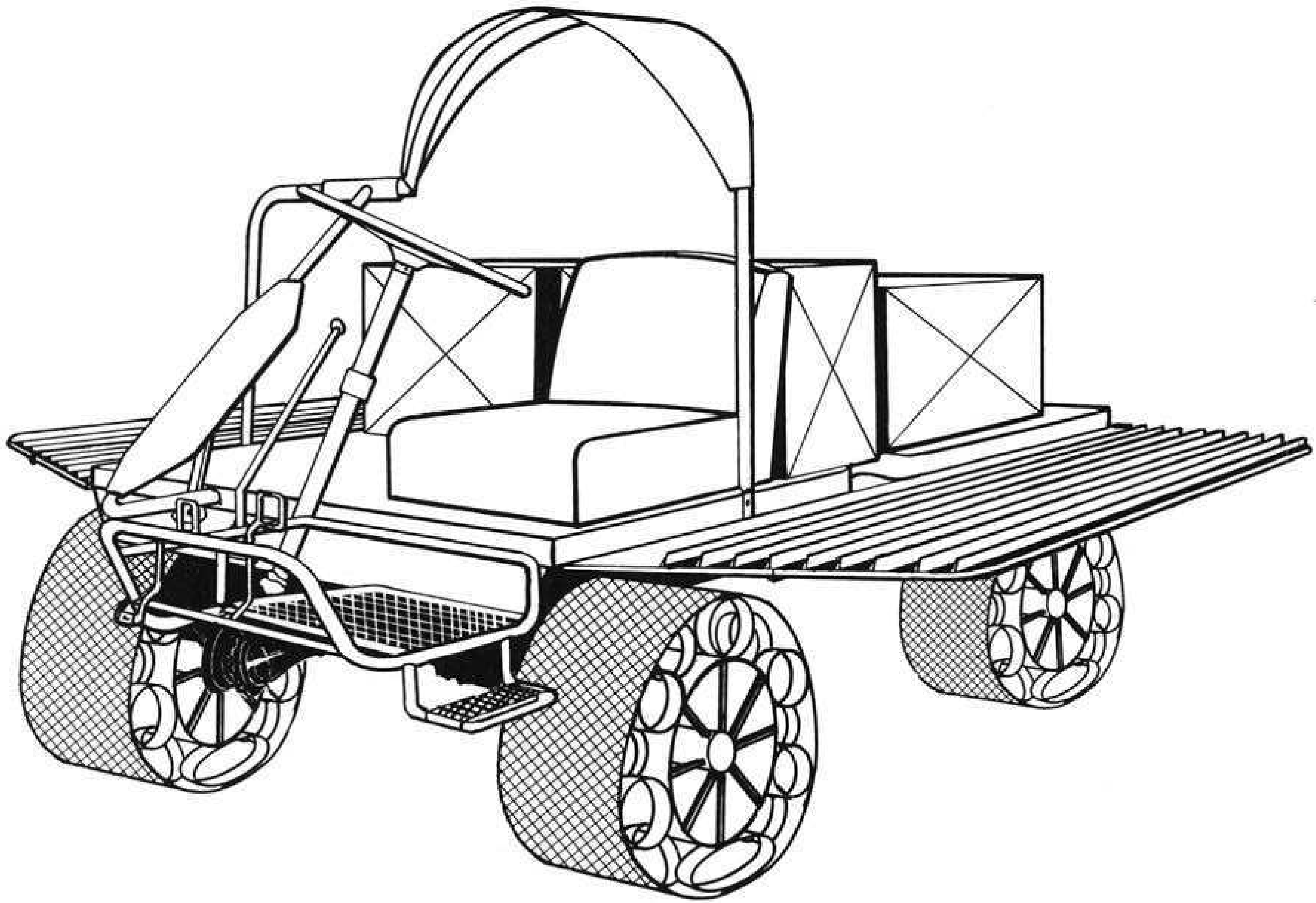


Figure 7.4-4. General Concept - Second Order Vehicle

	EXIST	NEW	COMPONENT ASSEMBLY	ACTIONS	SECOND ORDER
POWER	X		I.C.E. ENGINE	P	UNSATISFACTORY - REPLACED AS UNIT
		X	BATTERY PACK	A	NEW POWER SOURCE-4 BATT. PACKS
		X	BATTERY CHARGE & CONTROL	A	D.C. PULSE CONTROL
		X	RADIATOR	A	SPACE RADIATOR DESIGN
	X		FUEL TANK	R	NOT REQUIRED
MOBILITY & STRUCTURES	X		WHEEL ASSEMBLY	P	METALASTIC WHEEL
	X		ACKERMAN STEERING (4 WHEELS)	L	LUBRICATION & SEALING
	X		STEERING GEAR ASSEMBLY	L	" " "
	X		TRANSMISSION	L	LUBE.,SEAL,COOL
	X		TRANSFER CASE	L	" " " CHANGE HIGH GEAR
	X		CLUTCH	M	MANUAL TO CENTRIFUGAL
	X		PROPELLER SHAFT	L	LUBRICATION
	X		PARKING BRAKE	L	LINING MATERIAL CHANGE
	X		FRONT AND REAR AXLE	L	LUBE, SEAL
	X		WHEEL UNIVERSALS & BEARINGS	L	LUBRICATION
	X		PAYLOAD PLATFORM	U	46" X 94"
	X		FRAME ASSEMBLY	M	A/E SURFACE TREATMENT ONLY
	X		FRAME TUBES	M	A/E SURFACE TREATMENT ONLY
	X		BODY RAIL	P	REPLACED BY TIE DOWN POINTS
				-	---
		X	ELECTRIC DRIVE MOTOR	A	SINGLE MOTOR COUPLED TO TRANS.
				-	---
CREW STATION	X		SEAT	M	REDESIGN FOR SUITED MAN
	X		STEERING WHEEL	M	" " " "
	X		ACCELERATOR (FOOT)	M	" " " "
	X		CLUTCH PEDAL	R	NO PEDAL REQUIRED- CENTRI. CLUTCH
	X		THROTTLE, CHOKE, HAND STARTER	R	NOT REQUIRED
	X		BRAKE PEDAL	L	REDESIGN FOR SUITED MAN
	X		TRANSMISSION SHIFT LEVER	L	EXTENDED LENGTH
	X		TRANSFER SHIFT LEVER	R	NO SHIFTING-TRANSFER GEAR LOCKED
		X	INSTRUMENTATION	A	SPEEDOMETER, BATT. PRESS. CURRENT
		X	ROLL BAR	A	REQUIRED TO PROTECT MAN
	X		FOOT CONTROL BASKET	M	REDESIGN FOR SUITED MAN

ACTION CODE: A= ADDED COMPONENT
R= REMOVED COMPONENT
L= LUNARIZED COMPONENT
M= MODIFIED COMPONENT
U= UTLIZED WITHOUT MODIFICATION
P= REPLACED WITH NEW COMPONENT OR SYSTEM.

Figure 7.4-5. Second Order Modifications Chart

7.4.5 Third Order Modified Vehicle

7.4.5.1 Vehicle Configuration. The configuration concept of the third order modified vehicle is given in Figure 7.4-6.

7.4.5.2 Modification Actions Taken. The modification actions taken in the third order are tabulated in Figure 7.4-7. The modification action code designations are similar to those used in Figures 7.4-3 and 7.4-5.

In this modification, the internal combustion engine, the transmission, the transfer case, the axles, drop gear cases, the drive shaft, and the drive shaft brake of the basic vehicle have all been removed. Each wheel is now provided with an A.C. electric drive motor, harmonic transmission, reduction gear, suspension system, dampening, and brake integrated into a unit. Each wheel assembly is composed of a constrained-elliptical, elastic, metal wheel with drive and idler gears. The prime power is provided by three battery packs, each of which consists of two silver-zinc battery units. The ackerman type steering system has been modified and now consists of an electric motor driven ball-geared unit for only two wheel steer. The vehicle control system is entirely electric motor actuated. Reference to Figure 7.4-6 will show that the basic vehicle platform is retained. The foot basket, seat and roll-bars are the same units utilized on the first and second order modifications. The thermal control system radiators are again placed on each side of the cargo platform and utilized as fenders. However, the radiators are now only one-half the area of those utilized on the first and second order modified vehicles. The heat rejection load from the power system is one-half as large for this modified vehicle as that required on the first and second modified vehicles.

The vehicle control unit and instrument panel are shown placed forward and above the seat. The control unit may be removed from the instrument panel and the vehicle operated by the operator while he is walking or standing beside the vehicle.

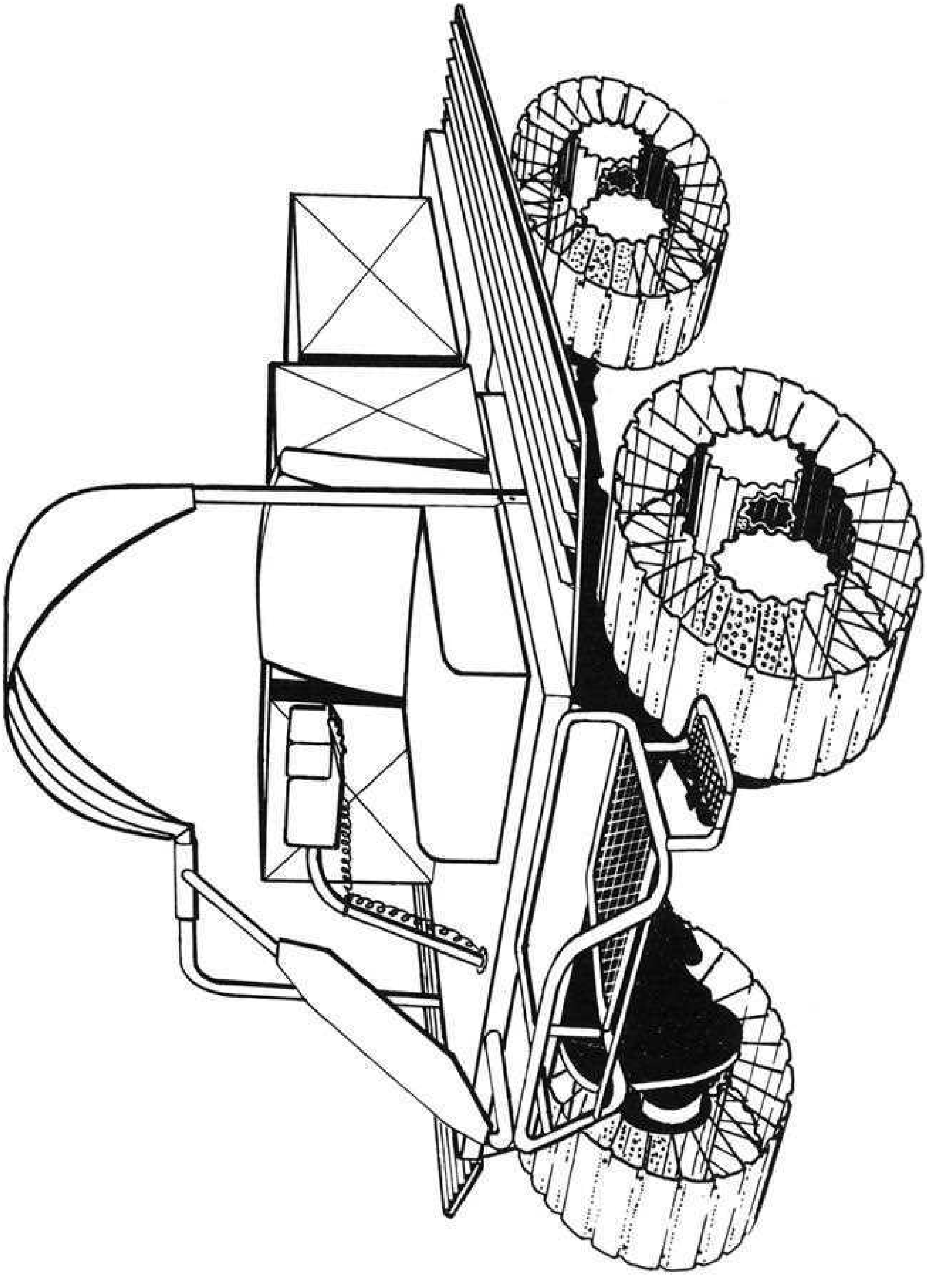


Figure 7.4-6. General Concept - Third Order Vehicle

	EXIST	NEW	COMPONENT ASSEMBLY	ACTION	THIRD ORDER VEHICLE
POWER	X		I.C.E. ENGINE	P	UNSATISFACTORY-REMOVED AS UNIT
		X	BATTERY PACK	A	3 BATT. PACKS NEW POWER SOURCE
		X	BATTERY CHARGE & CONTROL	A	AC SYSTEM
		X	RADIATOR	A	REDUCED RADIATOR SIZE
	X		FUEL TANK	R	NOT REQUIRED
MOBILITY & STRUCTURES	X		WHEEL ASSEMBLY	P	FORCED ELLIPTIC WHEEL
	X		ACKERMAN STEERING (4 WHEELS)	M	SIZE CHANGE FOR NEW SYSTEM
	X		STEERING GEAR ASSEMBLY	P	ELECTRIC REMOTE ACTIVATION
	X		TRANSMISSION	P	MOTORS HARMONIC DRIVES
	X		TRANSFER CASE	P	" " "
	X		CLUTCH	P	" " "
	X		PROPELLER SHAFT	P	" " "
	X		PARKING BRAKE	P	NEW BRAKE ON MOTORS (2 WHEEL)
	X		FRONT AND REAR AXLE	P	MOTORS & HARMONIC DRIVES
	X		WHEEL UNIVERSALS & BEARINGS	P	" " "
	X		PAYLOAD PLATFORM	U	46" X 96"
	X		FRAME ASSEMBLY	M	A/E SURFACE TREATMENT ONLY
	X		FRAME TUBES	R	FRAME BRACKET ON MOTOR DRIVE
	X		BODY RAIL	R	REPLACED BY TIE DOWNS
				A	TORSION BAR SYSTEM
		X	ELECTRIC DRIVE MOTOR	A	FOUR MOTOR DRIVE
				A	ELEC. BALL SCREW
CREW STATION	X		SEAT	M	REDESIGN FOR SUITED MAN
	X		STEERING WHEEL	P	HAND-REMOTE STEER
	X		ACCELERATOR (FOOT)	P	HAND CONTROL
	X		CLUTCH PEDAL	R	NOT REQUIRED
	X		THROTTLE, CHOKE, HAND STARTER	R	NOT REQUIRED
	X		BRAKE PEDAL	P	HAND BRAKE
	X		TRANSMISSION SHIFT LEVER	R	NOT REQUIRED
	X		TRANSFER SHIFT LEVER	R	NOT REQUIRED
		X	INSTRUMENTATION	A	REMOTE CONTROL CONSOLE
		X	ROLL BAR	A	REQUIRED TO PROTECT MAN
	X		FOOT CONTROL BASKET	M	REMOVE FOOT CONTROLS

ACTION CODE: A= ADDED COMPONENT
R= REMOVED COMPONENT
L= LUNARIZED COMPONENT
M= MODIFIED COMPONENT
U= UTLIZED WITHOUT MODIFICATION
P= REPLACED WITH NEW COMPONENT OR SYSTEM·

Figure 7.4-7. Third Order Modifications Chart

7.4.6 Power Requirements for Modified Vehicles

The power required to operate each modified vehicle, for one 24 km traverse, for each of the three orders of modification, is presented in Figure 7.4-8. The power required from the LEM/S power system for recharging the vehicles' battery units is also given. In addition, an alternate electric prime power system for the third order vehicle is shown.

7.4.7 Power System Concepts

Figures 7.4-9 and 7.4-10 summarize the power system utilized in the basic Mule vehicle and the systems concepts selected for the three orders of modification.

7.4.8 Wheel Concepts

Figure 7.4-11 summarizes the wheel type selections for the three orders of modification. The 26.3" wheel diameter of the basic vehicle has been maintained for each order; this dimension is the minor axis dimension of the third order wheel.

7.5 FIRST ORDER VEHICLE MODIFICATIONS

This section presents the detailed discussions on the first order vehicle modification.

7.5.1 Mobility and Structure

7.5.1.1 Wheel Considerations. The pneumatic tire used on the basic vehicle has been eliminated from modification considerations, see paragraph 7.3.1 of this section. The wheel choice for the first order modification is to be the simplest and most economical type consistent with the study criteria. A metal type wheel was the most obvious choice. From this category, rigid, hollow and foam-filled wheels have been considered. It was considered that a rigid wheel would be the simplest and least costly wheel choice. It would not be the best choice; however, in view of the lack of obstacle definition available in the study criteria, it has been selected.

VEHICLE	KWH FROM POWER PROFILE	FROM BATTERY	OTHER	REQUIRED FROM SHELTER FOR RECHARGE
MOD I AND II	12 KWH	4 BATTERY PACKS 71 AMP HRS/BATTERY 41% DEPTH OF NOMINAL DISCHARGE CAPACITY	NONE	16 KWH (54 VOLTS D.C., 0-24 AMPS)
MOD III	6 KWH	3 BATTERY PACKS 48 AMP HRS/BATTERY 28% DEPTH OF NOMINAL DISCHARGE CAPACITY	NONE	9 KWH (54 VOLTS D.C., 0-18 AMPS)
ALTERNATE MOD III	6 KWH	2 BATTERY PACKS 36 AMP HRS/BATTERY 21% DEPTH OF NOMINAL DISCHARGE CAPACITY (0-10 AMPS AT 42 VOLTS)	RADIOISOTOPE THERMIONIC GENERATOR (12 AMPS AT 42 VOLTS) 500 WATTS	NONE

Figure 7.4-8. Summary of Power Requirements

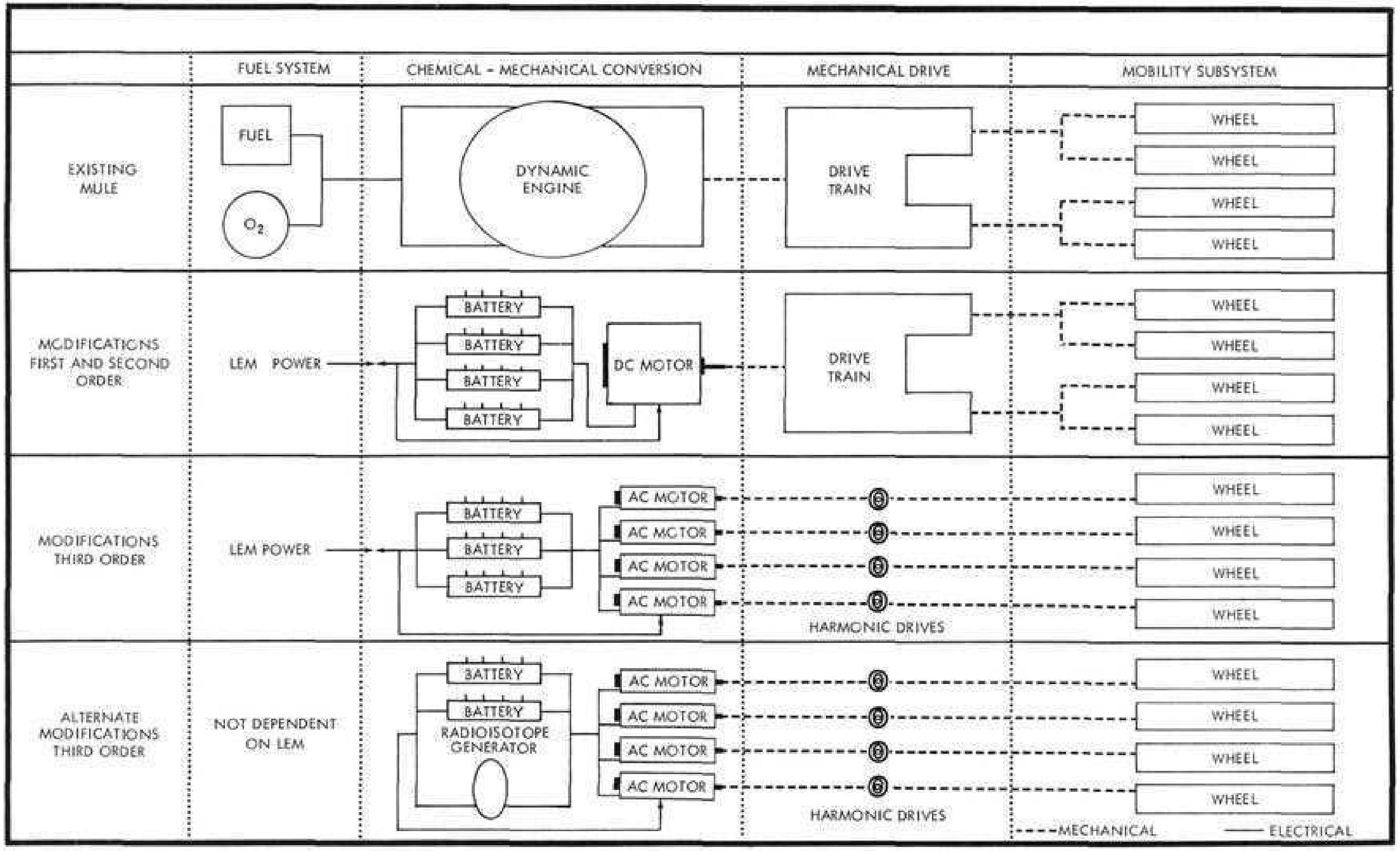


Figure 7.4-9. Power System Concepts

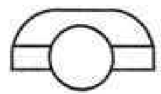
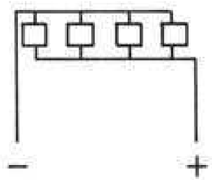
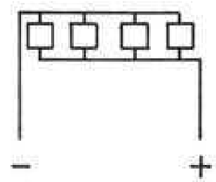
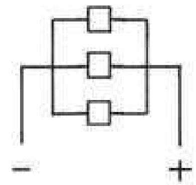
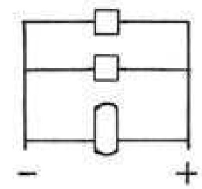
EXISTING	FIRST ORDER	SECOND ORDER	THIRD ORDER	THIRD ORDER ALTERNATE
DYNAMIC ENGINE 				
ADVANTAGE				
•LIGHT WEIGHT •DENSE POWER PACKAGE •DESIREABLE POWER PROFILE •EXTENSIVE ENGINEERING ON TERRESTRIAL ENGINES	•FLIGHT TESTED BATTERIES •REDUNDANT BATTERY SYSTEM •HIGH RELIABILITY •HIGH EFFICIENCY •REFINED CONTROL	•SAME AS FIRST ORDER	•REFINED CONTROL •BRUSHLESS MOTOR •REDUCED SIZE •REDUCED COMPLEXITY •LOWER COST NOTE: IMPROVED POWER SYSTEM IS THE RESULT OF WHEEL AND TRANSMISSION IMPROVEMENTS.	•SAME AS THIRD ORDER •INDEPENDENT OF LEM •INCREASED MISSION CAPABILITY •REDUNDANT POWER SYSTEM •REDUCED SYSTEM WEIGHT
DISADVANTAGE				
•TWO FUEL SYSTEM •HIGH FUEL CONSUMPTION •LOW RELIABILITY •EXTENSIVE DEVELOPMENT REQUIRED •COMPLICATED STORAGE •POOR CONTROL CAPABILITY •NON-REDUNDANT SYSTEM	•EXCESSIVELY HEAVY •TEMPERATURE SENSITIVE •LEM DEPENDENT	•SAME AS FIRST ORDER	•LESS REDUNDANT BATTERIES	•INCREASED TOTAL COST •ABORT PROBLEMS •STORAGE-THERMAL PROBLEMS •OPERATIONAL THERMAL PROBLEMS

Figure 7.4-10. Prime Power Concepts

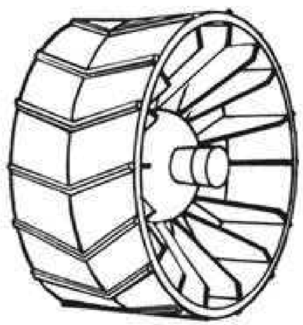
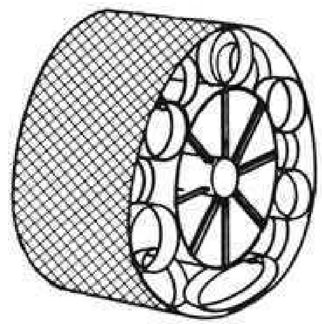
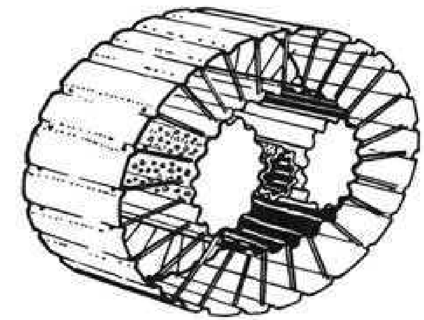
FIRST ORDER	SECOND ORDER	THIRD ORDER
 RIGID METAL	 METALASTIC	 FORCED ELLIPITIC
	ADVANTAGES	
•LOW COST •MINIMUM TEST REQUIRED •VERY RELIABLE •QUICK DEVELOPMENT	•MEDIUM COST •IMPROVED RIDE QUALITY •IMPROVED OBSTACLE QUALITY •REDUCED SINKAGE •REDUCED POWER REQ. IN SOFT SOILS	•LOWEST POWER REQ. FOR SOFT SOIL •LEAST SINKAGE •BEST RIDE QUALITY •GOVERNMENT DESIGN
	DISADVANTAGES	
•POOR OBSTACLE CAPABILITY •POOR RIDE QUALITIES •MAXIMUM POWER IN SOFT SOIL	•PROPRIETARY ITEM •HIGHER DEVELOPMENT COST •DESIGN UNPROVEN IN SMALL SIZES	•HIGHEST DEVELOPMENT COST •LEAST RELIABLE •REQUIRES MAXIMUM TEST •INCREASED STEERING EFFORT

Figure 7.4-11. Wheel Concepts - Mule Modifications

7.5.1.1.1 Rigid Metal Wheel—The rigid wheel has two undesirable, general characteristics:

- It has little or no energy absorbing ability.
- It would transfer impact loading to the (suspension and) frame of the vehicle almost instantaneously.

In the low lunar gravity environment and with no vehicle suspension system, the action of these characteristics upon the vehicle are compounded. Intuitively it would be expected that a rigid wheel on the modified vehicle would result in a very hard and bouncy ride. This can be demonstrated on a simplified basis. Consider a rigid wheel which is perfectly elastic (coefficient of restitution of 1) and which exerts a force on the lunar surface of 464 newtons (104 lbs force). This wheel is then allowed to roll, unrestrained, on the lunar surface over various vertical obstacles. Speeds of 4.8 km/hr (3 mph) and 8 km/hr (5 mph) and vertical obstacles of 2.5, 5.1, 7.6, 10.2, 12.7 and 15 cm (1, 2, 3, 4, 5 and 6 inches) were selected. The results of the wheel encounters at these speeds with these obstacle heights are shown plotted in Figure 7.5-1, in terms of height of bounce and time off the surface. These results are based only on a vector change of direction of wheel travel and no net change of energy.

These results do not reflect actual conditions since the wheel is, in fact, attached to the vehicle and is not perfectly elastic. It does show, however, that with the rigid wheeled vehicle and no suspension system, any obstacle encountered by the vehicle must be approached slowly and crossed at very reduced speed.

The rigid wheel concept will also impose heavy, sharp impact loads on the vehicle chassis since energy transfer, in this case, is dependent on the length of time of contact with the obstacle and hence wheel deflection. With the exception of smooth level surfaces, a very reduced vehicle speed must be accepted. Any attempt to maintain directional and speed control at 4.8 km/hr (3 mph) over a rock strewn surface would be virtually impossible.

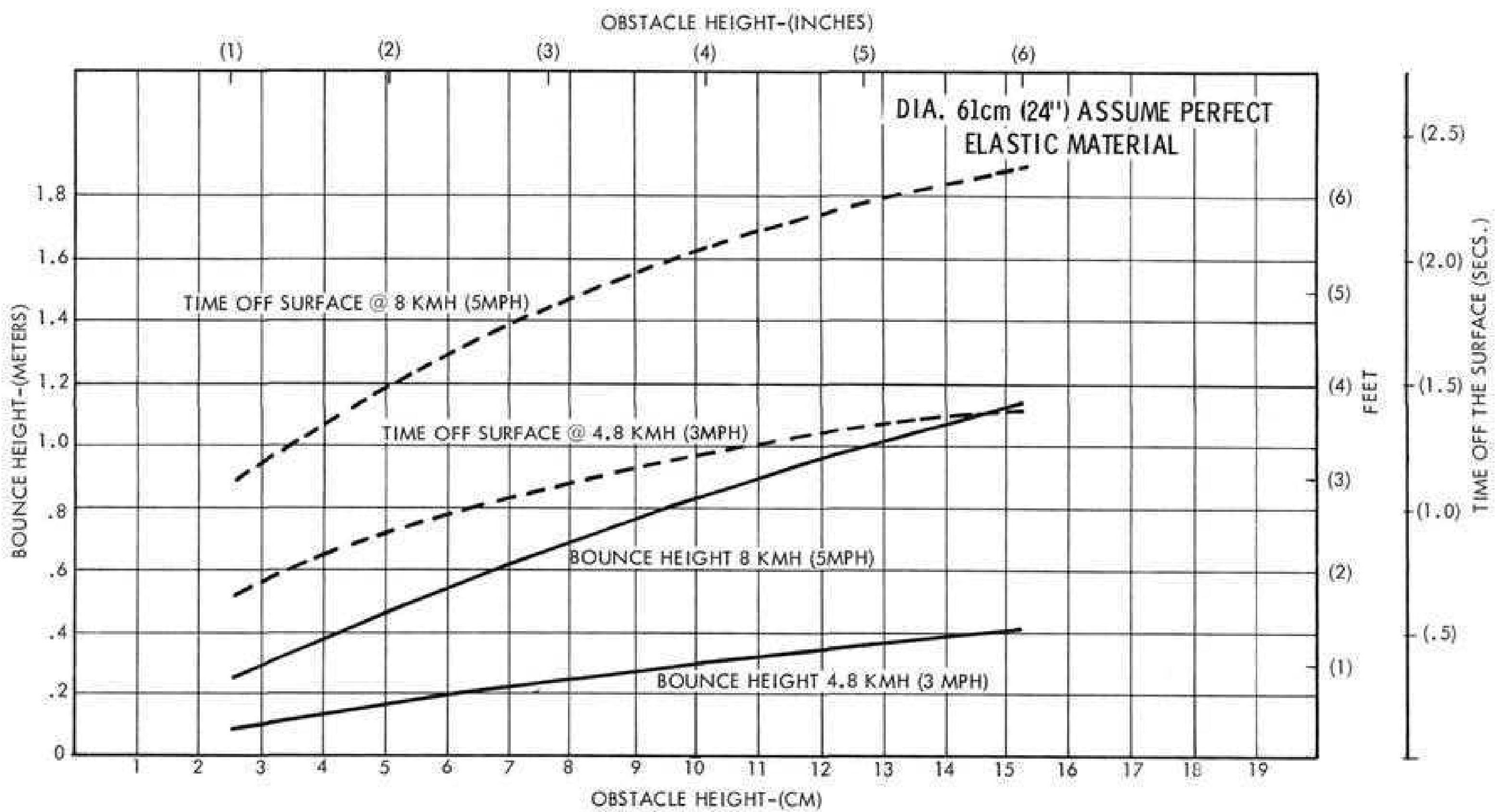


Figure 7.5-1. Theoretical Dynamic Behavior of a Rigid Mule Wheel

The rigid wheel selected for use on the first order modified vehicle is shown in Figure 7.5-2. An alternate rigid wheel is also shown on this figure in which an elastic polymer insert is placed in the wheel web section to dampen high frequency shock loadings.

7.5.1.2 Power and Energy Calculations. Reference is made to Figure 7.4-8, paragraph 7.4.6, which summarizes the power requirements for one traverse for the three orders of vehicle modification.

The determination of energy requirements for the modified vehicles has been calculated considering the following factors:

- Soft soil motion resistance computed by AMC, ATAC's Land Locomotion Laboratory mobility equations and the Bekker soil values system. The soil values used were from the ELMS model, see Table 6-1, Section 6.0, OPERATIONAL TERRAIN ANALYSIS, of this report.
- Gravitational effects on grades, an additional force exerted on a vehicle operating on slopes. The force may add or subtract from motion resistance.
- Rolling resistance work performed on flexing the wheel and frictional losses due to local slippage of the wheel on hard, smooth surfaces.
- Inertia resistance, a combination of translational and rotational inertia of the vehicle and its components. This resistance is always directed opposite the vector of acceleration.
- Motion resistance, a resistance due to surface roughness (micro-topography) and obstacle negotiation (macro-topography).
- Maneuvering, an additional resistance force developed in steering maneuvers.
- Mechanical efficiency of the vehicle drive line.

Detailed calculations may be generated by a comprehensive program of power and energy computation, as indicated in reference 3. The equations postulated in this reference are basic mobility equations (generally subscribed to by the AMC, ATAC Land Locomotion Laboratory) and elementary physics. These equations have been used in a similar manner in this study. We have utilized some simplifying

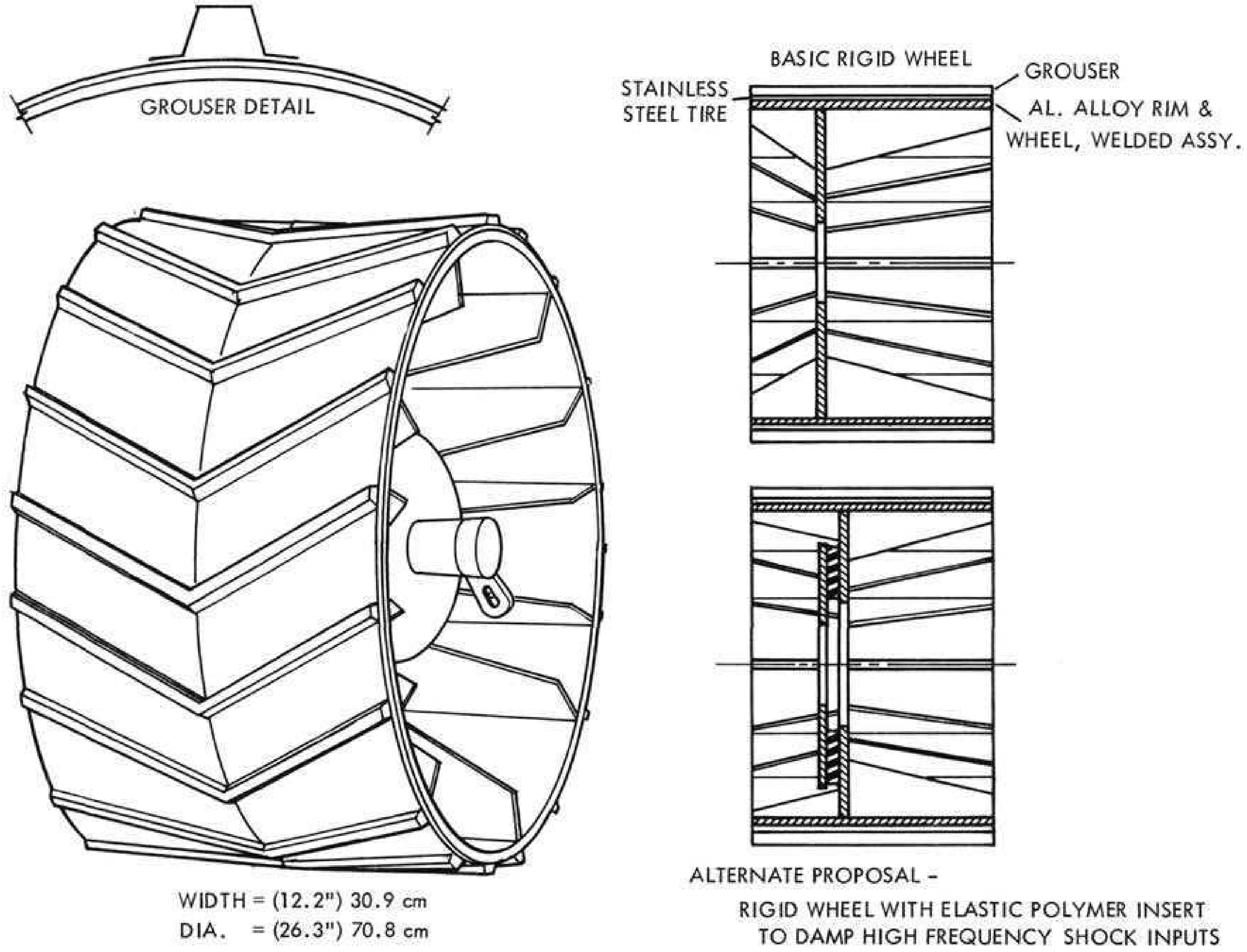


Figure 7.5-2. Rigid Wheel - First Order Modification

assumptions and reduced the depth of detail. It is felt that these simplifications are consistent with the terrain model utilized in this study. The following assumptions are utilized:

- Soft soil resistance is composed of a resistance due to compaction (R_c) and a resistance due to bulldozing (R_b). In this analysis R_b will not be considered as it is only significant with high wheel sinkages and is considered to occur only at the front wheels. Also, any such value for R_b calculated for a cohesionless soil is of doubtful value.
- All wheels will be considered to be loaded equally and no weight transfer will be assumed for grade situations.
- Rolling resistances are variously given as 0.01 to 0.04 of wheel loading. With a rigid wheel, this coefficient may be reduced further and, in fact, is ignored for the first order rigid wheel.
- The actual translatory and rotational inertia of the modified vehicle are unknown. However, a translatory inertia may be estimated by assuming a maximum mass for vehicle and cargo. A total of 2,500 lbs mass yields a value of 78 slugs. If the rotational inertia is assumed to equal 1.4 of the translational inertia, the resulting value of 109.0 slugs can be used to size the motor starting torque to provide a minimum acceleration rate of 0.023 m/sec^2 (0.75 ft/sec^2) in the weakest soil on a 4% grade. Assuming average terminal speeds of 3 mph (4.4 ft/sec), constant acceleration rates, and 124 starts per traverse, the total energy requirement amounts to only an additional 0.05 KWH at the wheels or 0.0625 KWH at the drive motor. This additional energy requirement is ignored.
- A 20% energy reserve will be provided for obstacle negotiation and surface roughness, as per the criteria.
- Energy requirements for maneuvering are dependent upon the soil strength at the site of the maneuver and thus cannot be predicted. Energy for this purpose will be provided for by use of a contingency allowance.

The vehicle energy requirement anslysis includes the effect of the following factors:

- Soil composition resistance

- Grade resistance
- Drive train mechanical efficiency
- Obstacle negotiation and surface roughness resistance for which a 29% (of total) allowance is provided.
- A contingency allowance of 10% to include an allowance for maneuvering.

The calculation procedure involved computation of tractive effort versus slip percentage for each soil group specified in the criteria. The graphs developed from these calculations are given in Figures 7.5-3, 7.5-4 and 7.5-5. Positive and negative slopes, within the range of slopes specified for each soil group in the typical ELMS model traverse, are shown.

Wheel horsepower and total drive motor input horsepower were calculated. Vehicle speed requirements were set by the criteria as at least 4.8 km/hr (3 mph) in the level soft soil group and at least 8 km/hr (5 mph) in the level compacted soil group. The drive motor input horsepower arrived at to achieve the required minimum vehicle speed in the level soft soil group was considered as the maximum (for any wheel configuration). Computed vehicle performance on the modified ELMS model typical traverse, figure 7.5-6A, was governed by this maximum input horsepower; it was either speed limited (on negative slopes) or power limited (on positive slopes). To illustrate how the modified vehicle will operate in a soil group other than the soft soil group, an example can be given. In the hard soil group ($k\phi = 3.0$, $n = 1.0$) on level ground with the 1.75 hp drive motor (necessary for minimum speed in the soft soil group), the vehicle can achieve a speed of 12.8 km/hr (8 mph), see Figure 7.5-21.

A chart has been developed for each slope-soil group division given in the ELMS typical traverse (as modified for 15^{o} maximum slope). Slopes of 0^{o} to $+4^{o}$ and -4^{o} are grouped in one chart with the soil characteristics group of $k\phi = 0.5$, $n = 0.5$. The next chart groups slopes of $+5^{o}$ to $+10^{o}$ and -5^{o} to -10^{o} with the soil group of $k\phi = 1.0$, $n = 0.75$. The last chart groups slopes of $+10^{o}$ to $+15^{o}$ and -10^{o} to -15^{o} with the soil group of $k\phi = 3.0$, $n = 1.0$. These charts are given in Figures 7.5-6, 7.5-7 and 7.5-8, respectively.

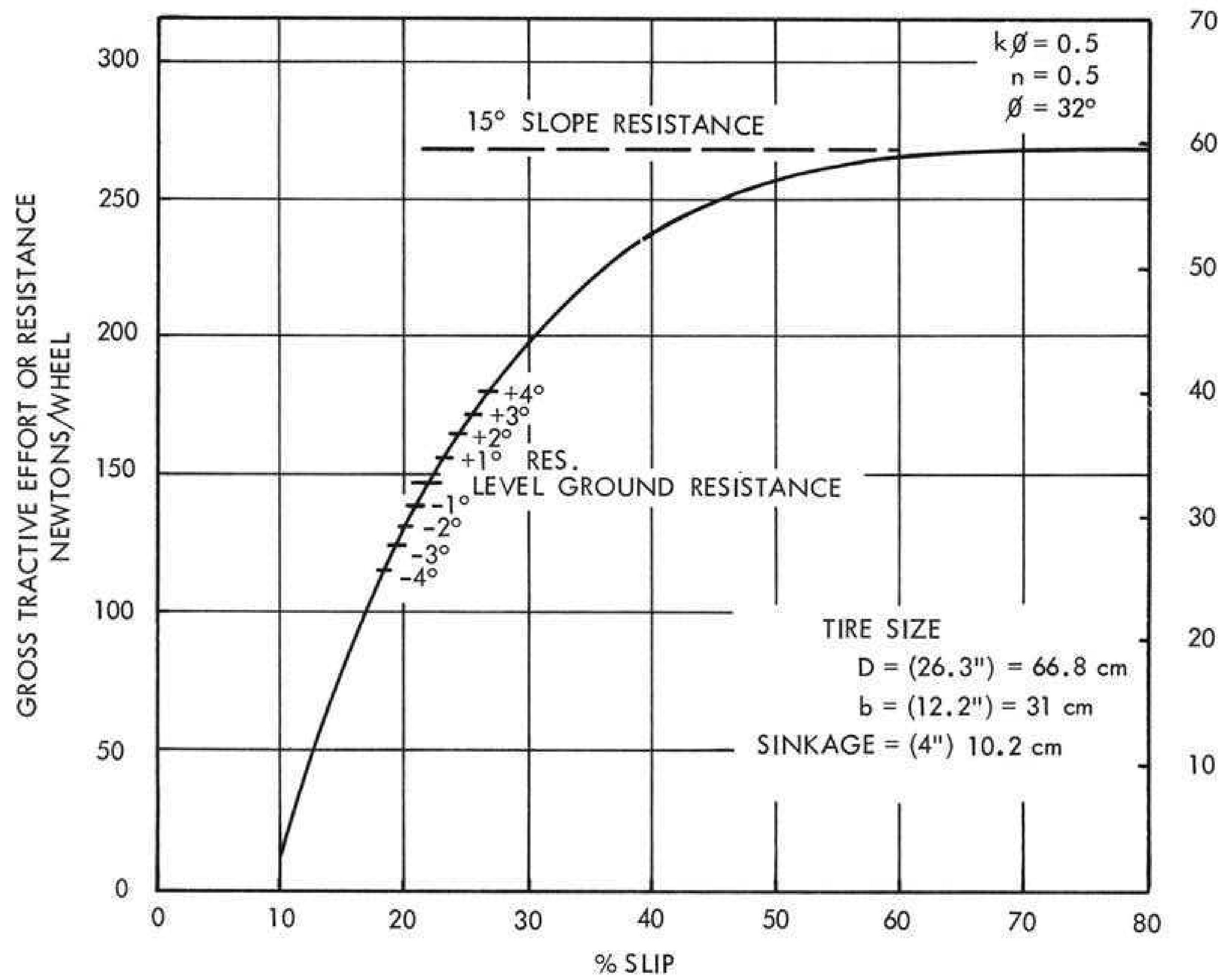


Figure 7.5-3. Tractive Effort vs Slip - First Order Vehicle, $K\phi = 0.5$

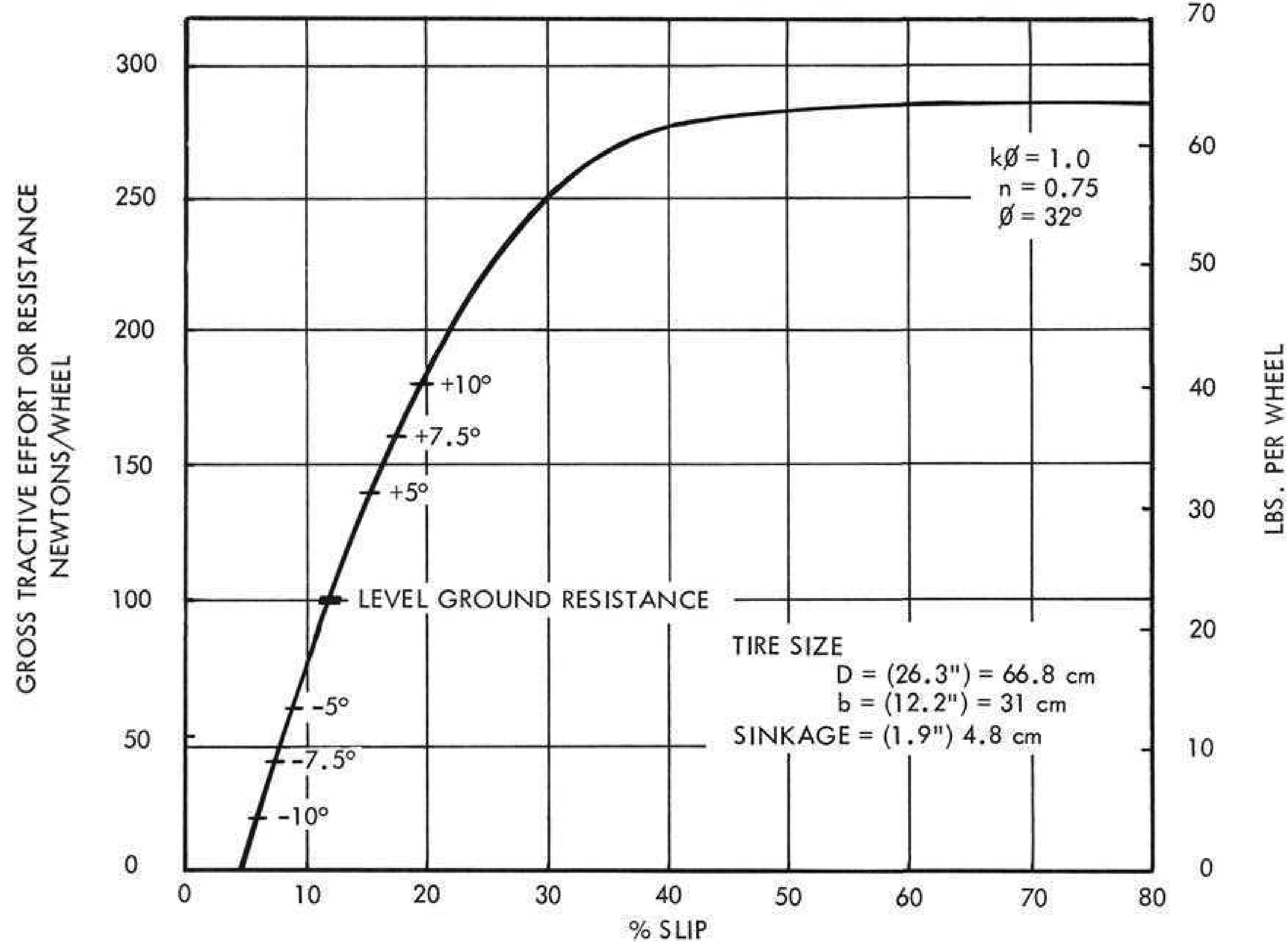


Figure 7.5-4. Tractive Effort vs Slip - First Order Vehicle, $K\phi = 1.0$

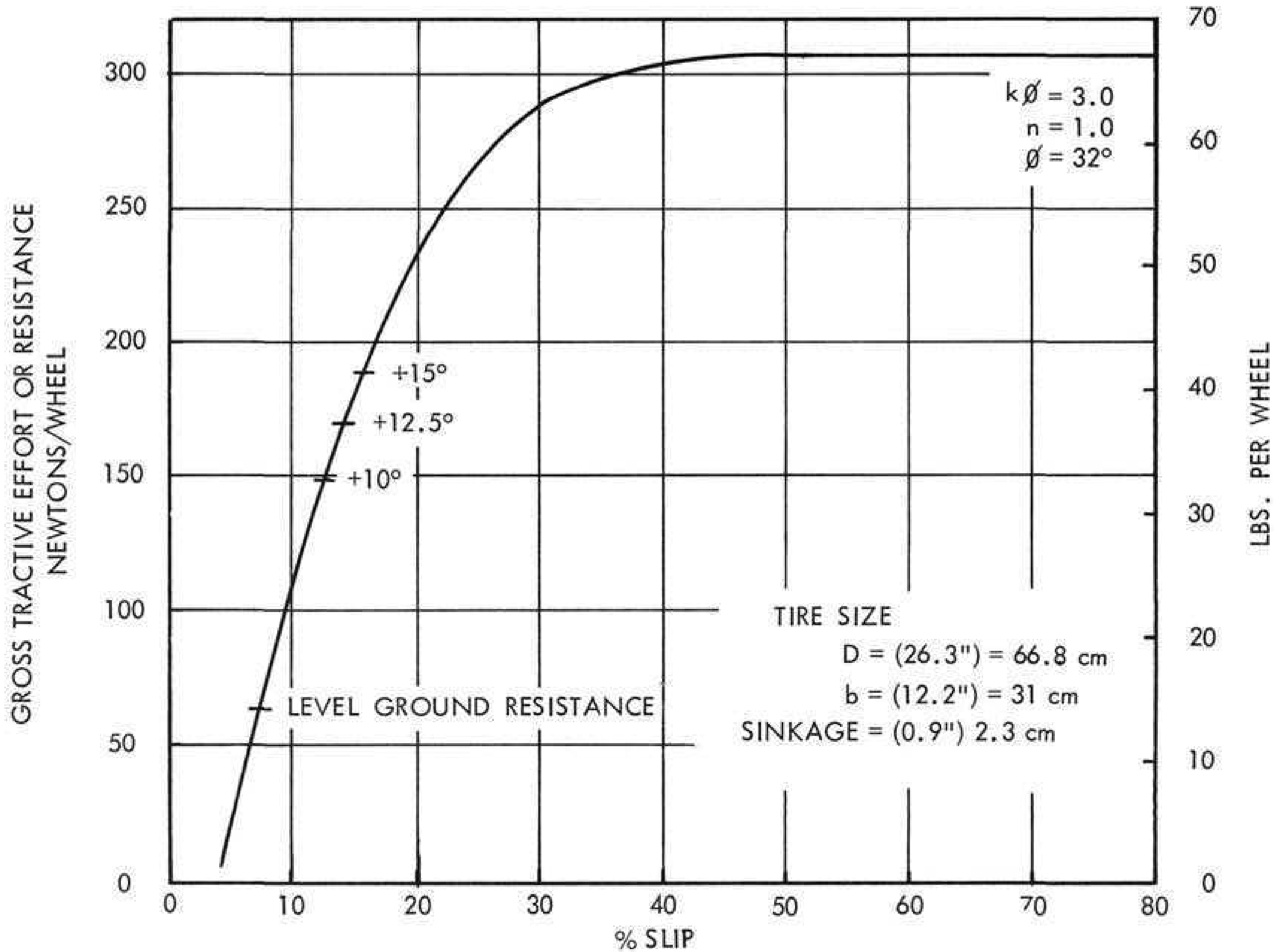


Figure 7.5-5. Tractive Effort vs Slip - First Order Vehicle, Kϕ = 3.0

Slope (Degrees)	% of Traverse	Total Distance	Kφ	n	φ
0	11.0	2.655 km	0.5	0.5	32^{o}
-1^{o}	13.0	3.138 km	0.5	0.5	32^{o}
$+1^{o}$	9.5	2.293 km	0.5	0.5	32^{o}
-2^{o}	12.8	3.089 km	0.5	0.5	32^{o}
$+2^{o}$	11.7	2.824 km	0.5	0.5	32^{o}
-3^{o}	5.5	1.328 km	0.5	0.5	32^{o}
$+3^{o}$	10.5	2.534 km	0.5	0.5	32^{o}
-4^{o}	3.0	0.724 km	0.5	0.5	32^{o}
$+4^{o}$	7.0	1.690 km	0.5	0.5	32^{o}
-5^{o}	3.0	0.724 km	1.0	0.75	32^{o}
$+5^{o}$	4.5	1.086 km	1.0	0.75	32^{o}
-7.5^{o}	1.8	0.434 km	1.0	0.75	32^{o}
$+7.5^{o}$	1.2	0.269 km	1.0	0.75	32^{o}
-10^{o}	.06	0.145 km	3.0	1.0	32^{o}
$+10^{o}$	1.2	0.270 km	3.0	1.0	32^{o}
-12.5^{o}	0.72	0.182 km	3.0	1.0	32^{o}
$+12.5^{o}$	0.48	0.126 km	3.0	1.0	32^{o}
-15.0	1.55	0.384 km	3.0	1.0	32^{o}
$+15^{o}$	0.95	0.241 km	3.0	1.0	32^{o}
		24.14 km			

Figure 7.5-6A. Terrain Profile for One Traverse

Grade	Req. Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req'd At Wheel/Wheel	x 4	÷ .8
0^{o}	32.7 x 1.095 = 35.8 ft lbs	22%	38.3 (3.00 mph)	49	.335	1.335	1.67 say 1.7
$+1^{o}$	34.51 x 1.095 = 37.8 ft lbs	23%	35.8 (2.80 mph)	46.5	.335	1.335	1.7
$+2^{o}$	36.33 (1.095) = 39.8	24 1/2%	34.1 (2.67 mph)	44.2	.335	1.335	1.7
$+3^{o}$	38.14 (1.095) = 41.7	25%	31.7 (2.50 mph)	42.2	.335	1.335	1.7
$+4^{o}$	39.95 (1.095 = 43.8	26 1/2%	30 (2.35 mph)	40.2	.335	1.335	1.7
-1^{o}	30.89 (1.095) + 33.8	21%	38.3	48.3	.31	1.24	1.55
-2^{o}	29 (1.095) 31.8	20%	38.3	48.0	.29	1.16	1.45
-3^{o}	27.3 (1.095) 29.9	19.3%	38.3	47.7	.272	1.087	1.36
-4^{o}	25.4 (1.095) 27.8	18.5%	38.3	47.0	.249	.995	1.24

Figure 7.5-6. Horsepower Requirements - Soil Group 1

Grade	Req. Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req'd At Wheel Speed	x 4	÷ .8
+5°	31 x 1.095 = 34	15°	38.3	45	.291	1.165	1.46
+7.5°	35.5 x 1.095 = 38.9	17°	38.3	46.2	.343	1.37	1.71
+10°	40 x 1.095 = 43.8	19.5°	32.5 (2.55 mph)	40	.335	1.335	1.7
-5°	13 (1.095) = 14.24	13%	38.3	44	.1195	.477	.595
-7.5	8.5 (1.095) = 9.3	7%	38.3	41.2	.073	.292	.365
-10	4 (1.095) 4.38	5.5%	38.3	40.5	.0338	.1352	.169
0	22 (1.095) = 24.1	12%	64.0 (5 mph)	72.5	.333	1.333	1.7
0	24.1	12%	38.3 (3 mph)	43.5	.1995	.796	.996

Figure 7.5-7. Horsepower Requirements - Soil Group 2

Grade	Req Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req'd At Wheel/Wheel	x 4	÷ .8
+12.5°	37.2 x 1.095 = 40.8	13%	38.3	44	.342	1.37	1.71
+15°	41.5 x 1.095 = 45.5	15.5	32.5 (2.55 mph)	38.6	.335		1.7
-12.5	Vehicle Coasts						
15°	Vehicle Coasts						
0	14.5 (1.095) = 15.9	7 1/2%	38.3 (3 mph)	41.5	.126	.503	.63
0	15.9	7 1/2%	64 (5 mph)	69.2	.21	.84	1.05
0	15.9	7 1/2%	102.2 (8 mph)	110.5			1.7
+10°	32.75 x 1.095 = 35.8 ft lbs	12 1/2%	38.3	43.5	.297	1.186	1.48
-10°	Vehicle Coasts						

Figure 7.5-8. Horsepower Requirements - Soil Group 3

The total energy requirement for each wheel configuration (each order of modification) was thus computed from the following:

- The required wheel power in each given soil-grade, wheel speed situation.
- Vehicle speed in each situation.
- Percentage occurrence of each soil-grade in the total traverse distance (for one typical traverse).
- The length of the single typical traverse.

These calculations resulted in the development of the power profiles utilized in sizing the power system. These profiles are given in the Power System discussions in each modification order.

The total energy requirements were computed for two different rigid wheel configurations. The first was dimensionally similar to the standard M-274 wheel with pneumatic tire, diameter 25.3" and width of tire 8.3". The second was dimensionally similar to the M-274 wheel with pneumatic tire (terra-type flotation tire), diameter 26.3" and width of tire 12.2". The maximum motor output was based upon that required to maintain 4.8 km/hr (3 mph) in the weakest soil group on level ground.

The standard M-274 wheel and tire dimensions would require a 2.45 hp motor and a total single traverse energy of 16 kwh. The terra-tire and wheel dimension required a 1.70 hp motor and a total energy of 12 kwh for a single traverse.

The use of the standard tire dimensions for the rigid wheel resulted in a larger motor (relatively), a larger battery recharge energy, a longer recharge time, higher soft soil sinkage and high steady-state slip-values in the soft soil group. Therefore, it was decided to utilize the terra-tired wheel dimensions for the rigid wheel for the first order modification.

7.5.1.3 Obstacle Negotiation. The prediction of vehicle obstacle negotiation and ditch crossing (crevasse) capability was investigated using a theoretical analysis which has been confirmed by experiment. The same techniques and information were utilized to predict the obstacle negotiation performance of the modified Mule vehicle.

The following assumptions were utilized:

- Coefficient of friction = 0.6
- Assumed equal weight distribution on all wheels

  W = weight of vehicle

  W_2 = weight on rear wheels
- Wheel diameter, D = 66.8 cm (26.3")
- Wheel base, S = 144.8 cm (57")
- S/D ratio = 66.8/144.8 = 2.16

Utilizing the S/D ratio and weight distribution given, an h/D ration of 0.187 is determined, where:

h = obstacle height

D = wheel diameter

Thus h = 0.187 x D = 12.4 cm (4.9") vertical step height.

Ditch crossing ability is intimately related to vertical obstacle negotiation. An h/D of 0.187 yields an ℓ/D of 0.78, where:

ℓ = width of ditch (crevasse)

D = diameter of wheel

thus, ℓ = 0.78 x D = 52 cm (20.5")

7.5.2 Power System

7.5.2.1 Prime Power System. The function of the prime power system is to provide the the electrical power requirements for the vehicle propulsion motors, the auxiliaries required for the proper functioning of the vehicle, and for other power using equipment associated with the mission, if required. The amount of electrical power required for the vehicle propulsion function for each traverse is shown in Figure 7.5-9 for Mod 1 and 2. This is the power profile for one traverse. From this power profile, the maximum electrical power which is required at any one time, and the total number of kilowatt hours per traverse were calculated. Figure 7.4-8 in paragraph 7.4.6 shows the KWH required for the three orders of vehicle modifications for each mission traverse.

7.5.2.1.1 Selection of Prime Power Systems—Extreme simplicity, maximum use of existing space qualified hardware, and the aim to minimize system engineering cost were the principal criteria for selection of the specific prime power system. Having a high reliability per component and the ability of the system to complete its mission with partial component failure were also given a very high weighting. Because of the additional requirement to maximize the energy density of the power system, the obvious choice for the energy storage requirement was the silver-zinc rechargeable battery.

7.5.2.1.2 Characteristics of Silver-Zinc Alkaline Batteries—At present, in the missile and space fields, the most useful high energy density system is the silver-zinc alkaline battery. In the earlier short time space missions, the battery was sealed in an outside container with each cell being of a vented type. It was therefore necessary to have a heavy case which could stand high pressures with some type of safety vent. In this type of unit there was an evolution of gas (hydrogen and oxygen) during stand or during discharge. After a few cycles the pressure would build up to a relatively high level. This pressure was principally due to hydrogen since the oxygen which was also evolved was more readily re-absorbed by the electrolyte and the zinc electrode. As requirements became more exacting, the construction of the cell was changed so that each cell would be capable of being sealed and would operate over many cycles without building up excessive pressure. Modifications were made to the cells to eliminate the formation of hydrogen on either charge or discharge and to allow the oxygen to more readily re-adsorbed by the zinc electrode. Hydrogen gas formation can be suppressed by including over capacity in the negative electrode so

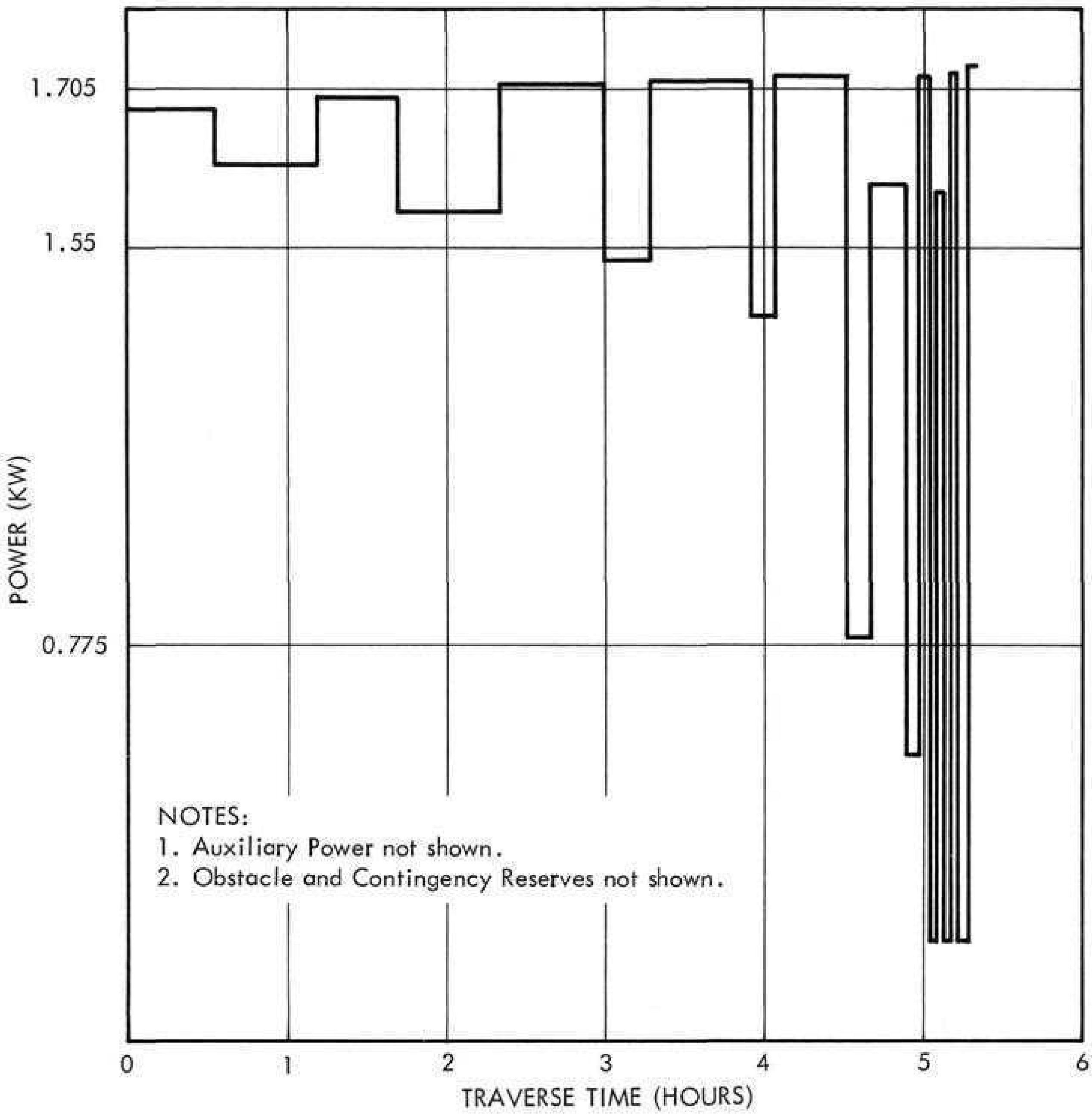


Figure 7.5-9. Power Profile for Traverse - First and Second Order Vehicle

that it is never fully charged and by limiting the final charging voltage. Based on several detailed silver-zinc battery investigations, which were sponsored largely by NASA, batteries have been constructed and rigorously tested which have a high reliability for 20-30 charge-discharge cycles.

One of the major problems in sealed cells is to obtain matched cells for each assembly of cells in a battery. There is always a slight variation in capacity in normal cells. These capacity differences become even more critical in sealed cells since only a reduced amount of electrolyte is used so as to allow most of the zinc surface area to be available for the absorption of oxygen to reduce internal pressures. Therefore, in the construction of sealed silver-zinc batteries it is necessary to prepare a large number of cells and cycle them after their formation period to pick cells precisely matched in capacity, discharge voltage and charge voltage so that completely balanced batteries can be built. After the cells are matched and assembled into groups for a battery, they are assembled into sturdy containers which will withstand the differential in pressure when the container is exposed to an extremely low pressure environment. It is particularly important to control the leakage rate from the cells to the outside vacuum environment because, due to the very limited amount of liquid electrolyte present in the cells, a small amount of drying out through loss of water vapor will greatly affect the performance.

Cycling of silver-zinc batteries is done in such a way that they are never fully discharged. The danger in fully discharging a silver-zinc battery is that one of the cells may go into reversal, i.e., gases being generated from the electrolyte at the electrode surfaces with dangerous pressures being built up inside these cell casings. The type of charging which has been found best for these silver-zinc cells is a modified constant potential charge where in its early stages it has a constant current limit. The maximum voltage for charging should be 1.95 times the number of cells in series. This will keep the battery below the gassing potential of each of its cells and will bring it to its fully recharged state. To avoid serious capacity losses, the batteries should be brought to a temperature of 288.6^{o}K to 294.4^{o}K (60-70^{o}F) before either charging or discharging of the cells is started.

7.5.2.1.3 Selection of Specific Silver-Zinc Battery—After surveying the sealed silver-zinc oxide batteries which have been qualified for space missions, it became evident that one battery, the Model 205 Surveyor battery, had the appropriate capacity and charge-discharge characteristics for this application. In fact, it was the only fully qualified system which was capable of giving the approximate ten cycles required and which had the appropriate ampere hour capacity for vehicle propulsion. The Model 205 Surveyor battery is sealed and consists of fourteen series connected, rechargeable, silver oxide-zinc cells. The cells are assembled in a magnesium canister and the total assembly weighs 21 kg (46.2 pounds). Each of the fourteen cells is connected to a unique common manifold which allows equalization of gas pressures in the battery cells while preventing the establishment of a common electrolyte path which could discharge the cells. This manifold is fitted with a pressure transducer which monitors the internal battery pressure. Whereas the standard operating pressure is between 0 to 1.03 bars (0 to 15 psia), it is necessary to determine that higher pressures are not periodically generated which may endanger the cell and lead to rupturing of the case. A pressure override is activated from the signal produced by the internal pressure measuring transducer which cuts the battery out of the circuit if an internal pressure in excess of 3.45 bars (50 psia) is encountered. This design enables the battery to be charged by a constant potential-current limited type charger, which is the most effective to fully regenerate the capacity of the cell with highest efficiency and in the minimum amount of time. The internal battery pressure does rise to a limited extent during the transition from constant current limitation to the constant potential charge but begins to decay very soon. This method of charging insures that the battery is always in the highest state of charge possible within the limitation of the mission power profile.

During the development of the Surveyor 205 battery, it became necessary to develop new methods for preventing vibration damage to the battery components. Special positive plates and negative grids were developed for this purpose. The positive plate has plastic struts attached to two of its corners. These struts are also attached to the cell case and thus limit relative motion between the positive plates and the cell case. The negative grid has a plastic frame attached to it which makes the negative plates less compressible during vibration and therefore insures maintenance of packed tightness and eliminates relative motion between the various cell components and the cell case. During qualification of this model 205 battery, these features have been proven adequate to enable it to meet a severe vibration requirement.

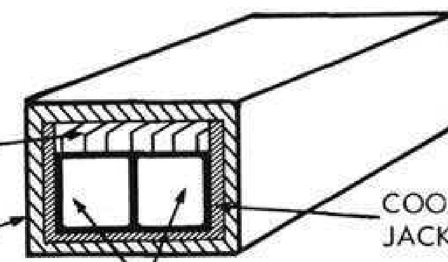
Two of these model 205 Surveyor batteries are placed electrically in series and packaged in thermal insulation and a cooling jacket to provide the "battery pack"; which pack will be the source of power for the vehicle propulsion, see Figure 7.5-10. Each battery pack, therefore, consists of 28 cells connected in series. Each cell is designed to discharge at 1.5 volts. The battery pack, therefore, delivers 42 volts D.C. at its design maximum discharge rate of 14 amps. Approximately 70 ampere hours are discharged from each battery pack during the traverse for the first modification vehicle. This corresponds to a 41% depth of nominal discharge capacity. Each battery pack is recharged at the 6 ampere rate until the constant potential of 54 volts (1.92 volts per cell) is reached and then maintained at this constant potential of 54 volts.

A coolant is circulated through the outer battery case to maintain a temperature of not less than 288.6^{o}K (15.6^{o}C or 60^{o}F) to not greater than 321.9^{o}K (48.9^{o}C or 120^{o}F) in the cooling jacket. Heat is generated by the action of the electrochemical cells. During charge the battery pack will generate 38 watts of heat at its maximum design rate of 14 amps (100^{o}F). During recharge at the 6 amp rate at 310.8^{o}K (37.8^{o}C or 100^{o}F), approximately 90 watts of heat must be dissipated by the coolant. During discharge at less than the 14 ampere rate or recharge at less than the 6 ampere rate, the amount of heat generated will be appropriately reduced. During startup, the cells will be allowed to warmup to a minimum of 288.6^{o}K (15.6^{o}C or 60^{o}F) before the coolant is circulated to the radiator.

Each battery canister can be exposed to the extremely low pressure encountered in space. The batteries when packaged in the outer shell of insulation and cooling jacket will have a volume of approximately 25.4 cm (10 inches) high by 33 cm (13 inches) wide by 38.1 cm (15 inches) long and have a mass of 44.6 kg (98.4 pounds). Based on the surveyor mission qualification, these Surveyor 205 batteries, and hence the battery

TWO MODEL 205 "SURVEYOR" SILVER-ZINC BATTERIES IN SERIES

THERMAL INSULATION AND COOLING JACKET



PHYSICAL CHARACTERISTICS
- WEIGHT 98.4 LBS. MASS
- LENGTH-38.1 cm (15")
- WIDTH-33.0 cm (13")
- HEIGHT-25.4 cm (10")

ELECTRICAL CHARACTERISTICS
- 42 VOLTS AT MAXIMUM DESIGN (14 AMP.) DISCHARGE RATE
- APPROX. 70 AMPERE HOURS DISCHARGE DURING TRAVERSE (41% DEPTH OF DISCHARGE OF NOMINAL CAPACITY)
- RECHARGED AT CONSTANT POTENTIAL OF 54 VOLTS (6 AMPERE MAXIMUM))

THERMAL CHARACTERISTICS
- COOLANT TEMPERATURE: 60°F TO 120°F
- HEAT GENERATED, DISCHARGE: 38 WATTS (14 AMP. RATE AT 100°F)
- HEAT GENERATED, CHARGE: 90 WATTS (6 AMP. RATE AT 100°F)

RELIABILITY:
- 0.998 80% CONFIDENCE (SURVEYOR MISSION QUALIFIED)

Figure 7.5-10. Power System Battery Pack

pack, have a reliability of 0.998 at 80% confidence. This battery is manufactured by the Missile Battery Division, of the Electric Storage Battery Company and is presently available as flight qualified hardware for this application.

7.5.2.1.4 Description of Prime Power System—The requirement on the power system is to deliver to the electric drive system from 0 to 56 amperes at 42 volts D.C. For protection of the battery power system a 65 ampere breaker is included in the bus line as seen in Figure 7.5-11. Four 52 volt D.C. battery packs are connected in parallel. With uniform load sharing, this would amount to each battery pack drawing a maximum of 14 amperes. A small series resistance is used with each of the battery packs so that total resistance in the battery circuits are equal and to provide uniform load sharing at the maximum load point. An override is used with the battery packs in the discharge mode. This override disconnects the battery pack from the parallel arrangement if the internal pressure in either of the batteries of the battery pack indicates that a gas pressure of greater than 3.45 bars (50 psia) has been reached. If this override is activated, a light on the control panel will indicate that this battery is not in circuit. This is the only indicator required to confirm proper functioning of the batteries.

During the recharge of the cell, the electrical power is provided through a cable from the LEM/S. At the beginning of recharge, 24 amperes at 54 volts are impressed across the parallel array of battery packs. The battery charge controller, Figure 7.5-12, associated with each battery pack, limits the recharge current which passes through any one of the battery packs to a maximum of 6 amperes at 54 volts. It also discontinues charging if the internal battery pressure of either of the batteries in the battery pack exceeds the value of 3.45 bars (50 psia). The exact voltage which must be obtained from the control box in the shelter will be a function of the length and size of the cable connecting the shelter with the recharging vehicle.

From the power summary shown in Figure 7.4-8 for the Mod 1 vehicle, it is seen that approximately 12 killowatt hours of electrical energy would be consumed per traverse. Approximately 16 kilowatt hours of electrical energy will be required per traverse from the shelter power supply to fully regenerate the battery system. A period of

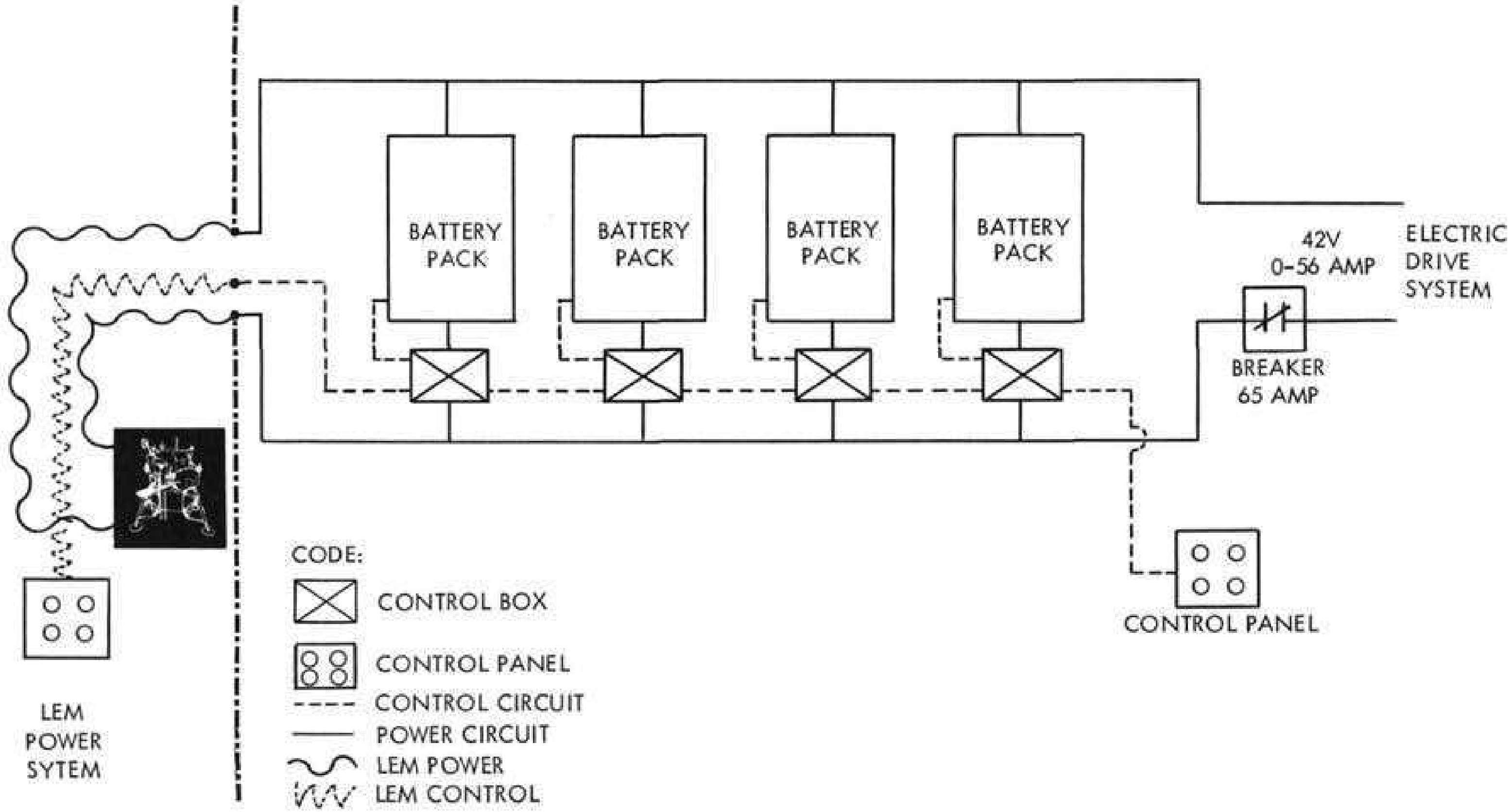


Figure 7.5-11. Power System - First and Second Order

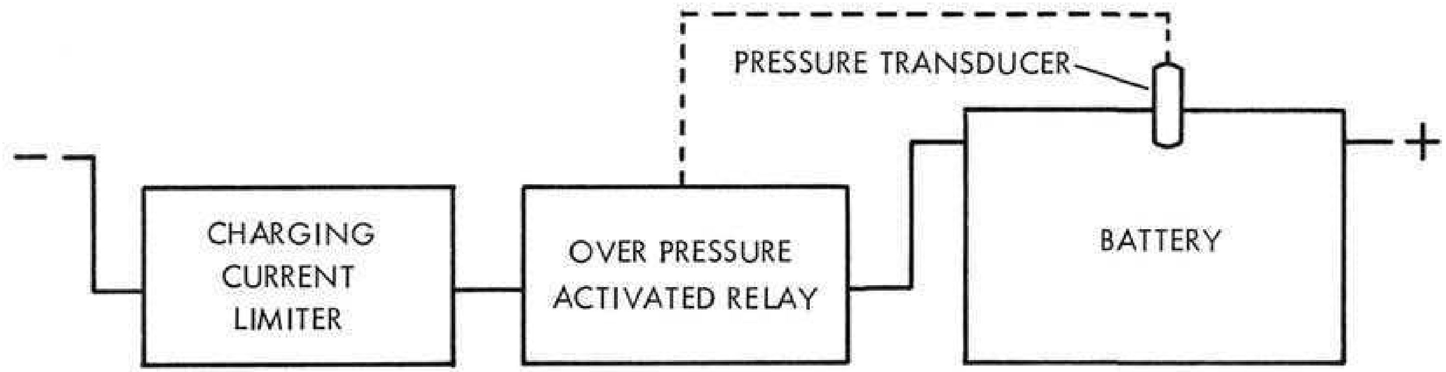


Figure 7.5-12. Battery Control Box - Power System - First and Second Order

from 16-18 hours will be required after a traverse has been completed and charging started from the shelter to fully regenerate the system so the next traverse can be undertaken. The mass of primary fuel which must be stored in the shelter to provide the approximate 16 kilowatt hours per traverse will be dependent on the type of energy conversion equipment which is used in this shelter and its electrical energy conversion efficiency. An estimate of this mass, dependent upon use of an H_2-O_2 shelter system is used for calculation of shelter fuel penalties. The total mass of the power system which is mounted on the vehicle includes four battery packs at 94.6 kg (98.4 lbs) each, and the total mass of the individual battery pack control boxes, the control panel and the bus bar is estimated to be approximately 11.3 kg (25 lbs). The total mass of the system is therefore approximately 190.5 kg (420 lbs).

The parallel arrangement of four battery packs, see Figure 7.5-11, provides a redundancy which can improve the system reliability. Each of the battery packs has a reliability of 0.998, 80% confidence, based on the use of the flight qualified Model 205 Surveyor battery. Since in the normal traverse only about 40% of the nominal capacity is being discharged, any two of the four battery packs which remain operational could provide the power needed to complete the traverse in progress. Two simultaneous battery pack failures would, therefore, not be sufficient to cause a failure of a single traverse. The much higher drain on a battery pack caused by the failure of one or two of the other packs would, however, reduce the reliability of the pack to perform future recharge and discharge cycles. Based on this, however, the reliability of the power system to complete any single traverse is extremely high because of this redundancy and its ability to successfully complete the total mission is higher than 0.998 at 80% confidence based on the battery component. The control components are very simple and can be designed and tested to insure component reliability equal to that of the cell and battery reliability.

7.5.2.1.5 Alternate Prime Power Systems—The power system incorporating the presently flight qualified Surveyor 205 zinc-silver oxide battery is by far the least complex, most highly reliable, and least costly system for providing the energy requirements of the vehicle for a traverse. Its principal disadvantages are that it is dependent upon the LEM/S for recharge power, and must be connected to the LEM/S with an electrical power cable for periods of 15 to 18 hours to accomplish recharging. An alternate approach would be to have the battery charging power generator on board the vehicle. In this mode of operation, primary fuels would be stored in the LEM/S and transported to the vehicle where they would be converted to electrical power to provide for the recharging of the secondary batteries. Again taking into account the simplicity of the proposed system, the use of previously flight qualified hardware and the lowest cost, a fuel cell-battery pack combined system is proposed as an alternate system, see Figure 7.5-13.

Fuel

Because of the large amount of experience presently available with hydrogen and oxygen in their cryogenic state, this fuel and oxidant were selected for use with a fuel cell battery charger to be mounted on the vehicle. Flight qualified Gemini cryogenic tankage would be used to contain both the hydrogen and the oxygen. Two oxygen tanks and two hydrogen tanks would be stored on the LEM/S until the vehicle is on the lunar surface and ready for its first traverse. At this point, one oxygen tank and one hydrogen tank would be manually removed from the lander and placed on, and connected to, the vehicle power system. These tanks would contain the hydrogen and oxygen required for 5 sorties. After the 5th sortie when the mission is half completed, the hydrogen and oxygen cryogenic tankage would be exchanged for the additional tanks stored in the LEM/S.

Fuel Cell System

The existing flight qualified fuel cell which was selected for this application was the three-hundred fifty watt FCO12A manufactured by the General Electric Company which has been developed for the Biosatellite, see Figure 7.5-14. This system operates on hydrogen and oxygen and consumes 0.4 kg (0.9 lbs) of reactants per kilowatt hour. In the proposed power system shown in Figure 7.5-13, two of these fuel cells in series would replace one of the battery packs. During discharge the fuel cells would be in

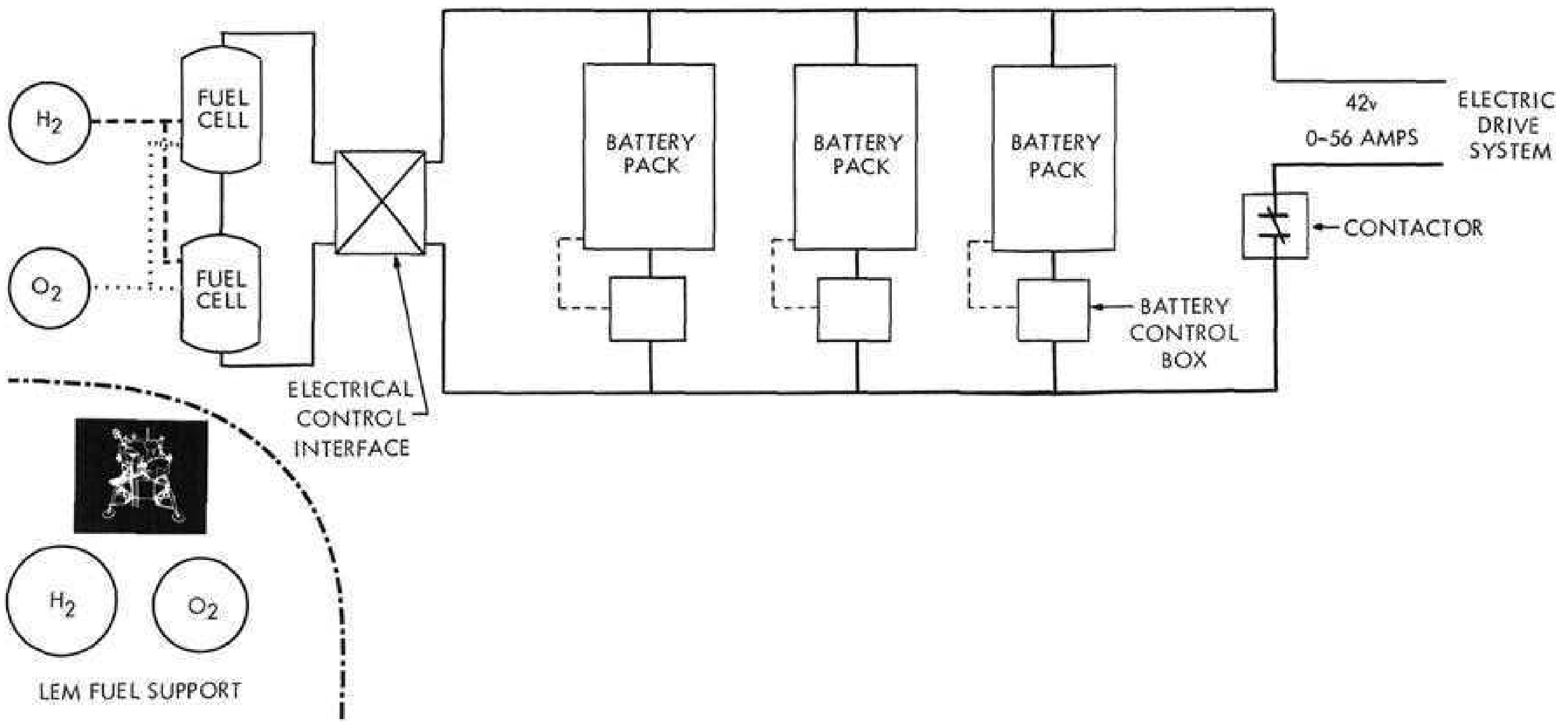
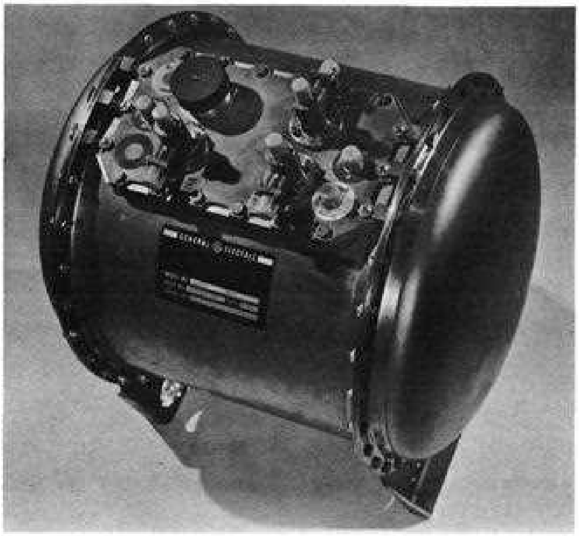
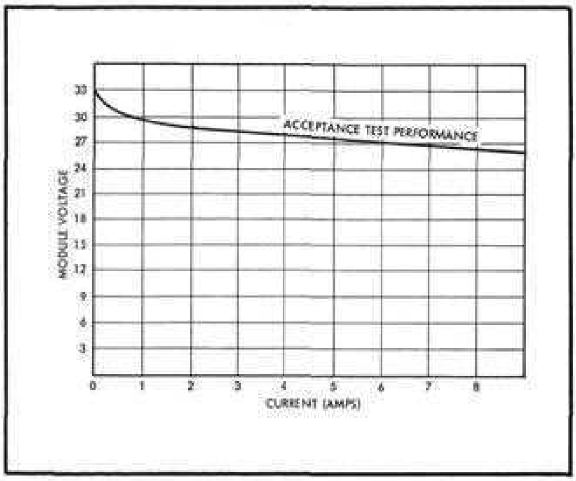


Figure 7.5-13. Battery - Fuel Cell Option - Power System - First and Second Order





Polarization Curve for 350-Watt Fuel Cell Battery

GENERAL DESCRIPTION

A practical power source for spacecraft equipment on missions from 30 to 60 days*. Operates on hydrogen and oxygen. Simplicity and inherent reliability are achieved by use of a shock-and vibration-resistant solid electrolyte. To operate independent of gravity with power output unaffected by orbit or vehicle attitude. Potable product water, useful for life support or equipment cooling, is removed by unique gravity-independent capillary system. Low temperature and pressure characteristics facilitate installation and start-up.

TECHNICAL DATA

Output

- Voltage: 27 ± 4V, DC. Peak: 350 Watts
- Operational Life (Test Results): 1200 hrs.*
- Product Water: One pint/kwhr

REACTANT CONSUMPTION

- Total Consumption: Approx. 0.9 lbs. /kwhr
- Hydrogen: ≃ 0.1 lbs./kwhr
- Oxygen: ≃ 0.8 lbs./kwhr

COOLING

- Type Coolant: Circulating liquid
- Inlet Temperature: Min.: 40°F Avg.: 75°F Max.: 100°F

ENVIRONMENTAL CONDITIONS

- Minimum Pressure: 10^{-8}mm Hg
- Temperature: −60°F to +120°F
- Rain: Protected per MIL-E-5272
- Humidity: 0-100%
- Shock: 15 "G"
- Vibration: 12g sinusoidal and PSD = $0.30g^2$/cps random.
- Zero-gravity
- Acoustic noise: 132 db
- Acceleration: 12.5 "G"
- Salt Sea Atmosphere: Protected per MIL-E-5272
- Sand and Dust: Protected per MIL-E-5272

PHYSICAL DIMENSIONS

- Length: 16.5″
- Diameter: 14″
- Weight: 35 pounds, including pressure regulator

*A function of load and operating parameters.

Figure 7.5-14. 350-Watt Fuel Cell Battery parallel with the batteries being discharged and during periods of no power draw the fuel cell would apply voltage across the battery pack and recharge the battery pack. Each fuel cell has the all over dimensions of 35.6 cm (14") in diameter by 41.9 cm (16-1/2") long. The mass of each fuel cell package is approximately 15 kg (33 lbs).

Two of these fuel cell canisters will be placed electrically in series. The proposed reactant supply and control system is shown in Figure 7.5-15. This fuel cell unit is being submitted to qualification tests for the NASA Biosatellite Vehicle.

Principal System Advantages

This system has the advantage that it can operate independent of any support from the LEM/S for electrical power for a total of 5 traverses. Since no special reconnection to the shelter is required before charging can be started, more efficient use is made of total non-power draw time for recharging the cells. The system control can be designed so that when no power is being drawn by the electric drive system the fuel cell is automatically recharging the batteries.

The advantage of the low temperature fuel cell is that it can easily be started up at the beginning of the lunar phase of the mission. The fuel cell would be run during the entire lunar mission, varying from part of the shared load during discharge to the low or maximum rate of battery recharge depending on the state of discharge of the battery.

The type of taper recharge current obtained from a fuel cell in recharging silver-zinc cells is largely self-controlling. At lower currents, higher voltages are obtained giving a modified constant potential charging.

System Disadvantages

Because the fuel cell is an active power generation unit the by-product heat must be rejected. During the maximum discharge encountered in a traverse, this adds an

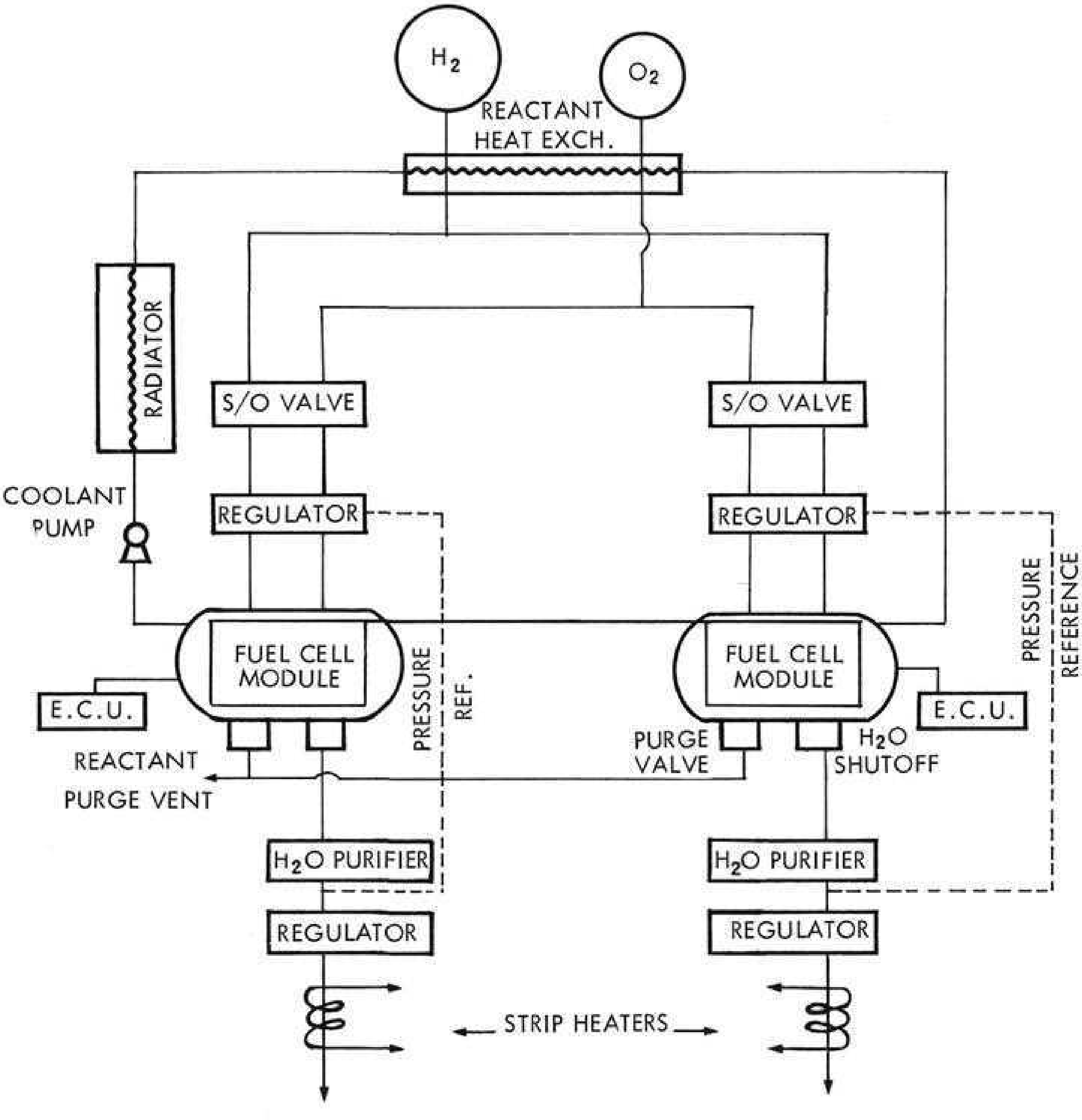


Figure 7.5-15. Fuel Cell System - Power System

additional 300 watts which must be dissipated in the vehicle radiator system. One of the disadvantages of the General Electric type of fuel cell system, which uses an ion exchange membrane, is the low temperature, $285.4^{\circ}K$ to $296.6^{\circ}K$ ($12.4^{\circ}C$-$23.6^{\circ}C$, or 80-$100^{\circ}F$), at which heat must be rejected. This heat rejection therefore requires a large increase in the size and mass of the vehicle radiator system. Taking this increase of radiator size into account, the total mass of a battery system alone (compared to the system of batteries plus fuel cell) is the lesser, see Table 7.5.2-1; principally because of the ground rule that the mass of tankage for hydrogen and oxygen used to generate electrical power in the LEM/S for recharge of the vehicle battery system would be assessed to the vehicle. Because of cargo carrying requirements, it is unattractive to provide special hydrogen and oxygen tankage for the on-vehicle mounted fuel cells (which of course are assessed to the vehicle mass). For this reason and the large radiator area (mass) required, see Table 7.5.2-1, no further consideration was given to the fuel cell plus battery system, and it is not recommended as the primary power system for the Mod 1, Mod 2, or the Mod 3 vehicles.

7.5.2.2 Electric Drive System. For the first order modification vehicle, the present engine and its fuel supply are replaced by an electric drive motor with associated controls, and a D.C. power source. The transmission system of the basic vehicle is functionally retained, and the motor is connected to it at the point of the present engine attachment. In order to be able to make maximum use of basic vehicle hardware, maximum input speed to the transmission is limited to 4200 rpm. Shift gears and transfer case of the basic vehicle are available for use as required.

7.5.2.2.1 Drive Selection—A variety of electric motors and control schemes are available for purposes of propulsion. For the first order modification vehicle, the guidelines of minimum cost and minimum complexity are considered to be paramount, at the expense of optimum performance if necessary.

The drive scheme of minimum technical complexity would involve a system in which power to the motor is either completely-off or full-on. The operator of the vehicle would control the time length of power flow to the motor. He thus would set average

Table 7.5.2-1

COMPARATIVE MASS TABULATION FOR SELECTED VEHICLE ENERGY SYSTEM, BATTERY PACKS FOR 1st AND 2nd ORDER MODIFICATION, AND ALTERNATE ENERGY SYSTEM BATTERY PACKS AND LOW TEMPERATURE FUEL CELL

System Component	Alternate System			Selected System		
	No. of Units	Mass		No. of Units	Mass	
		Kg	Lbs.		Kg	Lbs.
Battery (Unit)	6	136.5	300	8	182	400
Fuel Cell	2	34	75	-	-	-
Auxiliary Controls	-	-	-	1	9	20
Recharger	1	9	20	1	9	20
Fuel, H_2 & O_2	-	54.5	120	-	65.5	144
Recharge Umbilical	-	-	-	1	7	15
Fuel Tankage		68.5	150	*	*	*
Heat Radiator (System)	1**	71.5	157	1**	42.5	93
Pump		2	5		2	5
TOTALS		376 317	827 697		317.0	697
MASS PENALTY OF ALTERNATE SYSTEM		59	130		———	———

*By definition of the problem, tankage mass in the shelter is not charged
**Based on a/e = 0.133

vehicle speed as a function of maximum speed to which he permits the vehicle to accelerate while power is full-on; and of minimum speed to which he permits the vehicle to coast while power is off. This drive system would have required essentially only a motor, and a contactor to apply power to the motor, but had to be abandoned from a vehicle operational point of view because of the uncontrolled torque while power is full on. The vehicle would be expected, in this case, to dig a hole in the soft soil specified for a part of the lunar traverse rather than to move forward. Some sort of dynamic power and torque control of the motor is deemed essential to operation on the lunar soil.

The required control and propulsion characteristics are available with both A.C. and D.C. drive systems. In an A.C. system, the motor speed and hence the vehicle speed, is varied by varying the frequency applied to the motor. Voltage is varied as required to hold the saturation of the drive motor within desired or permissible limits. Control of voltage and frequency is accomplished from one operator command, and sets the torque level of the motor, thus controlling the motor and the vehicle. Since the available power supply is a D.C. source, a multi-phase inverter with a frequency range corresponding to the motor speed range, is required. For the vehicle speed requirements under consideration, 1.1 km/hr (0.68 mph) as a reasonable minimum to 8 km/sec (5 mph), this amounts to a relatively modest 7-1/3:1 drive range even without use of a gear shift. The rugged squirrel-cage induction motor, the "workhorse" of the industry can be used. However, it must be specifically designed to meet the requirements of a given traction application, even in terrestrial environment. There is no such thing as a "standard" A.C. traction motor, nor is there envisioned to be one within the time frame covered by this study.

In a D.C. drive system, the motor speed and hence the vehicle speed is varied by varying the average voltage applied to the motor. Starting with a D.C. supply of constant voltage, a means must be provided to obtain a smoothly variable voltage system. A motor with commutator and brushes is required, and an increased maintenance problem as compared to the squirrel-cage motor is involved. This problem can become particularly troublesome under adverse environmental conditions. However, "standard"

D.C. traction motors for terrestrial use are available. Thus in a D.C. system, motor development efforts can be concentrated on the lunar aspects of the application, yielding a minimum time and cost situation.

The major problems associated with operating in the lunar environment are proper cooling of the drive motor and proper operation of the commutator and brushes used. For the required 1.32 kw (1-3/4 hp) propulsion motor, time averaged losses to be dissipated during a traverse amount to approximately 320 watts. Peak loss during the traverse is 400 watts for 3.6 minutes. An additional loss of approximately 250 watts from the electrical drive and motor controls must be transferred out of the vehicle. These losses are too high to be transferred and radiated without use of a heat-transfer fluid. Utilization of such a fluid, carries the added benefit of permitting use of a sealed motor under some gas pressure, thus also establishing a feasible environment for proper functioning of commutator and brushes.

Based on the foregoing considerations, the first-order modification vehicle will be powered by one drive motor through the transmission of the vehicle. The unit will be a D.C. traction motor fed from a variable voltage D.C. supply. Heat will be carried away by a transfer fluid and cooled via a radiator to space.

7.5.2.2.2 System Description—The block diagram for the drive system is shown in Figure 7.5-16. The D.C. power source supplies an essentially constant voltage to the voltage control unit. Based on command from the operator of the vehicle, this voltage is converted to the proper average level as required by the D.C. drive motor for the desired speed and for the soil and grade conditions of the traverse. The drive motor is connected to the wheels through the transmission of the vehicle.

A schematic of the voltage control unit and its mode of operation are shown in Figure 7.5-16A. The D.C. voltage supplied from the power source is chopped by a power transistor T. Pulses are permitted to flow for a fixed period of time. Thus the frequency of pulse turn-on sets the average level of the output voltage E from this unit. In order to prevent pulsed current flow with its attendant problems of high peak values and

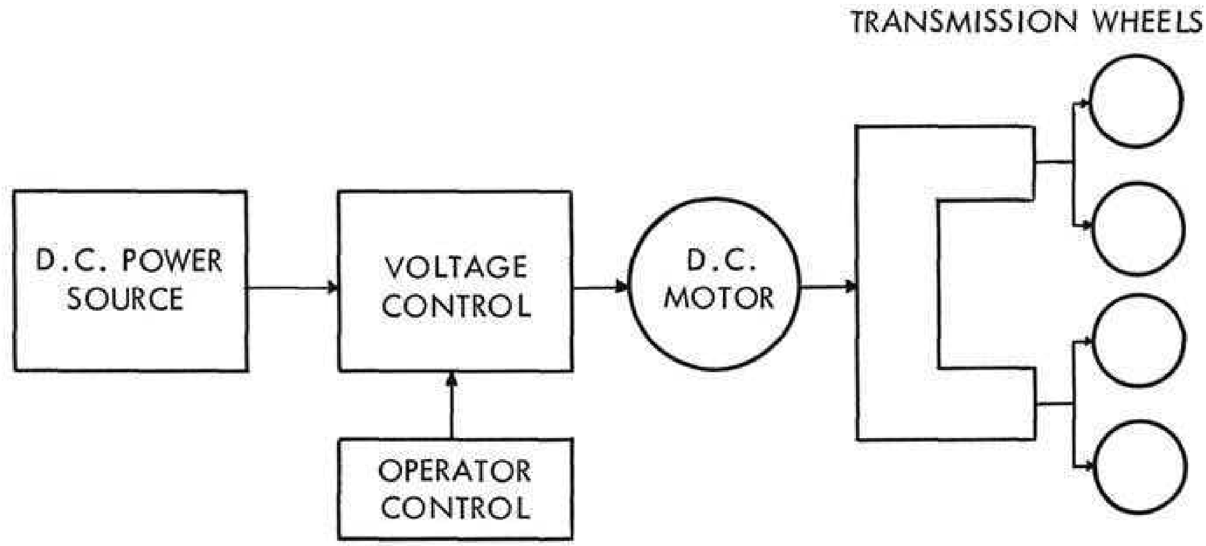


Figure 7.5-16. Electric Drive System - First and Second Order

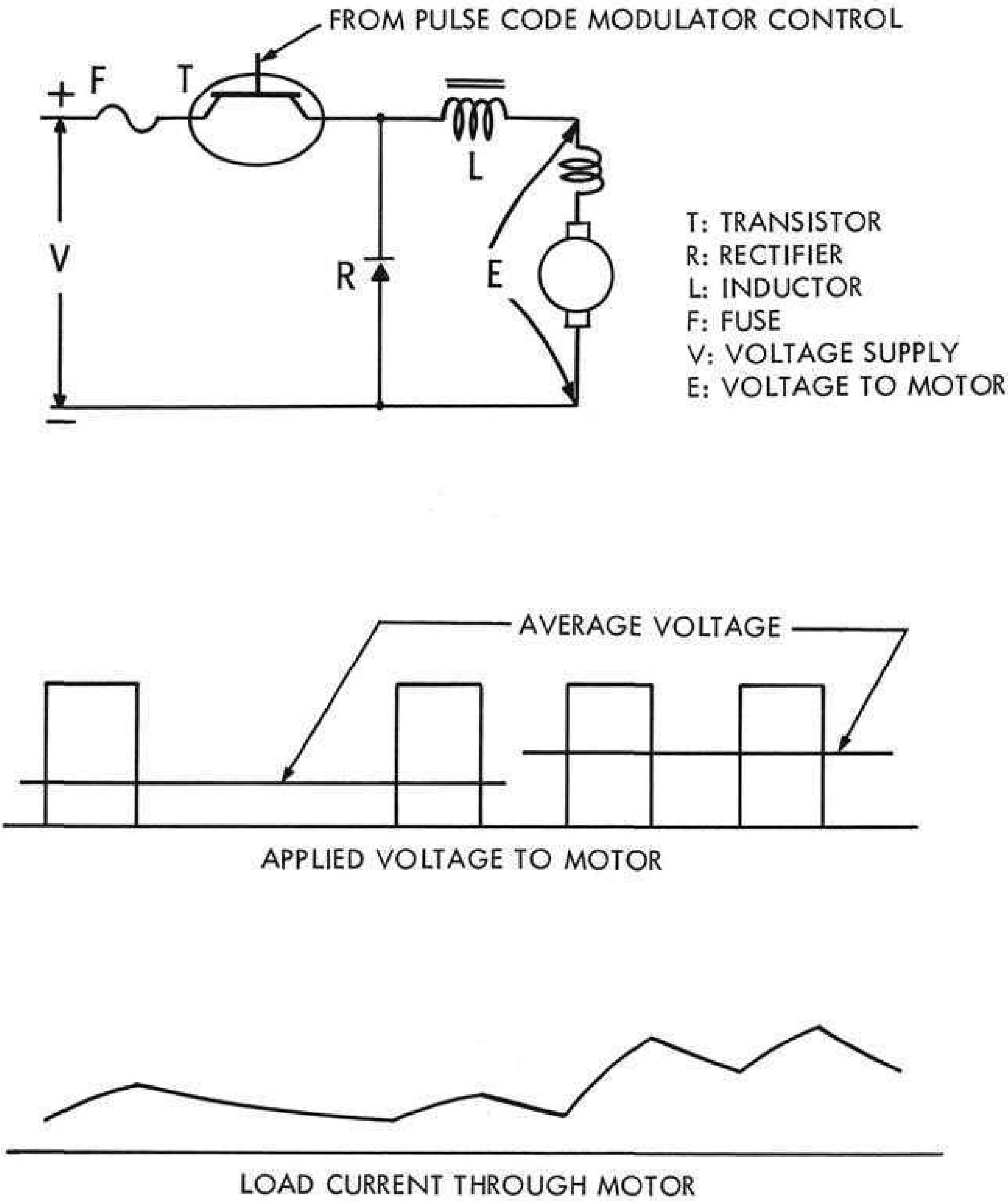


Figure 7.5-16A. Electric Drive Voltage Control - First and Second Order

multiply connected transistors, inductor L is added to the circuit. The inductor is used as an energy storage device, storing energy while pulse voltage is applied to the circuit and returning it during the voltage off period. Rectifier R permits the energy exchange mechanism to take place. The remaining two charts on this Figure illustrate the nature of the voltage and current applied to the motor.

Figure 7.5-17 shows a block diagram of the pulse code modulator control. This control feeds the base of the power transistor of Figure 7.5-16A setting up the pulse application of voltage to the propulsion motor. By changing the value of the two-gang potentiometer P, the operator of the vehicle sets the output (clock) frequency of a stable multivibrator. The wave is fed to a monostable multivibrator which permits output for a fixed period of time whenever it is triggered. The output controls the base of the power transistor, and is seen to be an exact replica of the voltage signal being fed to the motor. The desired voltage to be applied to the motor is thus generated at low power levels, and the power transistor is essentially used as a switching amplifier. A simple voltage regulator for the supply to the logic circuitry may be required.

Pulse code modulation systems of the general type described are being used in terrestrial applications. Although off-the-shelf units are not available for propulsion applications, the system is being used in developments for the military in ratings up to 11.2 kw (15 hp). The major developmental problem is thus its adaptation to the lunar environment.

In order to prevent excessive current flow to the motor when the operator of the vehicle calls for excessive voltage at a given speed, current limit circuitry is incorporated into the basic control. A voltage signal from a current shunt measuring motor current, is compared with a reference voltage representing the maximum allowable current. In case of excessive current flow, a logic "and" gate is used to cut off the transistor base signal and thus reduce voltage and current.

Dynamic braking will not be used. The Mule has adequate braking capability, and addition of the required resistor grids and contactors to affect dynamic braking, is not considered warranted.

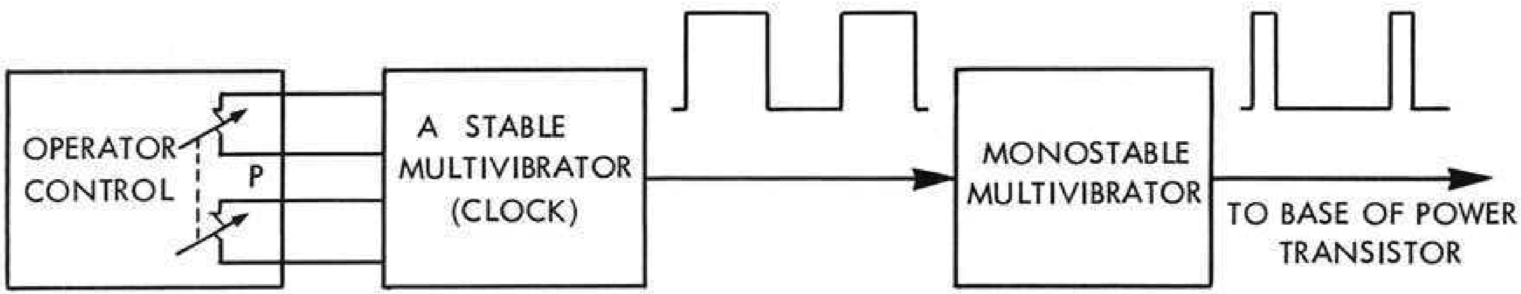


Figure 7.5-17. Pulse Code Modulator Control - First and Second Order

7.5.2.2.3 System Parameters—The first vehicle power requirements calculation for the designated lunar traverse was made with rigid metal wheels corresponding to the standard tire size (8.3" width) for the Mule vehicle. Based on the required power input to the vehicle transmission, the motor was sized at 1.86 kw (3.5 hp) between 4000 rpm and 3000 rpm dropping at constant torque to .3 kw (0.4 hp) at 500 rpm. Motor power, torque and efficiency for this situation are shown plotted versus speed in Figure 7.5-18, the applied voltage being varied to yield the designated power curve. Total energy per traverse to be supplied from the power source was calculated at 16 kw hours, plus energy for auxiliary use. The energy figure includes a 20% allowance for obstacle negotiation capability and a 10% margin to cover coolant pump drive and contingency allowance. The motor would weight between 22.7 kg (50 lbs) and 24.9 kg (55 lbs). The energy per traverse to be supplied by the power source was considered excessive, and wheels with increased width were included in the first-order modification vehicle (the equivalent of the terra-tire, 12.2" width).

Required motor output, using the wider wheels, is shown plotted versus speed in Figure 7.5-19. The circled points show calculated horsepower-speed requirements for the traverse and maximum speed conditions of the vehicles. The crossed points show corresponding minimum vehicle speed data, the minimum vehicle speed being chosen as 1.1 km/hr (1 foot per second) (0.68 mile per hour). The motor chosen for the application is shown by the curve marked "horsepower". This is a typical power profile for a propulsion motor, and is composed of a constant power and constant torque section. The torque curve is also shown on Figure 7.5-19.

The motor speed requirements were chosen on the basis of using gear ratios available in the present Mule vehicle. The ratios are 64 to 1 in low, 37.4 to 1 in medium and 21.8 to 1 in high. The motor speeds of Figure 7.5-19 are lower than those of Figure 7.5-18 due to the decreased wheel slip of the wider sheels. All the traverse points are handled in low gear and no gear shifting will be required during the specified lunar traverse. However, the specified vehicle speed of 8 km/hr (5 mph) on level terrain in the harder lunar soil group forces the use of second gear in order to remain below the maximum permissible 4200 rpm input speed to the vehicle transmission. As a side

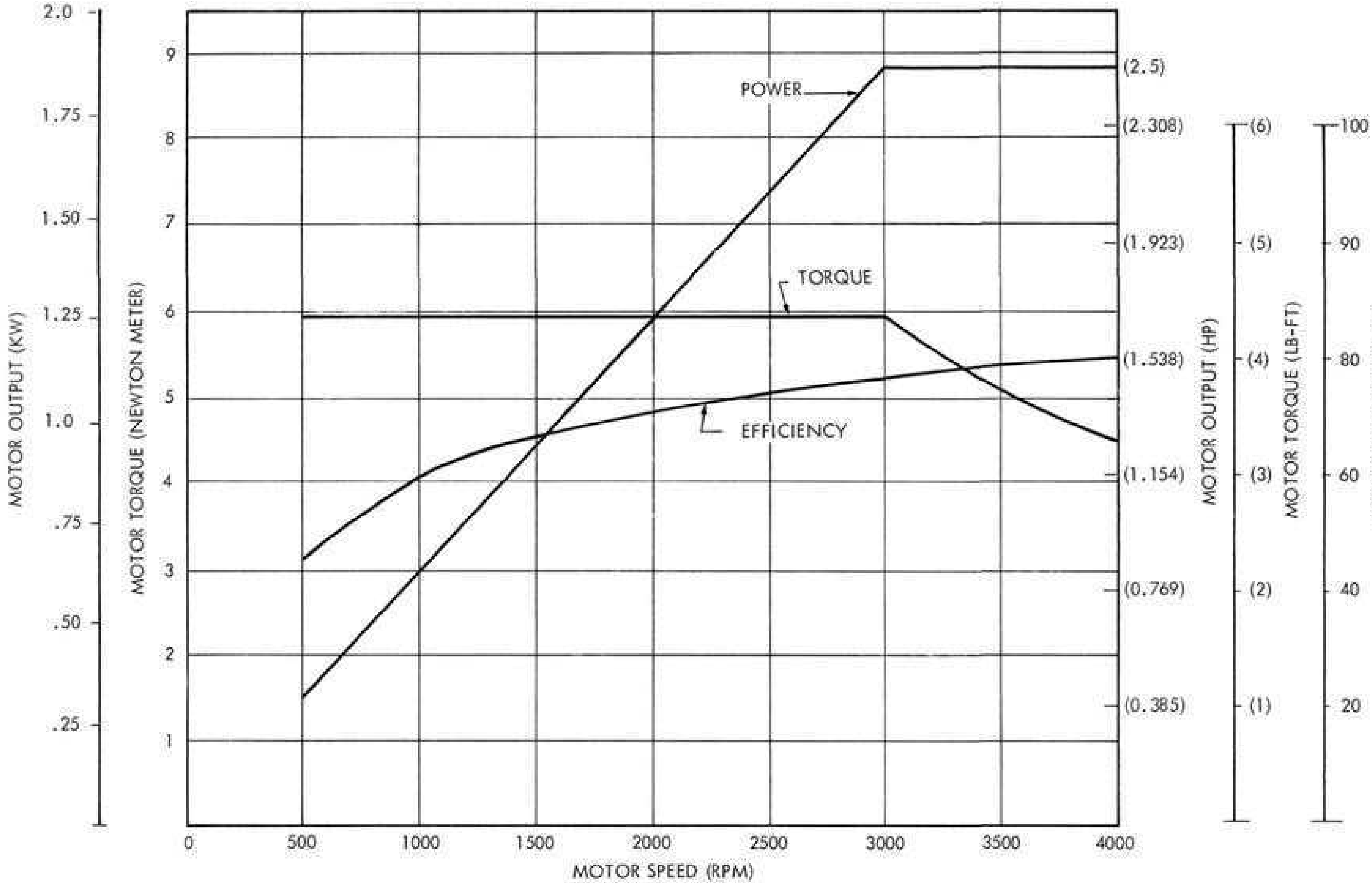


Figure 7.5-18. D. C. Traction Motor Performance

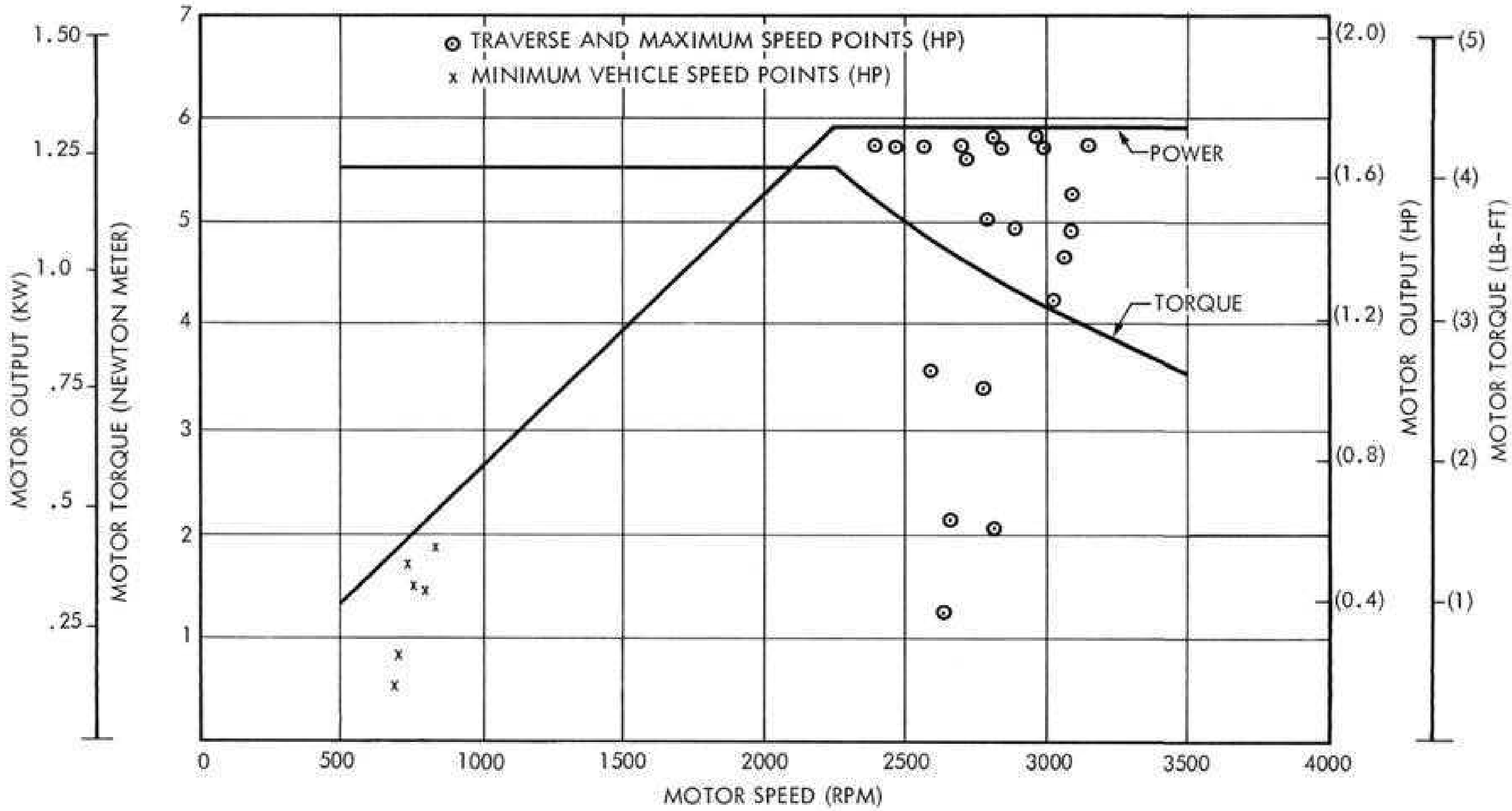


Figure 7.5-19. Traction Requirements - First and Second Order

benefit, use of the gear reduces the speed range of the motor and thus eases the motor and control problem. Since the gear shift mechanism is required, it will be used to back the vehicle. The gear shift lever will be equipped with a pushbutton switch which is used to control the contactor applying power to the motor. Thus the operator of the vehicle will perform the required function of unloading the motor prior to the gear shift operation, by interrupting the power supply to the motor.

The motor performance is shown on Figure 7.5-20. The curve marked "Voltage" shows the required voltage versus speed profile to yield the torque profile of Figure 7.5-19. Less than full power output at a given speed is obtained by decreasing the voltage below the voltage profile. Motor current and efficiency are also shown on Figure 7.5-20, and hold for the power profile of Figure 7.5-19. The vehicle with the power and torque profile of Figure 7.5-19 was chosen as the first-order modification vehicle.

Based on available gear ratios and required wheel speeds, giving due consideration to obtaining a reasonably small constant-power region for the motor, a top motor speed of 3500 rpm was chosen. This choice is reflected in Figures 7.5-19 and 7.5-20 where vehicle power requirements and motor performance are plotted versus motor speed. Motor speed is shown as a function of vehicle speed on Figure 7.5-21, for the hard lunar soil group on level terrain. For these conditions, gears must be shifted near 7.4 km/hr (4 mph) and again near 10.9 km/hr (6.8 mph). The maximum speed is near 13.0 km/hr (8 mph) at which speed the vehicle is power limited.

The required starting torque of the vehicle is based on an initial acceleration of 0.23m/sec^2 (0.75 ft/sec^2) under the worst soil and terrain conditions. The necessary starting torque of 6.8 newton-meters (5 lb-ft) will require a current flow of 57 amperes, and the current limit will be set at 65 amperes. Reference to Figure 7.5-20 shows a maximum current of 52 amperes with the motor fully loaded, a value compatible with the starting current and the current limit setting. Starting voltage is in the neighborhood of 7 volts. The lowest required voltage is based on the minimum desired starting torque. With a minimum starting torque equal to 10% of the maximum value, 2 volts minimum is required to start.

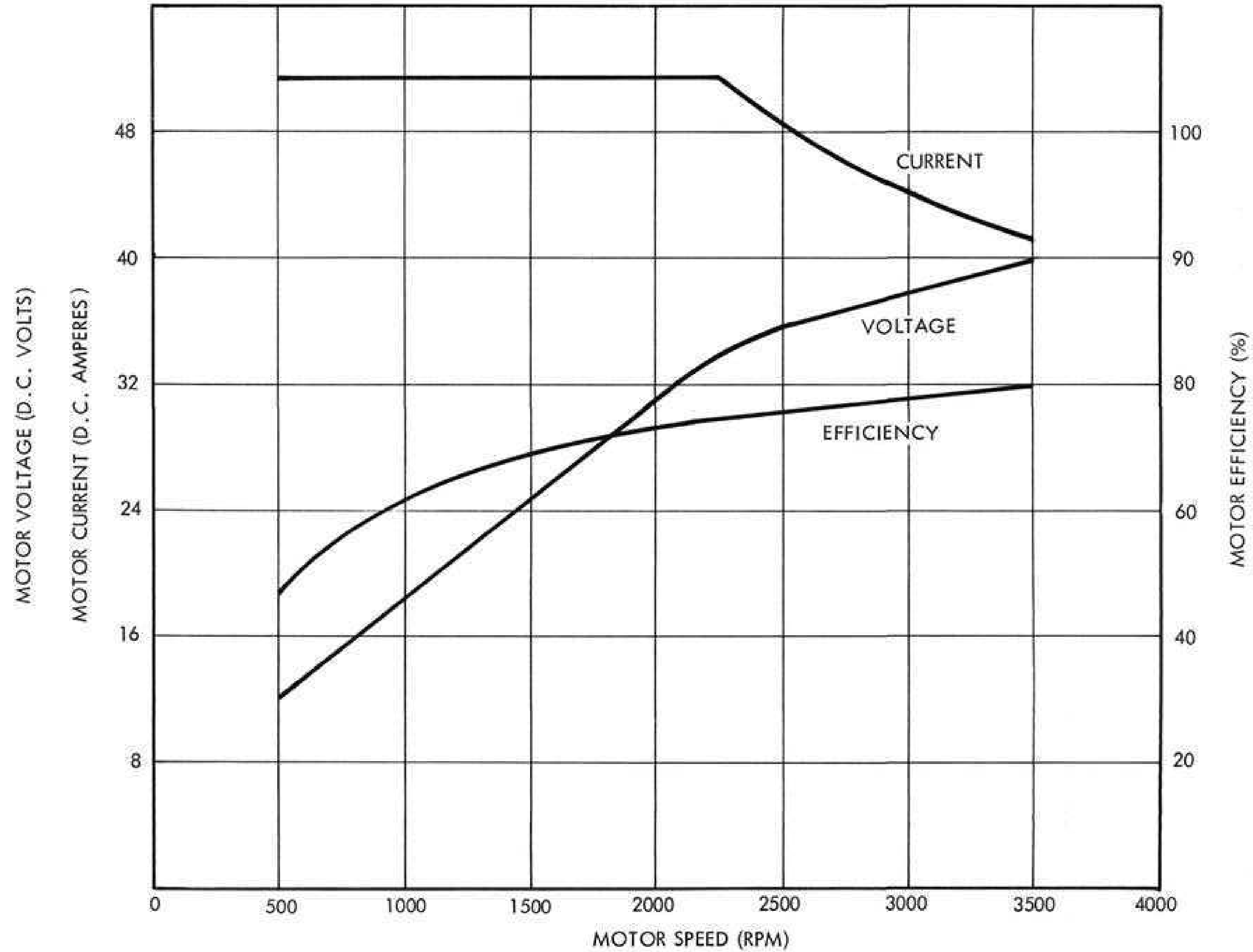


Figure 7.5-20. D. C. Traction Motor Performance

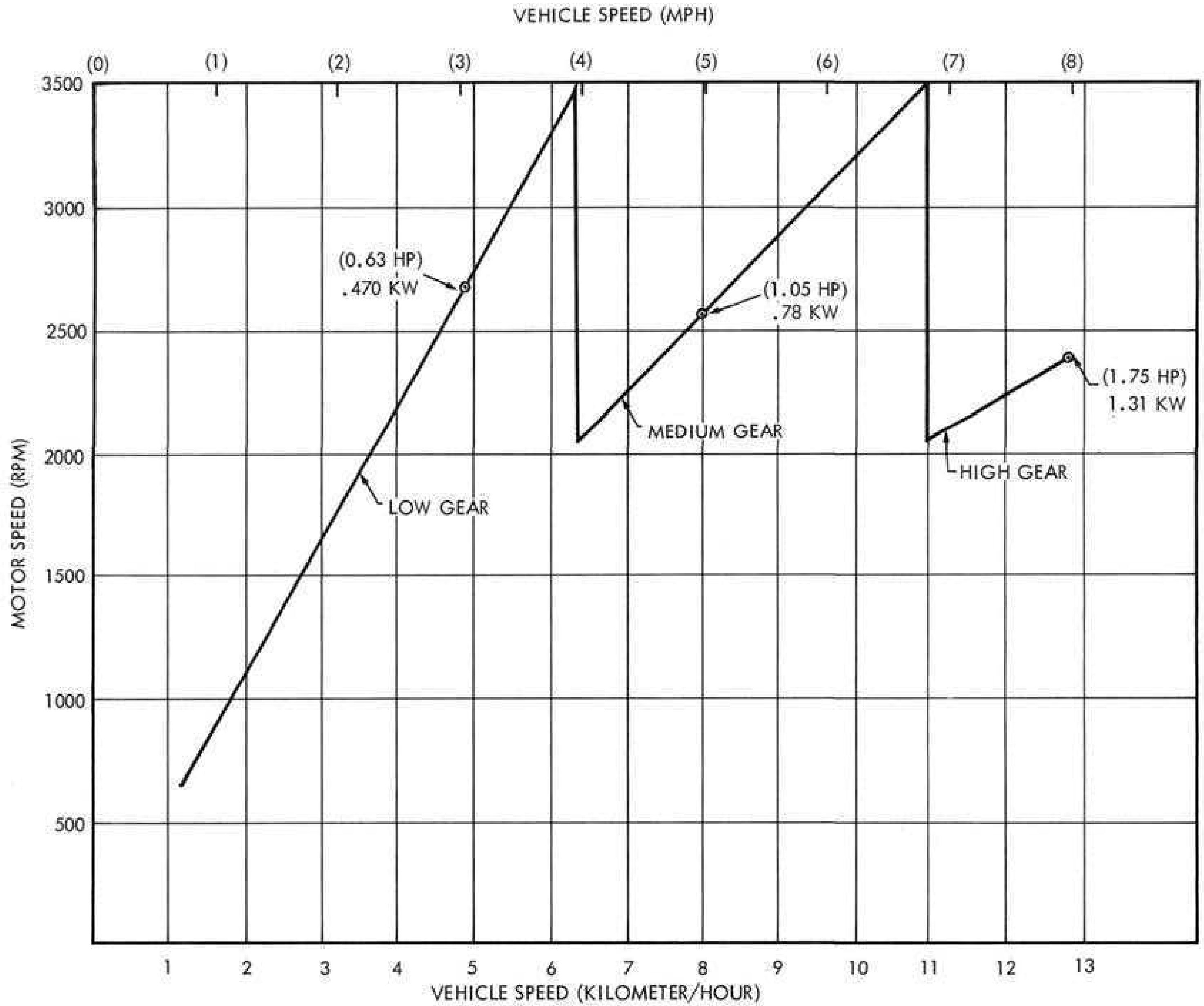


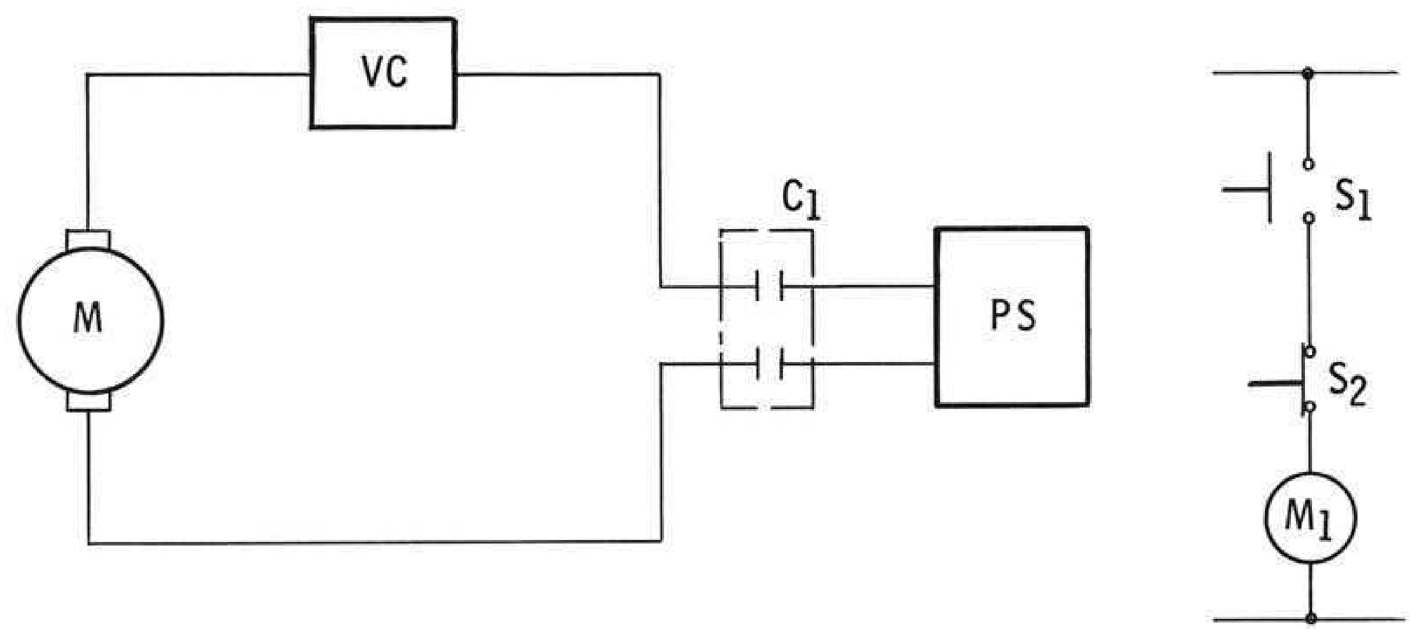
Figure 7.5-21. Motor Speed vs Vehicle Speed - First and Second Order

If the monostable multivibrator of the voltage control unit is set to conduct for 1 millisecond, a clock frequency range of between 50 cps and 1000 cps is required. Adjustment over this frequency range by potentiometer control as planned is entirely feasible. The choke is set for a 10 millisecond time constant to give continuous current flow to the propulsion motor.

The power profile for the traverse is shown in Figure 7.5-9. The energy requirement per traverse was calculated as 12 kw/ hrs. This value includes actual propulsion energy during travel; propulsion control energy only during stops; energy for fourteen starts; 20% reserve for obstacle negotiation capability; 10% for contingency and the energy for coolant circulation. This value does not include energy for other auxiliary functions. The maximum sustained average current during the specified traverse is 57 amperes, including obstacle negotiation reserve. Actual driving time of the traverse is 5.35 hours, allowing 2.65 hours for the specified stops during the eight-hour traverse.

7.5.2.2.4 Power Source and Propulsion System Interface—In order to keep motor currents within reasonable limits and thus be able to operate with a reasonable power transistor configuration, a 40 volt motor has been chosen. The battery power supply packages are 20 volt units, and must be used two in series in order to power the motor. The fuel cell charger proposed for the application and the power available at the LEM/S, are at 24 volts and thus cannot recharge the batteries when connected in series. It becomes necessary to connect the batteries in series for discharge and in parallel for charge.

For vehicle operation, the driver closes switch S_1 of Figure 7.5-22. Contactor C_1M_1 picks up and connects the power source to the motor voltage controller and the motor. Prior to a gear shift operation, momentary contact pushbutton switch S_2 on the gear shift lever is opened, interrupting power flow to the motor. Auxiliary contact A_1 is opened and contactor $A_2C_2M_2$ cannot be closed. Prior to a gear shift operation, pushbutton switch S_2 on the gear shift lever is opened, interrupting power flow to the motor. When the batteries are to be recharged at the completion of a traverse, switch



C_1M_1 : DOUBLE POLE, SINGLE-THROW CONTRACTOR

S_1 : START/STOP SWITCH

S_2 : GEAR SHIFT SWITCH

PS : D.C. POWER SOURCE

M : PROPULSION MOTOR

VC : MOTOR VOLTAGE CONTROL

Figure 7.5-22. Power Source and Propulsion System Interface - First and Second Order

S_3 is closed and contactor $A_2C_2M_2$ picks up. This connects all battery packs in parallel across the charger FC.

7.5.2.2.5 Components—The major components of the proposed electrical propulsion system will be described in this section:

Propulsion Motor

This is a D.C. series motor, with light compounding added to prevent runaway speeds in case of sudden unloading with power on. Mass and volume are determined on the basis of past experience with motors of this type. A plot of mass versus power per unit speed for D.C. motors is shown on Figure 7.5-23. The figures hold for relatively light-weight machines, and are based on determination of a sizeable number of aircooled units. If heat is carried away from the motor by another cooling fluid of comparable effectiveness, mass and volume data for lunar applications should remain comparable. It is of interest to note that the mass of the lunar drill motor being developed for NASA by the Systems Operations Division of the Westinghouse Electric Corporation, falls very close to the curve. The point depicting the drill motor is plotted on the curve. Based on the motor requirements shown on Figure 7.5-19, the motor will have a mass of 24.9 kg (55 lbs).

Volume is determined on the basis of previous experience and on the basis that the D^2L of the machine in cubic inches is essentially equal to the weight in pounds (D is the rotor diameter and L the axial length of the rotor iron). A conceptual schematic section of the proposed motor is shown on Figure 7.5-24. The diameter of the mounting bolt circle set at 30.5 cm (12 inches) is well above the value required from motor considerations. The large bolt circle is used to match the input mounting flange of the vehicle transmission. The mechanical arrangement follows along the lines of that of the lunar drill motor being developed for NASA. The cooling fluid enters the motor through the shaft at the right-hand side and flows along the shaft to pick up rotor heat losses. On leaving the rotor it enters a small storage chamber then flows through the pipes coiled around the stator. Here it picks up stator heat

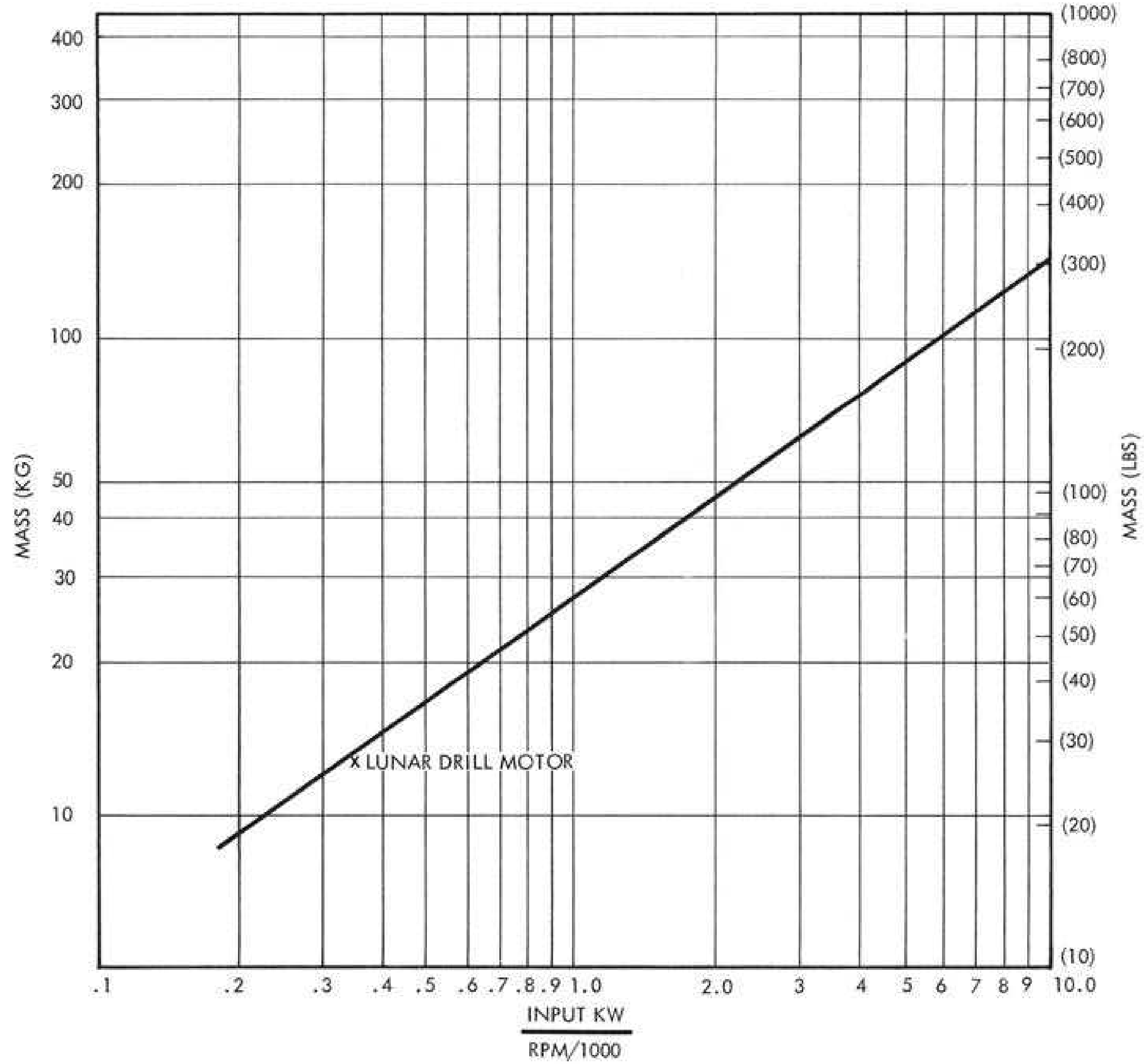


Figure 7.5-23. D. C. Motors - Weight vs Power/Speed

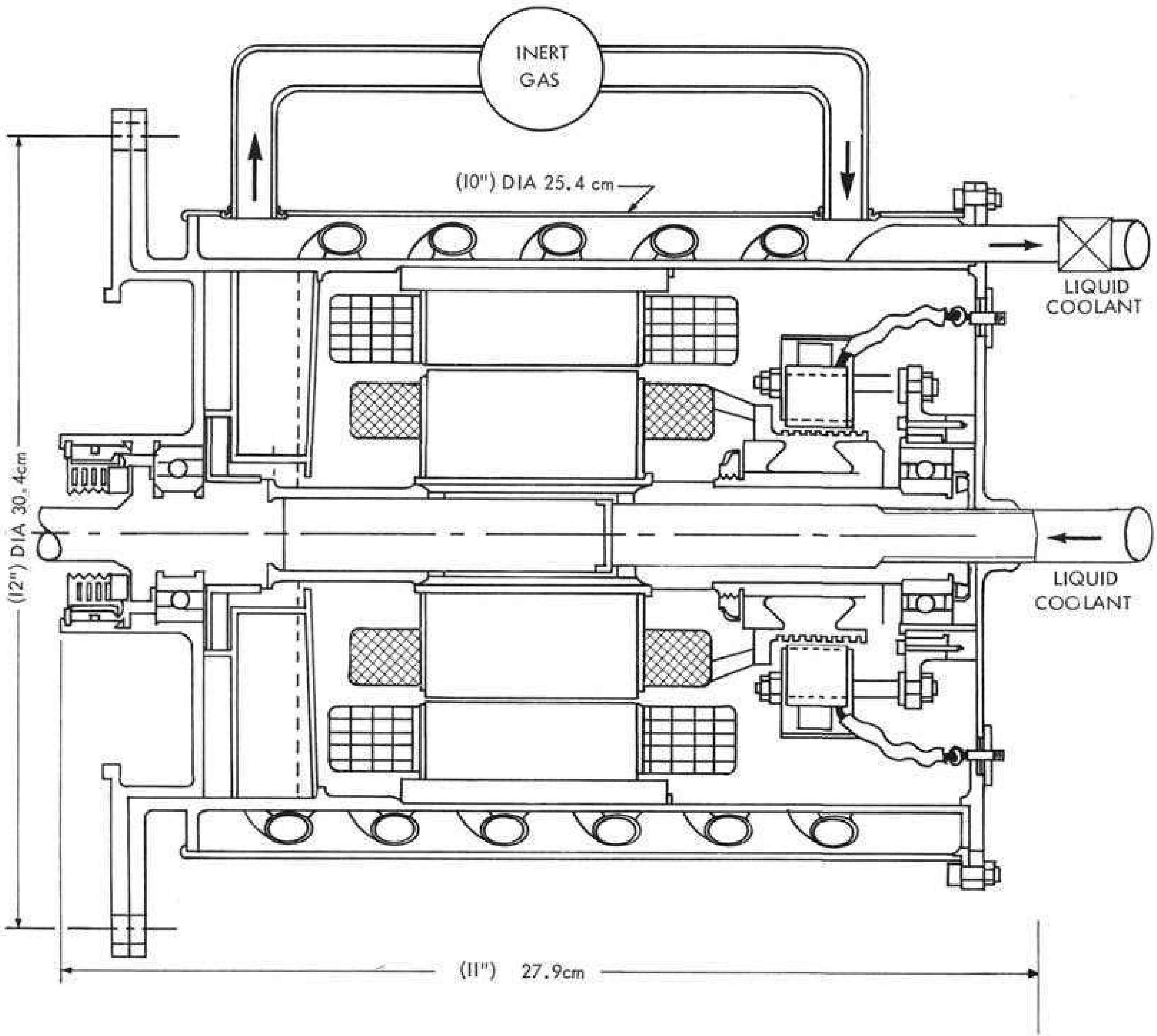


Figure 7.5-24. Propulsion Motor Outline - First and Second Order

losses and then leaves the motor. The motor will be equipped with class "H" insulation and thus be capable of withstanding a 2273°K (2000°C, 3632°F) total temperature in the windings. Due to the short life requirements, hot spot temperatures of the windings can actually exceed this value considerably.

The motor will be filled with an inert gas such as helium, nitrogen or argon. The gas has the dual function of holding down the vapor pressure of the cooling fluid and thus preventing its vaporization; and of providing an atmosphere in which the brushes and commutator of the D.-C. motor can function adequately. It is proposed to transport the gas in a container and fill the motor prior to use, in order to reduce gas leakage problems to a minimum. Holding sufficient gas pressure for the 14-day life cycle of the vehicle is considered feasible, and basically involves proper seal application and development. Approximately .045 kg to .067 kg (0.1 lb to 0.15 lb) of gas will be required to hold a 1034 millibar to 2068 millibar (15 to 30 psia) gas pressure in the motor.

With the proposed motor arrangement, provision of proper seals in considered to be the major developmental problem. These are vacuum seals and must be capable of holding a substantial portion of the gas for a 14-day life of the vehicle. Background experience of vacuum seal development is available from the MOLAB Program. The previously referenced development of a lunar drill motor involves the same seal problems; here Westinghouse reports a successful test of a rotary seal. The General Electric Company plans to charge lunar propulsion motors with helium or argon. It is thus apparent that confidence for a successful seal development for the motor is high. This is especially true in view of the relatively short specified life span of the vehicle.

Voltage Control

One of the problems in the solid state voltage control devices is heat rejection. In the Apollo systems this problem has been resolved by integrating tubes of the liquid cooling loops and heat inducting panels coated with epoxy resin to provide electrical insulation in the circuitry logic. In the voltage control unit, three or four such panels of 10 x 11.5 cm (4 x 4.5") will be required to accommodate the multivibrators, the current limiter, the voltage regulator and miscellaneous components, giving a control unit 15 x 15 x 15 cm (6 x 6 x 6") in volume and 2.3 kg (5 lbs) mass. The power transistor T and rectifier R of Figure 7.5-16A will be mounted on heat sinks and tied thermally to the cooling tube. The transistor will actually be a unit with three transistors in parallel, each of which is capable of carrying the load current of the motor; open-circuit protection of the transistor is thus provided. The mass of power transistor unit will be 0.90 kg (2 lbs), the power rectifier unit 0.45 kg (1 lb). All transistors and rectifiers of the voltage control will be sealed units. The semiconductor devices must be held below a total temperature of $373^{\circ}K$ or $100^{\circ}C$ ($212^{\circ}F$).

Inductor

The inductor L of Figure 7.5-16A will be mounted close to, or perhaps even inside the propulsion motor. It consists of a wound iron toroid. Cooling will be by transfer of the heat to the cooling fluid. If mounted external to the motor, it is proposed to bring the cooling tube through the center of the ring core. It will have a mass of 4.5 kg (10 lbs).

Operator Controls

The propulsion controls consist of two switches and a dual potentiometer. One switch (S_1 of Figure 7.5-22) is normally open and is used to close and open the coil circuit of the contactor C_1. It is the start-stop switch of the vehicle. The second switch (S_2) is mounted on the gear shift, is normally closed, and is used to unload the motor prior to shifting of the gears. Sealed switches are available from a number of companies, although special seal problems will be involved in view of the lunar environment. Masses are approximately 4 ounces per switch.

The two potentiometers are used to set the clock frequency of the drive, and thus to control motor voltage and vehicle speed. The potentiometers are of the order of

200,000 ohms each. It is proposed to use a two-gang potentiometer with 1/4 inch input shaft, to be fitted with a gear and driven from a ratchet operated by the accelerator pedal of the vehicle. An applicable unit, offered to military specification, has a mass of 0.25 kg (0.55 lbs).

Contactor

The contactor is a double pole, single throw unit. It is used to apply power to the motor. It must be able to interrupt the full 80 ampere motor current of 40 volts D.C.

Modifications of aircraft contactors are proposed for the application. Companies such as the Hartman Electrical Manufacturing Company are now actively engaged in developing contactors for use in zero pressure environment. Hartman is developing contactors for Grumman Aircraft on the LEM program, and has completed component qualification tests. A complete contactor has not yet been tested. Tests are being carried out down to 10^{-5} torr. The contactors are hermetically sealed by welding, and filled with an inert gas.

Instrument Panel

Three instruments are proposed to monitor the propulsion aspects of the vehicle. One is the vehicle speed indicator, described elsewhere in this report, for general information of the driver. The second is a speed indicator of the propulsion motor (Tachometer) to enable the operator of the vehicle to shift gears at 3500 rpm when accelerating and at 2100 rpm when decelerating. The third meter is an ammeter, to measure the load current of the propulsion motor. The motor speed indicator is a tachometer generator feeding a speed-calibrated voltmeter. This is a standard system for terrestrial applications. A trade-off subject to detail design analysis is the mounting of the tachometer inside the propulsion motor whence it would be provided with an atmosphere for its commutator and brushes; or mounting the tachometer outboard and outside the propulsion motor, with a gas atmosphere being separately supplied to it. Applicable meters, except possibly for required range, are being supplied by companies such as Weston Instruments for the Gemini and Apollo programs, and have passed qualification tests. The units are solder sealed and have an overlay of apoxy. They are filled with an inert gas.

7.5.2.3 Mechanical Drive System. The mechanical drive system on the first and second order modified vehicle is to be that of the basic vehicle with proper modification for the lunar environment. As previously described, the internal combustion engine has been replaced by a D.C. electric drive motor. The prime power is to be supplied by four battery packs. Figure 7.5-25, is a bottom view of the first and second order vehicles. The existing mechanical drive line from the D.C. electric drive motor to the wheels will be that of the basic vehicle, modified as required.

7.5.2.3.1 Major Components Modification Action—

Front Axle Unit

This unit will be sealed as in the standard Mule. Effort would be directed toward seal development and a general purpose lubricant for a lunar environment, see figure 7.5-26.

The severest environmental exposure would be experienced by those seals exposed directly to lunar conditions. On the Mule those seals fall into the following categories: lip type seals, "O" ring type seals, linear wiping seals and bellows type. The constant velocity universal joint seal (one at each wheel) is normally a rubber bellows but a metal bellows could be substituted. All other seal designs would be re-worked to utilize teflon sealing elements and thus be expected to withstand the temperature and vacuum environment for the limited life expectancy of the vehicle.

Recommended lubricants for the Mule have insufficient viscosity ranges to effectively operate at lunar environment extremes. A single existing low viscosity oil is suggested to replace all standard Mule liquid lubricants. This lubricant is MIL-L-7808, an aircraft turbine oil which has already been recommended for use in space environments (Lockheed Space Design Manual) as a general purpose gear lub.

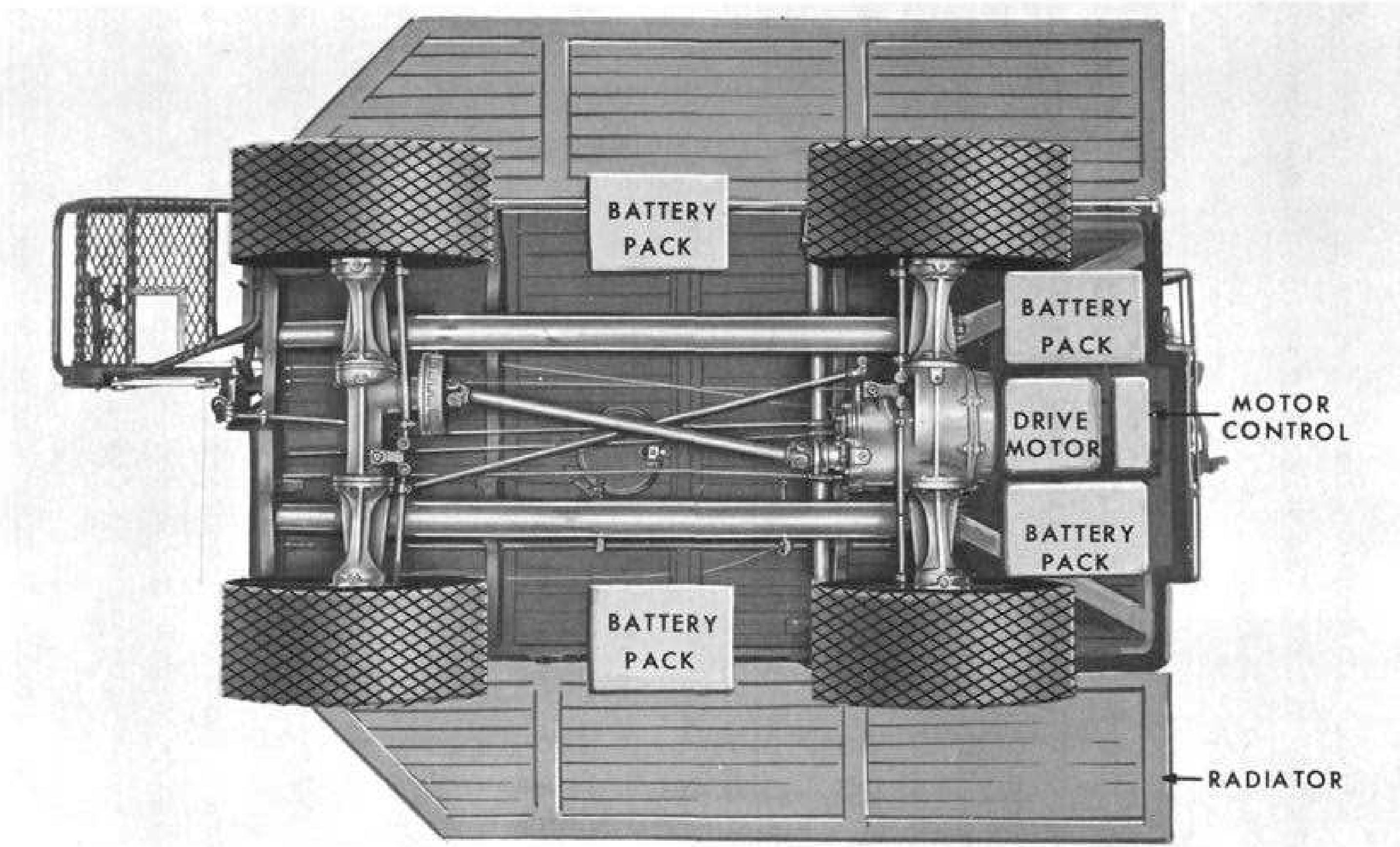


Figure 7.5-25. Power System Components Placement - First and Second Order

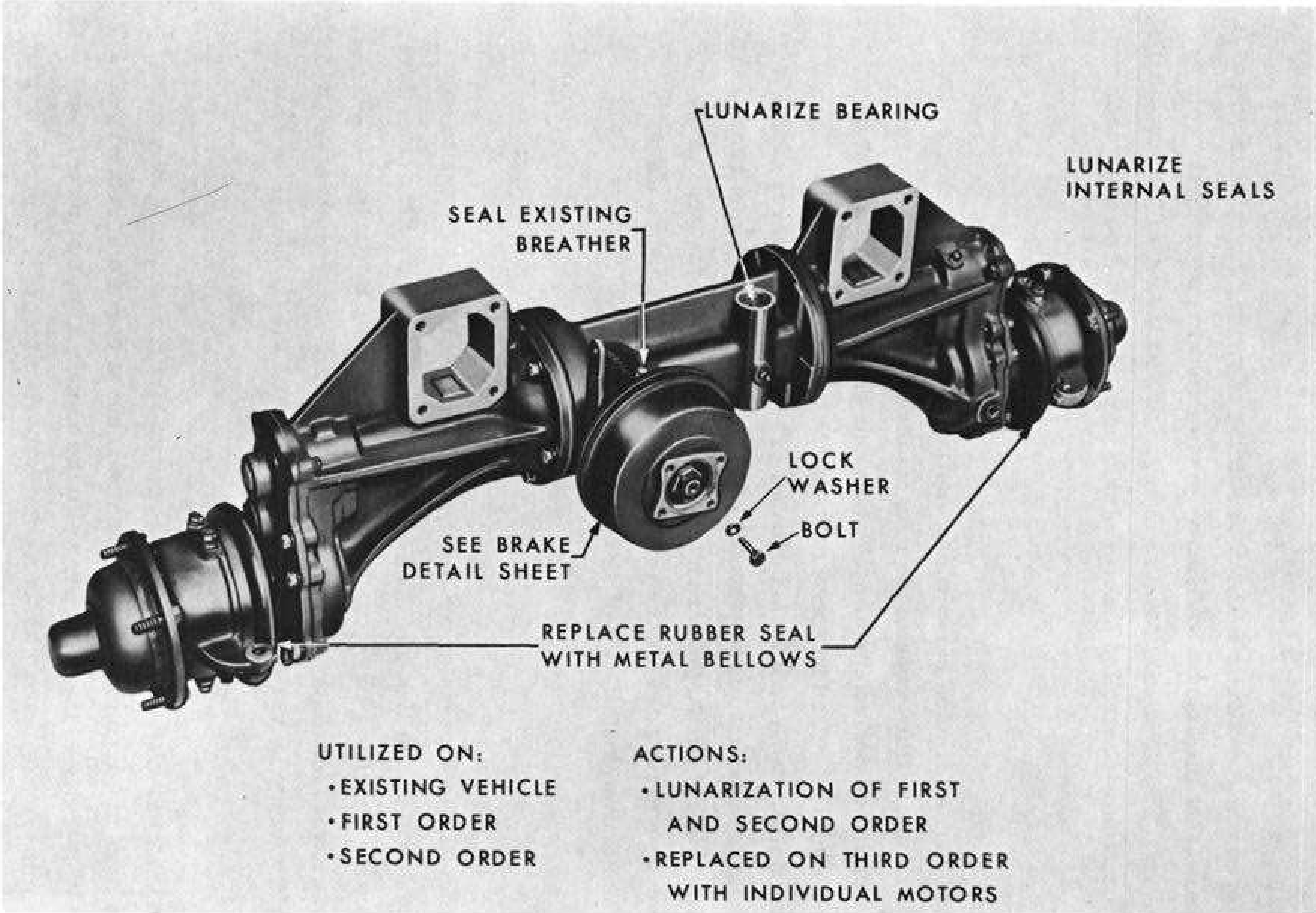


Figure 7.5-26. Front Axle Assembly

The remaining foreseeable difficulty is the cold welding of moving parts inside the axle. It is proposed that the axle be sufficiently pressurized to eliminate the cold welding phenomenon. The required pressure is very low and in fact will be supplied by lubricant sublimation. Controlled leakage of this pressure would be acceptable.

Rear Axle Unit

The rear axle unit may be treated in a manner similar to the front axle unit, see figure 7.5-27. The major difference is the integral transmission, transfer case and clutch unit. It will be possible to simplify this unit for the modified first order and second order vehicles. The low speed range of these modified vehicles eliminates the need for a two-speed transfer case. The gear ratio ranges of the transmission alone is sufficient. The transfer case shift linkage and high range gear set may be removed and the low speed transfer gear permanently engaged. Figure 7.5-28 shows the actions intended for the transmission unit. The figure shows schematically the intended installation of a cooling coil loop.

Clutch

The substitution of an electric motor for the dynamic engine allows elimination of the standard clutch and clutch linkage. The electric motor is fitted with a centrifugal clutch which does not require a linkage. This design allows the motor to start up unloaded and thus lowers maximum current flow. The clutch engagement speed will be chosen to provide a minimum vehicle speed of 1 ft/sec with the transmission in first gear.

For most of the lunar mission the Mule can be operated in first gear. However, at the upper speed limit of 5 mph plus it would be desirable to operate in second gear. Fortunately, gear changing at these speeds while the vehicle is in motion can be accomplished relatively easy even without a conventional clutch. It is proposed that an electric power interrupter to the drive motor serve as a "clutch" to unload the transmission gears; allowing the transmission to be shifted normally. The mule transmission

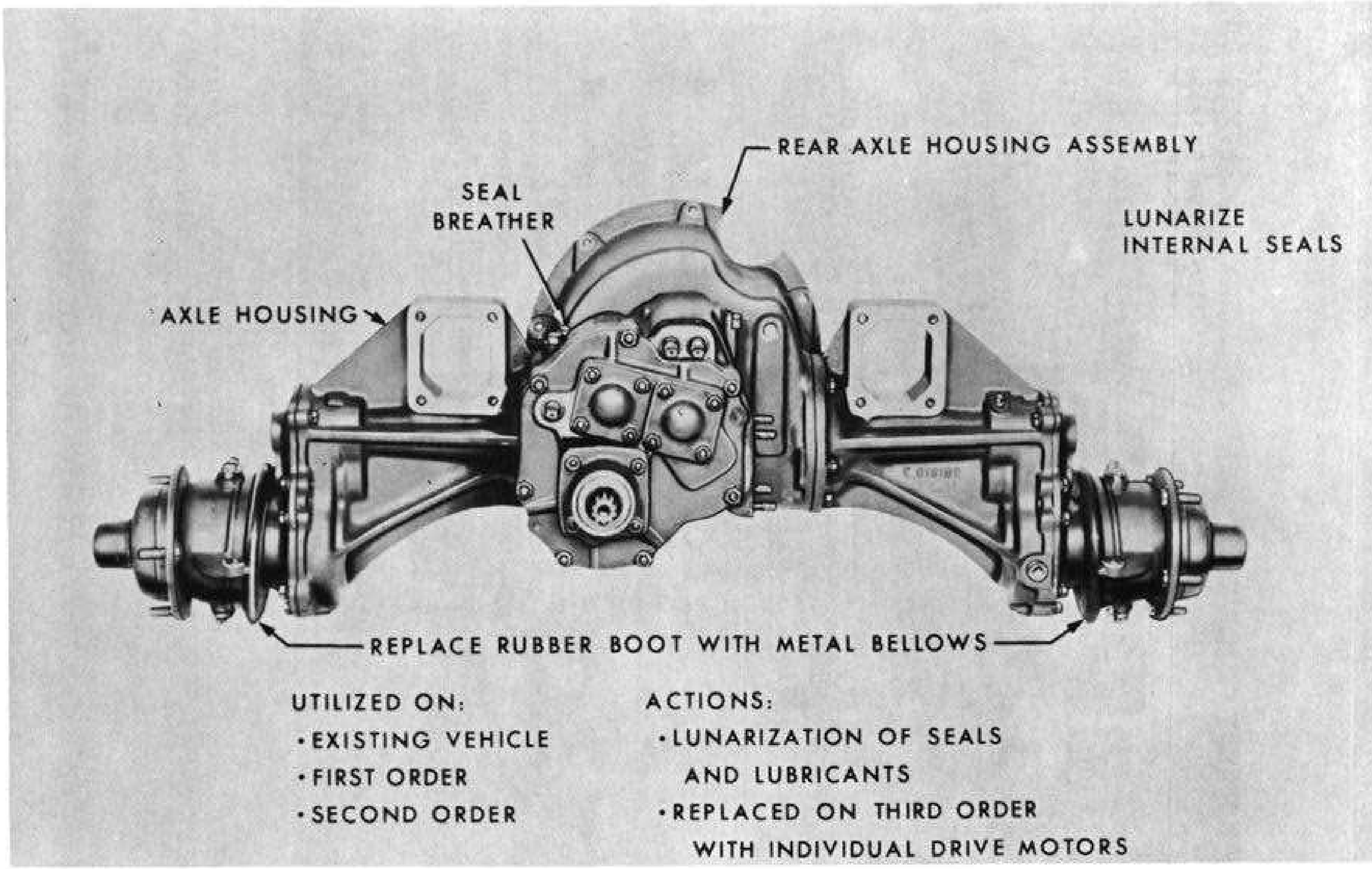


Figure 7.5-27. Rear Axle Assembly

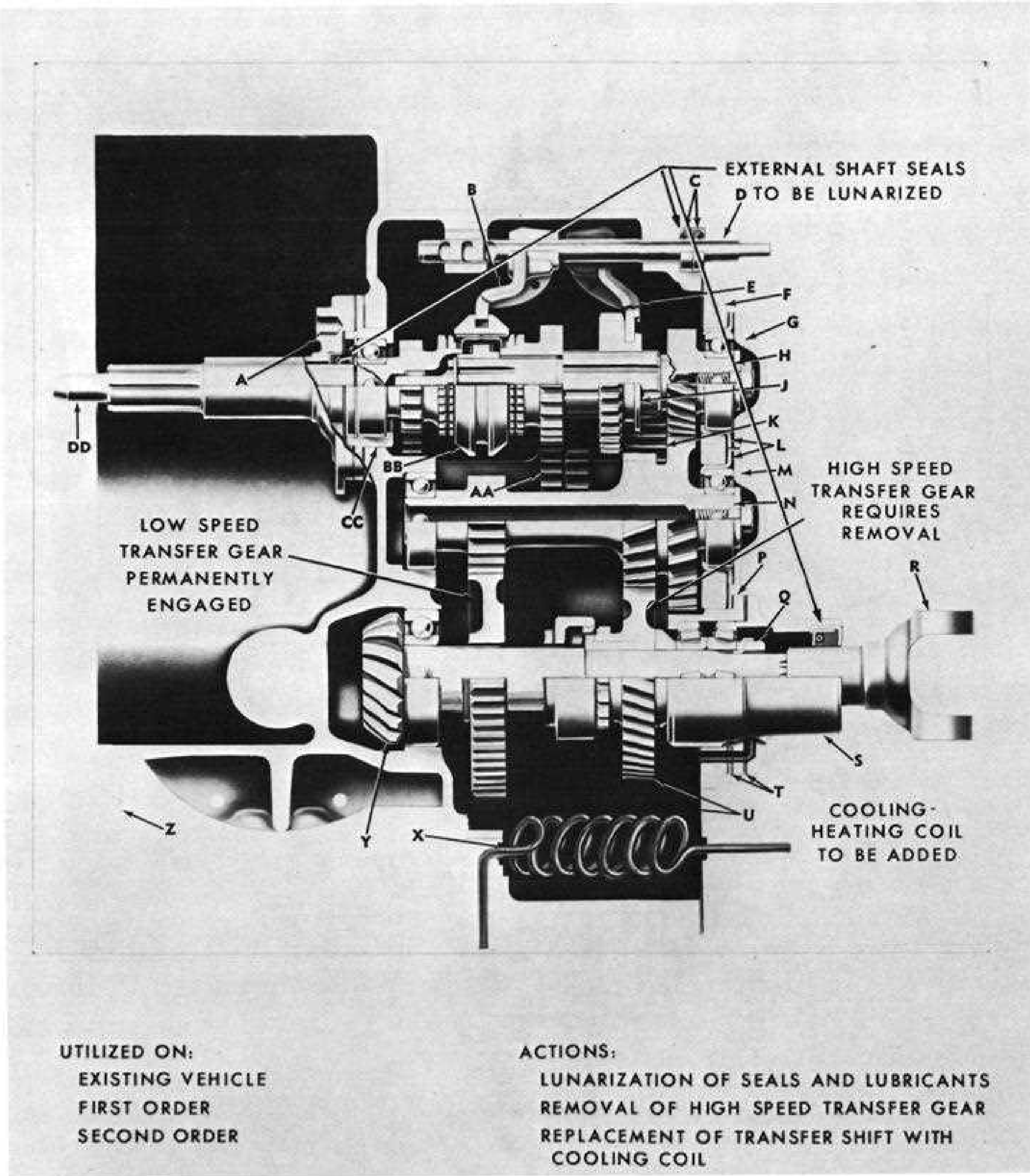


Figure 7.5-28. Transmission

is synchronized for shifts from first to second, from second to third and third to second. Shifts back to first gear will require bringing the vehicle to a halt, or training the operator to "double clutch" with the electric power interrupter. The transmission synchronizer rings are expected to handle the additional inertia loads of the drive motor armature for the relatively short design life of the vehicle.

Brake

Vehicle braking will not be provided for by use of the electric drive motor. The existing propeller shaft brake will be adequate for stopping and for parking of the vehicle. The selection of the proper brake shoe material will be a prime consideration. Figure 7.5-29 shows the intended modification actions.

Propeller Shaft

Figure 7.5-30 shows the drive shaft assembly and the intended modification actions. The only foreseeable difficulty will be the selection of a proper lubricant for the universal joints. The joints are of the cross type and it is thus only a matter of lubricating needle bearings oscillating through a small angle.

Steering Mechanism

The existing four wheel ackerman type steering mechanism will be retained and modified. The steering gear, figure 7.5-31, would be packed with a molykote type G lubricant. The steering linkage assembly is shown in figure 7-5-32. The tie rod ends are normally sealed ball and socket units that are greased periodically with chassis grease. For this lunar application, these would be replaced by unlubricated assemblies with a teflon socket and hard steel ball. Figure 7-5-33 shows the tie rods and associated parts. Figure 7.5-34 shows the proposed tie rod end modification. Figure 7.5-35 shows the comparative frictional performance between a standard lubricated joint and a teflon lined, nonlubricated joint.

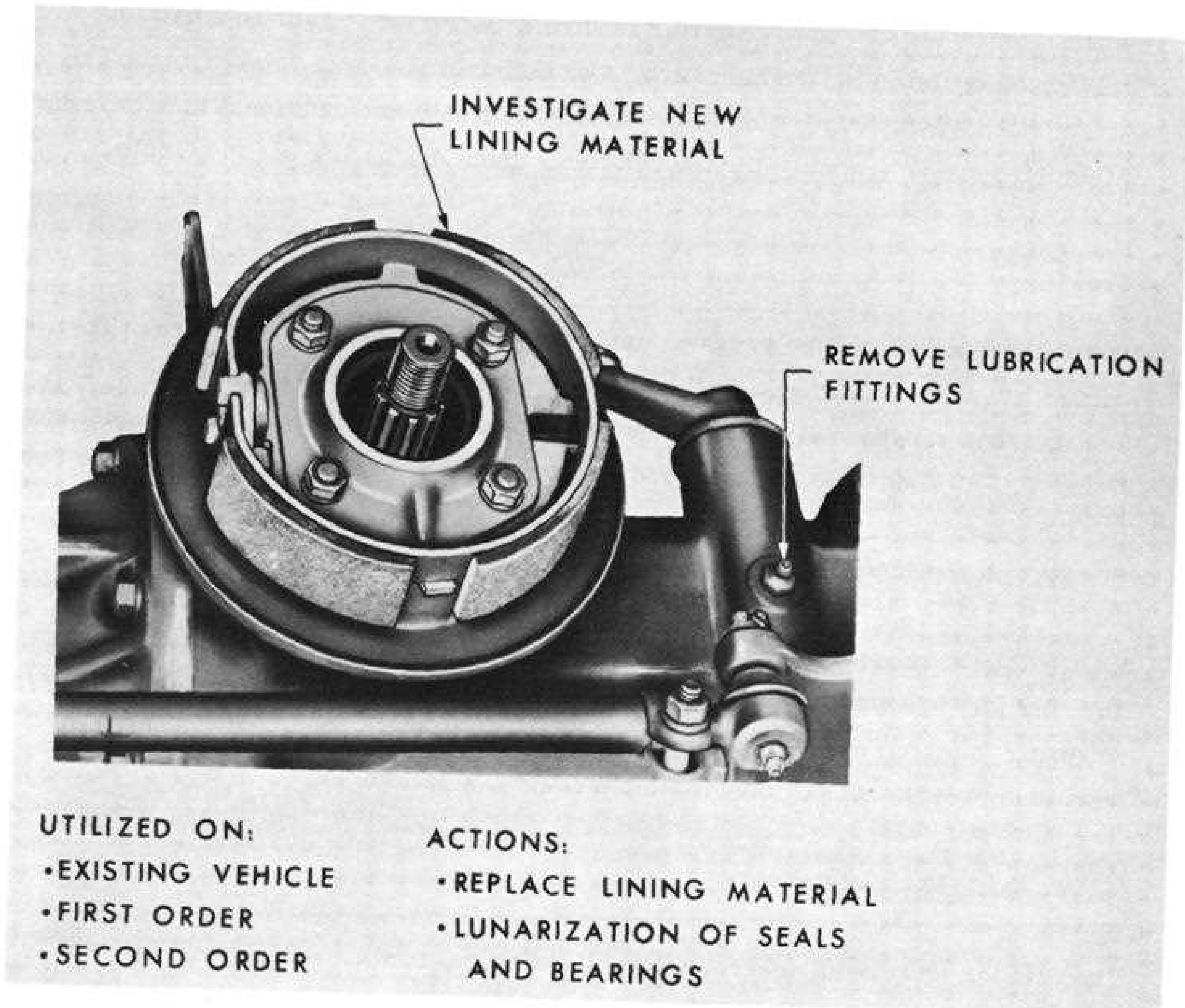


Figure 7.5-29. Shaft Brake Assembly

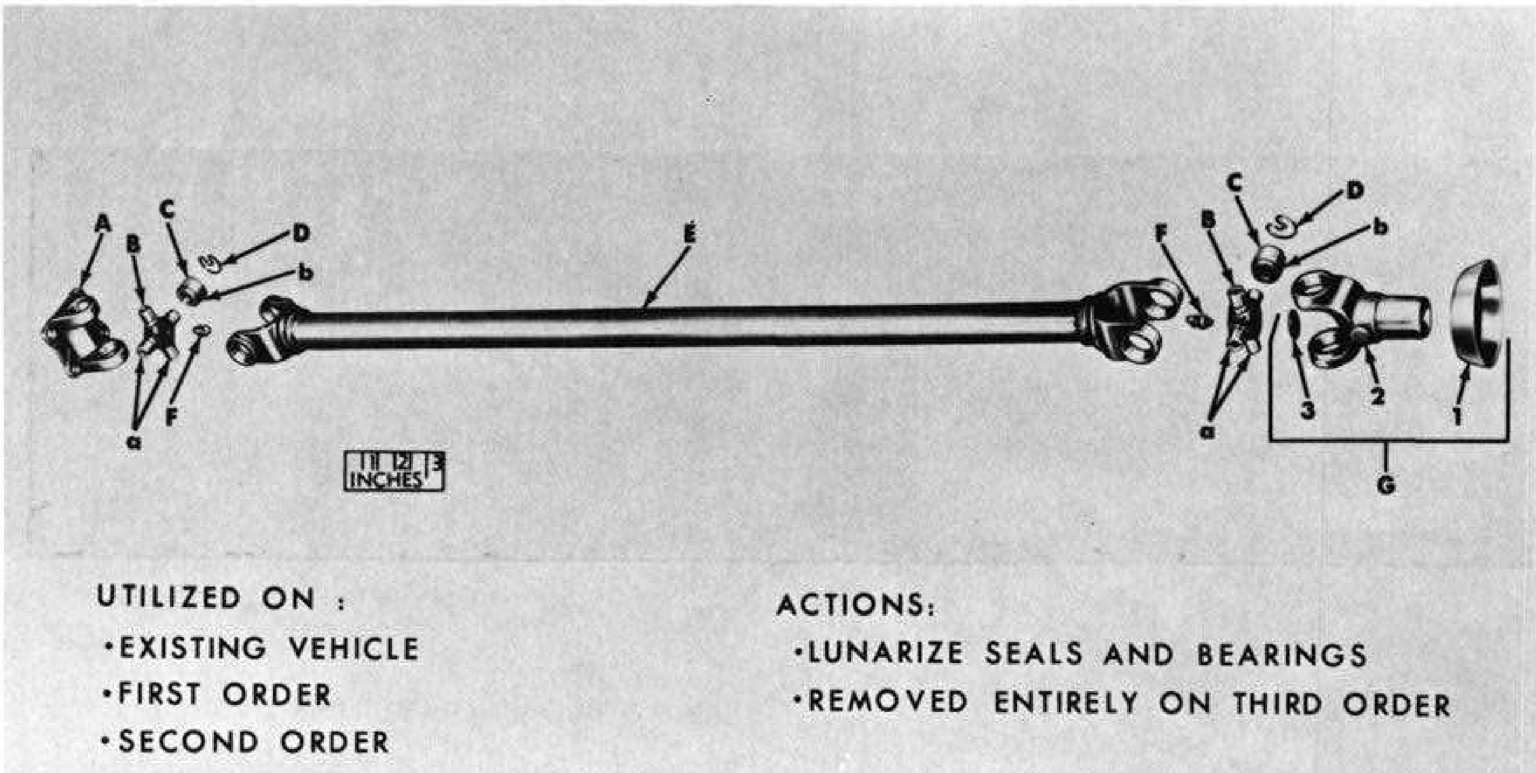


Figure 7.5-30. Drive Shaft Assembly

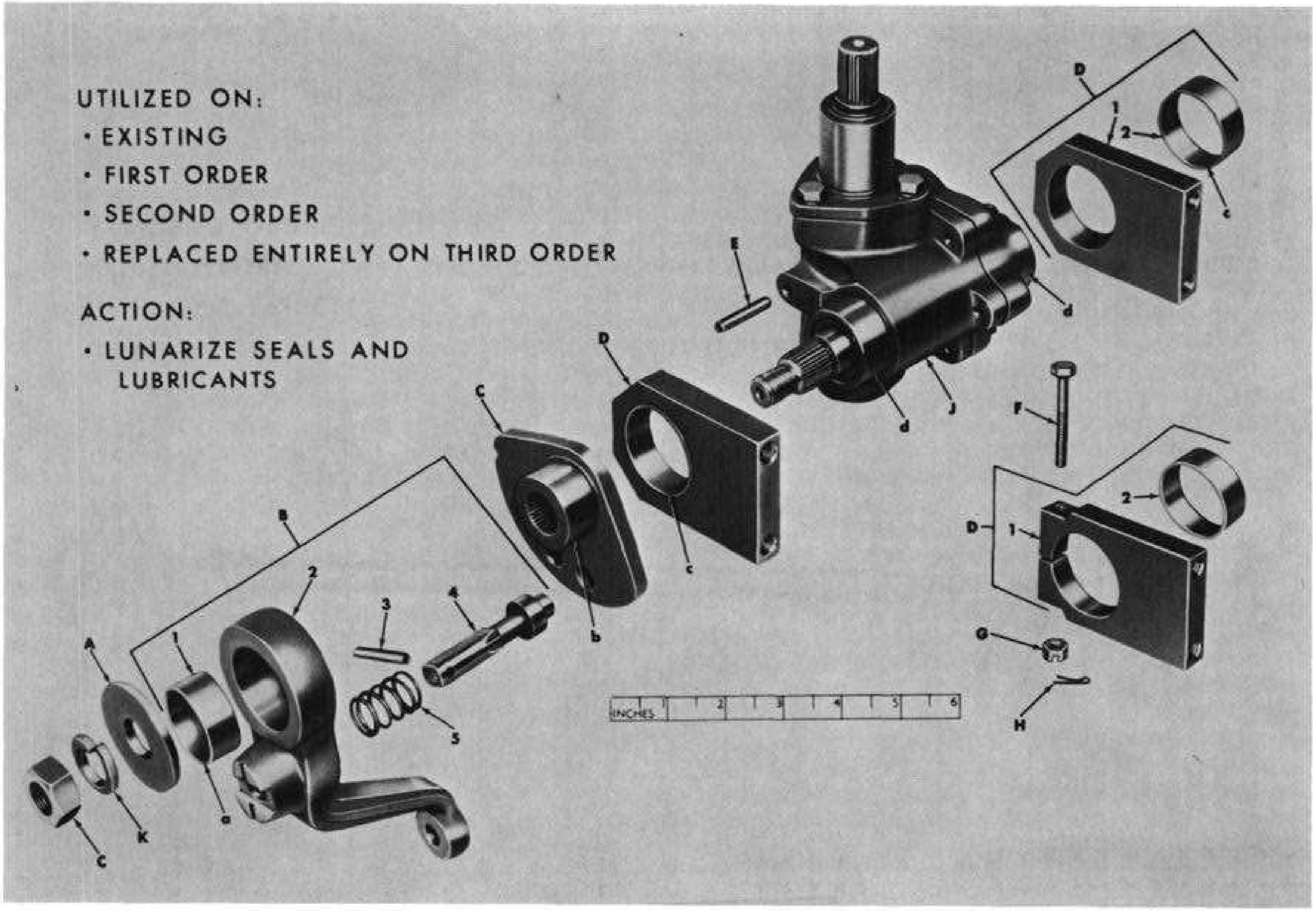
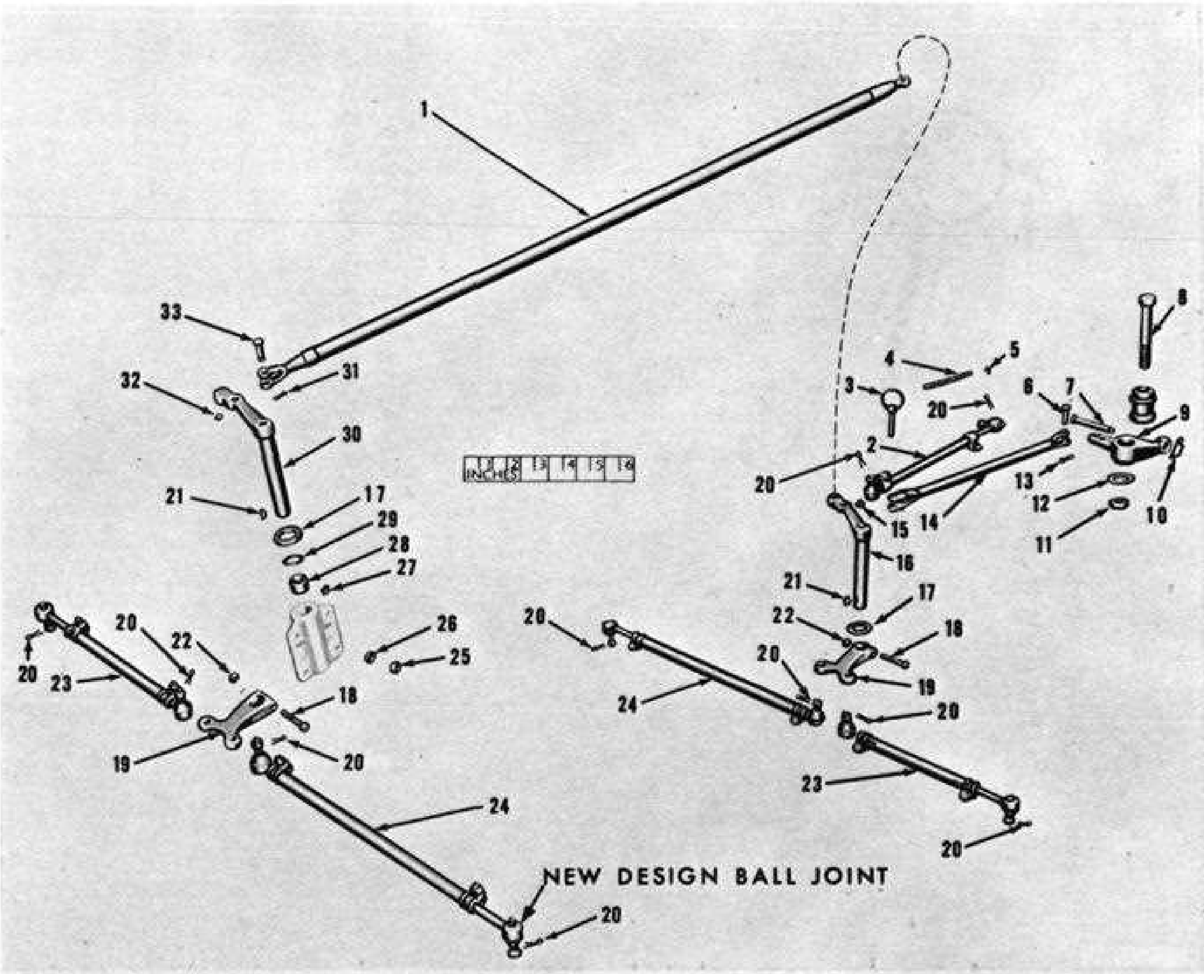


Figure 7.5-31. Steering Gear



UTILIZED ON:
- EXISTING VEHICLE
- FIRST ORDER
- SECOND ORDER
- THIRD ORDER

ACTIONS:
- LUNARIZE BEARING SURFACES
- CHANGE BALL JOINT DESIGN
- DIMENSIONAL CHANGES ON THIRD ORDER
- ELECTRIC ACTIVATION ON THIRD ORDER

Figure 7.5-32. Steering Linkage Assembly

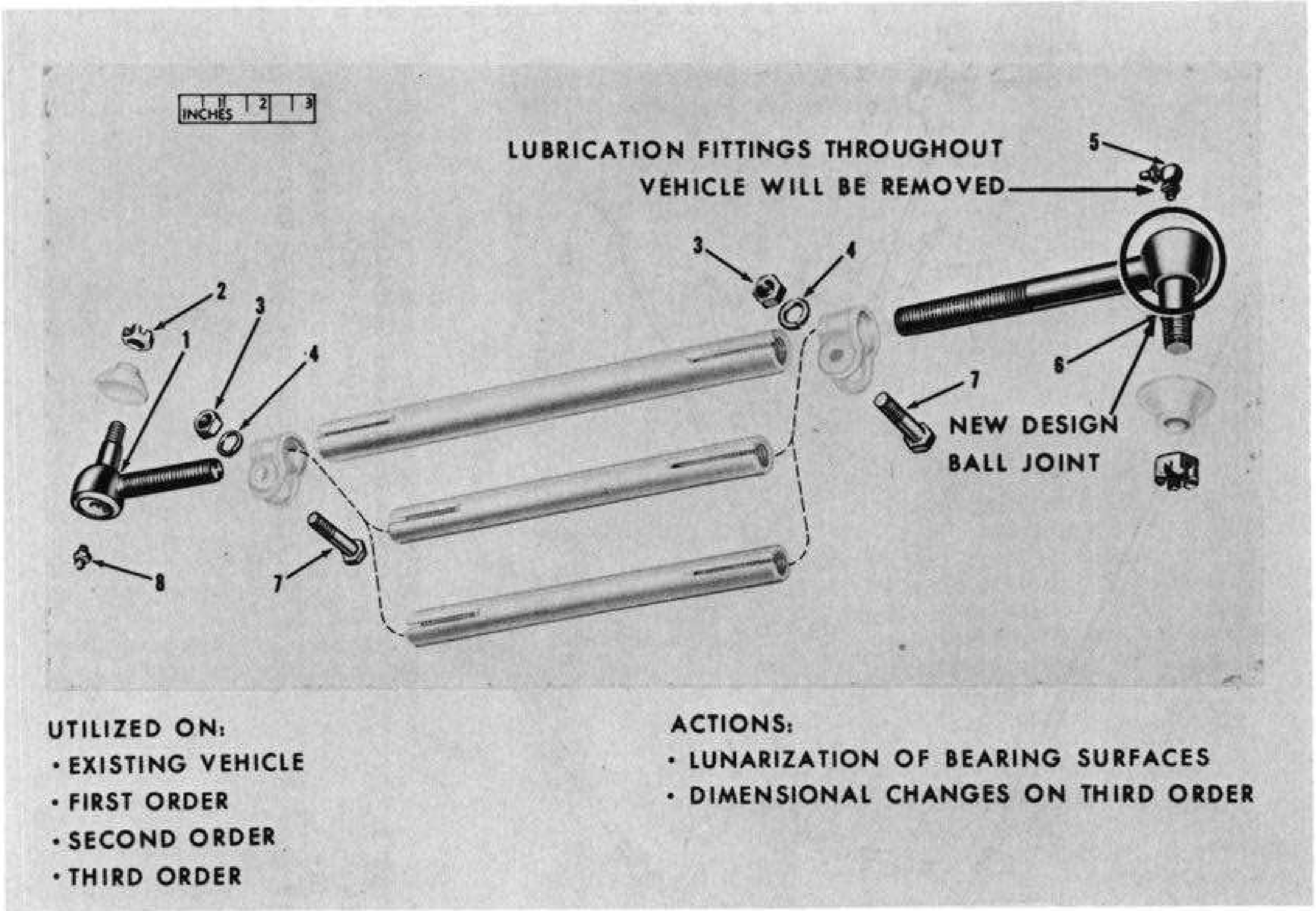


Figure 7.5-33. Tie Rods and Associated Parts

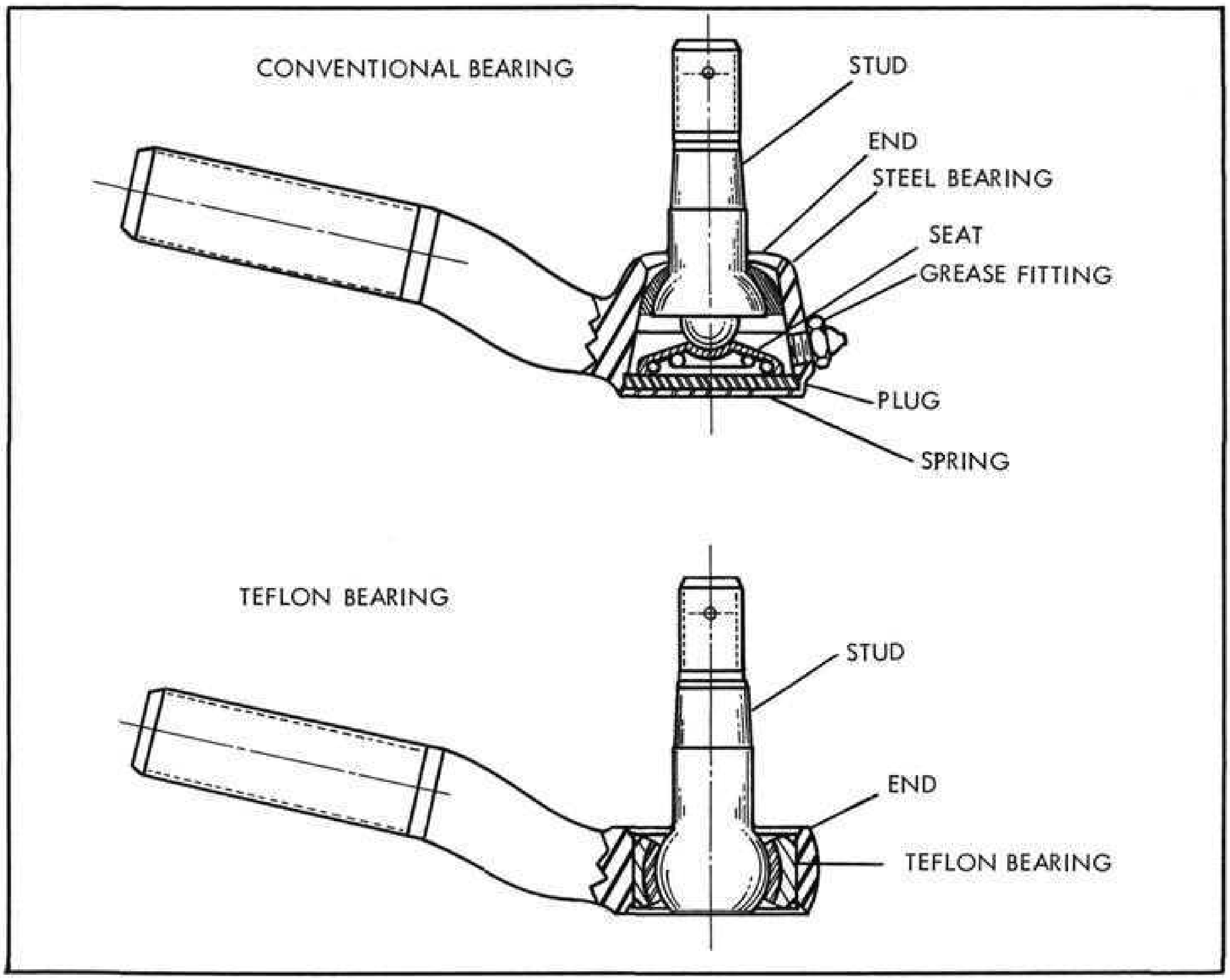


Figure 7.5-34. Tie Rod End

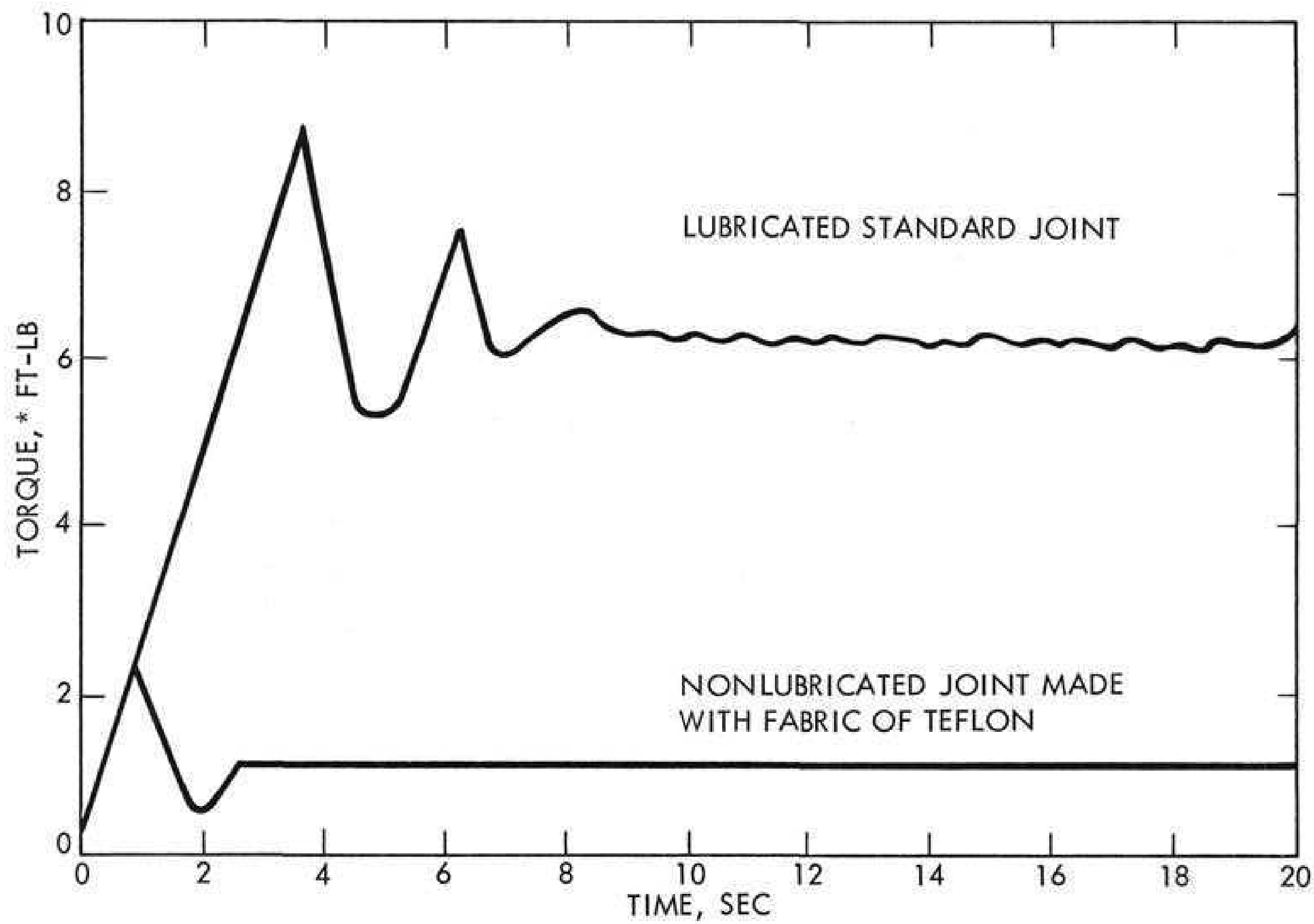


Figure 7.5-35. Fricitional Performance of Automotive Suspension Joints

Lubrication and Seal Points

All external lubrication points on the modified vehicles will be removed. No external lubrication will be required for the intended 240 km (150 miles) vehicle mission life. All seals will be lunarized as previously described for the front axle unit. Admittedly, seals will be a development problem, but no extreme difficulty is anticipated.

Structure

A primary consideration for the chassis is the inertia loadings during earth launch and lunar landing maneuvers. The stated requirement is for an 8g loading. The basic vehicle successfully withstands 30g loadings during air drop operations. Figure 7.5-36 indicates modification actions to be undertaken to the chassis assembly. In addition, the underside of the vehicle will be insulated. All parts exposed to sunlight will receive an a/e surface treatment.

7.5.3 Crew Station

7.5.3.1 Design Requirements. The requirement for crew station and PLSS storage are stated in the Guidelines and Constraints. This study is not concerned with personal protection against meteoroids and high energy nucleonic bombardment.

The problems in human factors design concern the crew station and controls modifications required. Since the first order modified vehicle is the modification concept involving minmal changes to the basic vehicle, it was desired to conserve those features compatible with operation in the lunar environment.

To evaluate compatibility of the basic vehicle with operation by an operator, attired in a pressurized (Apollo) soft suit, a special test demonstration was conducted by the Bio-Engineering and Astronautic Group at the Marshall Space Flight Center, Huntsville, Alabama.

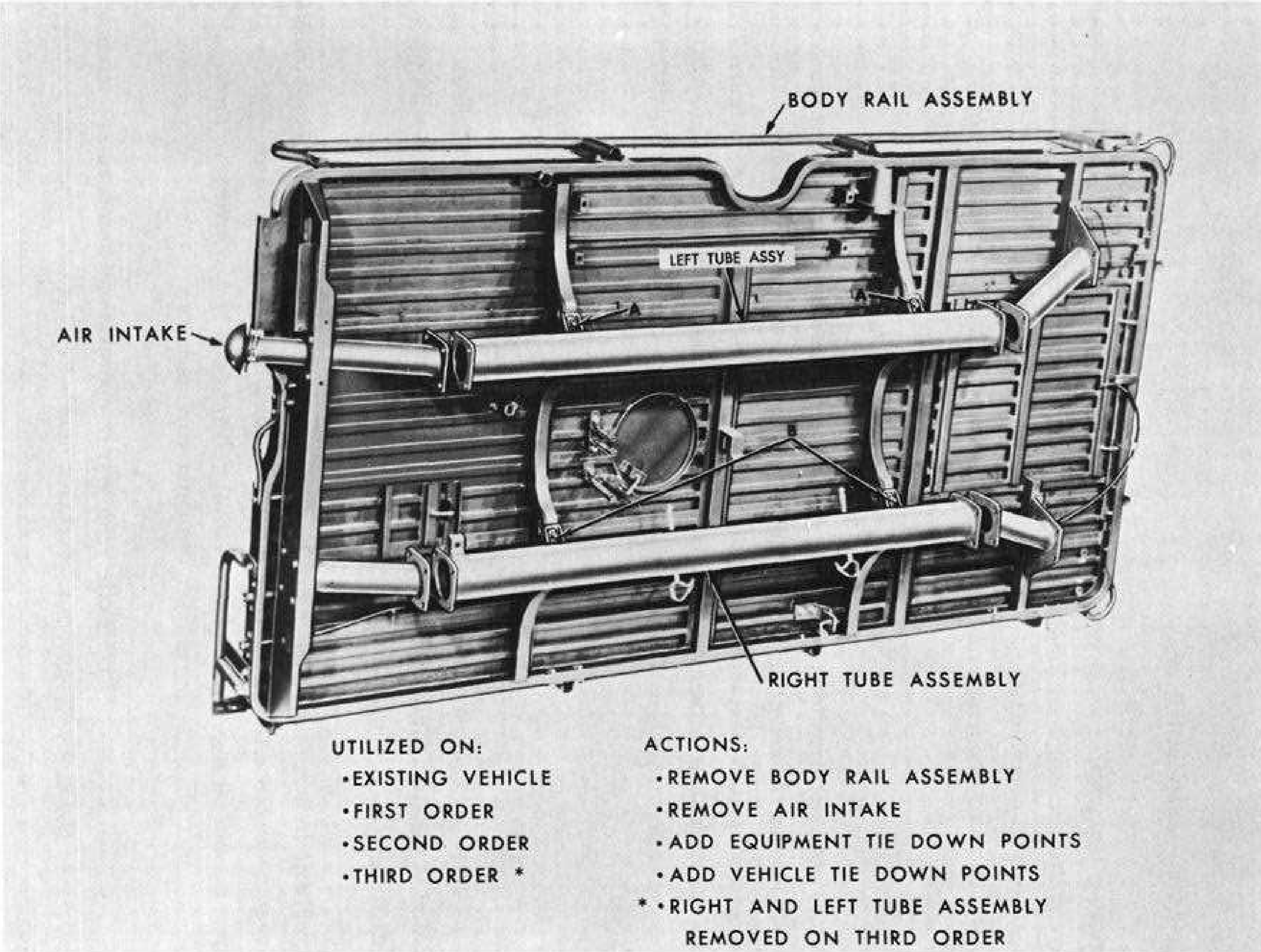


Figure 7.5-36. Chassis Aseembly

The demonstration included mounting and dismounting the vehicle by an operator in a suit pressurized to 240 millibars (3.5 psi). It included driving the vehicle, except that the gasoline fueled engine of the M-274 was started by a shirt sleeve operator. Such an engine will not comprise a component of a lunarized vehicle.

Figure 7.5-37 shows the operator seated on the M-274 vehicle. The suited operator, with suit under pressure, successfully mounted and dismounted the M-274 vehicle. He engaged the shift gears, started vehicle roll, stopped the vehicle, and reversed shift gears, and backed up the vehicle in reverse gear. Various activities were conducted with some difficulty, particularly mounting and dismounting. Dexterity in the pressurized soft suit was not so great that some effort was not required in steering. Pedal action appeared adequate. Motion pictures covering the demonstration are available. In view of all the observed activities, it was concluded that driving operation of the basic M-274 (exclusive of engine starting) is feasible; however, certain modifications appear desirable for improvement of the human factors design.

7.5.3.2 Modification Actions

Based on these conclusions, the crew station elements of the M-274 vehicle are evaluated for retention or modification in the first and second order modification vehicles as follows:

7.5.3.2.1 Steering Wheel Assembly—Retain the steering wheel assembly with the modifications noted in Figure 7.5-38 with the addition of a ball assist on the steering wheel as an option.

7.5.3.2.2 Foot Control Assembly—

- Remove the clutch pedal since a foot operated clutch will be replaced by a centrifugal clutch in the first and second order vehicle modifications, see Figure 7.5-39.
- Use a solid or open web foot rest area in the foot control base. The demonstration vehicle has been modified by placing a partial plate in the front basket base.

Figure 7.5-37. Suited Astronaut - Operating Mule Vehicle Test

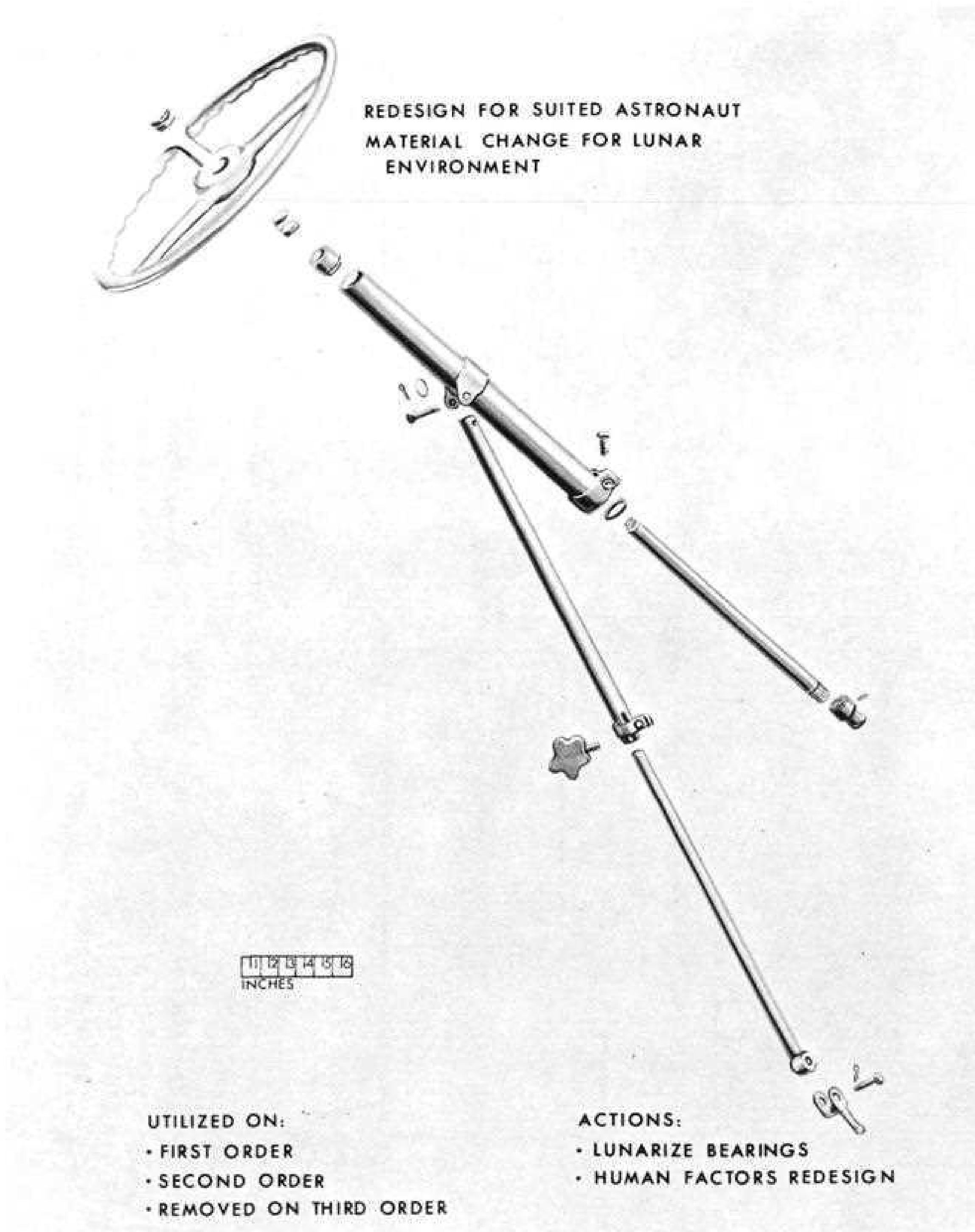


Figure 7.5-38. Steering Wheel and Column Assembly

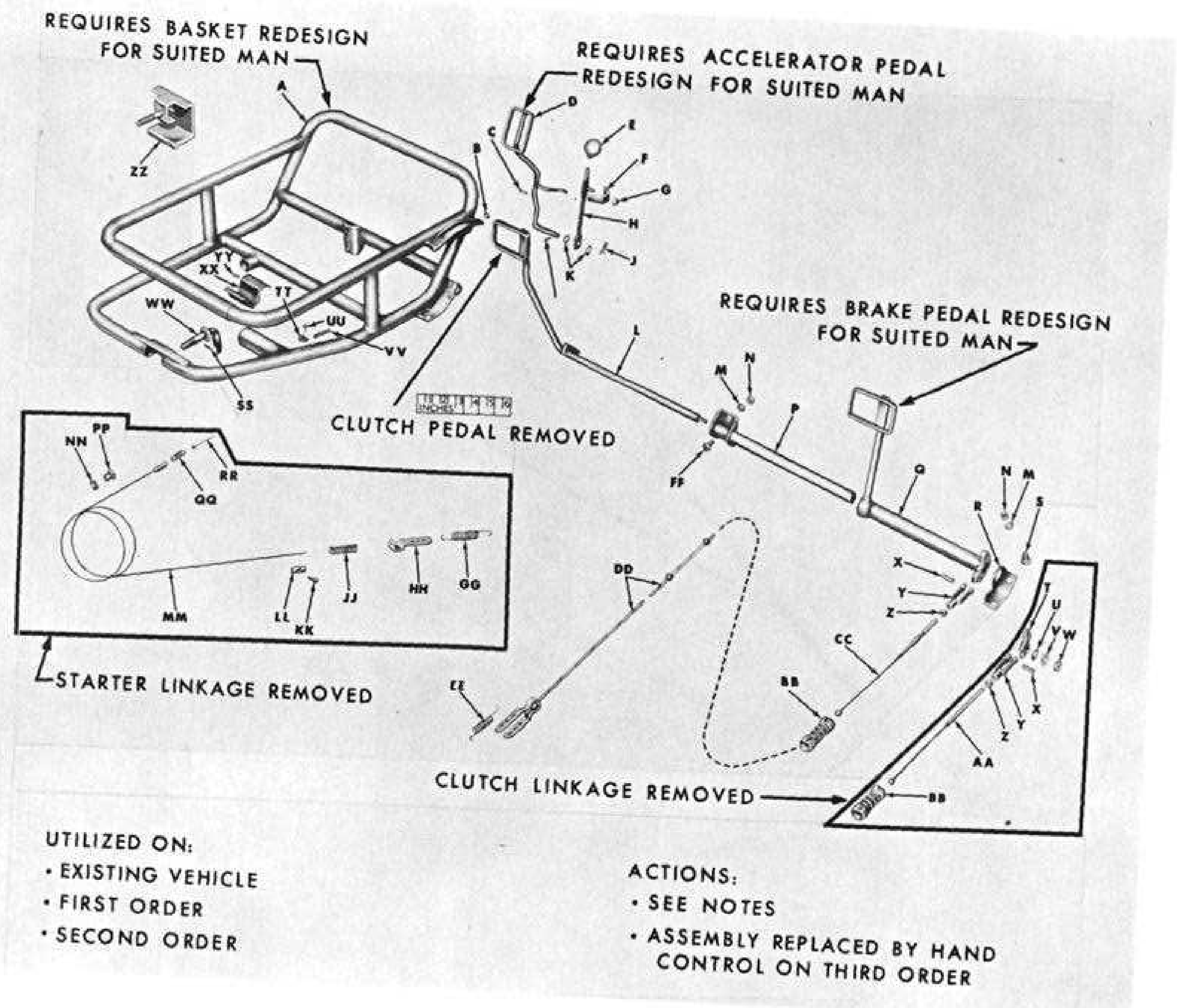


Figure 7.5-39. Foot Control Assembly

- Use a step on the foot basket for assist in mounting and dismounting.
- Use large pedal pads for brake and throttle control in lieu of the open face pedals of the basic M-274 vehicle.

7.5.3.2.3 Transmission, Transfer Case Shift Levers, and Parking Brake Handle—

- Remove the transfer case lever, since only one gear case range is required in the transmission train of the first and second order vehicle modifications, see Figure 7.5-40.
- Lengthen the gear shift lever to raise ball to a more convenient position for the suited operator. Install electric motor circuit interrupter switch on gear lever for gear change.

7.5.3.2.4 Body Rail Assembly—

- Remove the body rail assembly to facilitate mounting and dismounting the vehicle by an operator wearing a protective suit, see Figure 7.5-41. Part of the body rail had been removed from the demonstration M-274 vehicle evaluated as discussed above at the NASA Marshall Space Flight Center. Provide suitable cargo tie down devices on the cargo deck for the scientific mission packages.

Other design changes required to comply with specific constraints involving human factors compatibility include:

- Use of fender-guards structurally integrated with the thermal radiator supports, to protect the vehicle operator from debris thrown from the wheels.
- Use of longitudinal and lateral roll bars to protect the operator in the event of vehicle roll over, and to provide hand hold assistance in mounting and dismounting the vehicle.
- Use of a "deep back" seat rest on the operator seat to permit wearing of the PLSS back pack while operating the vehicle in the seated position. A safety harness would be employed.

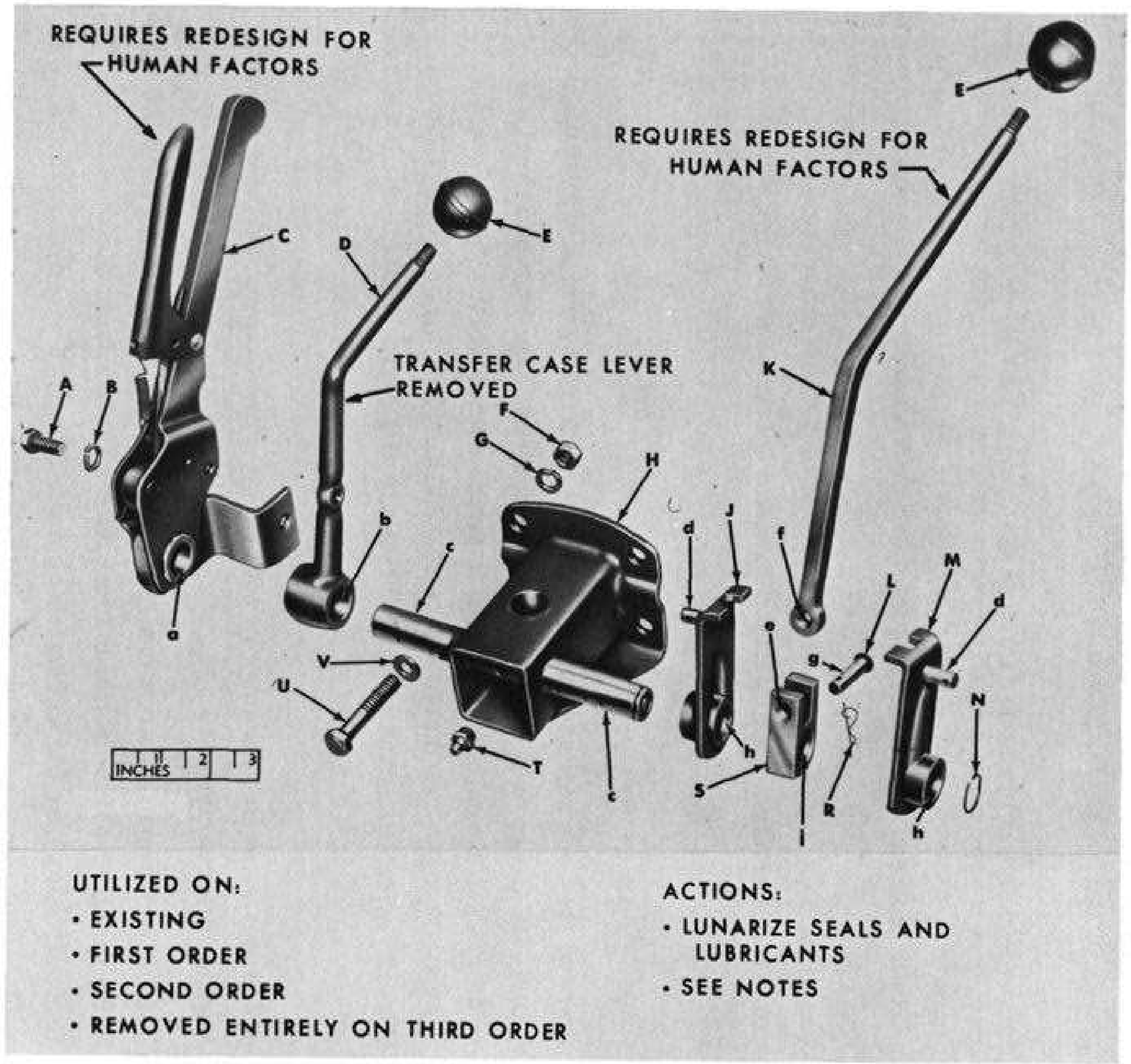


Figure 7.5-40. Transmission, Transfer Case Shift Levers

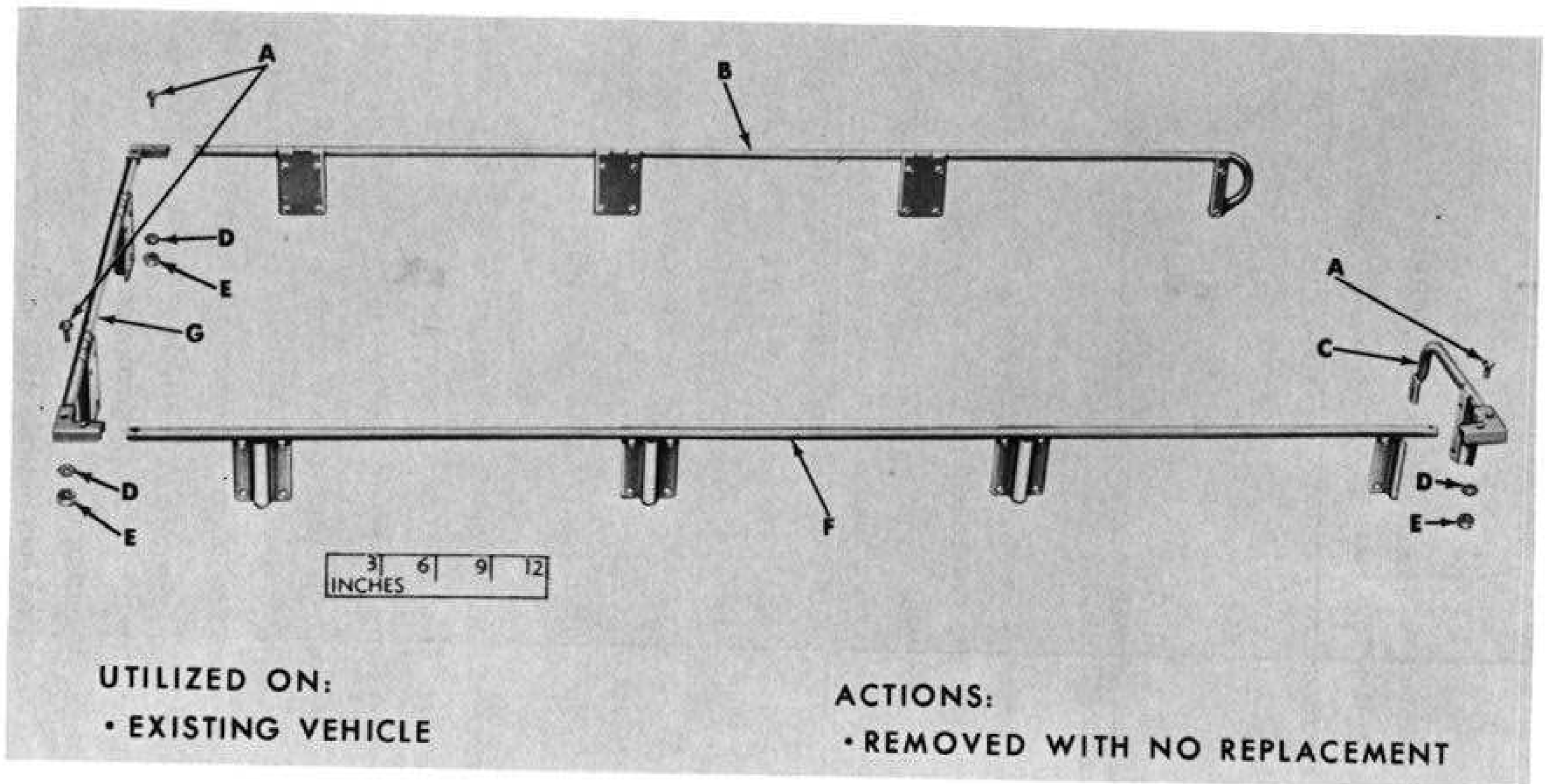


Figure 7.5-41. Body Rail Assembly

- Provide a tie down position for the second PLSS pack adjacent to the operator's seat, with an option for a second operator seat which would be packaged separately for mounting and use if desired, after vehicle delivery to the lunar surface.

7.5.3.2.5 Control System Summary—The minimum operator control system items proposed for the first and second order vehicle include:

- Steering wheel.
- Manual gear change lever, with attached manual electric motor drive circuit interrupter switch for clutchless gear change operation.
- Hand brake for parking.
- Foot throttle for control of electric motor drive power.
- Floor console, master (three position) push-button selector switch with "Drive", "Neutral", and "Reconnect" positions. The latter position is needed to engage the proper internal circuitry for the battery recharging mode of operation. The neutral position is provided to disconnect all circuits except the coolant liquid pump.
- Thermal radiator louver control manual over-ride.

In addition to the manual and pedal controls, instrumentation is required to enable the operator to assess the vehicle operating status at all times. The minimum instrumentation proposed for the first and second order modifications is:

- Speedometer, reading in kilometers per hour, to indicate vehicle speed.
- Tachometer, reading in revolutions per minute to indicate the electric drive motor speed. This item is required to enable gear shifting at indicated motor drive speeds. Due to wheel slip in soft soils, the speedometer cannot be taken as a precise indicator of drive motor speed.
- Side Inclinometer, reading in degrees rather than radians, to indicate the degree of vehicle roll about the longitudinal axis as it traverses side slopes. This instrument has been specified because of possible difficulty in operator self-orientation in locations where the horizon is irregularly defined by surface

features and also the effect of a reduced gravitational force. These may not be conducive to permit a clear operator sensation of a critical displacement from the vertical orientation.

- Odometer, in units of one tenths of kilometers per hour.
- Remaining Traverse Distance Estimator, in units of one tenths of kilometers, based on residual battery charge integration (watt-hour difference).
- Ammeter, reading in amperes, to indicate the drive motor current.
- Battery Overpressure Indicator Lights. These lights, one for each parallel battery circuit, illuminate when the individual battery circuit is disengaged because of battery overpressure.
- Radiator Coolant Liquid Outlet Temperature, reading in degrees Kelvin. This temperature will provide a means of evaulating the status of the coolant system operation.

Instrument readout may be presented in two locations: one on front roll bar console, and the other on a floor console beside the seat on the vehicle center line immediately to the right of the operator's seat. The speedometer, tachometer, odometer, and hour meter should be located on the roll bar console. Other vehicle condition readouts could be located in a floor console (battery pressure lights, radiator temperature) along with controls for radiator louvers.

7.5.3.2.6 Cargo Platform—The scientific payload packages, as specified in the criteria, are shown placed on the cargo platform in Figure 7.5-42. The packages can be placed easily on the area available. The area on the right-hand side of the operator's seat is available for a spare seat, spare PLSS units, or additional cargo.

7.5.3.2.7 PLSS Expenditure Rates—No specific study has been conducted on the metabolic power required for operation of the modified M-274 vehicle on the lunar surface. However, it is of interest to note the PLSS rated capability against the metabolic heat dissipation rates associated with other types of activities, see Table 7.5.3-1.

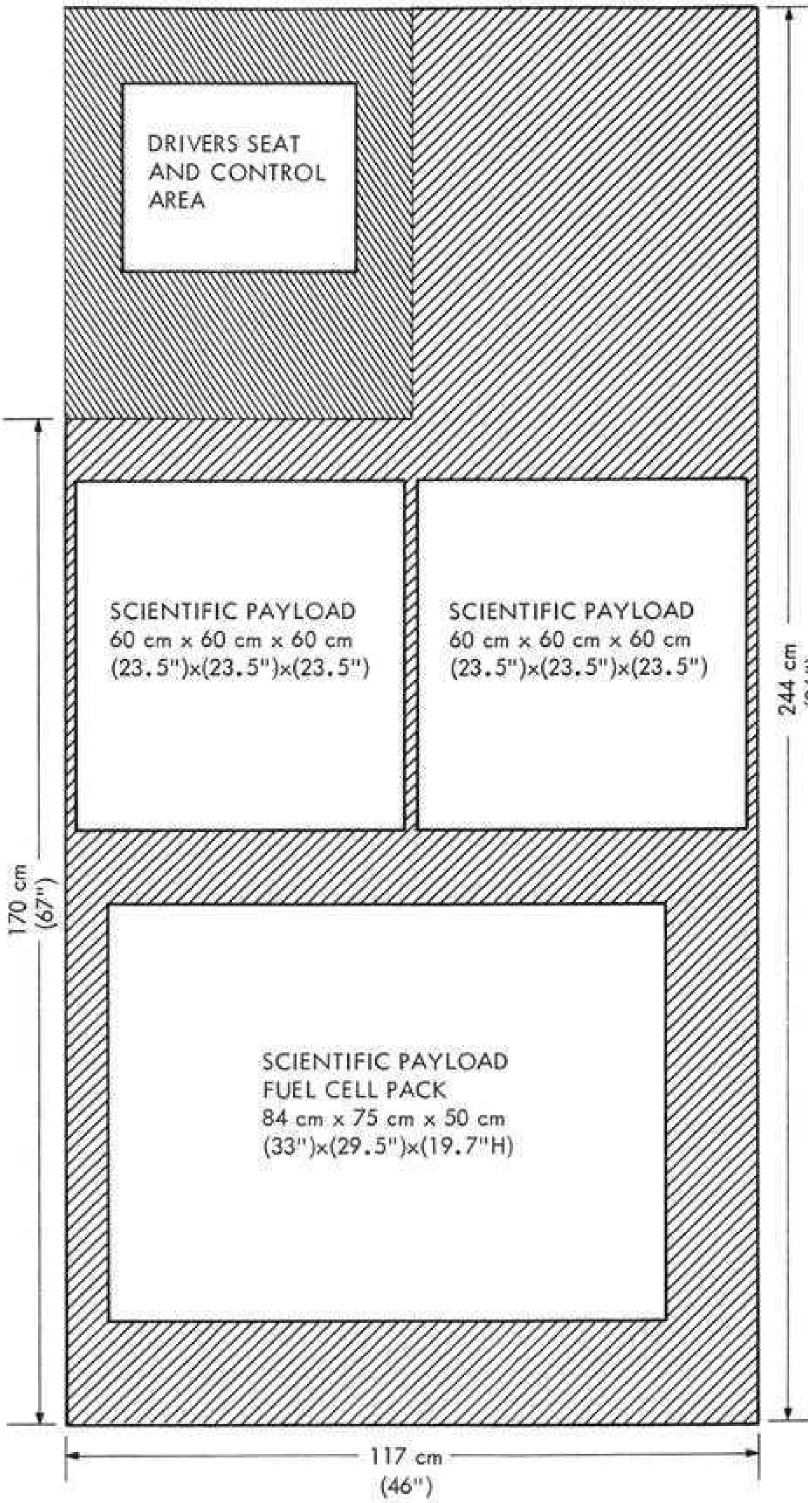


Figure 7.5-42. Scientific Packages - Available Platform Area

The PLSS has a total metabolic heat rejection capacity of 1410 metabolic watt-hours (1215 Kg Cal or 4800 Btu).

Table 7.5.3-1

ACTIVITY HEAT REJECTION REQUIREMENT vs INDICATED PLSS CAPACITY

ACTIVITY* (Shirt Sleeve)	HEAT OUTPUT Watts	Btu/hr	INDICATED PLSS CAPACITY Time Base, Hours
Moderatively Active, Sand Shovelling	475	1620	2.96
Combat Bomber Flight	206	700	6.85
Driving Truck	232	790	6.1
Walking in Deeply Plowed Field	544	1850	2.6
Moderate Trotting On Horseback	497	1690	2.84

*NASA Bioastronautics Data Book, Sp 3006, pp. 174-176, Washington, D.C. 1964.

Table 7.5.3-1 shows various activities and associated heat rejection rates in shirt sleeves, which are not precisely defined insofar as the level of activity is concerned. The criteria for the lunar surface is the ELMS model. This model defined no surface obstacle or roughness definition. Without such a definition, an expenditure rate of 700-800 Btu/hr seems adequate. With a rough lunar surface, even with a very low vehicle speed, the expenditure rate will increase. The allowable metabolic expenditure rate should be consistent with the ride characteristics of the vehicle over the surface.

7.5.4 Thermal Control

7.5.4.1 Radiator Design Criteria. In the lunar environment, heat rejection is governed by the Stefan-Boltzmann law. In free space the sink temperature is about $4^{\circ}K$, which for practical purposes may be considered as $0^{\circ}K$. However, on the surface of the moon and depending upon the heat rejecting radiator orientation, the following effects must be considered:

- effect of solar radiation.
- reflected solar radiation from the lunar surface.
- long wave radiation from the lunar surface.

A horizontal radiator has been chosen for this design estimate. The lowest effective sink temperature (without use of a ground reflector) is obtained with a horizontal radiator configuration. For this configuration, the most severe condition would be with the sun at the zenith where the incident solar radiation flux would equal the full value of 1396 watts per meter squared. The radiator surface would then be normal to the propagation direction of the solar radiation.

In calculations for determining the effective sink temperature, several sets of absorptivity, or a, and emissivity, or e, values are representative of current state-of-the-art. These are: a = 0.12, e = 0.900; a = 0.165, e = 0.900; and a = 0.197 and e = 0.900. Projected 1969 state-of-the-art values have been determined as a = 0.080 and e = 0.900. Table 7.5.4-1 gives the effective sink temperatures calculated for these data points.

Table 7.5.4-1. EFFECTIVE SINK TEMPERATURES

a	e	a/e	T_{es}
0.080	0.900	0.089	222°K
0.12	0.90	0.133	244°K
0.165	0.900	0.183	266°K
0.197	0.90	0.219	278°K

The mission constraints effectively require peak radiator performance at the end of the first seven days' exposure. Some degradation might be tolerated after this as solar illumination impinges more and more obliquely on the horizontal radiator surface. Use of a low α/ϵ ratio, available in the current state-of-the-art, such as 0.133 for the design estimate, is therefore justified.

For a liquid coolant radiator system, the radiator heat rejection rate equals the heat absorption rate in the coolant flow stream.

A coolant temperature drop in the radiator of $55.5^{\circ}C$ is used for the design estimates. This ΔT is consistent with other problem restraints and is subject to modification in a final design development.

Figure 7.5-43 is a presentation of the results of integration for the ΔT of coolant $= T_2 - T_1 = 55.5^{\circ}C$ under four sets of conditions, namely $T_{es} = 222^{\circ}K$, $T_{es} = 244^{\circ}K$, $T_{es} = 266^{\circ}K$, and $T_{es} = 278^{\circ}C$, which will be used for the baseline estimates of the vehicle radiator requirements. The gross heat rejection rates (i.e. black body rates) shown are actually $\dot{Q}/e_{Tb}n$ per unit area (square meter) where $\dot{Q}$ is the net heat rejection rate, in watts, e_{Tb} is the radiator emissivity and n is the radiator efficiency.

The above analysis assumes no temperature drop across the liquid coolant film wall, nor through the coolant tube wall of the radiator, and hence is only an approximation, valid for cases in which these temperature drops are negligible. The net heat rejection is obtained and, assuming as negligible temperature drops across the liquid coolant film and tube walls, the horizontal radiator area was determined. In addition to determining radiator area requirements, a louver exposure control will be required as it is necessary to keep the battery temperature above $289^{\circ}K$ ($26^{\circ}C$, $60^{\circ}F$). Near lunar sunrise, or lunar sunset, the effective sink temperature will approach $4^{\circ}K$ and, at $289^{\circ}K$, the black body radiation rate would approach 410 watts/m^2. At such a rate, the temperature drop would be quite rapid with little or no heat input. Therefore, for non-vehicle use periods and with operating times of little heat input, it must be possible to shut down the radiator without freezing the coolant. Use of louvers will allow the radiator efficiency to be controlled as desired.

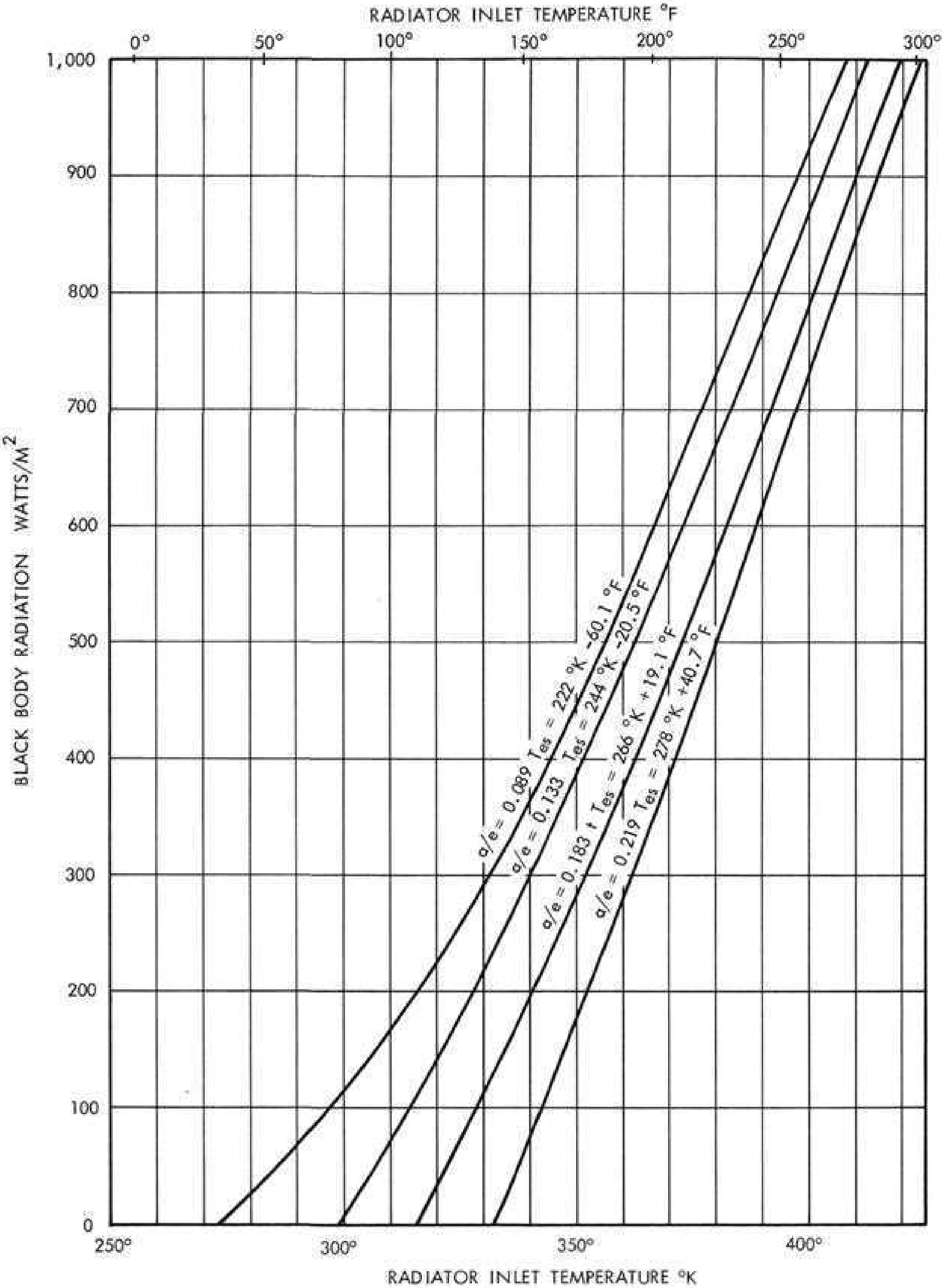


Figure 7.5-43. Equivalent Sink Temperatures, T_{es}

In the design estimations for the heat rejection radiators, the inlet radiator temperature at peak load will not be greater than approximately 360°K (87°C). This is lower than the lunar surface temperature with the sun at the zenith, hence the radiators may be located in view of the vehicle operator without adverse effect from long wave radiation. The radiator with a high emissivity will effectively reflect diffusively the incident solar radiation; hence, other factors being equal, an overhead radiator location would serve both to shade the operator, and would not provide input to, or be affected by the vehicle operator radiant energy balance. The overhead location has the disadvantages that it raises the center of the gravity of the vehicle, and requires lunar on-site assembly onto the vehicle, with flexible connect-and disconnect-couplings. In consideration of the disadvantages, for the purpose of this design estimate, an over-the-fender or over-wheel location for the heat rejection radiators is proposed.

A mass conversion factor of 13 kilograms per square meter radiator area is used to estimate the radiator underside insulation, louver, louver controls, supports, coolant liquid lines and coolant liquid masses. (This is equivalent to more than 2-2/3 lbs/ft^2. Previously used factors for the radiator mass have been as low as 1 lb/ft^2).

The specific heat loads on the radiator system are those actively rejected by the power system functional elements including the coolant liquid circulating pump. Horizontal surfaces will be passively heat rejecting by virtue of suitable reflective coatings (a/e similar to radiator surfaces). Side and underside surfaces must be insulated (specific mass of superinsulation is from 30 to 120 kg/m^3 [1.9 lb/ft^3 to 7.5 lb/ft^3]) to preclude long wave radiation leaks. Since an alternate energy conversion system has been outlined in a previous section, the following subsections will discuss the thermal control for the selected power system and the alternate system, respectively.

<u>7.5.4.2 First Order Modification - Selected Power System Thermal Control</u>.
Peak heat loads and temperature restraints for the selected power system are given in Table 7.5.4-2.

Table 7.5.4-2. PEAK HEAT LOAD SUMMARY, SELECTED SYSTEM, FIRST AND SECOND ORDER MODIFICATION

PEAK HEAT LOAD SUMMARY, MODIFIED M-274 VEHICLE HEAT REJECTION SYSTEM, 1st AND 2nd ORDER MODIFICATION WITH SELECTED ENERGY SYSTEM.

System	Unit	No. of Units	Peak Heat Load Each Watts	Total Peak Heat Load Watts	Minimum Temp °K	Maximum Temp °K
Energy	Battery	8	18.75	150	289 (60°F)	322 (120°F)
Energy Conversion	Motor Control	1	250	250	-	373 (212°F)
Energy Conversion	Electric Motor	1	400	400	-	473 (392°F)
Drive Train	Assembly	1	255	255	-	423 (302°F)
Heat Rejection	Coolant Pump	1	25	25	250 (9.5°F)	370 (306.5°F)
			1080			

In the design of heat exchange devices, many factors must be considered. In this estimate only those factors will be included which are deemed to be constraining in actual design. It will be assumed that a 10°C (18°F) temperature differential between coolant, and waste heat developing component will adequately allow for actual heat exchange system design. The power of 25 watts in the coolant pump is adequate for substantial system coolant flow rates. In roughly sizing the heat rejecting radiator, it is important to determine the maximum outlet temperature allowed. The batteries provide the lowest constraining heat rejecting temperature; which means that the coolant liquid heat capacity, within the design temperature change (equal to the ΔT through the radiator), must be adequate to absorb the battery heat within the temperature constraint imposed. 150/1080 or 0.139 of the total peak heat load must be absorbed by the coolant (below 322°K or 120°F) to maintain the battery temperature below the allowed maximum. The overall $\Delta T = 55.5°C$ (100°F), then the $\Delta T = (55.5)(0.139) = 8°C$ (14.5°F) is allowed from the radiator outlet to the battery heat exchanger outlet for absorption of the battery peak waste heat. Using a 10°C conduit wall temperature differential, the radiator outlet temperature is found to be 322-(8 + 10) = $304^{o}K$ ($88^{o}F$). The design inlet temperature then becomes 304 + 55.5 = $359.5^{o}K$ ($188^{o}F$).

For estimation of the horizontal radiator areas, the following parametric values are used. The radiating efficiency, allowing for interference by the rate control louvers, etc. is taken as 0.8, and the emissivity of the radiating surface is taken at 0.9. Figure 7.5-43 gives gross (black body) radiation rates for a horizontal radiator configuration in which the coolant temperature drop is $55.5^{o}C$, at three values of a/e. Using these values, and the above cited parameters, the radiator area is found as follows: Using the projected a/e = 0.089, the gross radiation rate is 530 watts/m^2. The net rate is (530) (0.9) (0.8) = 382 watts/m^2. The required radiator area is then 1080/382 = 2.84 m^2 (30.5 ft^2). Table 7.5.4-3 shows the radiator requirements for both projected art, and current art a/e ratios.

Table 7.5.4-3. HORIZONTAL RADIATOR AREA, ΔT COOLANT OF $55.5^{o}C$ ($100^{o}F$) FOR HEATLOAD OF 1080 WATTS, INLET TEMPERATURE OF $359.5^{o}K$ ($188^{o}F$), AS A FUNCTION OF A/E RATIO, e = 0.90 and η = 0.80

a/e	Tes ^{o}K	$\dot{Q}$ watts /M^2	Area Required M^2	Area Required Ft^2
0.089*	222 ($-60^{o}F$)	382	2.84	30.5
0.133	244 ($-20^{o}F$)	385	3.22	34.6
0.183	266 ($18^{o}F$)	260	4.15	44.2
0.219	278 ($41^{o}F$)	186	5.8	62.5

*Projected 1969 state of the art.

Based on the Table 7.5.4-3, a horizontal radiator of 3.25 square meters area with surface a/e ratio of about 0.133 is selected for the design estimate. The radiator will be installed over the vehicle wheels, in two sections, one on each side, of essentially equal areas. Louver control must be thermostatic, with manual override. Design configuration optimization can be made only in connection with systems development.

During charging operations, the battery heat rejection is 45 watts each, or a total of 360 watts. The total heat load during charging of batteries is less than at the peak vehicle heat rejection while moving, so that although the coolant pump must operate, the radiator being adequate for the peak load in motion will be more than adequate to dissipate the battery charging heat load.

The radiator mass (including coolant) is obtained allowing 13 Kilograms per square meter radiator area, hence the radiator mass is estimated to be 3.25 x 13 = 42.2 Kilograms or 98 lbs.

The coolant pump mass is estimated at 2.3 Kilograms or 5 lbs. The total mass is then 44.5 Kg or 98 lbs.

The schematic representation of the thermal control system is shown in Figure 7.5-44.

This view shows the Mods 1 and 2 coolant loop, heat input rates in watts, and the estimated radiator area, which is the sum of the two parts, mounted on respective sides of the vehicle.

7.5.4.3 First Order Modification - Alternate Power System Thermal Control. Peak heat loads and temperature restraints for the alternate system are given in Table 7.5.4-4.

Total ΔT of coolant = 55.5^{o}C. The minimum system temperature is the low temperature fuel cell temperature of 313^{o}K (104^{o}F), and the coolant must absorb 320 watts, 320/1370, or 0.234 of the total heat to be dissipated by the radiator. Allowing a 10^{o}C temperature difference between coolant and fuel cell, gives a maximum permissible temperature of 303^{o}K for the coolant; this less (0.234)(55.5) = 13^{o}C, gives the maximum allowable radiator outlet temperature of 290^{o}K and the radiator inlet temperature of 345.5^{o}K.

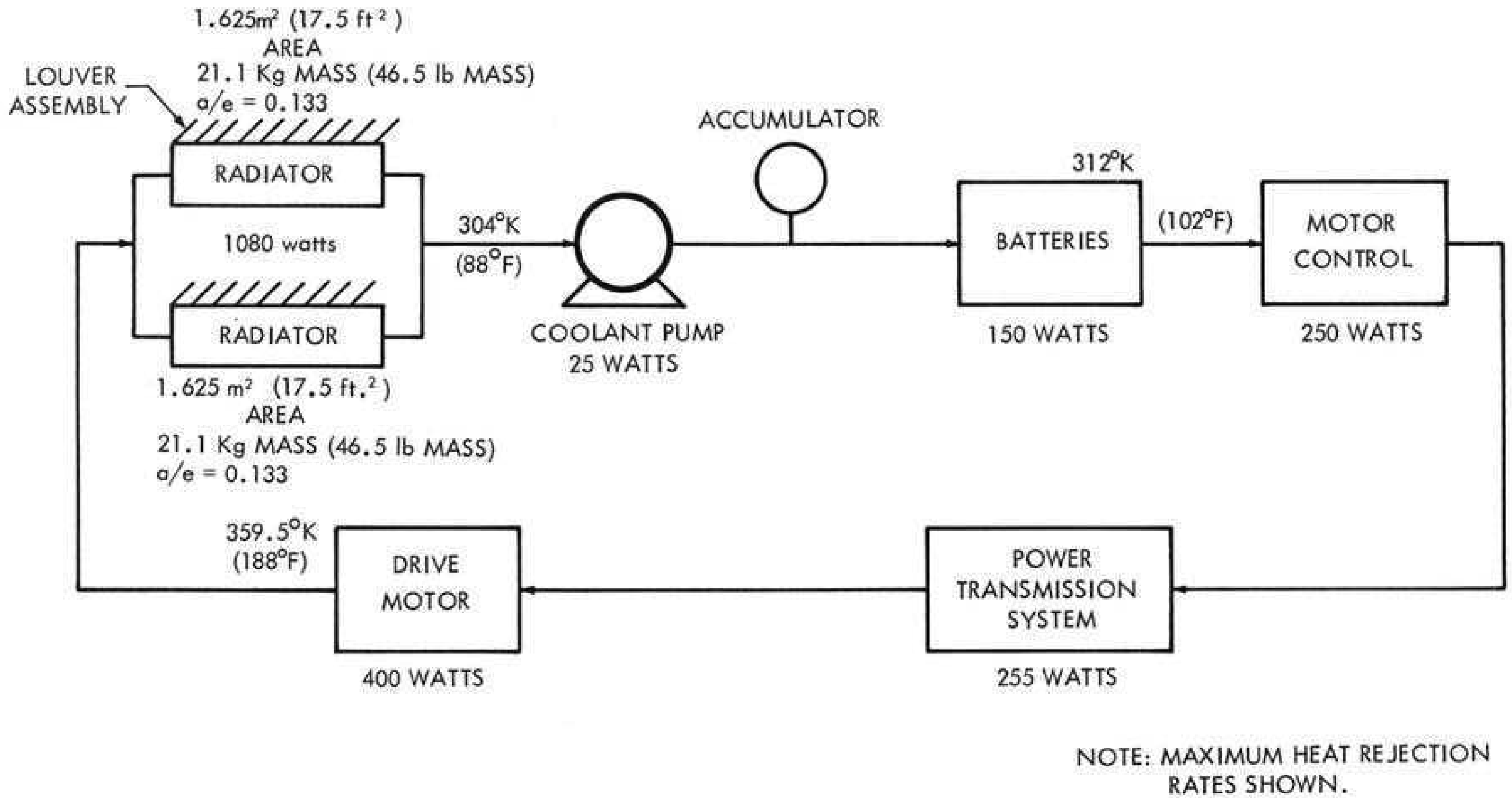


Figure 7.5-44. Coolant Flow Diagram - First and Second Order

Table 7.5.4-4. PEAK HEAT LOAD SUMMARY, MODIFIED M-274 VEHICLE HEAT REJECTION SYSTEM, 1st AND 2nd ORDER MODIFICATIONS WITH ALTERNATE ENERGY SYSTEM

System	Unit	No. of Units	Peak Heat Load Each Unit/Watt	Total Peak Heat Load Watts	Minimum Temp °K	Maximum Temp °K
Energy	Battery	6	20	120	289 (60°F)	322 (120°F)
	Fuel Cell	2	160	320	289 (60°F)	313 (104°F)
Energy Conversion	Electric motor	1	400	400	-	473 (392°F)
Energy Conversion	Control for Motor	1	250	250	-	373 (212°F)
Drive Train	Assembly	1	255	255	-	423 (302°F)
Heat Rejection	Coolant Pump	1	25	25	250 (-9.5°F)	370 (206.5°F)
				1370		

However, the maximum battery temperature allowed is 322°K, giving 312°K for the coolant. Since 240 watts comprises the battery load, the coolant ΔT in the battery cooling jacket is (240/1370) (55.5) = 9.5°C, giving a maximum entering temperature of 302.5°K which is 1/2 of a degree centrigrade or Kelvin lower than the maximum fuel cell coolant exit temperature, hence the radiator inlet and outlet temperatures must be reduced by 1/2°C, giving corrected values of 289.5°K (61.5°F) outlet and 345°K (161.5°F) inlet for the design estimate.

From Figure 7.5-43 and the above parameter, radiator areas are calculated, see Table 7.5.4-5.

Table 7.5.4-5. HORIZONTAL RADIATOR AREA, ΔT COOLANT OF 55.5°C, (100°F) FOR HEAT LOAD OF 1370 WATTS, (ALTERNATE ENERGY SYSTEM), INLET TEMPERATURE OF 345°K (161.5°F), AS A FUNCTION OF a/e RATIO WITH e = 0.90 AND n = 0.80

a/e	Tes °K	$\dot{Q}$ Watts/M^2	Area Required	
			M^2	Ft^2
0.219	278 (41°F)	97.5	1.40	150
0.183	266 (18°F)	175	0.78	84
0.133*	244 (-20°F)	250	0.55	59
0.089**	222 (-60°F)	375	0.365	39.3

*Design estimate values.
**Projected 1969 state of the art.

Using a/e = 0.133, the radiator mass is estimated as 0.55 x 13 = 71.5 kg or 157 lbs.

7.6 SECOND ORDER VEHICLE MODIFICATIONS

This section presents the detailed discussions on the second order vehicle modifications. This discussion presumes that the reader has read previously section 7.4, Summary of Vehicle Modifications.

7.6.1 Mobility and Structure

7.6.1.1. Wheel Consideration. The primary difference between the first order modified vehicle and the second order modified vehicle is the wheel design concept. It is proposed that instead of the first order rigid wheel, a high deflection, round metal elastic wheel be used. The primary purpose of this substitution is to provide an impact absorbing system in the vehicle chassis and hence to improve the dynamic response of the vehicle to terrain impact inputs.

The proposed elastic wheel would have the same proportions as the rigid wheel [diameter 66.8 cm (26.3") x 31 cm (12.2") wide]. A specific design would be chosen from the existing metal elastic wheels for which development is already in progress.

The metal elastic wheel replacing the pneumatic tired basic vehicle wheel will still be rigidly mounted to the vehicle. This concept carries important implications with regard to the modified vehicle's dynamic response in comparison with conventional terrestrial wheeled vehicle suspensions. In the typical terrestrial wheeled vehicle suspension system, an impact delivered to the pneumatic tired wheel (elastic in a sense) is partly absorbed by the wheel (internal friction). Locally the tire experiences very high loads and accelerations. However, the majority of the impact energy goes into accelerating the mass of the wheel and suspension control members. This mass, called the unsprung mass, is accelerated at a reduced rate over a large displacement (up to 6 inches in passenger cars). The sharpness of the impact is thus attenuated. The impact energy is transferred to the main mass of the vehicle, the sprung mass, by the suspension spring and damper (shock absorber) at a lower acceleration level over a relatively longer period of time. This causes the sprung mass, generally ten times the magnitude of the unsprung mass, to be accelerated slowly over a smaller displacement.

The disturbance the sprung mass experiences is dependent on the ratio of sprung mass to unsprung mass, the suspension spring rate, and the coefficient of damping of the shock absorber. Up to this point, the analysis has been concerned with the exchange rather than the dissipation of energy. Impact energy is dissipated only by friction. All actual suspension elements have internal frictional losses, but the shock absorber is the one element principally designed to dissipate impact energy. Kinetic energy is converted into heat through friction (primarily fluid friction in the usual shock absorber).

A desirable low acceleration, low displacement ride is achieved principally by low suspension spring rates, high wheel travel, optimum proportions of unsprung mass to sprung mass, and optimum damping. Typically this information has been determined experimentally for a given vehicle.

The rigidly mounted, elastic wheel, modified vehicle's suspension can now be analyzed in comparison. The unsprung mass consists only of the elastic rim and about half the springing material. The sprung mass is the rest of the modified vehicle. The ratio of sprung to unsprung mass is very high.

The suspension "spring" is the elastic portion of the wheel itself. The suspension travel equals the allowed maximum deflection of the rim. The only damping that may be introduced in this system is in the internal friction of the rim and springing elements.

An impact load applied to the metal elastic wheel causes an acceleration of a portion of the wheel rim and springing elements (a very small mass) over a distance dependent on the design of the wheel. This results in the acceleration of the large unsprung mass at a lower rate over a longer period of time than the original impact. The energy is transferred to the body of the modified vehicle by the elastic wheel in the same manner as a conventional suspension transfers energy to a terrestrial vehicle. However, the sharpness of the original impact is attenuated to a lesser extent. Energy dissipation (damping) is available only as internal friction in this wheel. Since damping is the only way in which a vibrating system may be brought to rest, it is an important quality. It would thus seem that a substantial amount of internal friction in the elastic wheel is desirable.

From the ride standpoint, this is true. However, wheel deflection occurs not only during obstacle encounter but also in the normal motion of the wheel over smooth terrain, and thus the internal friction also adds to the rolling resistance of the vehicle.

Another desirable quality of a terrestrial suspension system that would also be desirable in the modified vehicle elastic wheel is the absorption of inputs from microtopography (terrain roughness). On a terrestrial vehicle, this function is performed by the pneumatic tire. The tire tread in contact with the ground locally envelops small obstacles. The energy input is dissipated as internal friction in the tread and tire mass. The metal elastic wheel could simulate the same quality by an extremely flexible rim.

The major effects of the rigidly mounted elastic wheel on the dynamic response of the Lunar Mule may be summed up as follows:

- Significantly lower acceleration rates of the Mule unsprung mass, dependent on the spring rate of the wheel and the deflection.
- A reduction in bounce height as compared to the rigid wheel concept, dependent on the damping coefficient (internal friction) of the elastic wheel.
- A reduction of the transmission of low amplitude high frequency vibration (roughness-microtopography) dependent on the local flexibility of the rim.

The desirability of a high coefficient of damping in the elastic wheel for good ride quality is in opposition to a low rolling resistance requirement to minimize power input. These opposing requirements are the major objections to the rigidly mounted elastic wheel. A compromise could best be arrived at through a test and development program.

7.6.1.2 Power and Energy Calculations. The use of the elastic metal wheel on the second order modified vehicle has the effect of reducing the second order vehicle's power requirement. This reduction is only valid when the vehicle is operating in the soft soil group. In the soft soil, the vehicle utilizes approximately 80% of the power required to operate the first order vehicle with a rigid wheel. There is no appreciable power reduction when the second order vehicle operates in the harder lunar soil groups.

Utilizing the elastic metal wheel on the second order modified vehicle would allow an energy requirement for one (1) traverse of 14.5 KWH. This compares to 15.5 KWH for the first order vehicle. This is only a 6.5 percent reduction. Accordingly, no change in the first order vehicle modifications energy requirements is proposed for the second order vehicle.

7.6.1.3 Obstacle Negotiation. The metal elastic wheel performance for obstacle negotiation, due to the elasticity of the wheel, cannot be predicted. Therefore, the improvement or degradation of obstacle negotiation capability of the elastic metal wheel over that of the rigid wheel cannot be predicted.

7.6.2 Power System

The power system concept utilized for the first order modified vehicle is proposed without change for the second order vehicle.

7.6.3 Crew Station

The crew station concept utilized for the first order modified vehicle is proposed without change for the second order vehicle.

7.6.4 Thermal Control

The thermal control system concept utilized for the first order modified vehicle is proposed without change for the second order vehicle.

7.7 THIRD ORDER VEHICLE MODIFICATION

This section presents the detailed discussions on the third order vehicle modifications. This modification order is made to improve the mobility performance of the lunar vehicle to an optimum consistent with the criteria and retention of the basic vehicle's compactness and ruggedness. This order of modification will, therefore, be a maximum modification departure from the basic vehicle consistent with retention of the basic vehicle structure configuration. These discussions presume that the reader has read previously section 7.4, Summary of Vehicle Modification Concepts.

7.7.1 Mobility and Structure

7.7.1.1 Wheel and Suspension Considerations. The significant chassis features of the third order modified vehicle include substitution of a noncircular, elastic metal wheel for the basic vehicle's pneumatic tired wheel, and independent all-wheel suspension.

The proposed design of the wheel and suspension system for this third order modified vehicle is shown in Figure 7.7-1.

A magnitude of improvement in vehicle ride will be experienced with the introduction of a wheel suspension. It is of the long and short control arm type with a forward control strut on the lower control arm. A torsion bar spring is splined to the lower control arm at the inner pivot and anchored to the chassis. A rotary double acting damper is similarly splined to the lower control arm.

The outer ends of the control arms are attached to the electric drive motor case through ball joints. The motor case and gear case are thus used as structural members. The steering knuckle is also integral with the motor case. The design of all suspension pivots will be based on use of the development of the steering tie rod ends modifications as proposed for the first order vehicle.

The suspension configuration outlined is typical of existing automotive suspensions. Thus a vast background of engineering data is already available. A nearly optimum design in every respect could be arrived at on the first try.

With the introduction of a wheel suspension the modified vehicle takes on the desirable dynamic characteristics associated with "good" riding vehicles. As discussed previously,

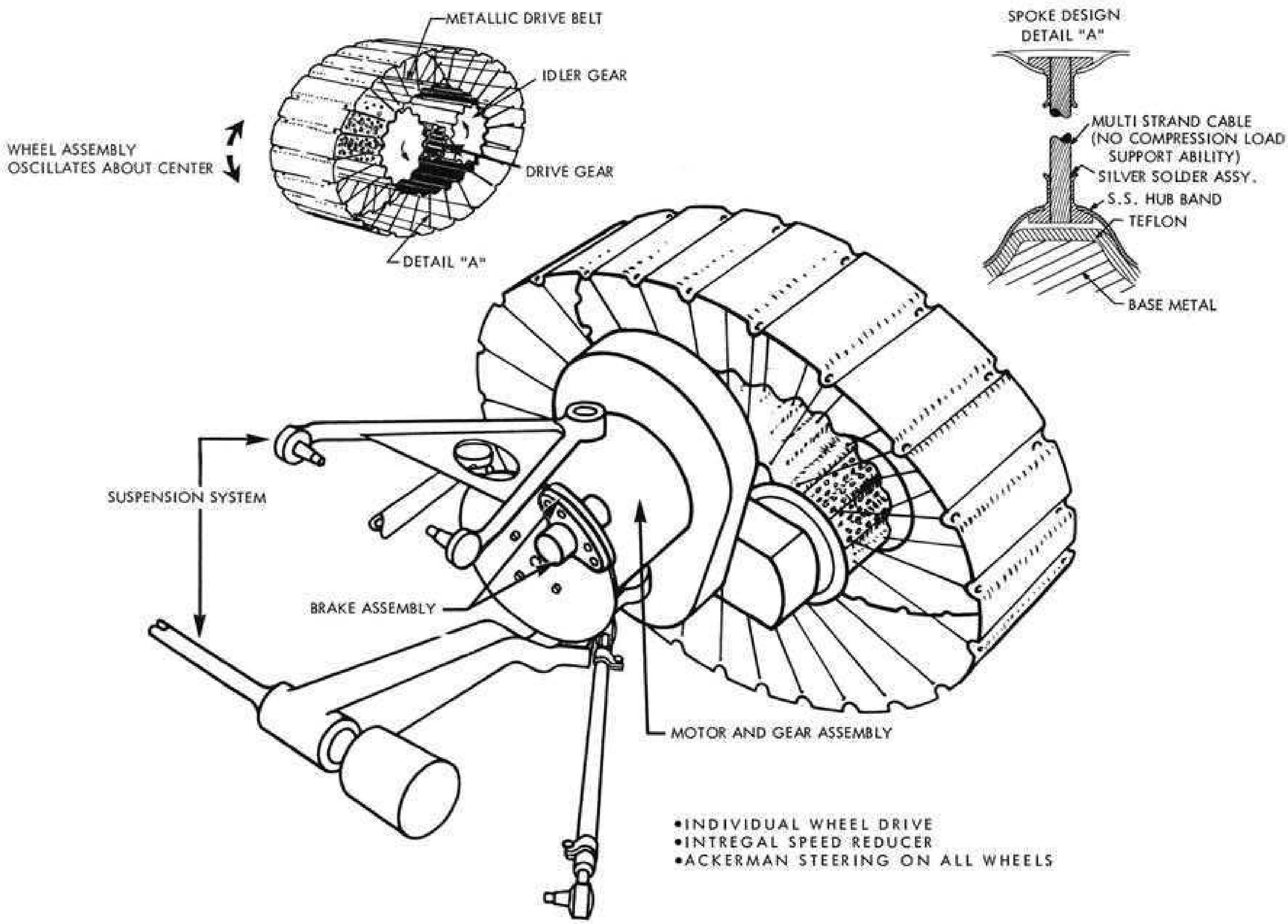


Figure 7.7-1. Constrained Elliptical Wheel - Third Order

for the second order vehicle, a low acceleration, low displacement ride is dependent on the attenuation of impact loadings through successive springing systems and the dissipation of the energy by frictional devices.

An impact load (obstacle encounter) applied to the wheel on the third order vehicle compresses the elastic rim but now must also accelerate the considerably greater unsprung mass of the wheel, gear box, motor, brake, and a portion of the control arms. Thus a considerable attenuation of the impact loading has been achieved by merely increasing the unsprung mass.

The motion of the wheel and its associated components is resisted by the sprung vehicle mass and the vehicle suspension spring. Thus impact energy is further attenuated in its transfer to the large sprung mass of the Mule. In this portion of the suspension, it is desirable to introduce a considerable damping force (shock-absorber) to dissipate the impact energy. The damping introduced at this point does not contribute to the rolling resistance of the vehicle.

It can be generally stated that the longer the suspension travel the better the ride quality. In the case of the modified vehicle a jounce and rebound travel of 4-5 inches would be suggested.

The proposed suspension for the third order vehicle would be feasible with any type of wheel, but in keeping with the effort for an optimum vehicle on the third order, a new concept is proposed. The new concept is basically the metal elastic wheel constrained to a noncircular configuration and so mounted that the low curvature portions of the wheel is always the ground contact surface. The advantage of this scheme is a larger ground contact area over the same wheel in a circular configuration. The larger contact area results in lower contact pressures, less sinkage in a given soil and hence lower rolling resistances. The magnitude of improvement is demonstrated by the fact that predicted soft soil performance for the noncircular wheel may be calculated from a circular wheel up to three times the minor diameter of the elliptic, or noncircular, wheel.

The use of the elliptic wheel is the principle reasons for significantly lower operational power and energy requirements in the third order vehicle.

The proposed design of the wheel for the lunar mule was shown in Figure 7.7-1. Basically the design consists of two concentric bands of stainless steel connected by a series of tension members (cables). The assembly is then deformed elastically to fit over two sprockets separated a fixed distance. The diameter of the sprockets and their distance apart determine the eccentricity of the wheel. The two sprockets are free to rotate about their own axes but the major axis of the wheel is constrained to remain approximately parallel to the ground.

The inner stainless steel band is corrugated to match the sprocket teeth. The outer band is corrugated at intervals to simplify attachment of the tension members and for mobility reasons.

The wheel may be powered or unpowered. The third order vehicle will have four powered wheels. The scheme is to introduce a pinion between the two outer sprockets and drive one of the sprockets. The other sprocket is allowed to rotate freely with respect to the driven sprocket. The purpose is to allow as much elastic deformation of the inner and outer stainless steel bands as possible. By inspection it can be seen that a substantial flattening of the contact surface of the wheel would cause a shortening of the corrugated inner band on one side and a corresponding lengthening on the other side. One free sprocket is necessary to allow this to happen without undue stresses developing in the entire system.

It is proposed that the wheel final drive be operated exposed directly to the lunar environment. As stated previously, the inner band is stainless steel. The two outer sprockets are teflon coated and the drive pinion is stainless steel. The alternating use of stainless steel and teflon is expected to prevent cold welding. The band, sprockets, and drive pinion would be sufficiently wide to keep tooth contact pressures at a low value. A shield would be provided to exclude large debris from the drive. A possible buildup of small particles on the sprockets would be minimized by perforating the inner band and the drive pinion. The scheme is to make these components self-cleaning.

The torque input through the drive pinion and sprocket is carried to the rim through the numerous tension members (cables) connecting the inner and outer bands. This is possible because the center line of a portion of the spokes (tension members) do not pass through any center of rotation and, therefore, can transmit torque as a tension force.

The noncircular metal elastic wheel has the same dynamic characteristics as the conventional metal elastic wheel. However, this particular concept offers additional impact absorbing qualities if properly mounted to the vehicle or suspension control arm. If the two outer sprockets, mounted on a rigid link, are allowed to oscillate about an axis parallel to the axis of the sprockets, and midway between them, then an additional "bogie" action is introduced. This "bogie" action by simple geometry helps to attenuate impact loading. It is proposed that the bogie mounting scheme be utilized on the third order vehicle. Some elastic centering device may be necessary to insure that the major axis of the wheel remains properly oriented with respect to the ground.

A working model of this wheel as proposed here for lunar surface use has not been built; however, a working model of the wheel for terrestrial use is shown in Figure 7.7-2. For terrestrial use, two strands of roller chain with extended side plates are used in place of the inner corrugated band. The model shown is fitted with rigid spokes but flexible spokes are preferred. This wheel is presently being developed by private industry for a terrestrial application. The concept, however, was originated by the U.S. Army.

The noncircular wheel represents a significant performance improvement over the other proposals in this study; however, it will require a greater development cost.

7.7.1.2 Structure. The basic vehicle chassis is closely integrated with the existing drive line system components, see Figure 7.5-25 and 7.5-36. In the third order vehicle all existing drive line components are removed. Consequently, the existing platform-chassis support structure will be reworked. A new frame platform support structure will be designed. The wheel suspension loads and torsion bar support points will be

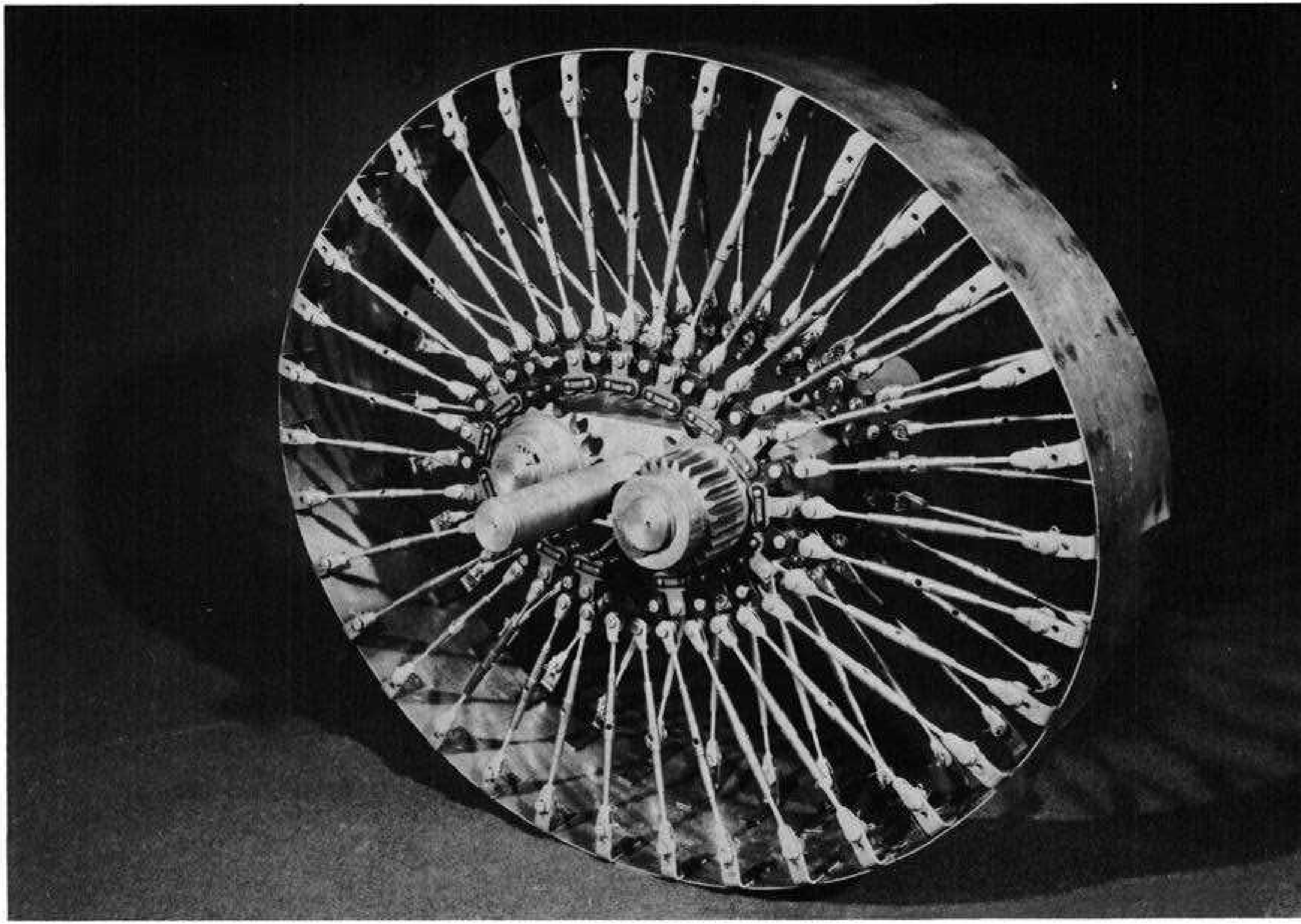


Figure 7.7-2. Test Model - Constrained Elliptical Wheel

carried by new frame members. The frame will be fabricated from magnesium or aluminum sections with possible use of titanium.

7.7.1.3 Power and Energy Calculations. Reference is made to Figure 7.4-8, section 7.4.6, which summarizes the power requirements for one traverse for the three orders of vehicle modification.

The assumptions and qualifications, as given in section 7.5.1.2, are utilized for this third order of modifications.

The calculation procedure followed the same procedure utilized in the first order modification power and energy calculation. The use of the constrained elliptic, elastic metal wheel, allows use of an effective wheel diameter of three (3) times the minor axis diameter of the actual wheel in power calculations. This is an approximation obtained by actual soil bed testing with wheel models in ATAC's Land Locomotion Laboratory.

The graphs developed for tractive effort versus slip percentage for each soil group are given in Figures 7.7-3, 7.7-4 and 7.7-5. Wheel horsepower and total drive motor input horsepower was calculated, see Figures 7.7-3A, 7.7-4A, and 7.7-5A. Vehicle speed requirements were again as set by the criteria as at least 4.8 km/hr (3 mph) in the level soft soil group and 8.0 km/hr (5 mph) in the level compacted soil group.

The total power required per single traverse was developed. This power profile is utilized in sizing the power system. A total of 6 KWH is required. This is one-half the single traverse power requirement for the rigid wheeled first order vehicle.

7.7.1.4 Obstacle Negotiation. The use of the noncircular, elastic metal wheel, could provide an improvement in obstacle negotiation performance and ditch crossing if the major axis of the wheel could be oriented selectively with respect to the obstacle or ditch.

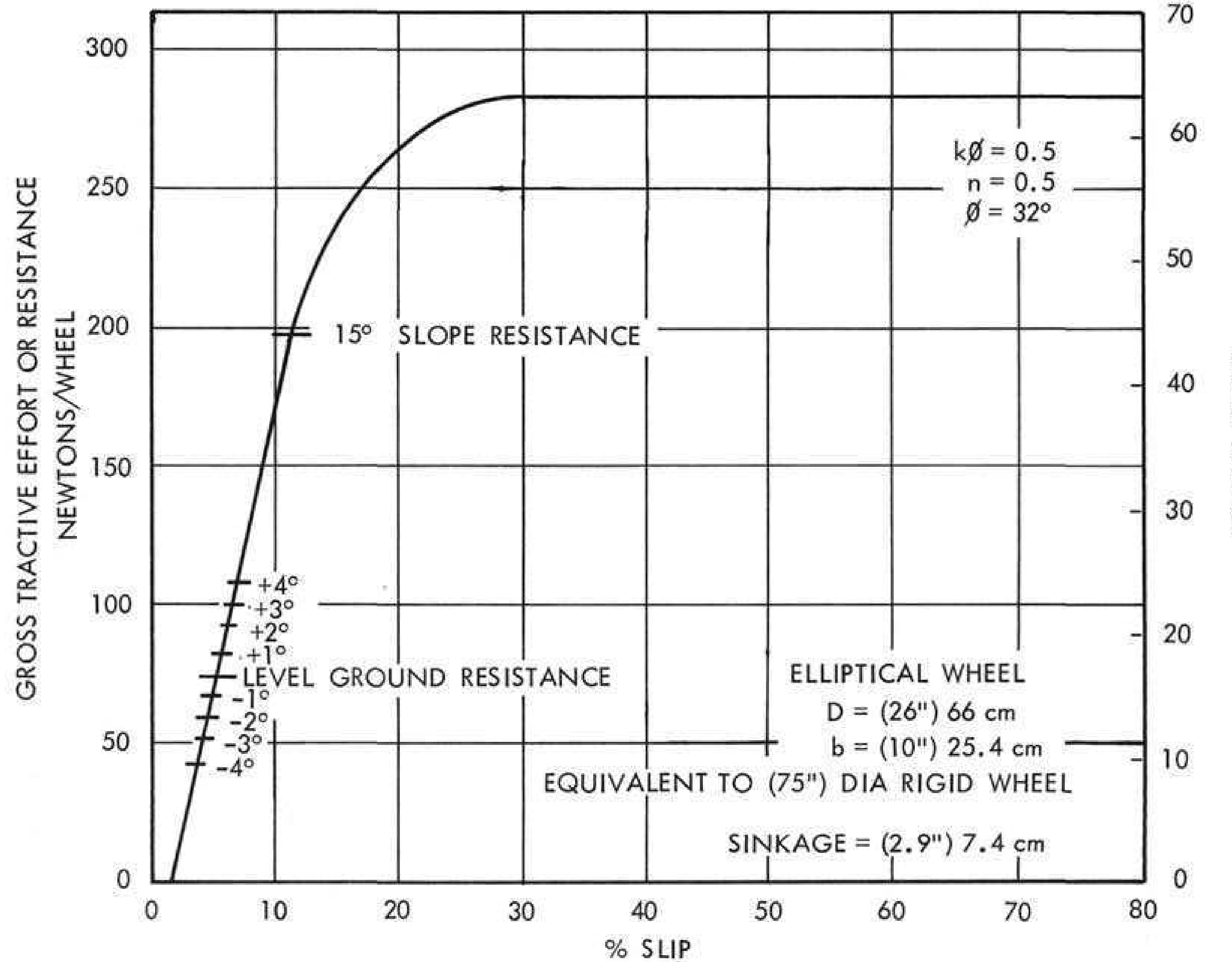


Figure 7.7-3. Tractive Effort vs Slip - Third Order, $K\phi = 0.5$

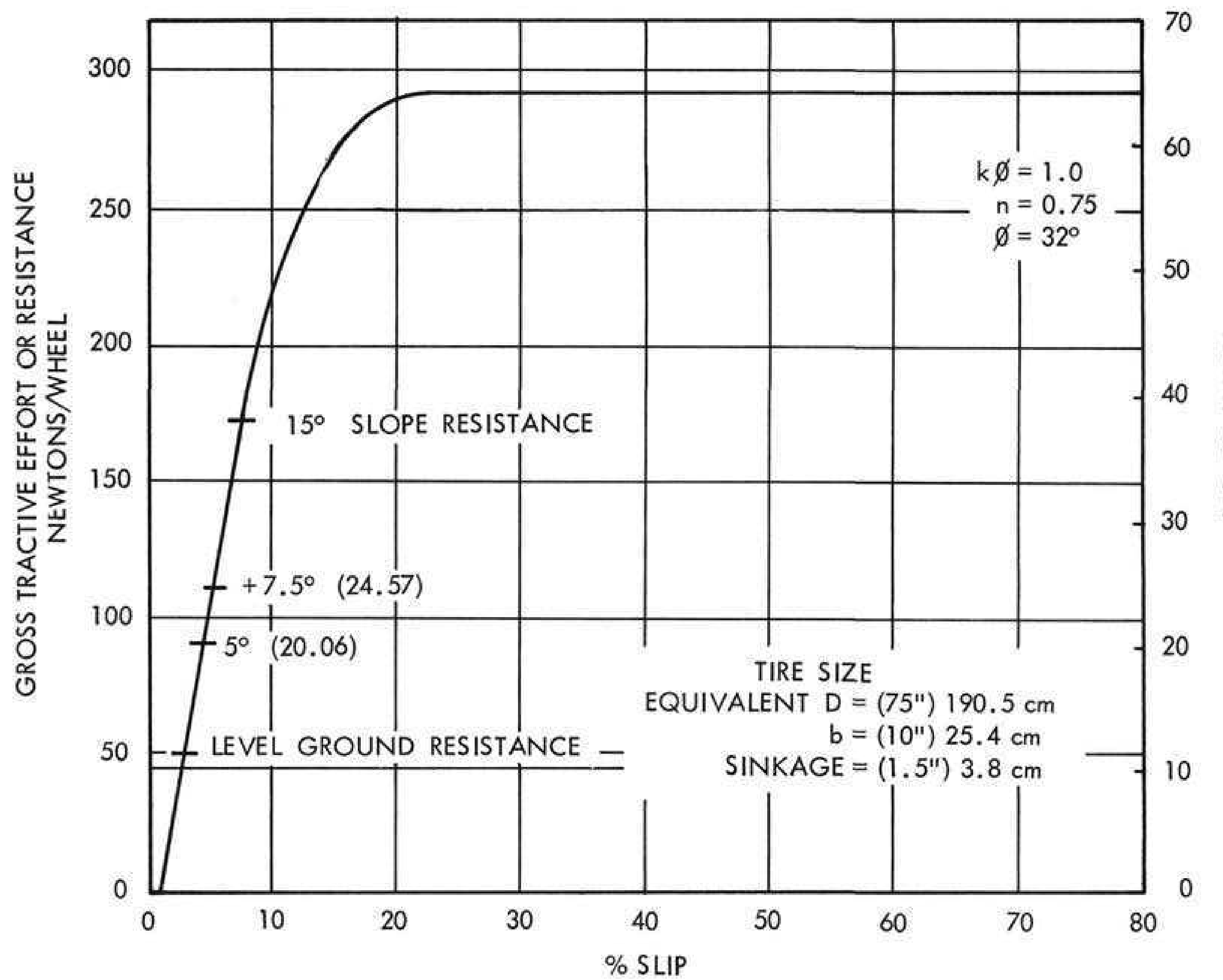


Figure 7.7-4. Tractive Effort vs Slip - Third Order, $K\phi = 1.0$

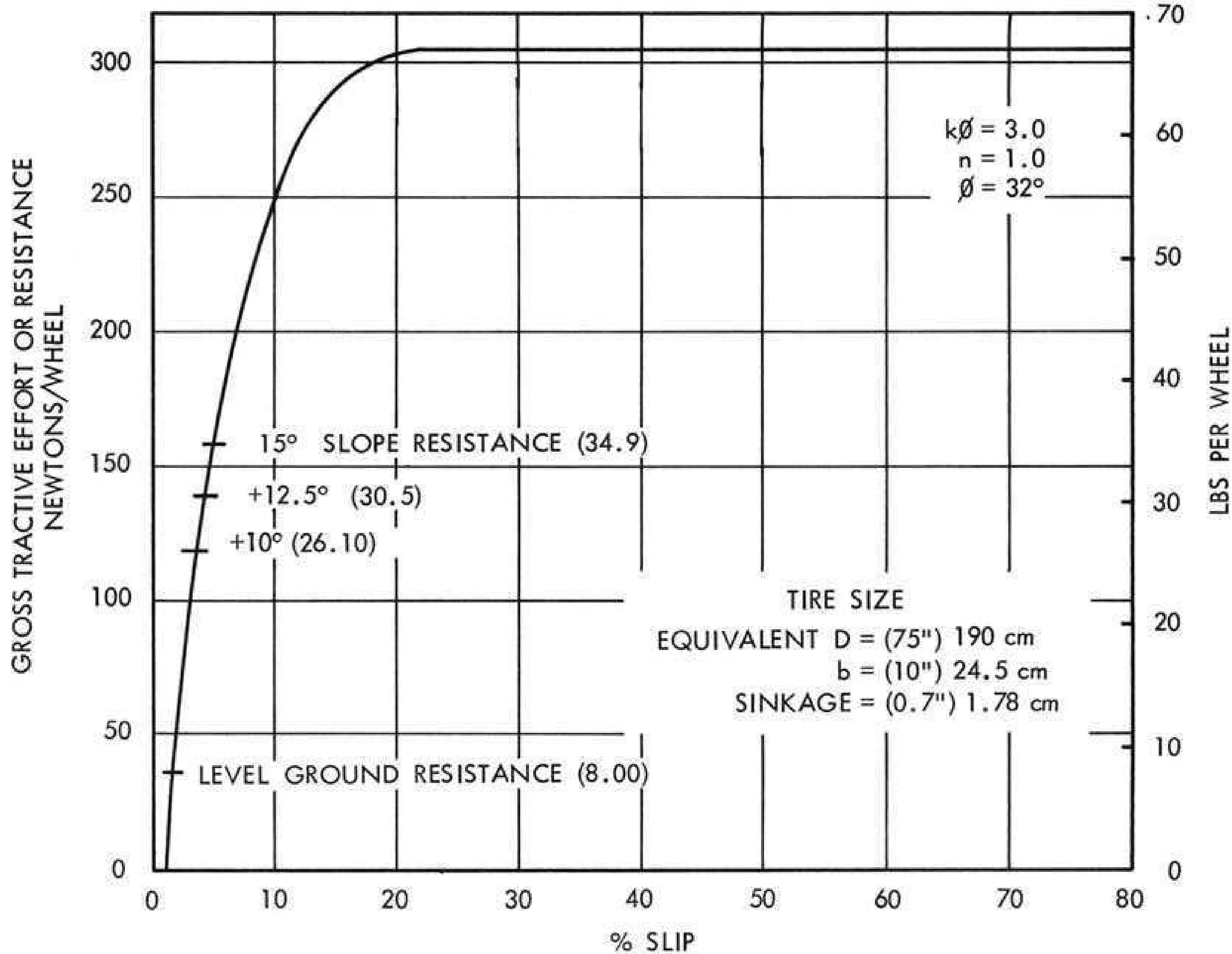


Figure 7.7-5. Tractive Effort vs Slip - Third Order, Kø = 3.0

Grade	Req. Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req'd At Wheel/Wheel	÷ .8	x 4
0	16.8	5.4%	35.1 (3 mph)	37.	.115	.15	.575 say .6
+1°	18.61	5.85%	31.9 (2.72)	33.9	.12	.15	.6
+2°	20.43	6.15%	29.1 (2.49)	30.9	.12		.6
+3°	22.24	6.55%	26.5 (2.25)	28.3	.12		.6
+4°	24.05	7.0%	24.5 (2.1)	26.2	.12		.6
-1°	14.99	5.0%	35.1	36.9	.105	.132	.528
-2°	13.17	4.8%	35.1	36.8	.092	.115	.46
-3°	11.36	4.2%	35.1	36.7	.079	.099	.396
-4°	9.55	3.95%	35.1	36.6	.067	.084	.336

Figure 7.7-3A. Horsepower Requirements - Soil Group 1

Grade	Req. Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req's At Wheel/Wheel	÷ .8	x 4
+5°	20.06 x 1 = 20.06	4%	30.2 (2.57 mph)	31.5	.12	.15	.6
+7.5°	24.57 x 1 = 24.57	4.7%	24.5 (2.1 mph)	25.6	.12	.15	.6
-5°	1.94 x 1 = 1.94	1%	35.1 (3 mph)	35.7	.0132	.0165	.066
-7.5°	Vehicle Coasts						
0	11 x 1 = 11	2.5%	35.1 (3 mph)	36.0	.0754	.0943	.377
0	11	2.5%	56 (4.78 mph)	57.3	.12		.6

Figure 7.7-4A. Horsepower Requirements - Soil Group 2

Grade	Req. Wheel Torque	Slip	Req. Veh. Speed In Wheel RPM	Actual Wheel Speed	HP Req'd At Wheel/Wheel	÷.8	x 4
+5°	26.10 x 1 = 26.1	3.7%	23.4 (2.00)	24.1	.12	.15	.6
+12.5°	30.5 x 1 = 30.5	4.3%	19.8 (1.70)	20.7	.12		.6
+15°	35.9 x 1 = 34.9	5%	17.2 (1.48)	18.05	.12		.6
-10°	Vehicle Coasts						
-12.5°	Vehicle Coasts						
-15°	Vehicle Coasts						
0	8 x 1 = 8	1 1/2%	35.1 (3 mph)	35.5	.0541	.0676	.271
0	8	1 1/2%	78 (6.66)	78.7	.12	.15	.60

Figure 7.7-5A. Horsepower Requirements - Soil Group 3

In respect to obstacle negotiation, with a favorable assumption of wheel major axis alignment, an obstacle height of 14.6 cm (5.75") is obtained by calculation. This is approximately a 20% increase over the first order wheel capability.

For ditch crossing, the major axis of the wheel can be assumed parallel to the ground surface. Calculation then shows that a 61 cm (24") ditch can be negotiated. This is an improvement of approximately 10 cm (4") in capability over that of the first order rigid wheel.

If preferential axis orientation could not be made effectively, the vertical obstacle negotiation capability will be reduced to that of the first order modification. The ditch crossing capability will be effective.

7.7.2 Power System

The third order modification is an attempt to define an optimum vehicle concept within the basic vehicle configuration. It was decided, in addition to elimination of the internal combustion engine, that the existing mechanical drive train of the basic vehicle would be eliminated and replaced. The proposed drive line consists of individual wheel drive electric motors and integral speed reducers at each wheel. The motors will be A.C. type. Power transmisison is through a harmonic drive speed reducer and a conventional gear drop case to the wheel axis. This concept is shown in Figure 7.7-1.

7.7.2.1 Prime Power System.

7.7.2.1.1 Selected Concept—The third order modified vehicle requires 6 KWH of electrical power output per traverse. The power profile for a single typical traverse is shown in Figure 7.7-6. The block diagram for the proposed power system is shown in in Figure 7.7-7. To maintain a power source redundancy, three battery packs will be used. These battery packs are the same type units discussed in first order modifications and as presented in Figure 7.5-10, under first order modifications. To provide the 0-30 amperes required at 42 v.d.c., up to 10 amperes will be drawn from each battery

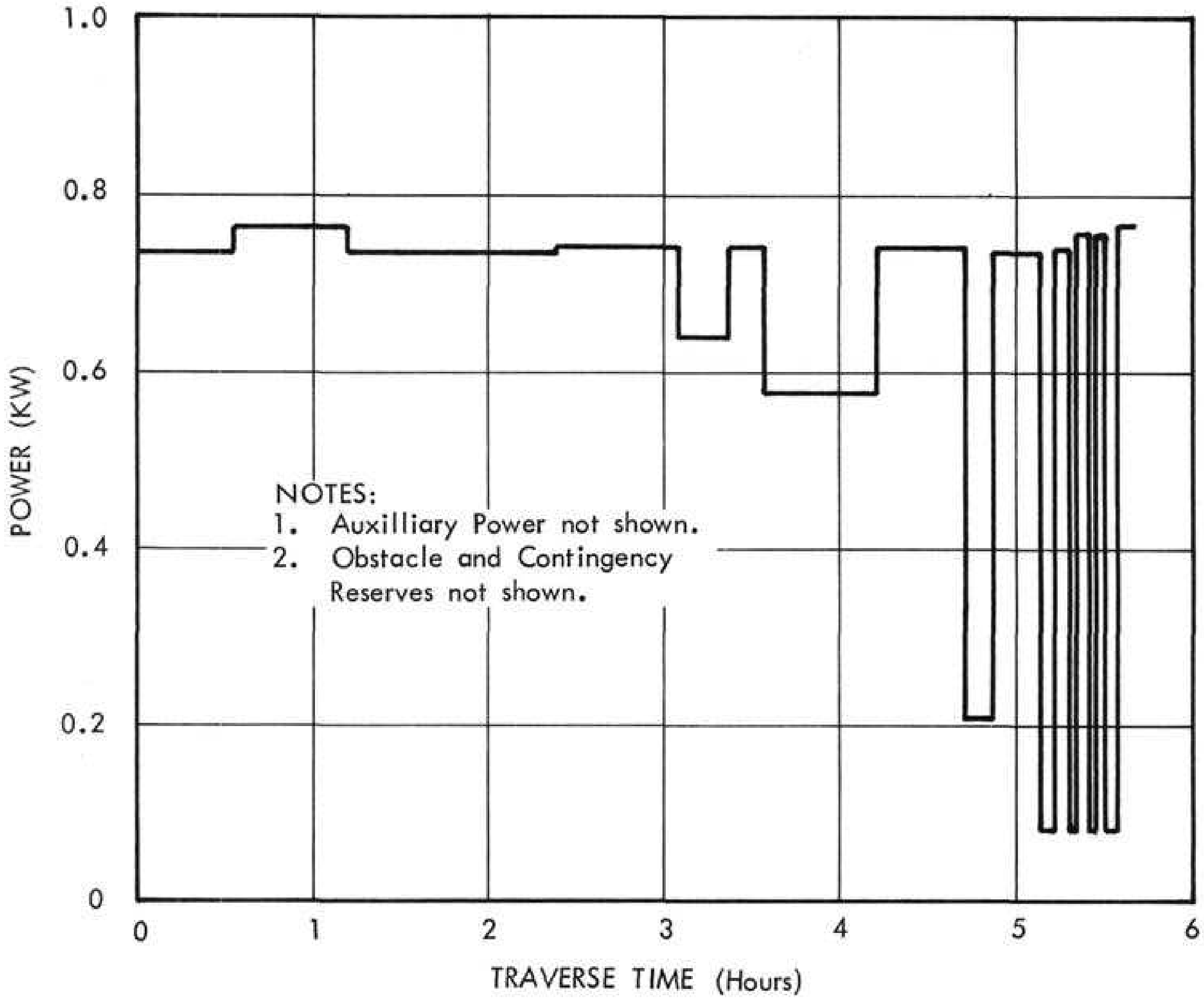


Figure 7.7-6. Power Profile for Traverse - Third Order

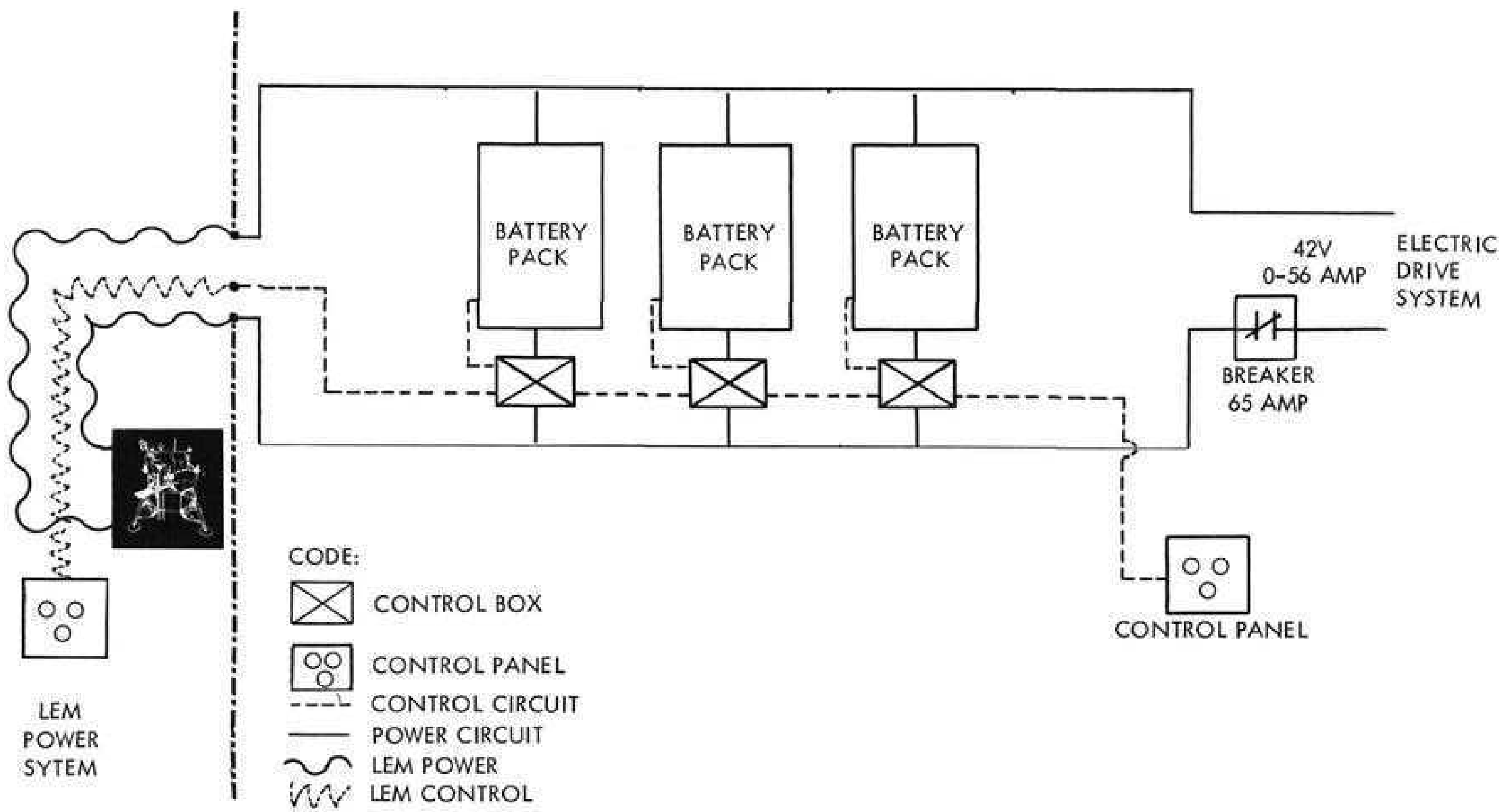


Figure 7.7-7. Power System - Third Order

pack. An estimated 48 amp/hrs will be drawn from each cell during each traverse. This is a 28% depth of nominal discharge capacity.

Any one of the three battery packs will insure the success of an individual traverse; and any two will insure total mission capability, 10 traverses. At the 28% depth of discharge, the battery reliability will be higher than the 0.998 (80% confidence level) established for this battery for the Surveyor mission. At this lower depth of discharge, a period of 10 to 15 hours will be required, after traverse completion, for complete recharging from the LEM/S power system.

7.7.2.1.2 Alternate Concept—Because of the much larger operational flexibility which could be gained by having the lunar surface vehicle independent of the LEM/S for battery recharge power, an alternate possible concept is presented. This alternate system concept will utilize a radioisotope powered thermionic generator to provide vehicle propulsion power and to provide recharge power to battery packs during non-load periods.

The block diagram of this alternate power source system is shown in Figure 7.7-8. The battery packs, two in number, are the same type units recommended for the Selected Concept. One of the three battery packs recommended for the Selected Concept is replaced in this alternate concept by the radioisotope generator. The battery packs are retained in this concept to provide a high reliability and to provide the surge current required for the motors during obstacle crossing and for higher speed operation. The isotope generator provides a constant level of 500 watts D.C. of electrical power, which is either used in load sharing for vehicle propulsion, for recharging the batteries, or is dissipated as heat when the vehicle requires no power and the batteries have been completely recharged.

On discharge the load is shared between the batteries in parallel and the radioisotope generator, each contributing approximately 10 amperes at 42 volts D.C. at peak load. In this system concept, each battery pack is discharged to the extent of 36 amp/hrs per traverse. This is 21% depth of nominal discharge capacity.

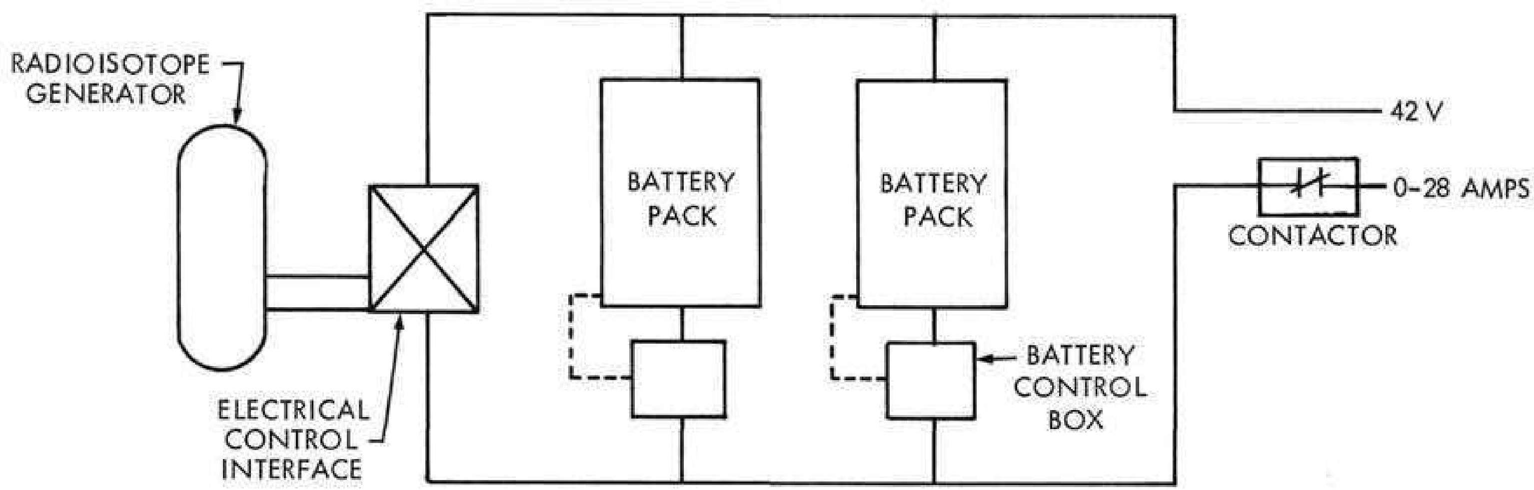


Figure 7.7-8. Battery - RTG Option - Third Order

The reliability of this configuration is essentially the same as the Selected Concept using only batteries. A single battery pack failure will neither effect the reliability of completing a specific traverse nor the total mission reliability. The recharge reliability will, however, be increased since, in this alternate concept, the battery charger is completely on board and connected permanently to the battery packs. In the case of the systems which are recharged from the power system in the LEM/S, any failure concerned with the umbilical cable, its connectors, the LEM/S power supply, or the cryogenic LEM/S tankage and delivery of the hydrogen and oxygen required to produce electrical power, would be a failure of the vehicle mission.

The use of the radioisotope powered thermionic generator would completely free the vehicle from any dependence on the LEM shelter for electrical power needs. Sufficient power would be produced by the 500 watt generator over the two week period to provide full electrical propulsion power and also to provide an amount of on board vehicle power for scientific mission use, if required. Since the power conditioning associated with the thermionic converter package would provide a continuous 500 watts of power appropriate for battery charging, or for load sharing during vehicle propulsion periods, the battery packs will be automatically recharged at any point where the total system load is less than 500 watts. This fact combined with the relatively low depth of discharge on the battery packs indicates that the vehicle will be immediately capable of a second traverse upon the return of the first traverse. This gives the astronauts a great flexibility in scheduling the specific traverses to be undertaken. The only requirement for the power system would be that over a total of several traverses, the total amount of time that the vehicle must remain stationary must be approximately three times the amount of time the vehicle is operated at full power.

Although this system using a radioisotope powered thermionic generator adds a great system flexibility and increased reliability, it was not proposed as the power concept for the simplest modification. This is because presently no space qualified hardware is available which will provide the comparable advantages of this radioisotope powered thermionic generator concept. This system, therefore, involves relatively high development costs compared to the other systems, where only presently space qualified hardware was incorporated and where the system integration engineering and testing costs should be minimal. Several types of radioisotope powered generators are now being investigated by NASA and being supported in various stages of research or development. The prime requirements for the radioisotope device for this vehicle application are a very compact size, light weight, and by far the most important, a high heat rejection temperature. Because of the physical restraints of placing a radioisotope generator on the vehicle, this particular radioisotope heated thermionic generator was selected because its heat radiator is self-contained and it is sufficiently light and sufficiently small in envelope volume to be arranged on the vehicle in a manner to maximize the total angle which it can dissipate heat to the space and lunar environments without radiating to other vehicle components.

A sketch of the physical dimensions of the generator, the radiation temperature and heat loads are given in Figure 7.7-9.

Description of Radioisotope Powered Thermionic Generator

The use of the decay energy of a radioisotope as a primary energy source has great attraction in space power systems. The heat can readily be converted to electricity and is also useful in controlling the thermal environment of the system of which it is a part. There are several isotopes which release very large quantities of thermal energy per unit of weight over extended periods of time, a most important factor in space power systems.

The thermionic energy converter is an attractive means of converting thermal energy to electricity in space where its light weight, low volume, relatively high efficiency and high heat reject temperature can be utilized to full advantage. However, until recently the use of radioisotopes with thermionic converters was impractical. Those isotopes with sufficiently high power densities to operate a thermionic converter have half-lives too short for practical missions. Conversely, the longer-lived isotopes have power densities too low for thermionic use.

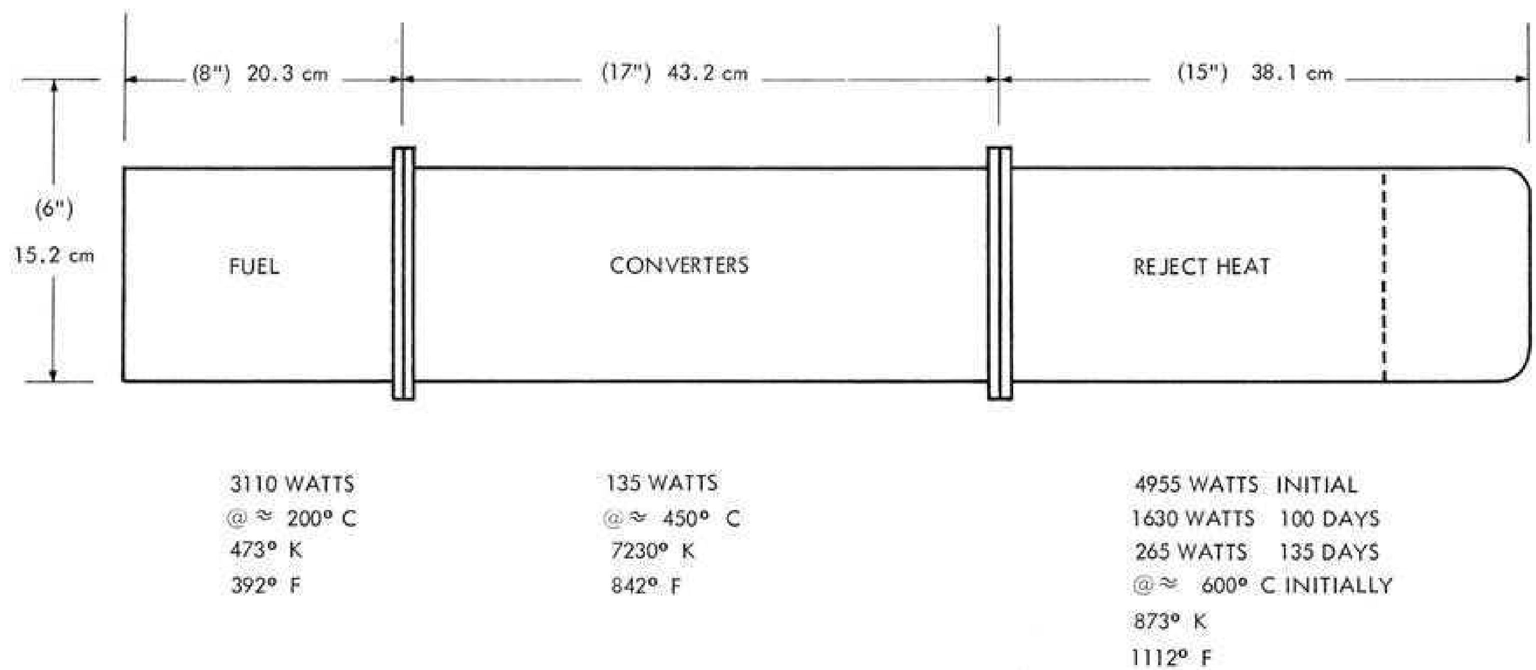


Figure 7.7-9. Radioisotope Thermionic Generator - Third Order, Alternate

The heat pipe is the "missing link" which has made isotope-thermionic space power systems practical for the first time. The heat pipe is a heat transfer device of great versatility. It couples the high heat transfer capabilities of evaporation and condensation with capillary return of the condensate to the evaporator. By suitably varying the ratio of the heat input and heat output areas of the heat pipe, the thermal flux density can be either diffused or concentrated to match exactly the otherwise unequal thermal characteristics of the isotope and converter. Thus, the low density energy from a long-lived isotope can be concentrated to the level required by a thermionic converter. Through an approach to heat pipe operation now being applied to isotopic systems for NASA, it is possible to deliver constant power to a thermionic converter at constant temperature through as many as three half-lives of an isotope. This operation is entirely static and automatic and makes possible the use of short half-life isotopes for extended missions. Since the power output of the thermionic converter is a rapidly varying function of temperature and power input, power flattening is required in any system intended to operate for an appreciable portion of a half-life.

The power flattening mode of heat pipe operation is the basis of the system selected. The isotope selected is Polonium 210, a high energy alpha emitter with good availability, low cost, and a half-life of 138 days. The nuclear radiation from Po^{210} is very low, making it safe for manned missions with little shielding beyond its containing capsule and normal surrounding structure. The system is designed to produce a continuous power output of 500 electrical watts for battery charging service on the lunar surface throughout a full half-life of the isotope, providing a 38-day safety margin beyond the 100-day mission life. Its mass, exclusive of any protective material required for abortive occurrences, is approximately 25.9 kg (57 pounds).

System Performance

Electrical power, watts	500
DC-DC conversion efficiency, %	85
Thermionic efficiency, %	15
Gross electrical power, watts	589

Parasitic heat losses, watts	300
Thermal power to diodes, watts	3920
Gross thermal power, watts	4220
Beginning of life thermal inventory, watts	8388
Overall efficiency, %	11.9
System life, days	138

The total estimated cost, including the delivery of three fully flight qualified systems, is $7,760,000. The period of performance is 3-4 years, determined primarily by the parallel progress made in development of the isotopic fuel forms.

7.7.2.2 Electric Drive System.

7.7.2.2.1 Constraints—In order to increase the reliability aspects of the vehicle, the mechanical transmission and drive train, including clutch and gear shift arrangements, is to be eliminated. Furthermore, again for reasons of reliability, the single drive motor of the first and second order modification vehicles is to be replaced by four individual wheel motors. The 40 volt D.C. power source of the previous vehicle modifications remains available, with energy and current capacities to be determined. Slip steering, i.e., steering via adjustment of the speeds of the individual wheel motors, is not desired. However, an emergency mode of steering, whereby individual wheel motors are stop-start controlled, will be used.

7.7.2.2.2 Drive Selection—It is possible for the operator command to act on one control system for all four wheel drive motors; to act on a separate control system for each wheel motor; or to act on any combination of motor controls between these extremes. The individual motor control is generally the ultimate system, offering complete flexibility of controlling the power flow to and the speed of the individual wheels. The single controller system places the same power and speed profile on to each motor. Thus, if surface or vehicle control parameters demand different wheel speeds, some of the wheels, after automatic speed adjustment to the maximum permissible current setting, would simply slip. For terrestrial use, and in the first and second order vehicle modifications, the wheels are driven in this manner; i.e., the drive train of the "Mule" vehicle does not have differential. Since this is by far the least complex four-motor drive arrangement, it is proposed to test such a system first. Should this system prove inadequate on a simulated lunar surface, the drive can then be extended to the more complex systems.

Motors of both the A.C. and the D.C. type are again feasible. Due to the absence of shift gears, the required speed range of the motors is well in excess of that for the first and second order modification vehicles. The increased speed range complicates the design and forces careful and optimum design as regards efficiency over the range. Experience and calculations appear to favor A.C. motors over the D.C. series units in this respect, easing the motor cooling problem and the burden on the power sources for A.C. motors. The A.C. motor has no commutator and brushes, thereby eliminating potential brush problems of the D.C. motor. A squirrel-cage induction motor can be used, an extremely rugged and reliable motor. Characteristics of the A.C. motors will be used in the further discussions. However, the field appears to be relatively open and D.C. motor systems are recommended for detail studies as possible alternates.

7.7.2.2.3 System Description—The block diagram for the system is shown in Figure 7.7-10. The D.C. power source supplies essentially constant D.C. voltage. The operator demand sets a voltage and frequency level for the output from the inverter. This in turn drives the squirrel-cage induction motors at a speed commensurate with the operator demand for the soil and grade conditions of the traverse. The motor is connected to the wheels through a fixed-ratio gear train.

In order to provide motor starting torque, a multiphase inverter is used, either two-phase or three-phase. The inverter is essentially a switch which permits current flow through the motor successively in both directions. For the power requirements of this application (see section 7.7.2.2.4, System Parameters), power transistors can be used as switches, thus presenting a relatively straight forward system. A simplified schematic of one phase of the inverter is shown in Figure 7.7-11. The transistors T are used as switches, being turned on and off as a function of the desired motor frequency.

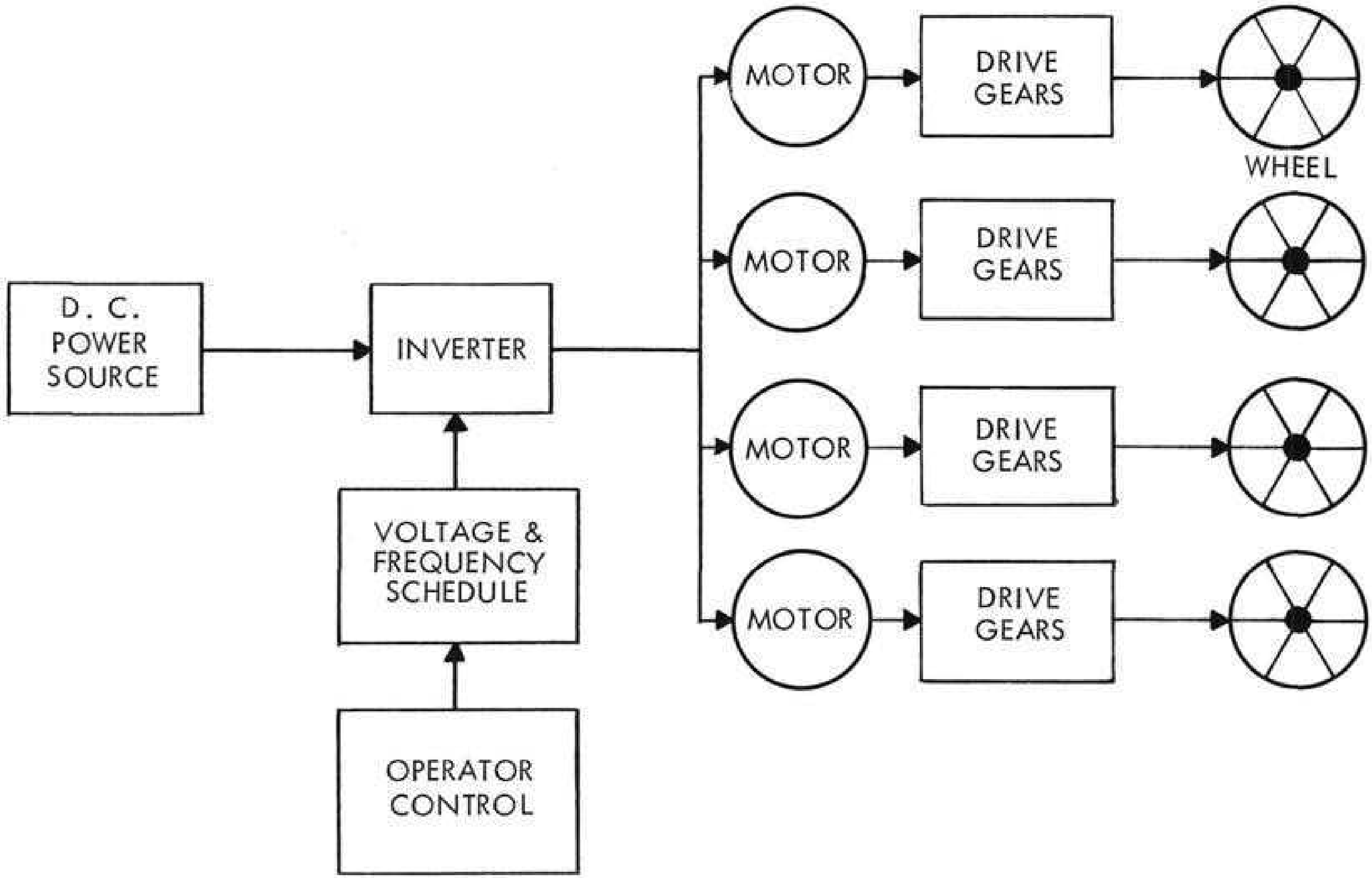


Figure 7.7-10. Electric Drive System - Schematic - Third Order

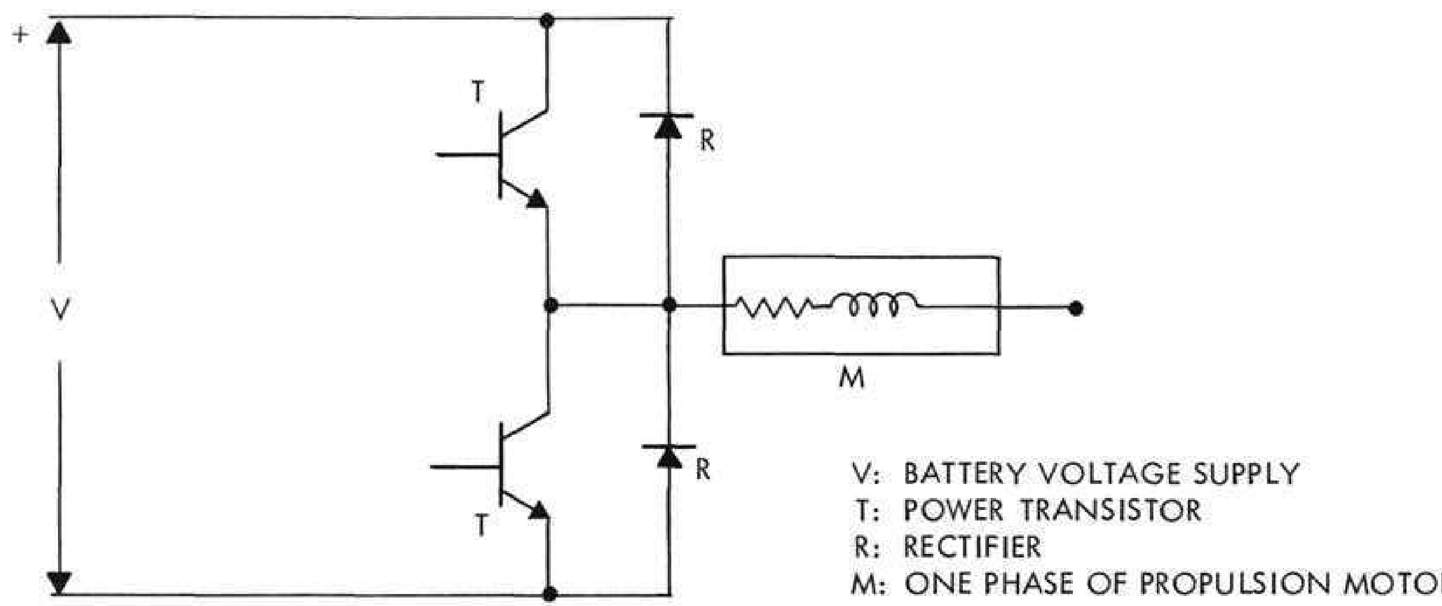


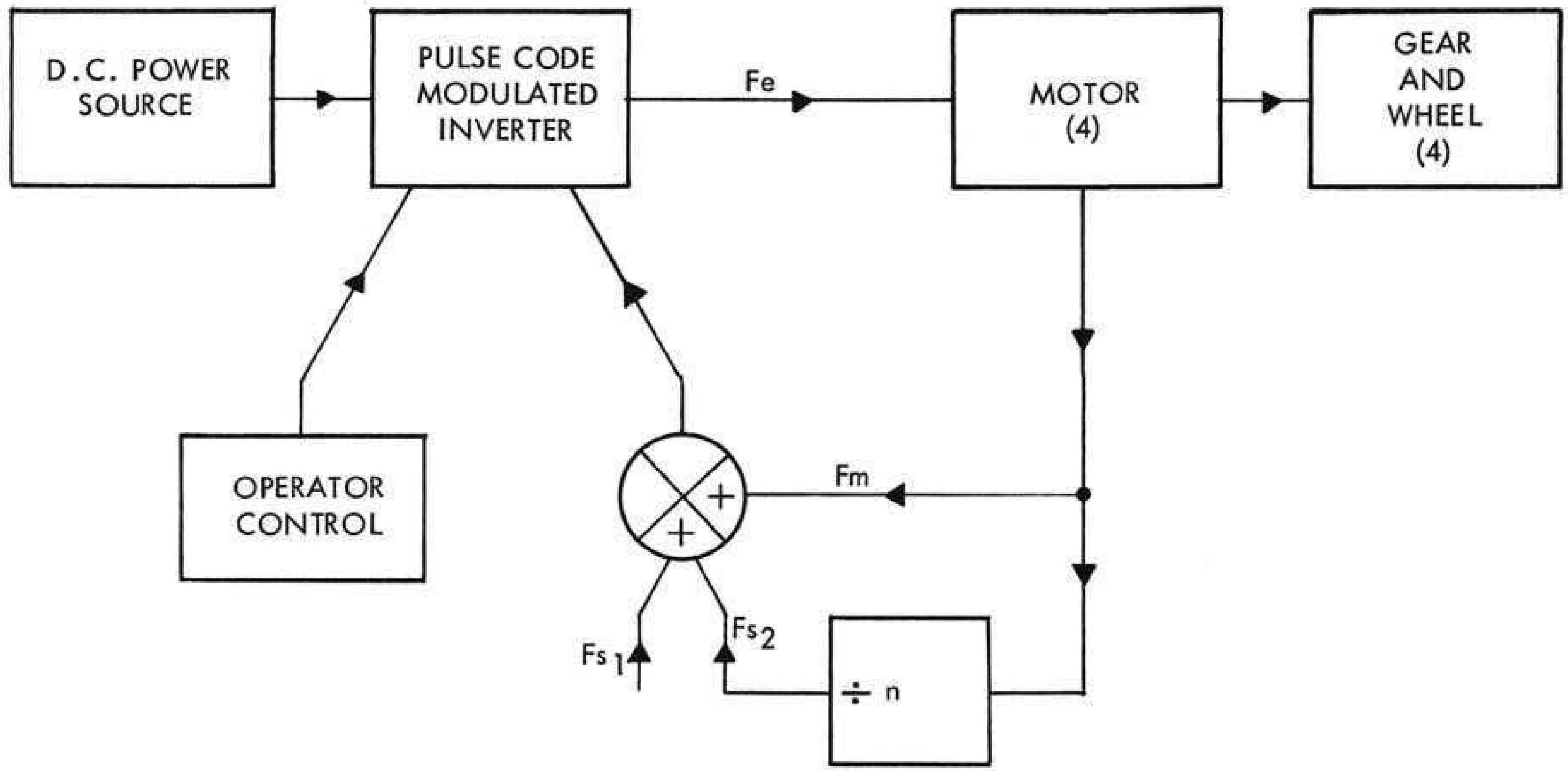
Figure 7.7-11. Electric Drive System - Inverter - Third Order

The rectifiers R permit energy interchange between the battery power source and the inductance of the motor, i.e., they enable flow of reactive current to take place.

A number of control schemes are applicable. The operator command may call for a specific voltage and frequency schedule; or it may call for a specific motor slip schedule and set voltage as a function of motor saturation; or it may call for a specific voltage schedule and set frequency as a function of actual motor speed and a fixed motor slip schedule. The latter scheme is perhaps the easiest to implement and is proposed for use in the lunar application. This system trades good transient response for excellent torque and speed control. This is a good trade-off since rapid vehicle response and acceleration characteristics are relatively unimportant considerations for the lunar application.

A block diagram of the control scheme is shown in Figure 7.7-12. The operator sets the voltage level of the inverter. When starting the vehicle, a fixed frequency control signal, fs_1, is fed to the inverter, so that the motor receives power at the demand voltage and frequency fs_1. As the motor turns, its speed f_m is measured. A frequency signal which is a fixed percentage greater than f_m is fed to the inverter, as a consequence of the divide by "n" circuit. The fixed frequency fs_1 is switched out near the end of the constant torque traction profile and the motor now operates on a constant percentage slip basis. This is a desirable arrangement for the constant power section of the traction profile. Without further protection, it would be possible for the operator to apply excessive voltage for a given motor speed point, and thus cause high saturation and excessive motor currents. In order to prevent this condition, a current limit is incorporated into the system. A signal from a D.C. current shunt is compared with a reference. In case of excessive current, a signal is fed to the voltage control of the pulse code modulated inverter, reducing the voltage applied to the motor.

Both voltage and frequency output of the inverter are set by controlling the bases of transistors T in Figure 7.7-11. The voltage control is accomplished by pulse code modulation (PCM), in a manner previously described for the first order modification vehicle. Depending on detail system design, it may be desirable to obtain sinusoidal



Fs 1: CONSTANT CYCLE SLIP FREQUENCY
Fs 2: CONSTANT PERCENT SLIP FREQUENCY
Fm : MECHANICAL FREQUENCY (SPEED) OF MOTOR
Fe : ELECTRICAL FREQUENCY TO MOTOR
n : DIVISION CONSTANT

Figure 7.7-12. Electric Drive System - Speed Control Flow Diagram - Third Order output from the inverter. This can be accomplished by varying the clock frequency of the PCM system in sinusoidal fashion for a given voltage demand signal. The maximum clock frequency of the PCM control will be set between 5 KC and 10 KC. The combination of lower average load currents and higher switching (clock) frequencies for the transistor, as compared to the first order modification vehicle, is expected to make the use of a smoothing inductor unnecessary. A shift register will be used to control the length of the pulse train to each transistor base, thus setting the output frequency of the inverter. A control is interjected at the shift register which will change the phase rotation of the inverter output voltage, and thus will reverse rotation of the motor. This control will be set by the operator and permit backing the vehicle.

Control systems of the type proposed and described here have been, and are being, used in various experimental armed forces vehicles. In these cases, power levels have been substantially in excess of what is involved in the lunar application. Nevertheless, considerable specific system development is envisioned for this multi-wheel drive, in addition to application to the lunar environment. However, the problems are well within the realm of technical know-how, device limitations, and state-of-the-art, and confidence for a timely and successful development effort is justified.

Dynamic braking will again not be used. Such a system cannot bring the vehicle to a complete stop and must be combined with mechanical brakes. Plugging to come to a complete stop is not considered appropriate because of the attendant high heat losses inside the motor, and because of the relatively uncontrolled decelerating torques. Braking requirements of the vehicle are small in any case, and the complications of dynamic braking at high speeds combined with mechanical braking at low speeds, do not appear warranted. The drive motors will thus not be used for braking purposes.

7.7.2.2.4 System Parameters—The wheels proposed for the third order modification vehicle reduce torque and power requirements considerably as compared to the first and second order vehicles. Required power and torque output per wheel drive motor are shown plotted versus motor speed in Figure 7.7-13, using an 80% transmission efficiency between the motor and the power applied to the ground. The circled points

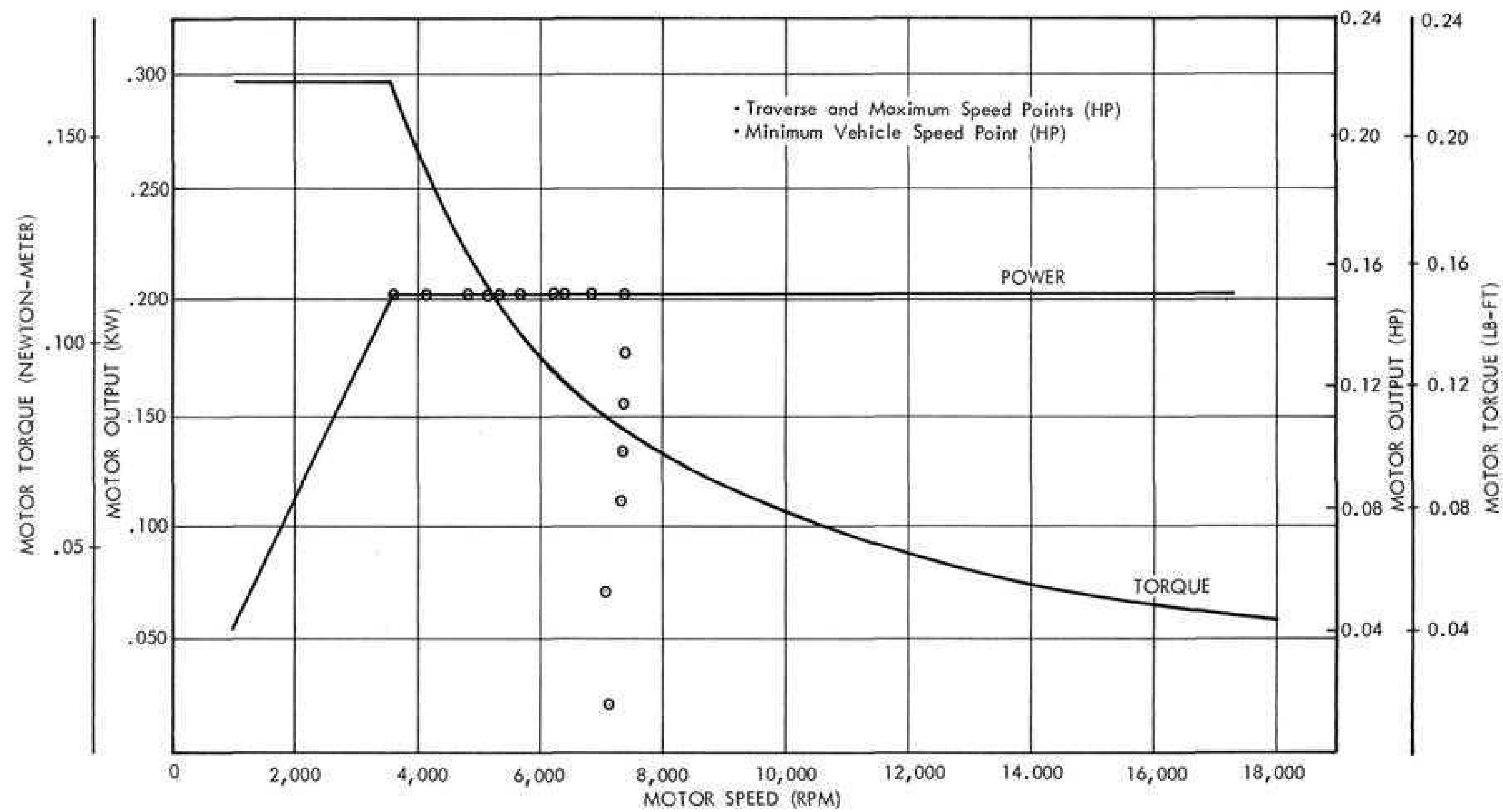


Figure 7.7-13. Electric Drive System - Traction Requirements Per Wheel - Third Order

show calculated horsepower-speed requirements for the specified lunar traverse and for the the maximum speed conditions of the vehicle. The crossed point shows the worst condition minimum vehicle speed point at 1.09 Km/hr (1 foot per second or 0.68 mile per hour). The motor chosen for the application is shown by the horsepower and torque curves of Figure 7.7-13. Comparison with Figure 7.5-19 shows substantially lower vehicle power requirements; and a substantially wider constant-horsepower region due to the absence of shift gears. The motor speed range reflected in Figure 7.7-13 was selected on the basis of a reasonably low number of poles in view of the small motor rating; a reasonable minimum speed in order to retain a manageable size and weight; and a reasonable ratio gear between the motor and the wheel. The gear is described elsewhere in this report; it has a fixed ratio of 200 to 1. Maximum vehicle speed on the hard lunar soil, zero grade is 10.72 km/hr (6.66 mph) at which speed the vehicle has reached the design power capability of the system.

The motor performance is shown on Figure 7.7-14. Voltage is shown on a percentage basis. Its exact value is unimportant from the point of view of this study, the available d.c. voltage from the power supply setting a reasonable value. In the constant torque region the voltage varies directly with speed, setting a constant flux pattern in the motor. In the constant power region, the voltage first slowly increases with speed, and then remains constant, gradually decreasing the saturation level of the motor. The current decreases rapidly with speed in the lower speed range, later increasing again due to lowered efficiency and power factor characteristics of the motor. The motor data shown on Figure 7.7-14 hold for the power and torque lines of Figure 7.7-13.

As for the first and second order modification vehicles, the required starting torque is based on an initial acceleration of .23 m/sec^2 (0.75 ft/sec^2) .8 kmph/sec (0.5 mph/sec) under the worst soil and terrain conditions. The necessary motor starting torque of 0.45 Newton-meter (0.33 lb-ft) will require an in-rush current only some 10% above the maximum current during the specified lunar traverse (hard soil, 15 degree grade). The starting current, the operational current, and the current limit setting, can thus be easily coordinated.

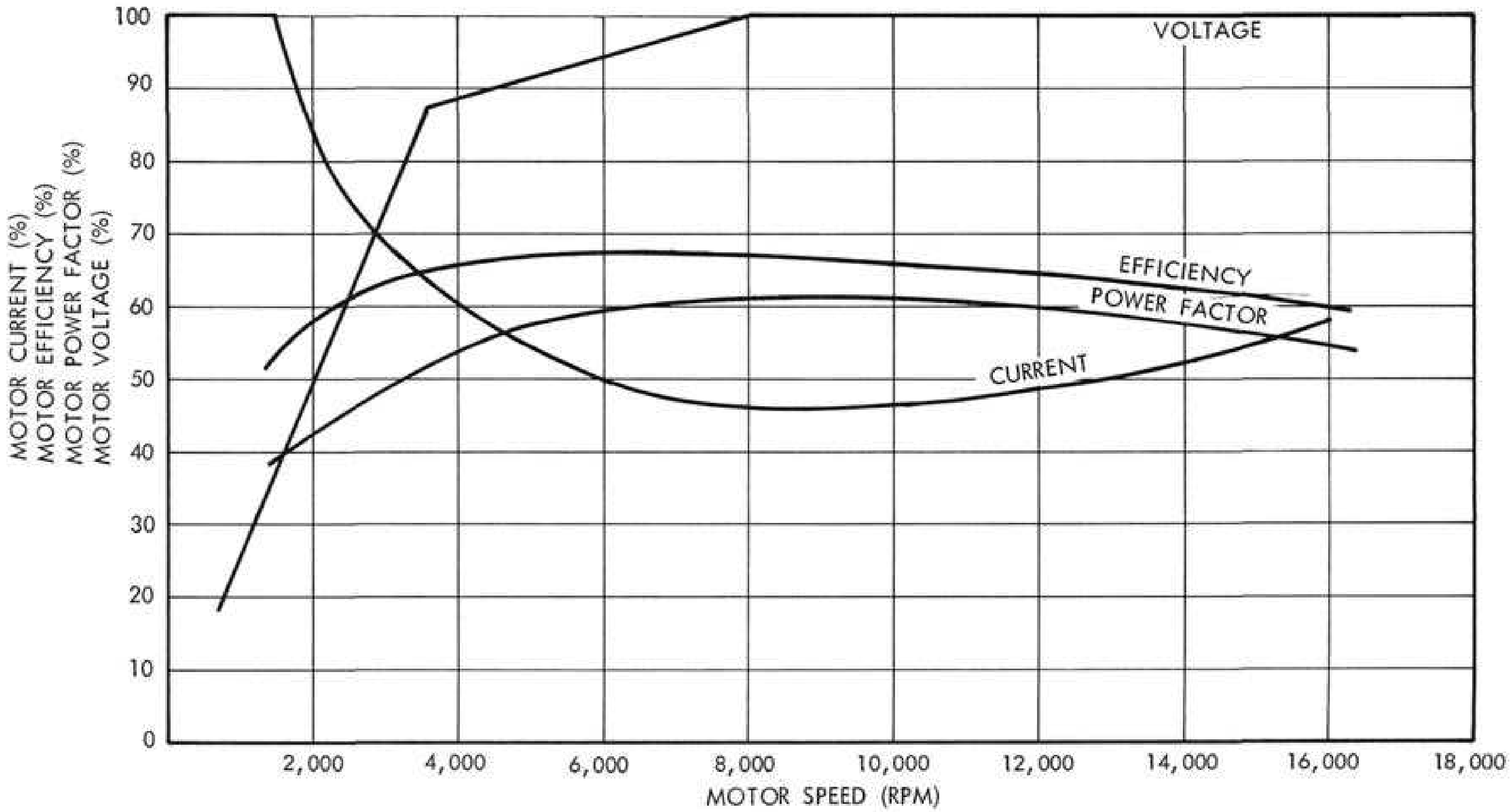


Figure 7.7-14. Electric Drive System - Motor Performance - Third Order

The power profile for the traverse is shown in Figure 7.7-6. The motor performance data of Figure 7.7-14 were used in the calculation of the profile, modified for part loading where necessary. Efficiency values for the controls and converter were taken from Figure 7.7-15. The energy requirements per traverse were calculated as 6 kw hrs. This value includes actual propulsion energy during travel; propulsion control energy only during stops; energy for fourteen starts; 20% reserve for obstacle negotiation capability; 10% for contingency and the energy for coolant circulation, but it does not include energy for auxiliary functions. The maximum sustained average current during the specified traverse is 25 amperes from the 40 volt power source. The current limit setting will be near 30 amperes. Actual calculated driving time of the traverse is 5.68 hours.

7.7.2.2.5 Power Source and Propulsion System Interface—The power source is a battery array (or in an alternate concept is supplemented by an isotope charger floated across it). The system is at 40 volts D.C. Proper matching of the batteries, and isotope charger if used, will be handled in the power source system.

A contactor will be connected between the power source and the inverter and used to apply power to the motors. The system will follow along the lines shown for the first order modification vehicle. The gear shift switch will be omitted in this modification.

7.7.2.2.6 Components—The major components of the proposed propulsion system for the third order modified vehicle will be described in this section:

Propulsion Motor

This is a squirrel cage induction motor, a rugged and dependable machine. The stator has standard multiphase coils wound in slots of the lamination stack. The rotor slots are filled with metal and shorted to one another either by a casting process or by inserting bars and connecting them with end rings.

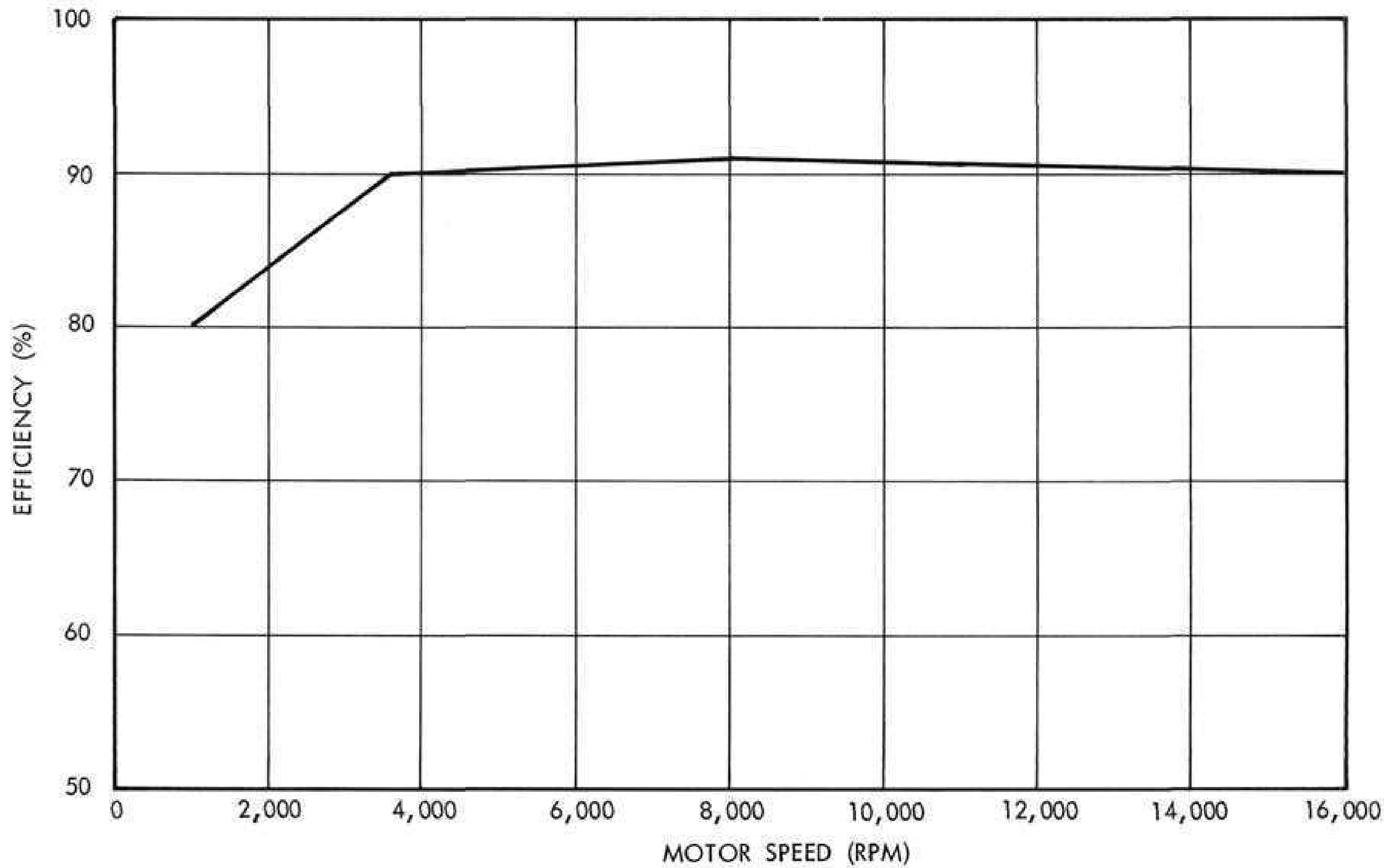


Figure 7.7-15. Electric Drive System - Controls and Converter Efficiency - Third Order

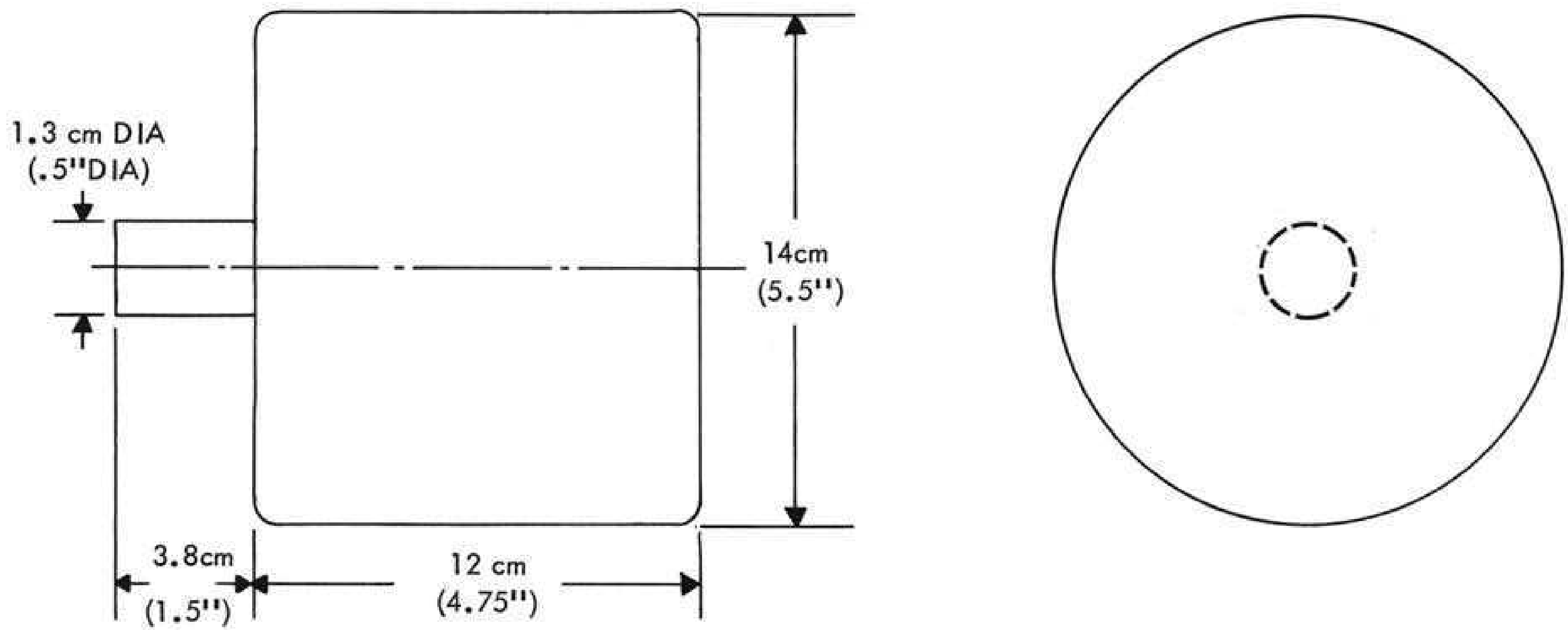
Size and mass of the motor are again based on previous experience. For the starting and for the load requirements shown in Figure 7.7-13 the wheel motor will have a mass of 3.68 kg (8-1/8 lbs). An outline drawing is shown in Figure 7.7-16. Motor losses in the order of 70 watts must be dissipated. Class "H" insulated coils will be used in the stator. Cooling coils will be wound around the stator core, and cooling fluid circulated to carry away the heat. Direct cooling of the rotor will probably not be necessary due to heat conduction through the rotor, radiation to the stator, reasonable rotor losses and ability of the rotor to operate at higher temperatures than the stator. It is not proposed to provide an atmosphere inside the motor. No particular problem would be introduced, however, by a later design decision to provide an atmosphere. As a matter of fact, it would be entirely feasible to seal the motor and gear in one unit, thus removing any necessity for a rotating seal. A speed pickup will be mounted in or at the motor to measure the mechanical feedback frequency f_m.

Drive Controls and Inverter

Packaging of the unit will be along the lines described for the voltage control of the first-order vehicle modification. The mass is estimated at 1.4 kg (3 lbs) for the logic and 1.4 kg (3 lbs) for the power transistors and rectifiers. Use of integrated circuits would cut the mass of the logic to 1 lb. Overall dimensions of the package will be 15 by 20 cm (6 x 6 x 8 inches). Cooling will again be accomplished by utilizing a tube carrying cooling fluid as a heat sink. Heat losses from the unit will be in order of 80 watts, with the silicon semiconductor devices to be held at or below a total temperature of 423° K (302° F, 150° C).

Operator Controls

The propulsion controls consist of one switch and a dual potentiometer. The switch is the start-stop switch of the vehicle. It is used to close the contactor applying power to the motors. The two potentiometers are used to set the clock frequency of the voltage control unit of the inverter, and are handled in identical fashion to the potentiometers of the first-order vehicle modification. The switch has a mass of .11 kg (4 oz), the dual potentimeter .25 kg (0.55 lb).



NOTE: 1) FOUR MOTORS REQUIRED
2) MOTOR TO BE CONFIGURED INTO DRIVE ASSEMBLY

Figure 7.7-16. Electric Drive System - Basic Wheel Motor Size - Third Order

Contactor

The contactor is a double pole single throw unit, capable of interrupting the maximum current of 30 amperes at 40 volts from the power source. The comments on contactors made in the discussion of the first-order modification vehicle apply fully. The mass and dimensions of the contactor will be approximately the same as for the first order modification vehicle; i.e., 0.60 kg (1-1/3 lb) and 8.3 cm (3-1/4") by 10.8 cm (4-1/4") by 7.6 cm (3").

7.7.2.3 Mechanical Drive System Components. Reference is made to Figure 7.7-1 which is the third-order wheel and suspension unit integral with drive components.

7.7.2.3.1 Brakes—Caliper type disc brakes are proposed on all wheels. The brakes will be electrically applied, the force will be proportional to voltage. The disc will be mounted on an extension of the motor shaft and insulated thermally from the motor shaft. The caliper housing will be mounted on the motor case. The brakes on the rear wheel will also be mechanically actuated for parking purposes.

7.7.2.3.2 Steering Mechanism—Two wheel Ackerman steering will be used in the third-order modified vehicle. The steering linkage will be operated and positioned by electromechanical devices rather than the standard mechanical steering gear.

The introduction of an electrically controlled steering actuation eliminates all mechanical linkages in the control of the third-order modified vehicle. This allows the design of a detachable control console and thus remote control of the vehicle by the operator. Remote control of the modified vehicle would be highly desirable in hazardous situations.

7.7.2.3.3 Harmonic Drive—A harmonic drive system is proposed for transmittal of the power from the wheel drive motor to the forced semi-elliptic wheel, on the third order vehicle. The harmonic drive would perform two major functions: the first is to provide a 200 to 1 speed reduction between the drive motor and the wheel, and the second is to act as a drop gear assembly. The drop gear is advisable as it will provide both high body ground clearance and allow retention of the small diameter wheels. The harmonic drive operates by utilizing a cam to deflect a thin circular ring gear into an elliptical shape. As the cam is rotated, the major axis of the elliptical gear is rotated. External gear teeth on this deformed gear engage internal teeth of an outside circular and concentric gear; since the outside gear has more teeth than the deformed inside gear, the outside gear moves through an angle, per cam revolution, proportional to the difference in the number of teeth between the two gears.

The advantages of using this gear system in lieu of more conventional transmissions are:

- Provision of a positive drive through a hermetic seal.
- Weight and space saving of approximately 50%.
- Use of a developed product with high reliability.
- Use of a high efficiency system with maximum overload capability.

The harmonic transmission would be designed as an integral assembly with the drive motor and brake assembly forming identical independent drives for each of the four wheels.

7.7.3 Crew Station

In the third-order modification, the crew seat proposed is identical to that for the first and second-order modifications, as well as the foot rest cradle, and the step located on the left side. There would, however, be no foot operated controls, or motor tachometers on the instrument read-out. The steering wheel would be replaced by a toggle switch steering control, and the gear shift lever would be eliminated. It would be replaced by "forward" and "reverse" pushbuttons on the console.

The brake pedal would be replaced by electric toggle switch braking operated as follows: On the console a multifunction toggle would control differentially the A.C. motor drive frequency, thereby controlling the vehicle speed as a function of forward displacement of the toggle. Movement of the toggle left or right would actuate the electric steering motor, to displace the front wheels to left or right of center, such that the angular displacement would be a function of the time the toggle was deflected in the turning direction, and the angle would remain fixed until the toggle was deflected in the opposite sense (left or right). Deflection of the toggle to the rear would actuate the electric actuated brakes. This toggle control as well as the "forward and reverse" pushbuttons would be located on a separable portion of the control console, which would be electrically attached to the vehicle by means of an umbilical cable. This configuration is shown in Figure 7.7-17. The configuration would provide the ability to control the vehicle while walking abreast of the vehicle, which could be advantageous in arduous obstacle crossing.

The separation of the two parts of the console would change (by an interlock device) the toggle neutral position to a brake actuating position, such that in the event the operator dropped or lost hold of the console, the vehicle would be stopped, thereby preventing runaway.

In all other respects, the instrument read-outs, parking (hand brake) and cargo platform area would be the same as proposed for the first-order modification.

7.7.4 Thermal Control

7.7.4.1 Radiator Design Criteria. The same criteria used in the thermal control design for the first-order modifications, Section 7.5.3, were used in the design estimate for the third-order modifications.

The peak loads and temperature restraints are given in Table 7.7.4-1.

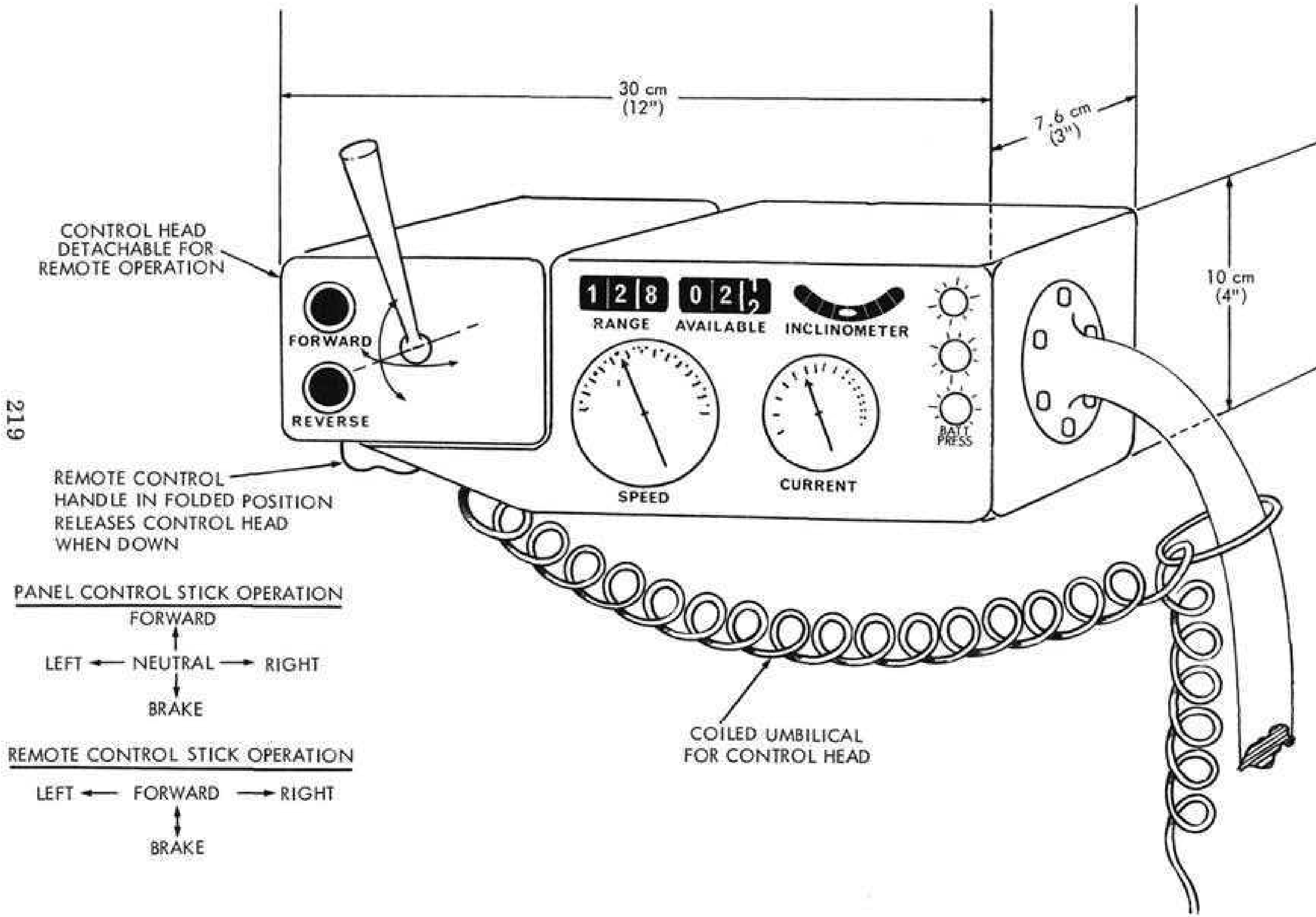


Figure 7.7-17. Instruments and Control Panel - Third Order

Table 7.7.4-1

PEAK HEAT LOAD SUMMARY MODIFIED M-274 CARGO CARRIER HEAT REJECTION SYSTEM, THIRD-ORDER MODIFICATION, WITH ALL BATTERY POWER OPTION.

SYSTEM	UNIT	No. of Units	Peak Heat Load Each Watts	Total Peak Heat Load Watts	Min. Temp. °K	Max. Temp. °K
Energy	Battery	6	15	90	289 (60° F)	322 (120° F)
Energy Conversion	Motor Control	1	80	80	-	373 (212° F)
Energy Conversion	Electric Motor	4	80	320	-	473 (392° F)
Drive Train	Assembly	4	22.5	90	-	423 (302° F)
Heat Rejection	Coolant Pump	1	25	25	250 (-9.5° F)	370 (206.5° F)
		TOTAL		605		

The total rate of radiation required is 605 watts, of which 90 watts must be taken from the system under 322° K. Allowing the 10° C temperature differential from the coolant to heat source, the maximum temperature of the coolant after absorbing the fraction of heat load corresponding to the 90 watt rate from the batteries (i.e. 90/605 = 0.148) is 312° K. Hence the radiator outlet temperature cannot exceed [312 - (0.165) (55.5)] = (312 - 8.25) = 303.7° K (87° F).

From the above criteria, and Figure 7.5-43, the radiator areas required for selected values of a/e are estimated as shown in Table 7.7.4-2.

Table 7.7.4-2

HORIZONTAL RADIATOR AREA, ΔT COOLANT OF 55.5°C (100° F), FOR HEAT LOAD OF 605 WATTS, INLET TEMPERATURE OF 359.2°K (187°F) AS A FUNCTION OF α/ϵ. e = 0.90, η = 0.80

a/e	Tes	Q Watts/M^2	Area Required M^2	Area Required Ft^2
0.089*	222 (-60° F)	378	1.60	17.2
0.133	244 (-20° F)	332	1.82	19.6
0.183	266 (18° F)	252	2.40	25.5
0.219	278 (41° F)	184	3.29	35.4

*Projected state-of-the-art.

Based on Table 7.7.4-2, a horizontal radiator of about 1.85 square meters area (19.9 ft^2) with a surface a/e ratio of about 0.135 is selected for the design estimate. The radiator location will be similar to that of the first-order modification, and the coolant pump operation will likewise be required during battery recharging.

The radiator mass will then be (1.85) (13) = 24 kg (53 lbs) and the coolant pump 2.3 kg giving a total radiator system mass, with coolant, of 26.3 kg or 58 lbs.

The schematic representation of the third-order thermal control system is shown in Figure 7.7-18.

7.8 REFERENCES

7-1 Burriss, W. L., Johnson, A. L., "Thermal Management of Manned Lunar Capsules", ASME Aviation and Space Conference, Los Angeles, California, 3-7 March 1963.

7-2 Wright-Patterson Air Force Base, "Investigation of a 3-KW Sterling Cycle Solar Power System", Volume I - Volume IX, Report No. WADD-TR-61-122, April-May 1962.

7-3 Pavlics, F., "Locomotion Energy Requirements for Lunar Surface Vehicles", Paper 660149, SAE Automotive Engineering Congress, 10-14 January 1966.

7-4 AiResearch Manufacturing Company, "Human Factors and Environmental Control - Life Support Systems", Report SS-3243, LESA Study Series, Vols 1-6.

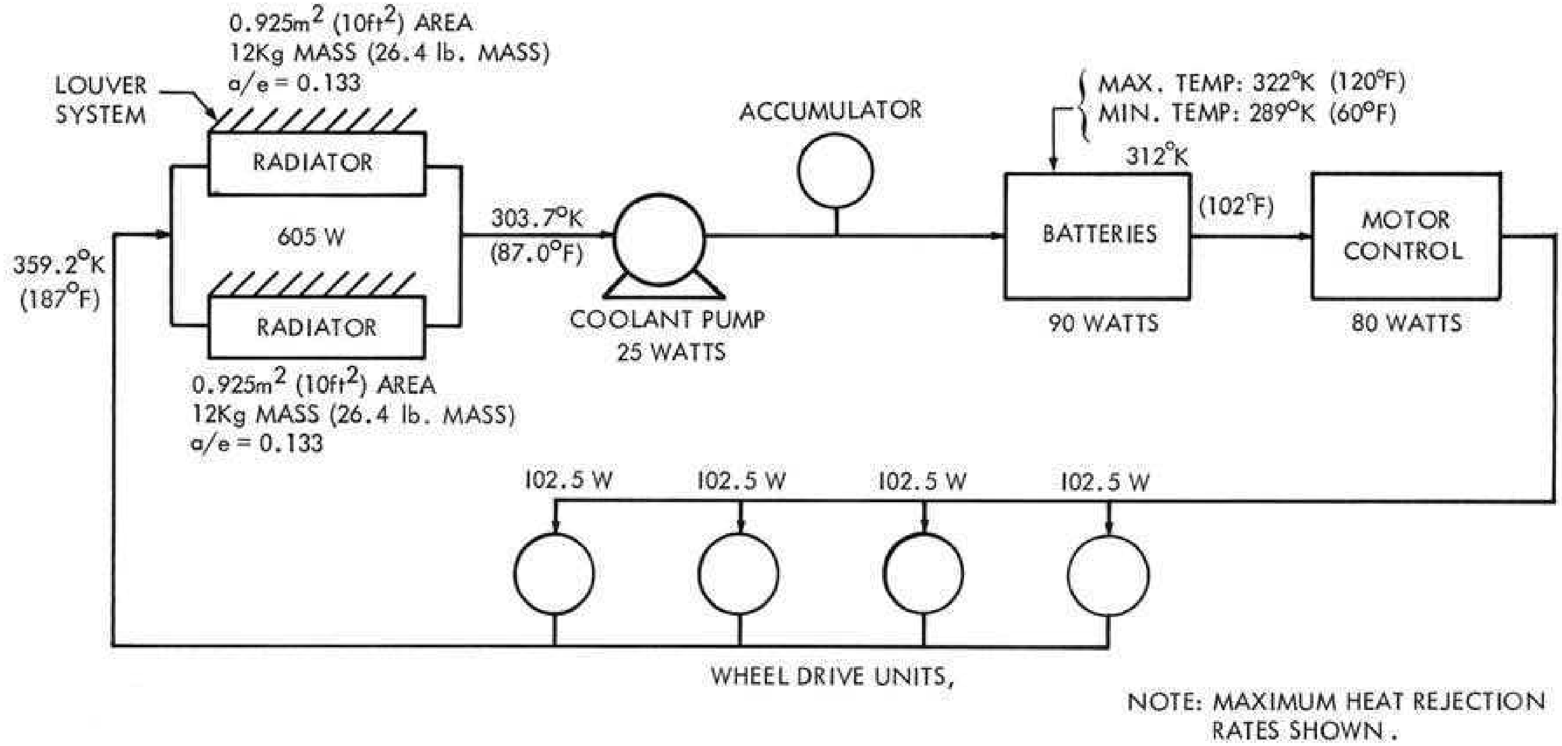


Figure 7.7-18. Cooant Flow Diagram - Third Order

8.0 PAYLOAD PACKAGING AND UNLOADING

The volumes in the LEM/S Adapter Areas, see Figure 1-1, Vol II - Part 2, APPENDIX, Section 1.0, GUIDELINES AND CONSTRAINTS, which are available for supplementary hardware payloads other than the LEM/S have been defined in the criteria.

Specifically used was Drawing, Code 26512, "Left Hand Payload Packaging Envelope", Grumman Aircraft Engineering Corporation, scale 1" = 10.0". (NOTE: The payload packaging envelope does not represent a physical or material enclosure or envelope, as such. It merely designates the limits of the volume available within the LEM/S Adapter Area volume for a supplementary payload).

8.1 Model Study

A three-dimensional, 1:10 scale, plastic working model of the left-hand payload envelope was prepared for use. It was used to adjudge the adaptability of similarly scaled models of the modified vehicle to positioning within the available volume. Paper models of the intended modified vehicle configuration were prepared to adjudge best fit in the envelope consistent with rigging restraints and unloading system concepts.

After a packaging and an unloading system concepts were selected, a demonstration model of the first order modified vehicle placed in a plexiglass model of the left-hand payload envelope was prepared.

Figure 8-1 is a photograph of the demonstration model. The view is an aft-looking view of the vehicle placed in the envelope in the packaged position. The unloading and positioning equipment is not shown in the model. Figure 8-2 is a photograph of the demonstration model with an outboard looking view of the model.

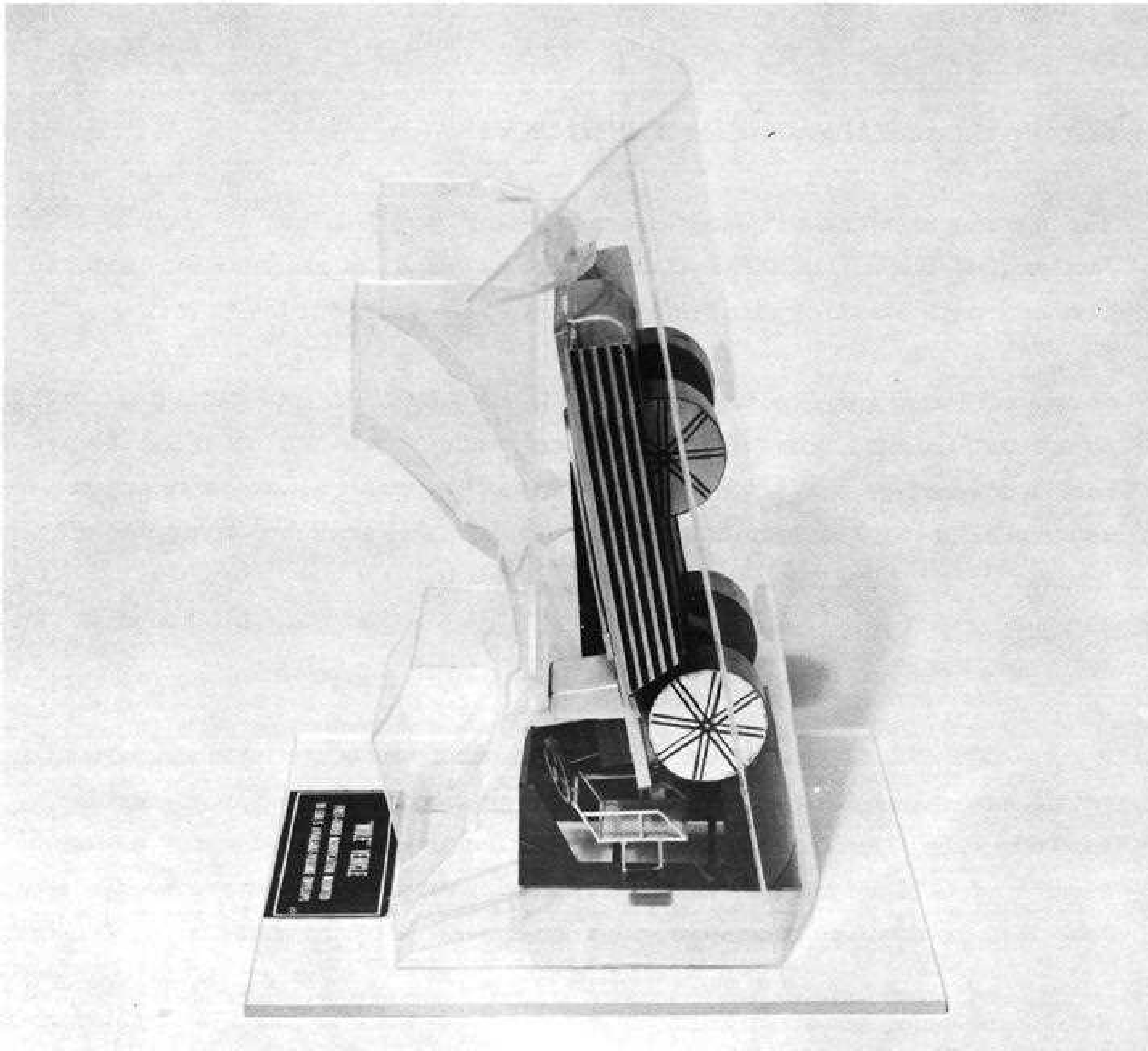


Figure 8-1. Model Vehicle in Payload Packaging Envelope - Looking Aft

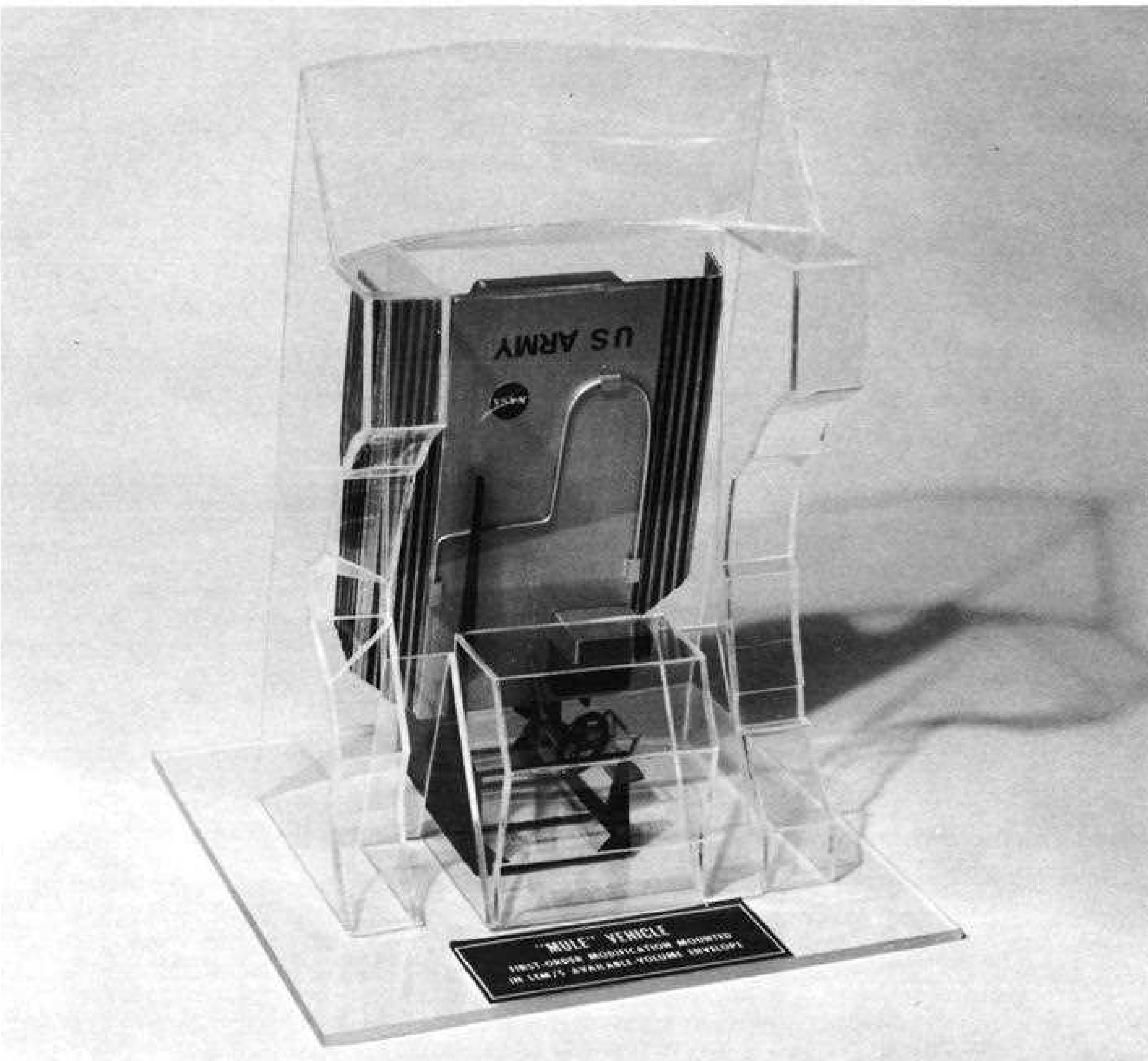


Figure 8-2. Model Vehicle in Payload Packaging Envelope - Looking Outboard

It will be noted that the horizontal radiators along the sides of the vehicle cargo platform have been hinged in the center, horizontally, to provide a more compact payload package. The hinging of the radiators will not be a requirement for the third order vehicle as radiator area requirements are one-half that of the first order vehicle. The foot control basket, forward of the vehicle platform is hinged and has been turned into an upright position along the platform.

The modified vehicle retains the compactness of the basic Mule vehicle. The vehicle is positioned as an integral, operationally configured unit within the payload. Only three simple auxiliary operator tasks will have to be performed:

- The roll-bars raised and locked in place.
- The foot basket dropped into place.
- The radiator sections raised level and locked in place.

8.2 Payload Packaging and Unloading Concept

Figure 8-3 is an elevation view of the LEM/S on the descent stage. The modified vehicle is shown positioned on the LEM/S descent stage. The vehicle is shown within the packaging volume envelope. The envelope was shown to allow correlation with the demonstration model view, Figure 8-1. The outside line of the packaging volume envelope corresponds to the 5-inch clearance line inside the Adapter Shroud which is blown free prior to the LEM/S-CM docking maneuver.

8.2.1 Storage Period Thermal Control

The vehicle will be required to remain upon the LEM/S descent stage after lunar landing for as long as three lunar day-night cycles (90 earth days). During this storage period the packaged vehicle must be maintained in a thermal and pressure environment suitable to allow vehicle operation at the termination of the storage period.

It is proposed that the vehicle be cocooned in an envelope of super-insulation. Within this envelope, plastic packages of water and a small wattage heat source (approx.

5 watt electric light bulb operated off the LEM/S RTG storage-phase power unit) would be placed and secured. Thermal control would be maintained by thermostatically controlling the electric light bulb on-off and cycling the cocoon temperatures to approx. 5 degrees above and below the freezing phase point of water.

This cocoon of super-insulation would be removed by the astronaut prior to vehicle off-loading.

8.2.2 Packaging and Unloading Concept

As shown in Figure 8-3, the vehicle will be placed in a packaging and off-loading hardware configuration which consists of light, metal, nested wheel troughs, a small motor driven (or hand cranked) winch with a cable-sheave system, and positioning restraints. The vehicle will be fastened into the wheel troughs with axle extension tie-downs.

The system off-loading operation would be conducted as follows:

- Packaging restraints and tie-downs would be removed.
- The vehicle center of mass will be outboard of hinge no. 1 in the outboard wheel trough connection to the lander. Assuming the lander is level, the gravity force of the vehicle will allow the vehicle and wheel troughs to move around and down about the fulcrum of hinge point 1 and hinge 2 in succession.
- The vehicle would move against the restraint of the off-loading winch and connected cable system.
- The outboard wheel trough would have a limit-stop about hinge point 2. The vehicle and the inboard wheel trough would slide, in the outboard trough down to the lunar surface.
- If the lander were not level (left side of the lander deck in Figure 8-3 down relative to right side), the winch and cable system would be so rigged (mine-hoist) as to allow the vehicle to be winched over the fulcrum of hinge point 1.

This system concept would provide some advantages equal to incorporation of an azimuth orientation device in the lander deck.

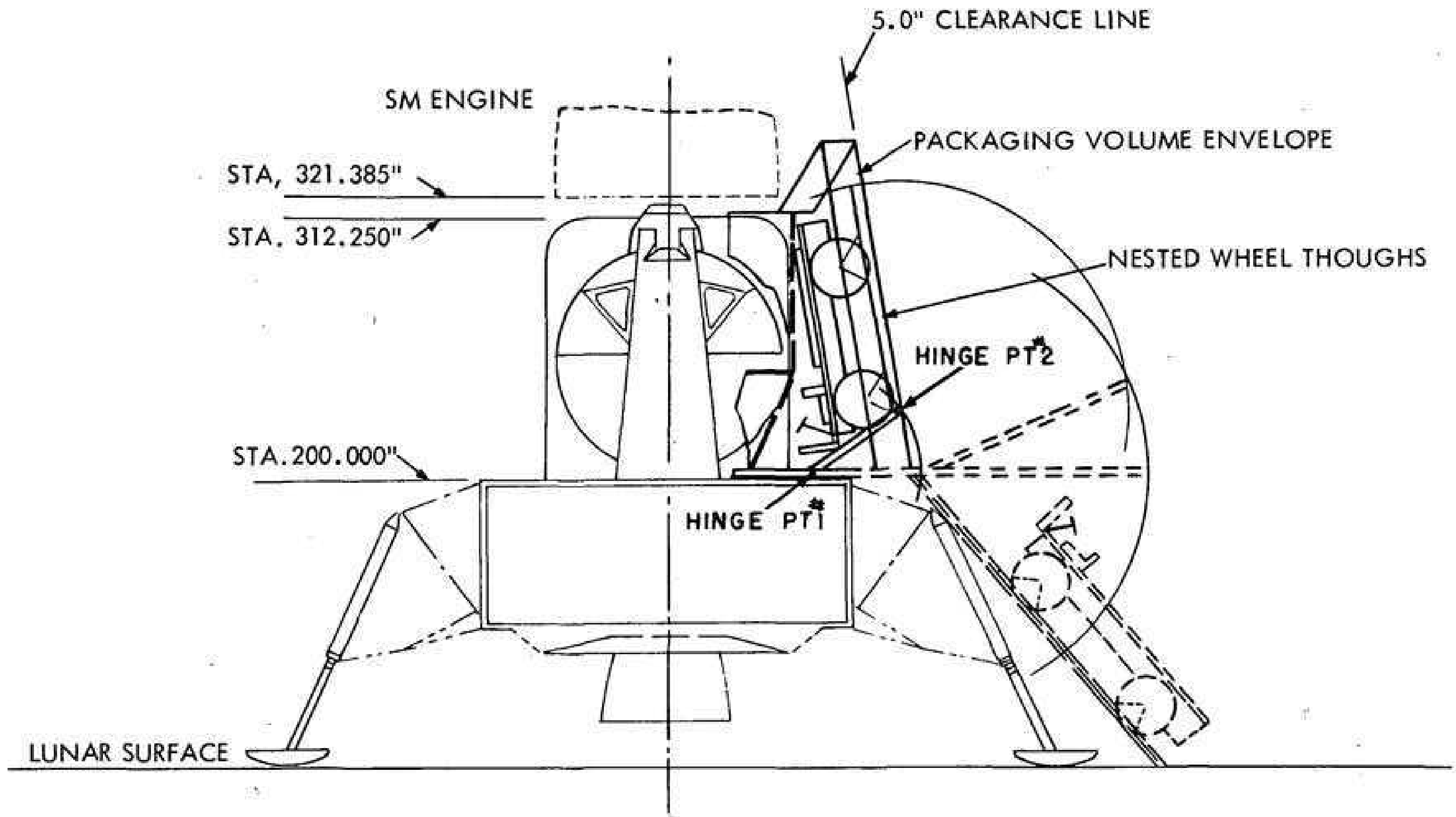


Figure 8-3. Vehicle Offloading and Packaging Configuration - LEM/S

9.0 VEHICLE MASS SUMMARY

The study guidelines and constraints contained the LEM/S supplementary hardware payload allowance. 2500 lbm was the nominal used for mission planning purposes. An allowance of 1500 lbm is included in this 2500 lbm total for a mobility device. It was stated further that the 1500 lbm allowance for the vehicle (mobility device) is the total mass allowance for:

- The vehicle (curb-side)
- The total fuel (power) required for a 240 km (150 mile) mission. If fuel is expended by the LEM/S power generation system to recharge the vehicle prime power source, the fuel mass will be charged to the vehicle mass allowance.
- All LEM/S equipment mass required primarily for vehicle power recharging will be charged against the vehicle mass allowance.

The vehicle mass allowance will not be charged for the mass of the LEM/S equipment required to off load the vehicle.

The unit of measurement will be the mass unit of the items rather than a weight unit. This quantitative scalar expression of inertia frees the tabulation from the need of consideration of a gravitational constant.

Table 9-1 is a tabulation of mass summary for the first and second orders of vehicle modification. This summary indicates that these orders of vehicle modification result in an overage of approximately 59 lbm. Although this indicates some overage above the allowance of 1500 lbm, the 59 lbm could easily be extracted from a redesign of the vehicle chassis and mechanical drive train. These items are excessively overstrength for lunar application. The use of the step-stress method in demonstrating component reliability will be useful in reducing this unnecessary overstrength.

Table 9-2 is a tabulation of the mass summary for the third-order vehicle modification. This order of modification results in a total vehicle mass charge which is 270 lbm under the vehicle allowance of 1500 lbm. Primarily, this mass savings is the result of elimination of the internal combustion engine, the vehicle's mechanical device train and the lower power requirement for the third-order vehicle (to negotiate the specified terrain traverse model). The lower power requirement reduces the amount of fuel necessary for battery recharge and the battery mass (3 battery packs versus 4 battery packs for first and second order). The third-order wheel and wheel gear is estimated at 43 lbm as against 25 lbm for the first-order vehicle wheel.

TABLE 9-1

VEHICLE MASS SUMMARY
(In Pounds of Mass)

First and Second Order Modifications

Original Vehicle	Curb Mass		955	
Removals from Original Vehicle				
Engine	170			
Wheels & Tires	124			
Total Removals		294		
Additions to Original Vehicle				
Power System				
Battery Packs 4 @ 98.4	393.6			
Drive Motor	60.0			
Controls	25.0			
Power System Total		478.6		
Mobility System				
Wheels		100.0		
Crew Station				
Seat Differential	10.0			
Radiators	90.0			
Roll-Bars	25.0			
Instruments & Panel	25.0			
Crew Station Total		150.0		
Total Additions		728.6		
NET (Additions-Removals) 728.6 - 294			435	
VEHICLE TOTAL				1,390.0
Shelter Penalties				
Battery Charger	20.0			
Fuel (For Recharging)				
Fuel Cell Consumption	134.0			
Umbilical	15.0			
Total Shelter Penalty				169.0
Total Vehicle Mass Charge				1,559.0
Vehicle Mass Allowance				1,500.0
Net Over Allowance				59.0

TABLE 9-2

VEHICLE MASS SUMMARY
(In Pounds of Mass)

Third Order Modification

Original Vehicle		Curb Mass		955
Removals from Original Vehicle				
Engine		170		
Drive Line		400		
Wheels & Tires		124		
Total Removals			694	
Additions to Original Vehicle				
Power System				
Battery Packs 3 @ 98.4		295.2		
Wheel Drive Units				
Motor	27			
Drive, Housing & Brake	27			
Suspension	10			
Unit Total	64			
4 @ 64		256.0		
Controls		25.0		
Power System Total			576.2	
Mobility System				
Wheel Units				
Wheel	38.4			
Gears	4.5			
Unit Total	42.9			
4 @ 42.9			171.6	
Crew Station				
Seat Differential	10.0			
Roll-Bars	25.0			
Radiators	45.0			
Instruments & Panel	25.0			
Crew Station Total			105.0	

TABLE 9-2 (Cont'd)

VEHICLE MASS SUMMARY
(In Pounds of Mass)

Third Order Modification

Total Additions		852.8		
NET (Additions-Removals) 852.8 - 694			159	
VEHICLE TOTAL				1,114.0
Shelter Penalties				
Battery Charger	20.0			
Fuel (For Recharging)				
Fuel Cell Consumption	81.0			
Umbilical	15.0			
Total Shelter Penalty				116.0
Total Vehicle Mass Charge				1,230.0
Vehicle Mass Allowance				1,500.0
Net Under Allowance				270.0

10.0 RELIABILITY PROGRAM PLANNING

10.1 Reliability Goals and Test Demonstration Methods

The problems associated with determining the vehicle reliability concern two areas. The first concerns the estimate of the design reliability of the modified M-274 vehicle proposed for use on the lunar surface, and the other concerns the justification of the test program designed to demonstrate the estimated reliability of the actual modified vehicle.

In the first problem, having the assigned reliability goal, the subsystem and component reliability goals may be assigned, bearing in mind the component mission criticality (Reference 2). In components of existing systems, established reliabilities may be compared with the mission profile and reliability goals, and adjustments made optimizing the reliability factors such that development cost is minimized. In any event, the reliability goals are indicators with respect to the extent of development and testing required.

The desired reliability is 0.90 with 80% confidence. In design of the proof testing of the final design modification, the reliability criteria will be used to justify the test program. In time based testing, for example, at 4.5 km/hr, the cumulative time for the basic modified M-274 Cargo Carrier to traverse 240 km will be 53 hours (See Table 10-1), and for the third order modified vehicle will be 55 hours. Hence, the operational lifetime may be considered to be 55 hours. Assuming random failures,

$$R = 0.9 = e^{-\tau/\theta}$$

where, R = reliability, e = naperian log base, τ = operational lifetime, and θ = mean time before failure.

Table 10-1

AVERAGE SPEED COMPUTATION FROM ELMS-SOIL AND SURFACE CHARACTERISTICS

ELMS MODEL					FIRST & SECOND ORDER				THIRD ORDER			
Soil Character			Slope	Fraction	Speed		Weighted Speed		Speed		Weighted Speed	
K_ϕ (1)	η (2)	ϕ (3)	Degrees (4)	Of Traverse (5)	km/m (6)	MPH (7)	km/hr (8)	MPH (9)	km/hr (10)	MPH (11)	km/hr (12)	MPH (13)
0.5	0.5	32°	0	0.110	4.83	3.00	0.53	0.33	4.83	3.00	0.53	0.33
0.5	0.5	32°	-1	0.130	4.83	3.00	0.63	0.39	4.83	3.00	0.63	0.39
0.5	0.5	32°	1	0.095	4.51	2.80	0.428	0.266	4.83	2.72	0.415	0.258
0.5	0.5	32°	-2	0.128	4.83	3.00	0.618	0.384	4.83	3.00	0.618	0.384
0.5	0.5	32°	2	0.117	4.30	2.67	0.502	0.312	4.00	2.49	0.468	0.291
0.5	0.5	32°	-3	0.055	4.83	3.00	0.266	0.165	4.83	3.00	0.266	0.165
0.5	0.5	32°	3	0.105	4.02	2.50	0.423	0.263	3.62	2.25	0.380	0.236
0.5	0.5	32°	-4	0.03	4.83	3.00	0.145	0.090	4.83	3.00	0.145	0.09
0.5	0.5	32°	4	0.07	3.78	2.35	0.266	0.165	3.38	2.10	0.237	0.147
1.0	0.75	32°	-5	0.03	4.83	3.00	0.145	0.09	4.83	3.00	0.145	0.09
1.0	0.75	32°	5	0.045	4.83	3.00	0.217	0.135	4.09	2.54	0.187	0.116
1.0	0.75	32°	-7.5	0.018	4.83	3.00	0.072	0.045	4.83	3.00	0.087	0.054
1.0	0.75	32°	7.5	0.012	4.83	3.00	0.058	0.036	3.38	2.10	0.041	0.0252
3.0	1.0	32°	-10	0.006	4.83	3.00	0.0029	0.0018	4.83	3.00	0.0016	0.001
3.0	1.0	32°	10	0.012	4.83	3.00	0.058	0.036	3.06	1.90	.0367	0.0228
3.0	1.0	32°	-12.5	0.0072	4.83	3.00	0.035	0.0216	4.83	3.00	.0348	0.0216
3.0	1.0	32°	12.5	0.0048	4.83	3.00	0.023	0.0144	2.73	1.70	.0541	0.0336
3.0	1.0	32°	15	0.0155	4.83	3.00	0.0748	0.0465	4.83	3.00	.0748	0.0365
3.0	1.0	32°	15+	0.0095	4.10	2.55	0.039	0.024	2.38	1.48	0.225	0.014
					AVERAGE		4.533	2.815	AVERAGE		4.37	2.710

Solving for θ, $\theta = T/\log_e 0.90$, $\theta = 55/0.10536 = 520$ hours, mean time before failure.

Table 10-1 utilizes the modified ELMS model traverse as shown in Table 6-1, Chapter 6.0, OPERATIONAL TERRAIN ANALYSIS, as a basis for calculation of average traverse vehicle speed. The vehicle's basic maximum horsepower requirement was determined as that required to maintain 4.83 km/hr speed in the soft soil group ($k\phi = 0.5$, $n = 0.5$) on level (zero slope) ground. The vehicle then was determined as being speed (power) limited on all positive slopes by use of this maximum basic horsepower.

Using the usual relationship (Reference 1) for determining the lower one-sided confidence limit for the mean time before failure (θ), in terms of cumulative test time (T), and solving for test time, the cumulative test times necessary to verify the required MTBF are derived, see Table 10-2. The left hand column of Table 10-2, represents the number of failures during the test and the right column shows the total test time that must have been accumulated, during which these failures occurred, in order to verify the mean time before failure θ, (or the reliability) at the required confidence. These cumulative test times are high.

There are a number of assumptions and limitations inherent in this approach. The only failures that may occur are those that are breakdowns caused by the sudden (and random) accumulation of stresses acting on and in the item without preceding deterioration symptoms. Failures due to degradation or wearout, or due to weak substandard components not being caught by the normal inspection methods, invalidate this approach.

Allocation of reliability goals to the component level, starting with a 0.9 overall system reliability, results in high reliability goals (0.99 or better for certain components). In consequence, to directly demonstrate the reliability by time-based tests at the anticipated stress level, long accumulated test times at the anticipated environment stress level would be required.

An alternate approach to time-based testing is proposed for application in this study. This concerns application of a test technique associated with the margin of safety.

TABLE 10-2

$\theta = 520$ hours $\qquad \theta/2 = 260 \qquad f = 1/\theta = 1920$

r = number of failures	X^2 0.2, 2r+2	T = $\theta/2\ X^2$ 0.2, 2r + 2 time hours
0	3.22	836
2	8.56	2120
4	13.44	3490
6	18.15	4720
8	22.8	5920
10	27.3	7100
14	36.3	9440

First, in such a technique, it will be necessary to determine the various stress levels and their distribution, that is, the standard deviation of the distributions. Then by strength testing, i.e. step level stress testing at values above the mean anticipated stress, a distribution for component strength with respect to the particular environmental stress is determined. With the anticipated stress distribution and the strength distribution as a function of that stress level, techniques are available for use in prediction of reliability, since this area, common to the two distributions, defines an anticipated failure distribution under the anticipated normal stress.

The nature of the problem must be fully appreciated and, at all times, careful judgment must be exercised in evaluating and isolating the environmental stresses. One reason is that it is virtually impossible in terrestrial test configurations to replicate the totality of the lunar environment, such that the effects of combined stresses, thermal and low pressure for example, may be difficult to appreciate and anticipate. In the stress level technique, effects of aging processes are not normally considered.

In the development of the test program, choice is available between the step stress method, and the time-based tests. The basis of choice will be the relative costs in given instances based on the reliability level required. As cited above, the time-based test time (hence cost) will be least for the final integrated system. This then provides a promising program base for the integrated system tests.

The assumption of random failure rates, also cited above, likewise requires careful consideration. In the proposed program, where the final integrated system is being tested under the anticipated environmental stresses, though random failures may be the basis for establishing a total test time goal, a given individual failure will not be considered prima-facia as random. It will be assumed to be a result of one or more of the following:

- Design Error
- Component incompatibility
- Material defects

A careful study will be made in all cases to prove or disprove the assumption. Hence, following such a failure, necessary corrections will be made, then the test continued. By testing a reasonable number of integrated design models (2 or 3 in the simulated environment, with simulated dynamic stressing), total simulator time may be minimized, since it is reasonable to assume that the type failure cited above would be distributed among the test models, and testing on the non-failed item would continue.

10.2 Reliability Program for the M-274 Modifications

10.2.1 Reliability and Failure Rate Goal Allocations

In order to allocate reliability goals, certain programs have been developed previously such as one reported by J. T. Hinely, Jr., and B. F. Shelley (Reference 2). However, in this vehicle study simplifying procedures have been used in view of the lack of information correlating failure rates with actual mission failures. Since reliability techniques are essentially tools for the attack of problems, essentially unsoluble by virtue of the fact that all possible modes of failure have not yet been reduced to mathematical models, the techniques used vary in sophistication with the field of application. For example, in the field of electronics, the greatest degree of understanding and practice has been developed with respect to reliability analysis. Air frame fabricators are in a favored position by virtue of the fact that civil and military aircraft failures are extensively investigated and documented regardless of whether or not the failure is catastrophic. In ground vehicle practice, sophistication in failure mode documentation and analysis is not this extensive but is currently being developed. In military vehicles, past emphasis has been upon high mortality parts designation and maintainability parameters. Because of severe environmental conditions and long operating lifetimes, fatigue failure rather than random failure (the assumption on which many of the reliability techniques of analysis are founded) proves to be dominant.

Thus, for this preliminary analysis, an unsophisticated approach was selected to explore the implications of the established reliability goals set by the problem constraints.

First, as seen in Table 10-1 the total mission time is approximately 55 hours. The first assumption is then that wear and fatigue are not significant factors in anticipated failure modes. Second, effective quality control is assumed, since relatively few produced items are involved, so that extensive inspection procedures may be justified. Therefore, apparent random failures may be assumed, that is, the failure rate and reliability are related by:

$$R = e^{-\tau F}$$

R = probability of success (reliability)

e = Naperian logarithm base

τ = mission time (55 hours)

F = failure rate (expressed as number of failures per million hours of operation)

Where $F = 1/\theta$, and θ = the mean time before failure (hours in this example).

The above cited paper (Reference 2) used allocation factors as a basis for allocating reliability goals for the system components based on the overall system reliability goal. This factor was taken as the quotient of a complexity-failure parameter, divided by an importance factor. Derivation of this latter factor required a knowledge of the correlation of mission failure rate with the component or subsystem failure rate in question. These data are not available insofar as any lunar vehicle is concerned. However, correlations may be anticipated based on the system and component criticality.

In the following analysis, the essential system, major subsystems and subsystems will be examined in the first order approximation with the assumption that all are of equivalent potential insofar as the mission criticality is concerned. In the previously cited paper (Hinely, et. al.) this factor varied from unity to a maximum of three for an aircraft system, much more complex than the vehicle analyzed here; hence, for a relatively simple system, which has been designed to eliminate marginally functional components, a common criticality parameter (except for the wheel assemblies, since they perform in part redundantly) appears to be a valid approximation.

The complexity-failure parameter may be taken as actual experienced failure rates, which for convenience are expressed as failures per million hours of operation.

Since there have been no lunar vehicle histories, in the systems interfacing with the lunar environment, only judgment (particularly in this preliminary feasibility estimate) can be exercised in tabulating numerical values for failure rates. In cases involving components not directly exposed to the environment, batteries, fuel cells, etc., and which have been flight qualified, actual data may be used. In most instances "analogous" values only can be tabulated.

This exercise is considered justified, however, since it serves to highlight problem areas, illuminate generally the system feasibility, and gives guidance in test design and estimation.

An apparent random failure distribution may be visualized as shown in Figure 10-1. In this figure, the distribution of failures represents the area common to two different distributions, the first being characteristic of the stress environment, where a random stress intensity is distributed about a "mean" stress value, and the other is distributed about a higher "design" stress for the component in question. A philosophy of reliability design is to keep this common area of two distributions within the bounds of the allowable failure rate. This is done by adequate design and quality assurance; hence, as noted previously in this section, a failure encountered in tests, such as stress step tests, will be examined to determine if deficiencies in design or quality exist.

Table 10-3 presents an allocation of reliability goals. The breakdown to significant minor assemblies indicates no unreasonable requirements, insofar as the first and second order vehicle modifications are concerned. This is true except for the radiator, and coolant pump, which are exposed in operating for the full mission period of 14-days. Study will be required on redundant flow path configurations in the radiator, coupled with some means of closing any loops opened by meteoroid puncture. A designed redundancy in the integral coolant pump components can also be considered.

In the third order modification, the component assembly with the least history is the forced elliptical wheel assembly. Although prototype demonstrations have shown the design feasibility, no actual applications have been made, so that its reliability could not be compared with that of similar items.

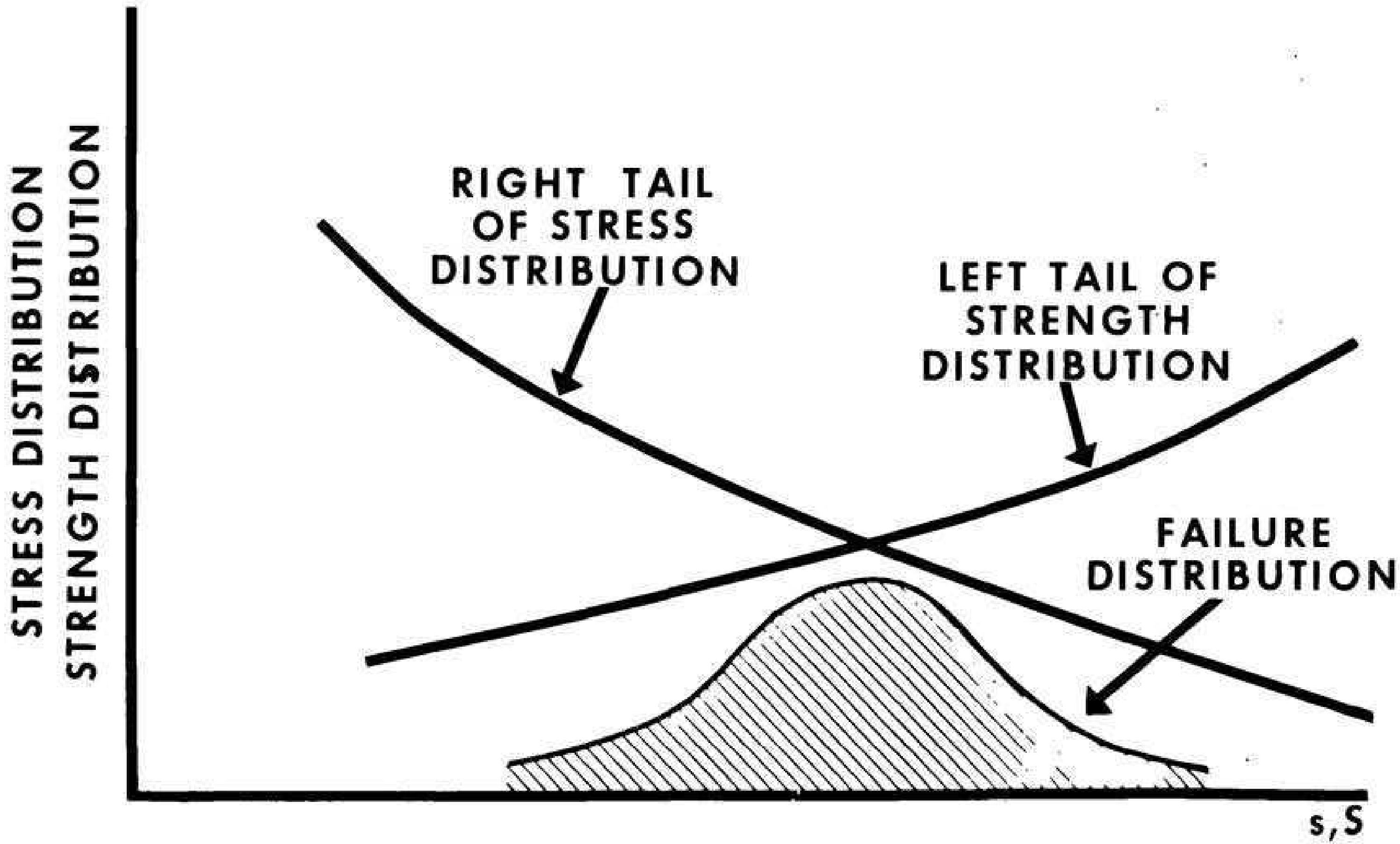


Figure 10-1. Failure Distribution - Step Stress Method

In the reliability goal allocation, only two areas of real redundancy exist, one in the battery system, and the other in the electric motor drives of the third order modification. In any battery system shown, one parallel circuit is redundant to the mission operation. In the four motor individual wheel drives, each motor has an overload capability, and the harmonic drive is reversible. Therefore, with one motor failure, and with some sacrifice of overall efficiency, the remaining three motors can power the vehicle. A short circuit failure mode in the individual motors requires overload electric current protection.

In all of the degrees of modification, the four wheel system represents a partial redundancy, in that the basic vehicle, with proper load adjustments, can operate with only three wheels. This is not a true redundancy, since, though a sortie may be completed on three wheels only, operational safety would no doubt preclude initiating a sortie with only three wheel systems in proper operating order. To allow for partial redundancy, the importance of the individual wheel drive assemblies was derated by a factor of 0.75, designated as an importance or criticality factor, such that in summation of the failure rate goals of these components to obtain the overall failure rate goal, a factor of 0.75 is applied.

10.2.2 Reliability Demonstration Test Selection Criteria

In Table 10-4, the estimates of minimum times are in hours for the time-base testing required to demonstrate the reliability goals for the major subsystems in the first and second order vehicle modifications. The minimum times are based on the assumption that testing to no more than six failures in the major subsystem groups will develop the desired overall confidence. In actual tests, environmental modes A and B (low pressure and thermal) can be imposed in the same test regime, i.e. a space simulation chamber. In addition, modes C (vibration), E (impact) and F (static torsion and beam loading) can be applied in conjunction with modes A and B. This combination of modes is deemed important in demonstrating the reliability of seals in the lunar environment. The mode group CDEF can be imposed terrestrially in driving tests over selected terrains, see Vol II - Part 2, APPENDIX, Section 3.0, TERRAIN ANALYSIS.

TABLE 10-3

FAILURE RATE AND RELIABILITY ALLOCATION CHART

I. INTEGRATED SYSTEMS GOAL:

R = Active mission probability of success = 0.900 = $e^{-\tau F}$

τ = Mission Time, 55 hours

F = Integrated System failure rate = $-(\log_e 0.900)/\tau$ = 1920 per 10^6 hours

Allowing 60 failures per mission hours to other than mission traverse operations, yields the vehicle systems goal of no more than 1860* failures per million hours.

II. MAJOR SUBSYSTEMS ALLOCATIONS:

	1st and 2nd Order Modified Vehicle		Third Order Modified Vehicle	
	Failure Rate Per 10^6 Hour	R	Failure Rate Per 10^6 Hour	R
A. Structures, fixed bed and mechanical group	110	0.994	110	0.994
B. Steering System Group	300	0.9835	395	0.9785
C. Energy Storage and Conversion Group	350	0.9809	254	0.9857
D. Drive Line Group	1100	0.9413	1101	0.9836
TOTAL SYSTEM	1860	0.9045	1860	0.9048

* The storage, unloading, and idle time operations, except as noted, are excluded in this allocation.

TABLE 10-3 (Cont'd)

	1st and 2nd Order Modified Vehicle		Third Order Modified Vehicle	
	Failure Rate Per 10^6 Hour	R	Failure Rate Per 10^6 Hour	R
III. SUBSYSTEMS ALLOCATIONS:				
A. Steering System Group				
1.A. Steering wheel and shaft*	5	0.9997	-	-
B. Steering motor drive**	-	-	100	0.9945
2.A. Steering wormgear and follower*	100	0.9945	-	-
B. Steering ball screw drive**	-	-	100	0.9945
3. Steering gear case and seal	50	0.9973	50	0.9973
4. Steering arm linkages including teflon ball joints	145	0.9920	145	0.9920
TOTAL SUBSYSTEM	300	0.9836	395	0.9785

* A - Alternate applies to 1st and 2nd order modified vehicle

** B - Alternate applies to third order modified vehicle

TABLE 10-3 (Cont'd)

	1st and 2nd Order Modified Vehicle		Third Order Modified Vehicle	
	Failure Rate Per 10^6 Hour	R	Failure Rate Per 10^6 Hour	R
B. Structures, Fixed Bed and Mechanical Group				
1.A. Pedal and hand control Linkages*	50	0.9973	-	-
B. Electric steering and hand controls**	-	-	50	0.9973
2. Foot rest platform, frame attachments and foot step	5	0.9997	5	0.9997
3. Radiator mounts	10	0.9995	10	0.9995
4. Cargo platform (chassis)	5	0.9997	5	0.9997
5. Drive line element supports	15	0.9972	15	0.9992
6. Fasteners and weldments	25	0.9986	25	0.9986
SUBSYSTEM TOTAL	110	0.994	110	0.994

*A - Alternate for 1st and 2nd order modification

**B - Alternate for third order modification

TABLE 10-3 (Cont'd)

	1st and 2nd Order Modified Vehicle		Third Order Modified Vehicle	
	Failure Rate Per 10^6 Hour	R	Failure Rate Per 10^6 Hour	R
C. Energy Storage and Conversion Group				
1.A. Battery bank, 4 parallel circuits of 2 series units each (unit r = 0.998), three parallel circuits required for operation.	4	0.9998*	-	-
B. Battery bank, 3 parallel circuits of 2 series units each (unit r = 0.998) two parallel circuits required for operation	-	-	2	0.9999**
2.A. Electric motor drive, 1 unit (driving power distribution train)	100	0.9945	-	-
B. Electric motor drive, 4 units each (driving reduction gear assembly)	-	-	2	0.9999**
3. Electric motor controls	70	0.9962	70	0.9962
4. Radiator & coolant lines	150***	0.9918	150***	0.9918
5. Coolant pump & electric motor drive	30****	0.9984	30****	0.9984
SUBSYSTEM TOTAL	350	0.9809	254	0.9862

*Based on cycling charge and discharge

**Three only required for mission accomplishment

***For 14 days exposure, the goal is 25 per 10^6 hours.

****For 14 days operation, continuous, the goal is 5 per 10^6 hours.

TABLE 10-3 (Cont'd)

	Failure Rate Per 10^6 Hours	R
D. Alternate A for Drive Line Group for 1st and 2nd Order Vehicle Modification		
1. Clutch & housing	80	0.9956
2. Transmission and transfer case	60	0.9964
3. Rear axle & housing	60	0.9964
4. Propeller shaft and propeller shaft universals with housings	165	0.991
5. Front axle, brake assembly, pull-pin seal, and housing	80	0.9956
6. Wheel drive assemblies*		
a. Half axle and housing	10	0.9995
b. Drop case assembly	45	0.9975
c. Constant velocity (C. V.) drive joint, and flexible housings & bearing	55	0.9970
d. Vacuum seals	50	0.9973
3. Wheel	25	0.9986
Total One Assembly	185	0.990
Weighted total for four assemblies, using failure criticality for 0.75**	555	0.970
Subsystem Summation	1100	0.9413

*There are four each assemblies, one for each wheel, three of which are deemed necessary for the completion of the sortie mission with the cargo properly counterbalanced.

**Assuming one of four assemblies redundant; however, although three wheel operation may allow completion of a given sortie, continuance of the 10 sortie mission is improbable, hence a 0.75 criticality factor is used to allocate a subsystem goal within the grouping rather than assuming redundancy.

TABLE 10-3 (Cont'd)

	Failure Rate Per 10^6 Hours		R
	Each	System of 4 (Weighted)	
D. Alternate B for Drive Assemblies and Individual Wheel Suspension Groups for the Third Order Vehicle Modification. (Although the motor is a subsystem in these groups, it is tabulated under energy conversion.)			
1. Shock absorber (importance factor 0.5)	15	30	0.9984
2. Torsion bar suspension (importance factor 0.75)	5	15	0.9992
3. Steering pin yoke, motor and drive gear housing assembly and pin bearings (importance factor 0.75)	50	150	0.9918
4. Harmonic drive, drop pinions and drive gears (importance factor 0.75)	100	300	0.9945
5. Drive unit seal (importance factor 0.75)	50	150	0.9918
6. Brake subsystem, service, parking, with housing and seal (importance factor 0.75)	30	90	0.995
7. Forced elliptical wheel assembly (importance factor 0.75)	122	366	0.980
		1101	0.9413

Table 10-4

TIME-BASED TEST CORRELATION BY MAJOR VEHICLE SUBSYSTEM GROUPS FOR 1st & 2nd ORDER MODIFICATION

Major Vehicle Subsystem Group	Failures				
	Rate $f \times 10^6$ goal	Primary Failure Environmental Modes*	θ MTBF Hours	Time Based Minimum Test Time Range Hours (for zero to 6 failures)**	Reliability Goal
Fixed Bed and Mechanical	110	BCDEF	9,100	14,600 to 82,500	0.994
Steering System	300	ABCDEFG	3,330	5,660 to 31,900	0.9835
Energy Storage & Conversion	350	BCDF	2,860	4,600 to 25,900	0.981
Drive Line	1100	ABCDEF	910	1,460 to 8,250	0.9415
Integrated System	1860	ABCDEFG	538	835 to 4,720	0.9+

*Failure Environmental Modes:

A - low pressure
B - thermal
C - vibration
D - dust
E - impact
F - combined static torsion beam loading mode
G - dynamic effects in reduced gravitational environment

**Based on $\frac{\theta}{2} X^2$ (0.2, 2r+2) (See reference 1)

Table 10-4 shows that time-base testing for the fixed bed and mechanical group required a time period significantly greater than any of the other major subsystem groups. This group is one in which step stress testing should prove advantageous; since, provided a distribution of the environmental stresses can be predicted, a relatively low number of stress tests to component failure can be the basis for predicting a distribution of failure as a function of stress.

10.2.3 Reliability from Step Stress Distributions

Provided the assumption may be made that there is no aging effect on components in the stress tests, in the course of weaker part failure, the overlap between stress and strength distribution represents possible failure.

The technique for the environmental failure analysis is essentially as follows:

The probability of a stress, s, occurring is equal to the area A_1 (Figure 10-2),

$$P\left(s_1 - \frac{ds}{2} \le s \le s_1 + \frac{ds}{2}\right) = f(s_1)ds = A_1$$

The probability of strength exceeding stress, s, is equal to the area A_2, or

$$P\ (S > s_1) = \int_{s_1}^{\infty} f(S)dS = A_2$$

The probability of no failure occurring at this stress is the product of these two probabilities:

$$dR = f(s_1)ds \int_{s_1}^{\infty} f(S)dS$$

The reliability of the component would then be the probability of strength being greater than any possible values of stress.

$$R = \int dR = \int_{-\infty}^{\infty} f(s) \left[\int_{s}^{\infty} f(S)dS\right] ds$$

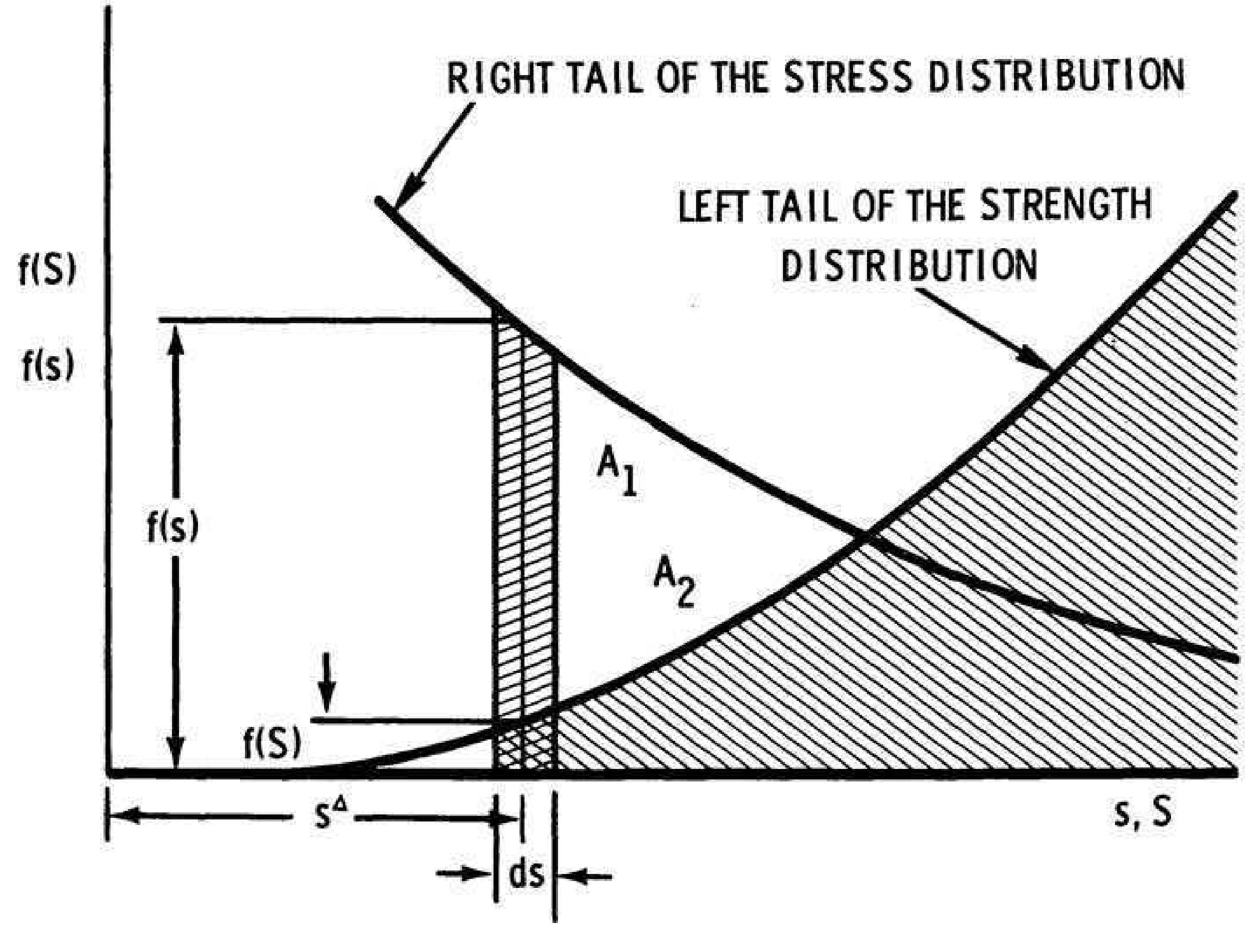


Figure 10-2. Reliability Determination - Step Stress Method

Another expression can be derived by using a different, but similar; approach. In this case, the probability of stress remaining less than a given strength is considered. Similar development of this approach is:

$$R = \int dR = \int_{-\infty}^{\infty} f(S) \left[\int_{\infty}^{S} f(s)ds \right] dS$$

The probability of failure can be derived using a similar analysis and the fact that failure will occur when strength is less than a given stress or when stress exceed a given strength.

$$Q = \int_{-\infty}^{\infty} f(s) \left[\int_{-\infty}^{s} f(S)dS \right] ds$$

or

$$Q = \int_{-\infty}^{\infty} f(S) \left[\int_{S}^{\infty} f(s)ds \right] dS$$

From these, the failure density function can be shown as

$$f(s) = \frac{dQ}{ds} = f(s) \left[\int_{-\infty}^{s} f(S)ds \right]$$

or

$$f(S) = \frac{dQ}{dS} = f(S) \left[\int_{s}^{-\infty} f(s)ds \right]$$

This distribution is represented by the shaded area shown in Figure 10-1 (not drawn to scale).

Exact solutions to the preceding equations are shown below only for normal stress and strength distributions. The actual form of the distributions in question must be determined, and appropriately handled for this technique to be valid.

Since the distribution of the difference between the two normal distributions assumed is also normal, let D = S - s, then

$$f(D) = \frac{1}{\sigma_D \sqrt{2\Pi}} \exp\left[-\frac{1}{2}\left(\frac{D-\mu_D}{\sigma_D}\right)^2\right]$$

Where $\mu_D = \mu_S - \mu_s$ and $\sigma_D = \sqrt{\sigma_S^{\ 2} + \sigma_s^{\ 2}}$

As long as S>s, there can be no failure, so reliability is P(D) > 0 or

$$R = \int_0^\infty f(D)dD$$

$$R = \frac{1}{\sigma_D\sqrt{2\pi}} \int_0^\infty \exp\left[-\frac{1}{2}\left(\frac{D-\mu_D}{\sigma_D}\right)^2\right] dD$$

transforming, letting

$$z = \frac{D-\mu_D}{\sigma_D}$$

where z is a standardized variable, giving new limits of integration:

for $\quad D = \infty,\ z = \dfrac{\infty - \mu_D}{\sigma_D} = \infty$

and for $\quad D = 0,\ z = \dfrac{0-\mu_D}{\sigma_D} = \dfrac{-\mu_D}{\sigma_D}$

also, $\quad f(D)d(D) = f(z)dz$

therefore,

$$R = \frac{1}{\sqrt{2\pi}} \int_{-\frac{\mu_D}{\sigma_D}}^{\infty} \exp\left(-\frac{z^2}{2}\right) dz$$

unreliability can be derived either based on D > 0 or on Q = 1 - R, either method giving:

$$Q = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{-\frac{\mu_D}{\sigma_D}} \exp\left[\frac{-z^2}{2}\right] dz$$

10.3 REFERENCES

10-1 Bozovsky, I., "Reliability, Theory and Practice", Prentice Hall, 1961.

10-2 Hinley, J. J., Shelley, B. F., "Methods of Allocating Reliability Goals To Aircraft Subsystems and Component", Reliability & Maintainability Conference, 1963.

Page

11.0 DEVELOPMENT PROGRAM

11.1 Army Development Program Procedures

Before any development is undertaken by the Army, a formalized statement of the requirement must exist. Any individual, or organizational element in the Army can initiate the formalization of a development requirement. The initial action takes the form of a proposal, under one of three categories:

- QMDO. The first is the QMDO, or Qualitative Materiel Development Objectives. In this category, proposals are drafted for improved capabilities, rather than proposing development of specific equipment, since the state-of-the-art may not permit a definite equipment proposal.
- QMR. The second category comprises the QMR, or Qualitative Materiel Requirements, which are sufficiently detailed to describe a distinguishable item of equipment.
- SDR. The third category is the SDR, or Small Development Requirements, which are similar to the QMR, except they describe items low in procurement cost, and of such nature to be of low cost in development.

The format for the proposals in each of these categories has been prescribed, and the responsibility for staff study and coordination of all information supporting the requirements rest with the CDC, Combat Development Command of the Army. In the CDC staff study, the requirements for the proposed equipments are evaluated, and with the help of the developing and using elements of the Army, the developmental feasibility, training requirements, etc., are evaluated prior to submission of staff recommendations to Army Headquarters for approval. QMR, in the instances of vehicle development, must be prepared and approved before the developing agency, the AMC, or Army Materiel Command for vehicle development, can initiate a development program. In independent actions, funding for development is programmed, and the technical aspects of the development program are approved. This is done in the form of a development project, by the Army Technical committee and the Army Chief of Research and Development. The studies required by the developing agency, the Army Materiel Command, Automotive and Tank Center, in the case of light wheeled vehicles for example, are initiated by the Combat Development Command's requirement for an assessment of developmental feasibility. This is indicated on Figure 11-1, which is a typical development schedule for a light wheeled vehicle, as the Merit Analysis. This continues to cover the studies required to develop the development project plan, and funding justifications, up to the time of the Project Approval, also as noted in Figure 11-1. The Development Program phase, noted in Figure 11-1, in some instances run concurrently. For example, component engineering evaluation runs concurrently with the Prototype Production, and Final Production model fabrication is initiated prior to the completion of the engineering and service tests. In addition, the final Production Package Release (final specifications) is not made until after the first production model has been delivered and accpeted. These concurrent activities usually require a three to four year program, from the Technical Committee Approval to Type Classification (formal recognition of the item as a standard item of issue, in established tables of organization and equipment). They are still sufficiently flexible to allow advantage to be taken of all the results of test and evaluation during the initial stages of the final product fabrication.

The Disposition and Findings, D&F, as noted in Figure 11-1, are elements in obtaining funding authority, and the Tables of Equipment, TE, concern the equipment required for the field maintenance.

The major part of the Engineering Development, the Prototype Design and Fabrication, as well as the fabrication of the Production Vehicle is accomplished on a contract basis with qualified industrial sources. The engineering and service testing are U.S. Army functions. Some of the engineering testing may be conducted in-house by the Army Materiel Commands' developing center; however, service testing is the responsiblity of the Test and Evaluation Command, TECOM. Vehicle service testing and combined

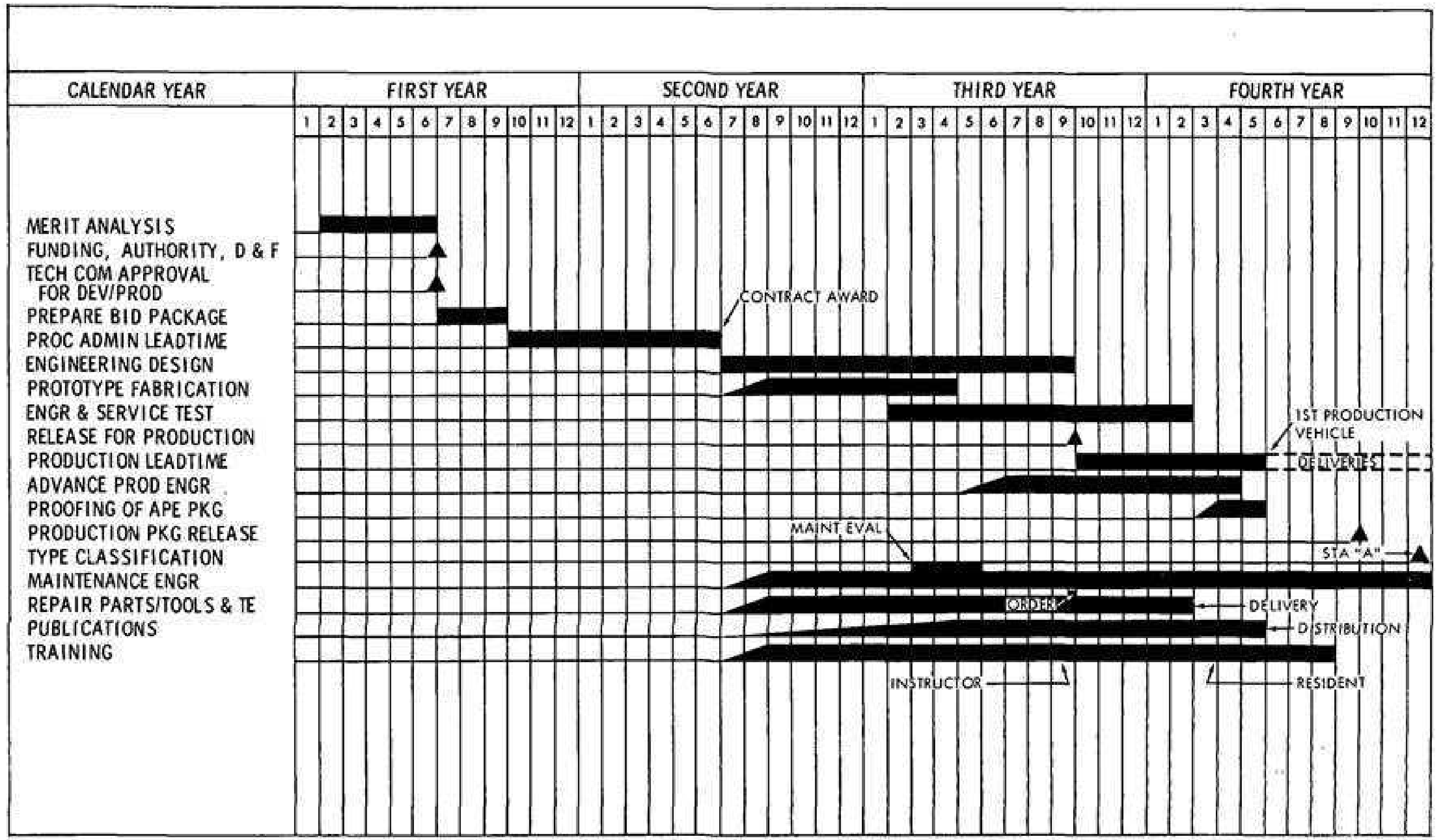


Figure 11-1. Typical Schedule - Light, Wheeled Vehicle

engineering and service tests are conducted, for the most part (excepting tropical and arctic climate tests), at the Aberdeen Proving Grounds, or at Yuma, Arizona. The combined engineering service tests comprise the cross country mobility and endurance testing. Part of the test objective is to develop experience on maintainability, including identification of the high mortality parts, so that provision for proper field support may be initiated, as well as final product improvement. The terrain characteristics available at the Yuma test facilities are discussed in Vol II - Part 2, APPENDIX, Section 3.0, TERRAIN ANALYSIS, of this study report.

11.2 Modification of the M-274 Vehicle

In estimating the development program requirements, a 1972 launch date was assumed as a program completion date. A program start date was assumed as calendar year 1967. This allows a five year development program. This is somewhat longer than the usual Army program of three to four years. The five year program has been so scheduled to permit component environmental testing, systems environmental testing and composite systems check-out environmental testing. The environmental testing required is unconventional with the U.S. Army standard practice. Facilities for environmental testing were assumed for this study to be U.S. Government facilities and available for use as required. No lead time is included for any such facilities' procurement or construction.

The five year program is not considered to be critically tight scheduling and could perhaps be compressed to four years. This would allow either a one year delay in program initiation or permit a one year earlier vehicle operational capability.

The overall cost would not change significantly by changing from a five to a four year program. Any further compression of the program (other factors being considered as feasible) would result in increased cost.

A simplified development program schedule is shown in Figure 11-2 for any of the three orders of vehicle modification. An item of note on this schedule is the Mobility Test Bed Gravity Simulation Test. It is proposed that the vehicle mobility system only be

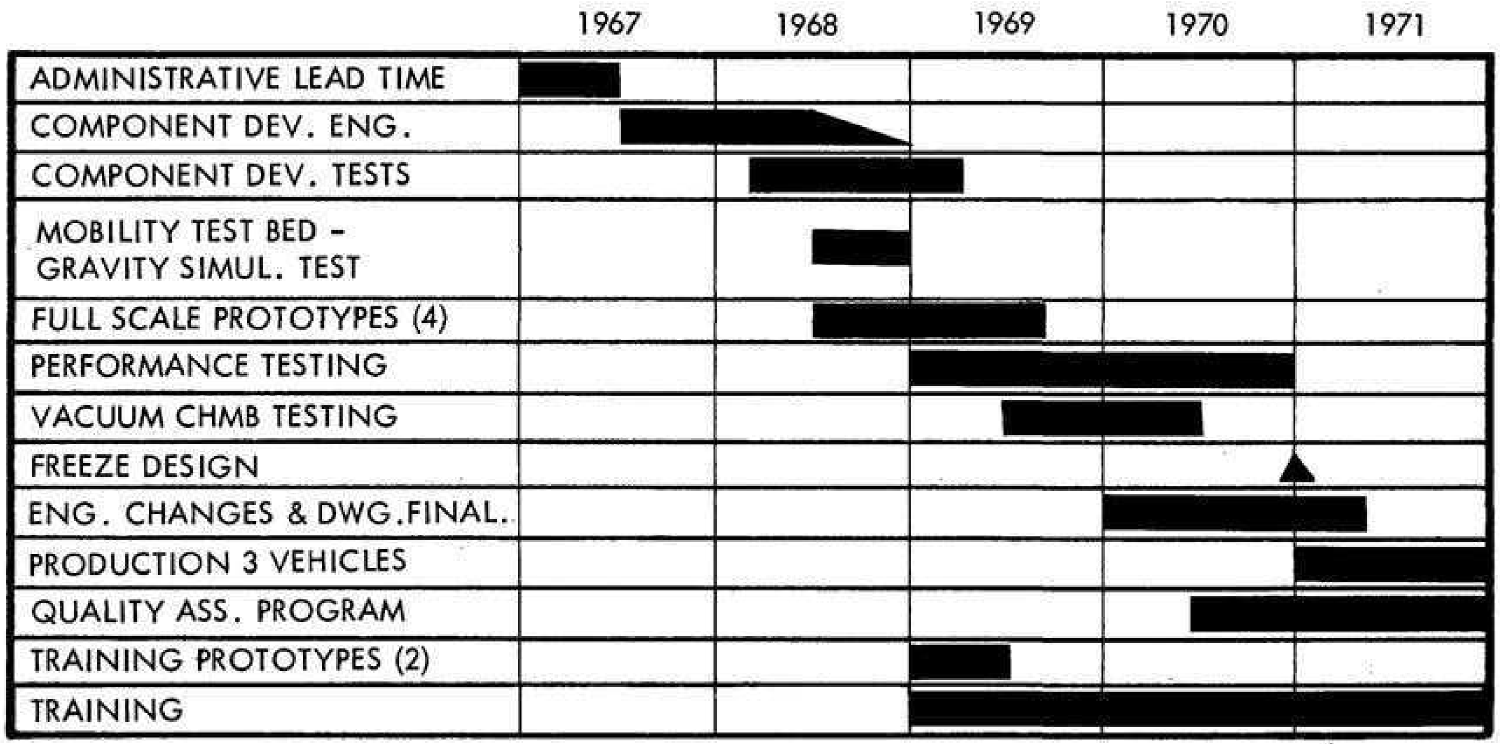


Figure 11-2. Simplified Development Schedule - Any Order

evaluated in the program in this manner. A mobility test bed, full-scale or scaled model, should be evaluated for vehicle dynamic response to the lunar gravitational environment and the lunar surface model.

11.2.1 First and Second Order Modification Programs

In the first and second order vehicle modifications, there will be the minimum component testing effort required relative to the third order vehicle modification. Figure 11-3 is a test program for the first or second order modification programs. This program shows the inter-relationship between the tests for the three main vehicle areas (power, mobility, and structure, and crew station) and the LEM/S power system and the payload packaging and off-loading equipment. The tests below the dotted line correspond to those tests designated as Engineering Tests. These tests results are used in a large measure as inputs to design development. In these areas, individual item testing will include step-stress tests to define reliability performance, as well as compliance with operational requirements. The environmental test program will follow the philosophy of conducting multiple component tests in each test event. This follows the step-stress method at the component level philosophy. This permits minimum time use of environmental facilities. The major systems testing and combined systems testing philosophy reflects the philosophy set forth by NASA, OMSF, in the "Test Philosophy and Outline Test Program", Apollo Logistics Support Systems Payloads, May 1964.

11.2.2 Third Order Modification Program

The objectives of the test program for the third order modification program are the same as those given for the first or second order modification program. The primary difference stems from the more extensive component development required. The test program format, as given in Figure 11-4, is similar to that given in Figure 11-3; however, the organization within the major systems areas is composed to meet differing component organizations.

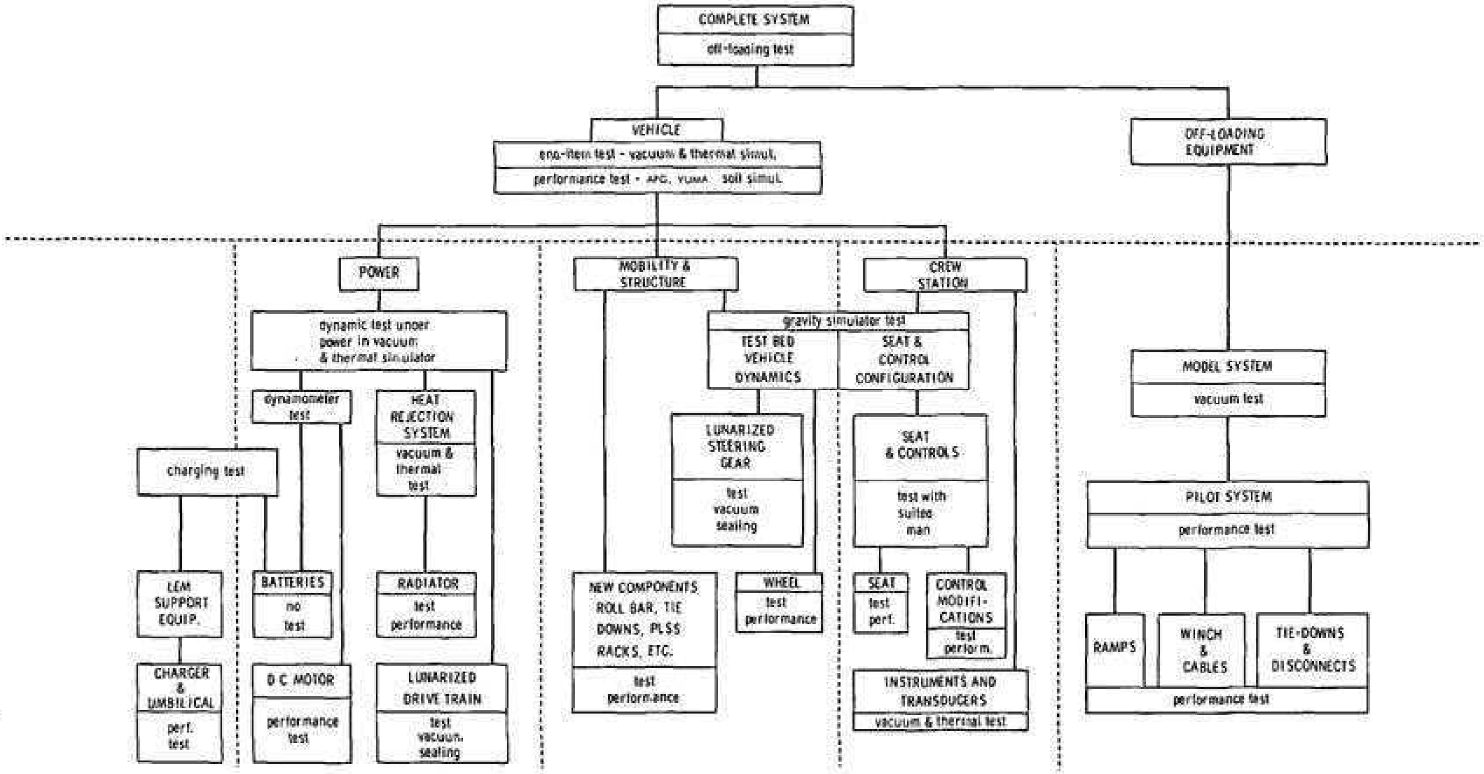


Figure 11-3. Test Program - First and Second Order Vehicle

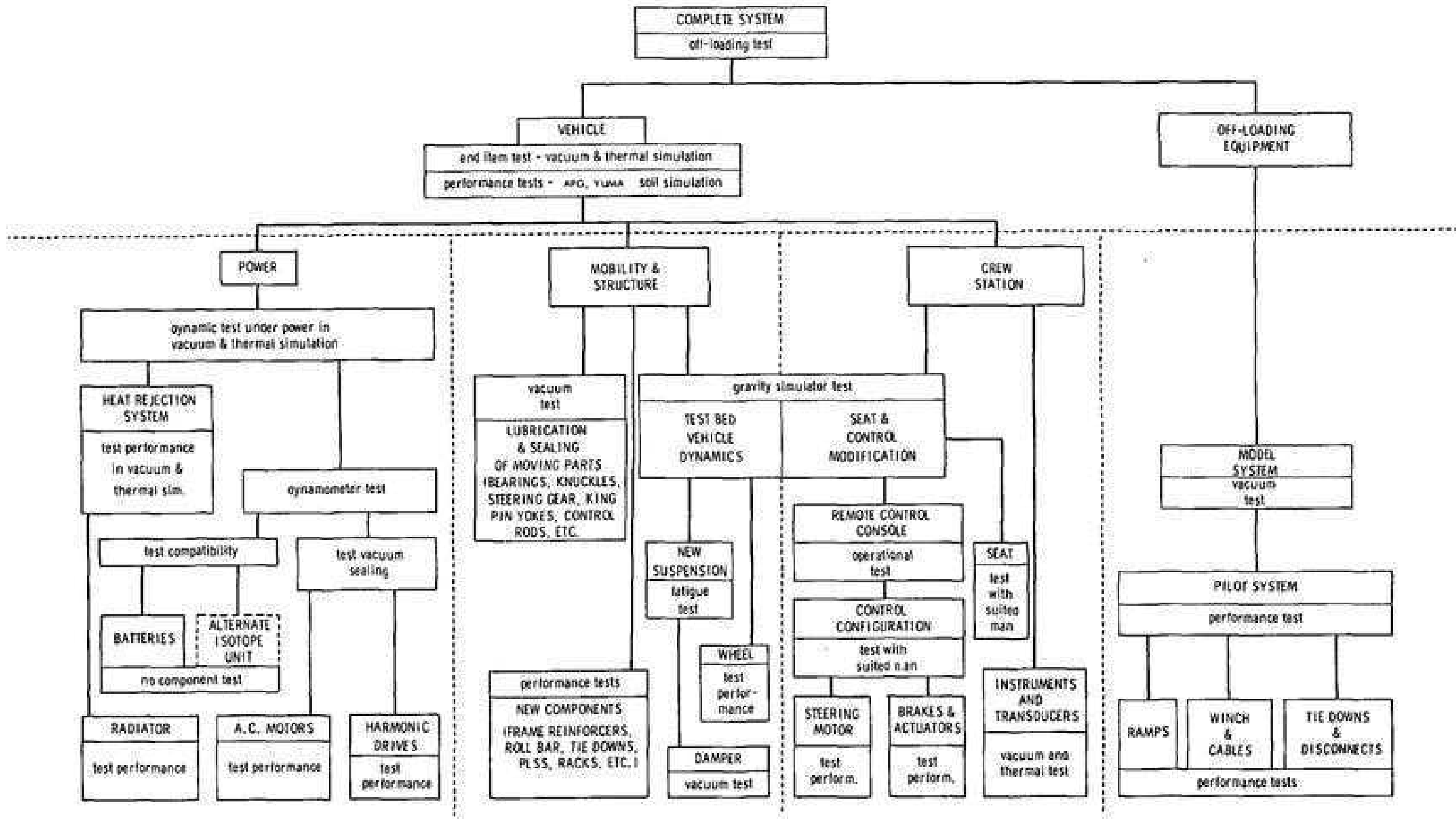


Figure 11-4. Test Program - Third Order Vehicle

In order to show, in detail, how the more complex program for the third order modification program would proceed, Figure 11-5, third order Development Schedule, was prepared. Again a five year program time span was utilized. In this program schedule, the Heat Rejection System, Battery Adaptation, Seat, and Off-Loading Equipment scheduling would be common to all orders of modification. Otherwise, the systems development shown are unique with the third order modification program.

In the third order program, the largest part of the developmental phase will be required in an 18 month time span beginning in the middle of calendar year 1967. This will require a larger proportion of total program funding in this time period than for the first or second order modification program.

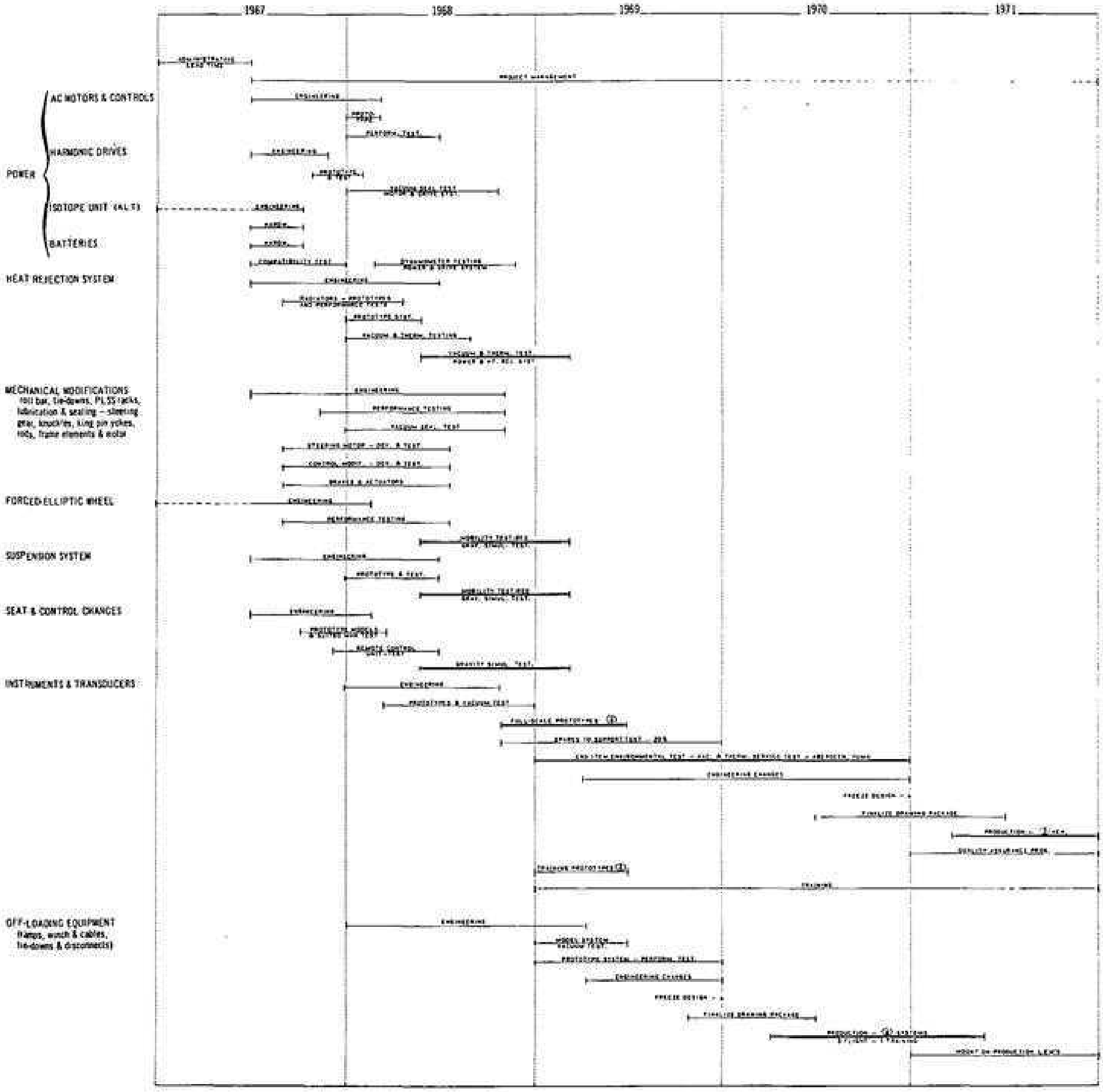


Figure 11-5. Development Schedule - Third Order Vehicle

12.0 COST ESTIMATES

The development program, as described in Section 11.0, DEVELOPMENT PROGRAM, includes the production of three flight vehicles, the LEM/S offloading and packaging equipment, and the LEM/S power system items associated with vehicle battery recharging.

The development program did not concern itself with launch and checkout equipment concepting, development or costs. These equipments will have a similarity for any vehicle concept.

12.1 Terrestrial Vehicle Cost Summary

In response to a NASA request, official and restricted Army data on vehicle development and costs and cost trends are being furnished under separate cover. Some development cost information on two particular Army vehicles is discussed in paragraph 3.3, Section 3.0, VEHICLE REVIEW AND SELECTION

12.2 Resources Data Format

The program costs developed in this study for the three orders of vehicle modification have been furnished to NASA separately, in the special data format requested in the work statement.

12.3 Cost Estimates

The program costs developed in this study for the three orders of vehicle modification are also presented in this report. The presentation format employed is one which follows the development program schedules.

Table 12-1 is a tabulation of program costs for the development of the first order modified vehicle. Table 12-2 is a tabulation of the program costs for the development of the second order modified vehicle. Table 12-3 is a tabulation of the program costs for the development of the third order modified vehicle.

These program costs for the three orders of vehicle modification are summarized by program:

Program	Program Cost
First-Order Vehicle	\$13,826,000
Second-Order Vehicle	\$14,347,000
Third-Order Vehicle	\$21,100,000

These program costs are summarized also in Figure 12-1. It will be noted that the primary difference in cost between the three programs is in the component development and testing phase of each program.

Table 12-1

DEVELOPMENT-PROGRAM COSTS

($ in thousands)

First Order Modified Vehicle

1.	COMPONENT DEVELOPMENT AND TESTING			$ 5,069
	a. D.C. Motor and Controls		1,250	
	(1) Engineering & Hardware	900		
	(2) Testing	350		
	b. Vacuum Sealing of Drive Train		1,100	
	(1) Engineering	600		
	(2) Testing (vacuum)	500		
	c. Batteries		50	
	(1) Engineering	0		
	(2) Hardware (10)	50		
	d. Dynamometer Test, Batteries, Motor & Drive System		35	
	e. Heat Rejection System		1,125	
	(1) Radiator-Develop & Test	200		
	(2) System Engineering	400		
	(3) Pilot System	25		
	(4) Vacuum & Thermal Test	500		
	f. Vacuum & Thermal Test - Power System with Heat Rejection		850	
	g. Mechanical Modifications (roll-bar, tie-downs, PLSS racks)		225	
	(1) Engineering	200		
	(2) Performance Testing	25		
	h. Seat & Control Modifications		205	
	(1) Seat - development & test	200		
	(2) Controls-functional test with suited man	5		
	i. Rigid Metal Wheel		6	
	(1) Development	5		
	(2) Performance Test	1		

Table 12-1 (Cont'd)

	j. Instruments & Transducers		223	
	(1) Engineering	20		
	(2) Hardware (about $500 ea.)	3		
	(3) Vacuum Test	200		
2.	TEST VEHICLES			$ 3,550
	a. Mobility Test Bed		150	
	b. Full Scale Prototypes (4)		1,000	
	c. Training Prototypes (2)		400	
	d. Spares to Support Test - 20%		310	
	e. Mobility & Steering Test (Gravity Simulator)		90	
	f. Vacuum Chamber Test		1,100	
	g. Service Testing APG, Yuma		500	
3.	ENGINEERING CHANGES & FINAL DRAWING PACKAGE			$ 1,200
4.	TOOLING			$ 600
5.	PRODUCTION - 3 VEHICLES @ $200			$ 600
6.	QUALITY ASSURANCE PROGRAM			$ 800
7.	OFF-LOADING EQUIPMENT			$ 675
	a. Engineering		300	
	b. Model System & Vacuum Test		210	
	c. Pilot System & Performance Test		75	
	d. Engineering Changes		30	
	e. Production - 4 Systems (3 flight + 1 training)		60	
8.	LEM SUPPORT EQUIPMENT (Charger and Umbilical			$ 75
	a. Engineering & Test		60	
	b. Hardware - 4 sets		15	
		TOTAL		$12,569
		10% PROJECT MANAGEMENT		$ 1,257
		PROGRAM TOTAL		$13,826

Table 12-2

DEVELOPMENT-PROGRAM COSTS

($ in thousands)

Second Order Modified Vehicle

1.	COMPONENT DEVELOPMENT AND TESTING			$ 5,543
	a. D.C. Motor and Controls		1,250	
	(1) Engineering & Hardware	900		
	(2) Testign	350		
	b. Vacuum Sealing of Drive Train		1,100	
	(1) Engineering	600		
	(2) Testing (vacuum)	500		
	c. Batteries		50	
	(1) Engineering	0		
	(2) Hardware	50		
	d. Dynamometer Test, Batteries, Motor & Drive System		35	
	e. Heat Rejection System		1,125	
	(1) Radiator - Develop & Test	200		
	(2) System Engineering	400		
	(3) Pilot System	25		
	(4) Vacuum & Thermal Test	500		
	f. Vacuum & Thermal Test - Power System with Heat Rejection		850	
	g. Mechanical Modifications (roll-bar, tie-downs, PLSS Racks)		225	
	(1) Engineering	200		
	(2) Performance Testing	25		
	h. Seat & Control Modifications		205	
	(1) Seat - development & test	200		
	(2) Controls - functional test with suited man	5		
	i. Metalastic Wheel		480	
	(1) Development	400		
	(2) Performance Test	80		

Table 12-2 (Cont'd)

	j. Instruments & Transducers		223	
	(1) Engineering	20		
	(2) Hardware (about $500 ea.)	3		
	(3) Vacuum Test	200		
2.	TEST VEHICLES			$ 3,550
	a. Mobility Test Bed		150	
	b. Full Scale Prototypes (4)		1,000	
	c. Training Prototypes (2)		400	
	d. Spares to Support Test - 20%		310	
	e. Mobility & Steering Test (Gravity Simulator)		90	
	f. Vacuum Chamber Test		1,100	
	g. Service Testing APG, YUMA		500	
3.	ENGINEERING CHANGES & FINAL DRAWING PACKAGE			$ 1,200
4.	TOOLING			$ 600
5.	PRODUCTION - 3 VEHICLES @ $200			$ 600
6.	QUALITY ASSURANCE PROGRAM			$ 800
7.	OFF-LOADING EQUIPMENT			$ 675
	a. Engineering		300	
	b. Model System & Vacuum Test		210	
	c. Pilot System & Performance Test		75	
	d. Engineering Changes		30	
	e. Production - 4 Systems (3 flight + 1 training)		60	
8.	LEM SUPPORT EQUIPMENT (Charger & Umbilical)			$ 75
	a. Engineering & Test		60	
	b. Hardware - 4 sets		15	
	TOTAL			$13,043
	10% PROJECT MANAGEMENT			$ 1,304
	PROGRAM TOTAL			$14,347

Table 12-3

DEVELOPMENT-PROGRAM COSTS

($ in thousadns)

Third Order Modified Vehicle

1.	COMPONENT DEVELOPMENT AND TESTING			$10,832
a.	A.C. Motors and Controls		2,500	
	(1) Engineering & Hardware	1,800		
	(2) Testing	700		
b.	Harmonic Drives		24	
	(1) Engineering	20		
	(2) Pilot	1.5		
	(3) Test	2.5		
c.	Vacuum Seal Test - Motor, Drive and Wheel Bearing Assembly		320	
d.	Batteries		50	
	(1) Engineering	0		
	(2) Hardware (10)	50		
e.	Compatibility Test - Batteries		5	
f.	Dynamometer Test - Batteries and Motor Drive System		35	
g.	Heat Rejection System		1,125	
	(1) Radiator Development & Test	200		
	(2) System Engineering	400		
	(3) Pilot System	25		
	(4) Vacuum & Thermal Test	500		
h.	Vacuum & Thermal Test - Power System with Heat Rejection		850	
i.	Mechanical Modifications (frame, motor mounts, roll-bar, tie-downs, PLSS racks, steering gear, knuckles, king pin, yokes, control linkages, etc.)		3,020	
	(1) Engineering	2,700		
	(2) Performance Testing	120		
	(3) Vacuum Test - Lubrication and Sealing	200		

Table 12-3 (Cont'd)

j.	Seat & Control Modifications		1,750	
	(1) Steering Motor - dev. & test	600		
	(2) Brakes & Actuators - dev. & test	150		
	(3) Controls, Umbilical & Remote Console - dev. & test	800		
	(4) Seat - dev. & test	200		
k.	Suspension System		330	
	(1) Engineering	150		
	(2) Pilot System	10		
	(3) Vacuum Test - Damper	150		
	(4) Performance Test	20		
l.	Forced Elliptic Wheel		600	
	(1) Engineering	500		
	(2) Pilot	60		
	(3) Testing	40		
m.	Instruments & Transducers		223	
	(1) Engineering	20		
	(2) Hardware (about $500 ea.)	3		
	(3) Vacuum Test	200		
2.	TEST VEHICLES			$ 3,550
a.	Mobility Test Bed		150	
b.	Full Scale Pilots (4)		1,000	
c.	Training Pilots (2)		400	
d.	Spares to Support Test - 20%		310	
e.	Mobility & Steering Test (Gravity Simulator)		90	
f.	Vacuum Chamber Test		1,100	
g.	Service Testing APG, YUMA		500	
3.	ENGINEERING CHANGES & FINAL DRAWING PACKAGE			$ 1,500
4.	TOOLING			$ 800
5.	PRODUCTION - 3 VEHICLES @ $250			$ 750
6.	QUALITY ASSURANCE PROGRAM			$ 1,000
7.	OFF LOADING EQUIPMENT			$ 675

Table 12-3 (Cont'd)

a.	Engineering	300	
b.	Model System - Vacuum Test	210	
c.	Pilot System - Perf. Test	75	
d.	Engineering Changes	30	
e.	Production - 4 systems (3 flights + 1 training)	60	
8.	LEM SUPPORT EQUIPMENT (Charger & Umbilical)		$ 75
a.	Engineering & Test	60	
b.	Hardware - 4 sets	15	
	TOTAL		$19,182
	10% PROJECT MANAGEMENT		$ 1,918
	PROGRAM TOTAL		$21,100

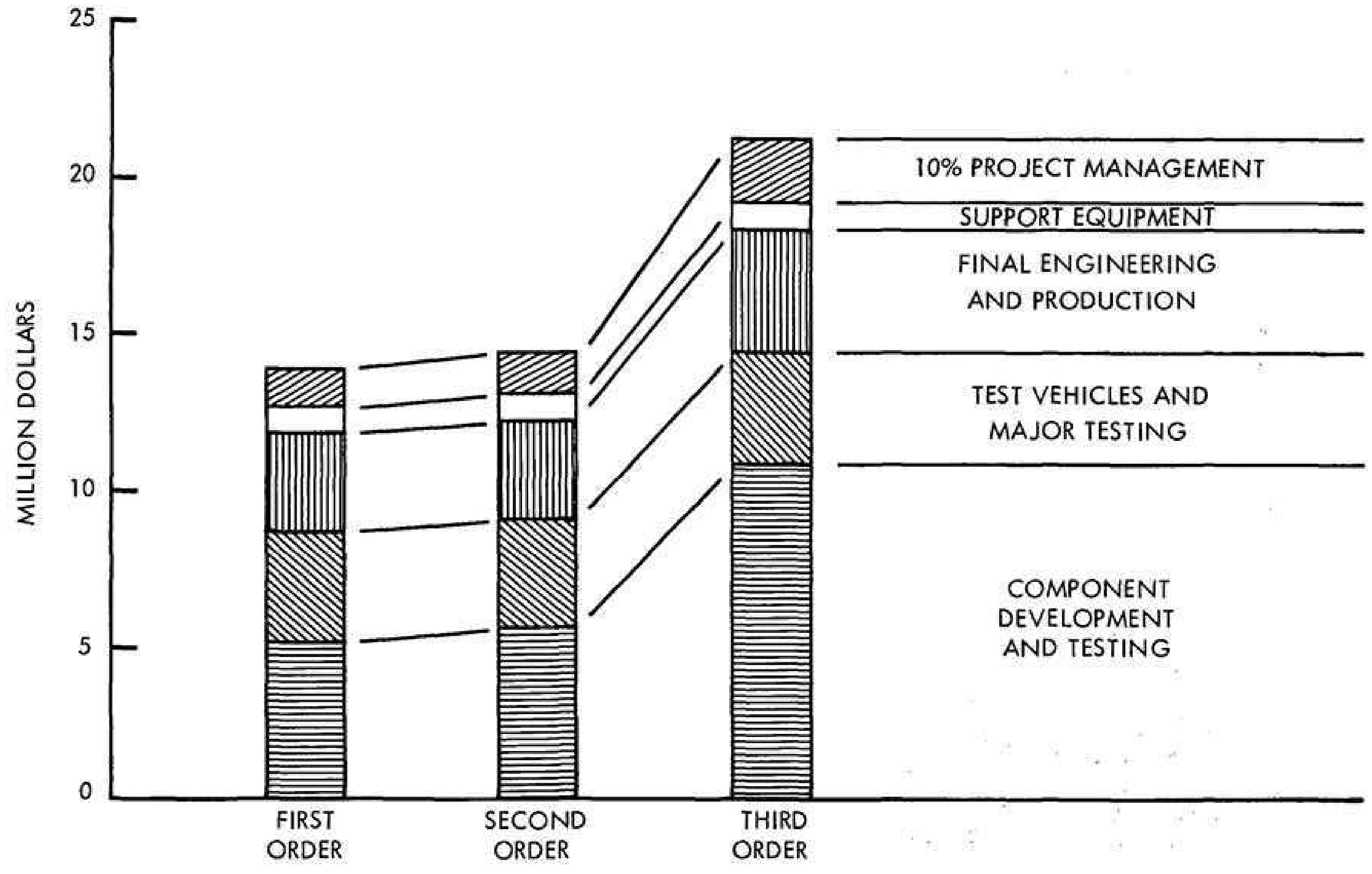


Figure 12-1. Comparative Program Costs - Modifications Programs

13.0 PROGRAM FEASIBILITY ANALYSIS

The Work Statement requests an analysis of the practicality of conducting a development/ modification program of this type to produce a lunar surface vehicle. The study conducted demonstrates the feasibility of modifying a particular terrestrial vehicle for operation on the lunar surface. To evaluate the desirability of accomplishing any of the three orders of vehicle modifications presented, some more definite mission requirements would have to be established.

The Army believes that a development program for a lunar surface vehicle by the modification route is practical and feasible. The use of any one particular terrestrial vehicle as the modification departure vehicle does not appear, however, to be the best answer. Rather, the program would be best attempted by a vehicle concept synthesis of systems and components. This synthesis would utilize to the fullest extent systems and components already developed or under development for terrestrial cross-country vehicles. The modification process would then be applied to lunarize these systems and components.

To illustrate the logic of this approach, the Mule vehicle modification orders can be considered. The third order Mule vehicle modification leaves only the vehicle platform and part of the steering system of the original vehicle. If a modification program were being attempted, and a lunar surface vehicle with a high order of terrain mobility capability were a necessity, it would be desirable to add to this third order vehicle an additional mobility improving device, an articulation joint. If this were added, none of the original mule vehicle would remain and, effectively, a vehicle would have been synthesized from systems and components.

In no case, however, would it be desirable to start the program with a majority of new vehicle systems and components which have not yet been developed or are not yet under development for use on terrestrial vehicles or space hardware. Almost every vehicle development program the Army undertakes is actually a modification program. The amount and the direction of modification is constrained by past experience as to how far it is possible and practical to modify. Long years of vehicle development experience has shown that the modification process is the surest, least costly, the most productive of results and the most predictive method.

The extremes of the lunar environment are, naturally, beyond the environment limits to which present Army vehicles are developed to operate. However, the Army develops vehicles to operate in exotic and harsh terrestrial environments. Army vehicles must operate at 325 °K (125 °F) in desert dust or in tropical rain forests. They must start and operate at 219 °K (-65 °F) in arctic climates. It would be logical, in order to produce a vehicle to operate in the lunar environment, to proceed to modify and extend this harsh terrestrial environment capability.

In the last twenty-five years, the Army has been extending their wheeled cargo-carrying vehicle capabilities for operation in marginal terrain. Each new vehicle incorporates added capability; but this is almost always a modification of a predecessor vehicle, an evolutionary process.

Over the years about 750 different light, marginal terrain vehicles have been examined by the Army. Hundreds have been built, or bought, and tested. The Army constantly examines new ideas for possible use. Promising systems and component concepts are usually developed and tested. Electrical, hydraulic and hydropneumatic drive lines have been developed and tested. For terrestrial environments these are not yet competitive economically and operationally with the internal combustion engine and the mechanical drive line. Suspension systems of all types, steering systems of all types, articulation devices of all types and wheel systems of all types have and are being developed and tested. The state-of-the-art in these areas are changing constantly. Soon many of the new and more promising devices and systems may become economically feasible for terrestrial use.

With use of developed components or components under development for space hardware and vehicle systems and components now developed or under development for terrestrial use, an excellent vehicle for use on the lunar surface could be synthesized; a vehicle with excellent inherent cross-country mobility and cargo carrying capability.

We would recommend that a lunar surface vehicle, of the weight and required cargo carrying capacity for the lunar surface missions intended, be essentially a modification program.

As an example, an excellent modified vehicle to accomplish essentially the requirements and meet the constraints of the Mule vehicle study would consist generally of the following systems:

Vehicle Frame	Designed for the vehicle use.
Articulation Joint	Modify terrestrial type joint already in latter stage of development.
Wheels	Modify terrestrial wheels already under development for terrestrial use.
Suspension System	Modify standardized terrestrial system.
Power Supply	Batteries already developed for space use.
Wheel Drive Motors	New development.
Wheel Drives	Modify terrestrial drives.

This vehicle development program would cost out close to that of the third order modified mule vehicle. It would result in close to an optimum cross country cargo carrying lunar surface vehicle for its weight and volume.

There are certain observations based upon U.S. Army experience with rough terrain vehicle operation which may be of help in lunar surface vehicle design:

- A vehicle's capability to outperform the operator (operator tolerance to vehicle performance) goes up quickly as a vehicle goes from on-road to off-road operation.

- The risk of operator injury increases rapidly as the vehicle goes from on-road to off-road operation (the consequences of errors in operator judgment becomes more severe.
- An acceptable operator risk level should, therefore, be matched with the vehicle performance capability. Excess vehicle capability is not required and for lunar surface use may be highly undesirable.

GLOSSARY

DEFINITION OF TERMS

AAP	Apollo Applications Program (formerly AES-Apollo Extension System). A concept for continuing exploration of the Moon and near-earth space through maximum utilization of the existing Apollo hardware.
Bekker Soil Values	Expressions of soil characteristics in parametric values utilized to determine soil-vehicle relationships in evaluation of a vehicle's mobility in soils by methods proposed by M. G. Bekker in numerous publications.
CM	The Apollo Command Module
Constraint	A general guideline which must be followed in the study.
Criteria	Complete instructions for the conduct of a task.
ELMS	Engineering Lunar Model Surface. A document prepared by NASA-MSFC which defines a relationship between lunar surface soil characteristics and surface slopes. It also establishes a distribution of soil characteristics and slopes with distance for traverses on the maria surfaces.
LEM	The Lunar Excursion Module of the Apollo program.
LEM/S	The Lunar Excursion Module modified into a shelter-laboratory consisting of a modified LEM ascent stage delivered unmanned to the lunar surface by a modified version of the LEM descent stage.
LEM-Taxi	A modified version of the LEM which is used to transport two astronauts to and from the lunar surface in conjunction with AAP mission.
LSSM	Local Scientific Survey Module. A surface roving vehicle designated to furnish transportation and power for carrying and emplacing scientific experiment packages.
Lunar Flying Vehicle	LFV—or "lunar hopper" is a system designated to act as an abort capability from a surface vehicle or a transportation adjunct vehicle.

MTBF	Mean Time Before Failure. Utilized in derivation of cumulative test times necessary to demonstrate reliability.
Mission	Those experiments specified to be performed at a particular location within a designated time period.
Mobility System	A roving vehicle or a flying vehicle or a combination of vehicles to provide transportation on or over the lunar surface for one or more astronauts and experiment package cargo.
Module	A discrete embodiment of all or a portion of the equipment required for performance of a specified function.
Payload	The aggregate of equipment and supplies to be delivered by a hardware transportation unit.
PLSS	Portable Life Support System, personal life support pack utilized by astronaut in space suit.
Radiator	Exchanger for rejecting excess heat by means of radiation to space.
Radius	Maximum radial distance for LEM/S permitted on a vehicle traverse.
Sink	A body (deep space) utilized for the disposal of heat in a thermodynamic process.
Structure	Refers to the vehicle chassis and structural frame.
Super Insulation	Multilayer insulation, such as Linde SI-44 which consists of alternate layers of aluminim foil and submicron glass fiber paper (35-70 alternate layers) with intervening spaces evacuated of atmosphere to a low order of pressure.
Traverse	An excursion over the lunar surface in a vehicle for the purpose of conducting scientific experiments or of emplacing scientific packages on the lunar surface.

SYMBOLS AND UNITS

Å	Angstroms, unit of length, equal to 10^{-10} meters.

A	Area, square meters or square centimeters.
Bar	Unit of pressure, 10^5 newtons/meter^2
C	Calorie, unit of thermal energy, see J, below.
C_p	Specific heat, such that C_p J expresses the proper MKS units of Joules per kilogram.
c	Coefficient of soil cohension, lbs-force/in^2.
D	Diameter, meters or centrimeters.
d	Infinitesimal differential operator
E	Energy, Joules or watt hours.
erg	Unit of energy, 10^7 ergs equals one joule.
e.v.	Electron-volt, unit of energy, 6.24 x 10^{18} ev equals one joule.
°F	Degrees Fahrenheit.
g	Acceleration due to gravity.
J	Mechanical equivalent of heat, 4186 joules per kilogram calorie.
Joules	Unit of energy (MKS) or newton-meter.
°K	Degrees Kelvin.
K_c, k_c	Modulus of soil deformation governed by cohensive character of soil (lbs-force)/(inch)$^{n+1}$.
$K\phi$, $k\phi$	Modulus of soil deformation governed by internal frictional character of soil (lbs-force)/(inch) $^{n+2}$
Kg	Kilogram, basic MKS unit of mass.
km	Kilometer, unit of distance, 1,000 meters.
L	Length, meters or centimeters.

M	Mass, kilograms or grams.
$\dot{M}$	Time derivative of mass, mass rate per unit time, kilograms per second.
m	Meter, basic MKS unit of length.
n	Exponent parameter governed by soil characteristic, applied to sinkage in sinkage-soil load function, stated to reflect soil stratification, see Bekker Soil Values.
n	Newton, MKS unit for force.
p	Pressure, bars.
Q	Energy content, joules
$\dot{Q}$	Time derivative of energy or power, joules/second or watts.
r	Radius, in meters or centimeters.
s	Distance, in meters.
T	Temperature, absolute scale, in degrees K (or as stated).
t	Time, seconds, minutes, or hours.
v	Velocity, meters/sec or kilometers/sec.
w	Mass flow rate, kilograms/second
z	Deformation of soil, in the Bekker Soil Values, expressed as inches.
α, a	Alpha, absorptivity of solar radiation, decimal fraction of incident radiation absorbed at body temperature.
ϵ, e	Epsilon, emissitivity, decimal fraction of *theoretical black body radiation radiated at* body temperature.
η	Eta, efficiency.
θ	Theta, angle between normal to a surface illuminated by solar radiation and the incident propagation path.

μ	Mu, viscosity, poise or centipoise.
π	Pi, the number of radians in 360 degrees of arc.
ρ	Rho, density, kilograms per cubic meter, or grams per cubic centimeter.
σ	Sigma, Stefan-Boltzmann constant, 5.71×10^{8} watts/m^2 $°K^4$
ϕ	Phi, angle of internal friction, in non-cohensive soils generally regarded as equivalent to angle of repose.
ϕ	Phi, flux, solar radiation flux, watts per square meter.
ϕ_s	Incident solar radiation flux.
ϕ_r	Reflected solar incident radiation flux.
Ψ	Psi, fluid transport modulus.