LMSC/HREC A783335
Revision A

LOCKHEED MISSILES & SPACE COMPANY
HUNTSVILLE RESEARCH & ENGINEERING CENTER
HUNTSVILLE RESEARCH PARK
4800 BRADFORD DRIVE, HUNTSVILLE, ALABAMA

THIRD INTERIM REPORT

PRELIMINARY DESIGN STUDY OF A LUNAR GRAVITY SIMULATOR

25 October 1966

Contract NAS8-20351

Prepared for George C. Marshall Space Flight Center
Huntsville, Alabama

APPROVED BY: R. B. Wysor
R. B. Wysor
Project Engineer

R S Paulnock
R. S. Paulnock
Mgr., Systems Engineering

R S Paulnock
fr J. S. Farrior
Resident Manager

FOREWORD

This report was prepared by the Lockheed Missiles & Space Company, Huntsville Research & Engineering Center, to document the accomplishments of the final study period for the Preliminary Design of a Lunar Gravity Simulator, Contract NAS8-20351. The study is being conducted by the Systems Engineering Organization at HREC under the direction of Mr. R. S. Paulnock, Manager, and Mr. R. B. Wysor, Project Engineer. Other contributors to this report and the study efforts during this third interim reporting period are Dr. Wolfgang Trautwein and Messrs G. O. Floyd, Z. V. Adams, E. L. Saenger, D. J. Wilson and G. E. Malone. This report was published by the Technical Publications Organization at HREC under the supervision of Mr. J. E. Coleman.

The study program is sponsored by the Advanced Systems Office of Marshall Space Flight Center under the technical direction of Mr. Herbert Schaefer, Principal COR, and Mr. Robert R. Belew, alternate COR.

Technical data in this report will be delivered to NASA/MSFC technical personnel at an informal presentation scheduled for 27 October 1966.

CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF FIGURES (continued)

Section 1
INTRODUCTION

The Lunar Gravity Simulator (LGS) is a system being considered for the evaluation of full scale Lunar Surface Vehicles (LSV) over simulater lonar terrain in an Earth's gravity environment. The purpose of such a system will be to:

- Substantiate the LSV mobility system performance parameters under simulated loading conditions as may occur from the 1/6 g environment and the anticipated obstacle, slope and velocity combination.
- Establish the confidence level of the ability to design the mobility systems for various LSV configurations.
- Determine the effect of the vehicle dynamic behavior on the vehicle operator and the man-machine relationship in a 1/6 g environment.
- Train astronauts in handling LSV's in a 1/6 g environment.

With these objectives in mind, LMSC has conducted a Preliminary Design Study of a Lunar Gravity Simulator System under contract to the Marshall Space Flight Center. The concept developed during this study is illustrated in Figure 1.1. This study effort encompassed a 15-week period which was subdivided into three five-week intervals. This report describes the tasks and accomplishments of the final period and the conclusions and recommendations for subsequent study efforts.

Figure 1.2 depicts the overall LGS study program plan (Task, Schedule and Manloading). The first ten-week period was devoted to a study of the two-dimensional LGS and the results of the first portion of this study were described in the First and Second Interim Reports (References 1 and 2). The completion of the three-dimensional study is described in this document. Efforts during this reporting period were devoted to the following tasks:

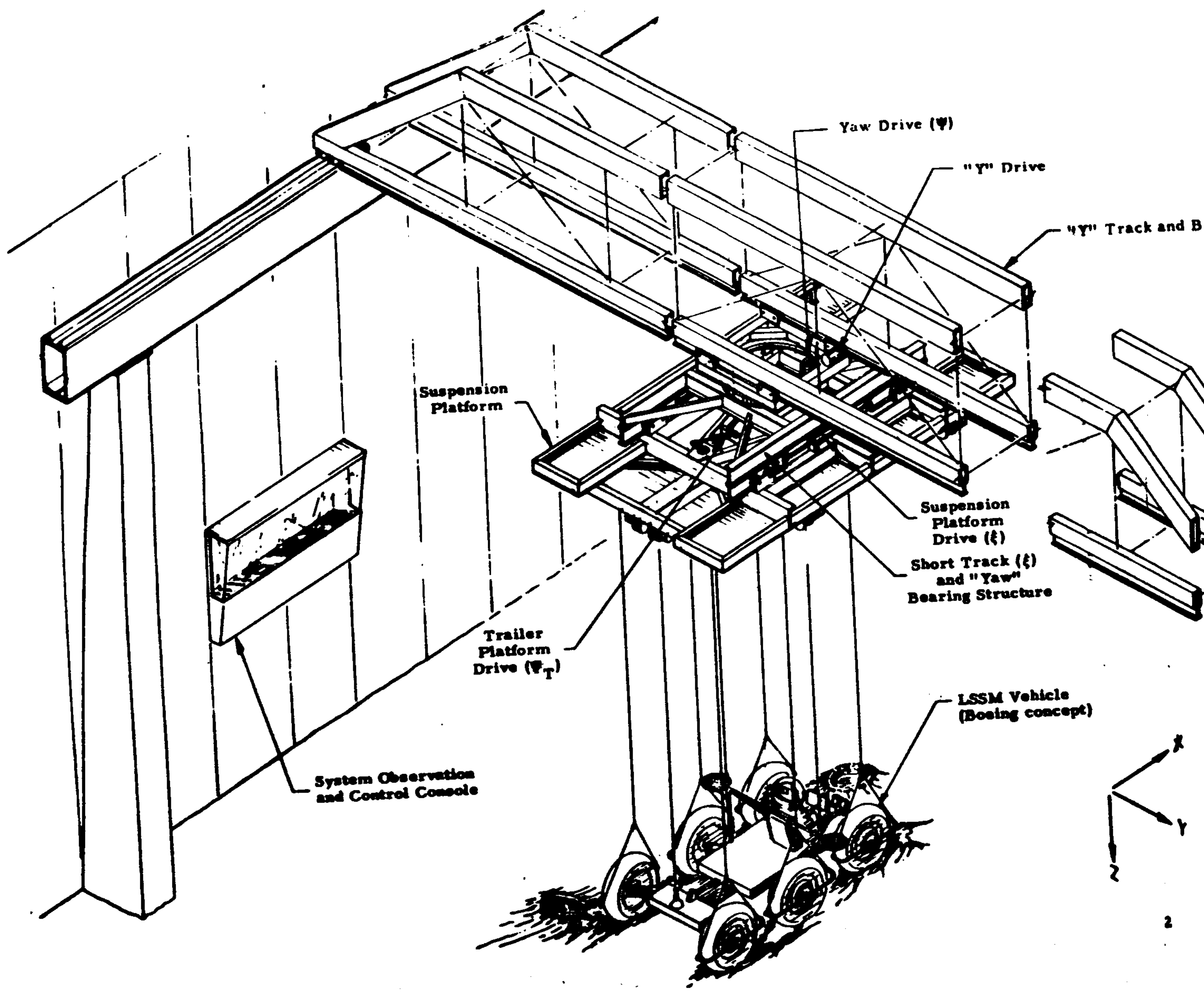


Figure 1.1
LUNAR GRAVITY SIMULATOR
FOR
LUNAR SURFACE VEHICLES

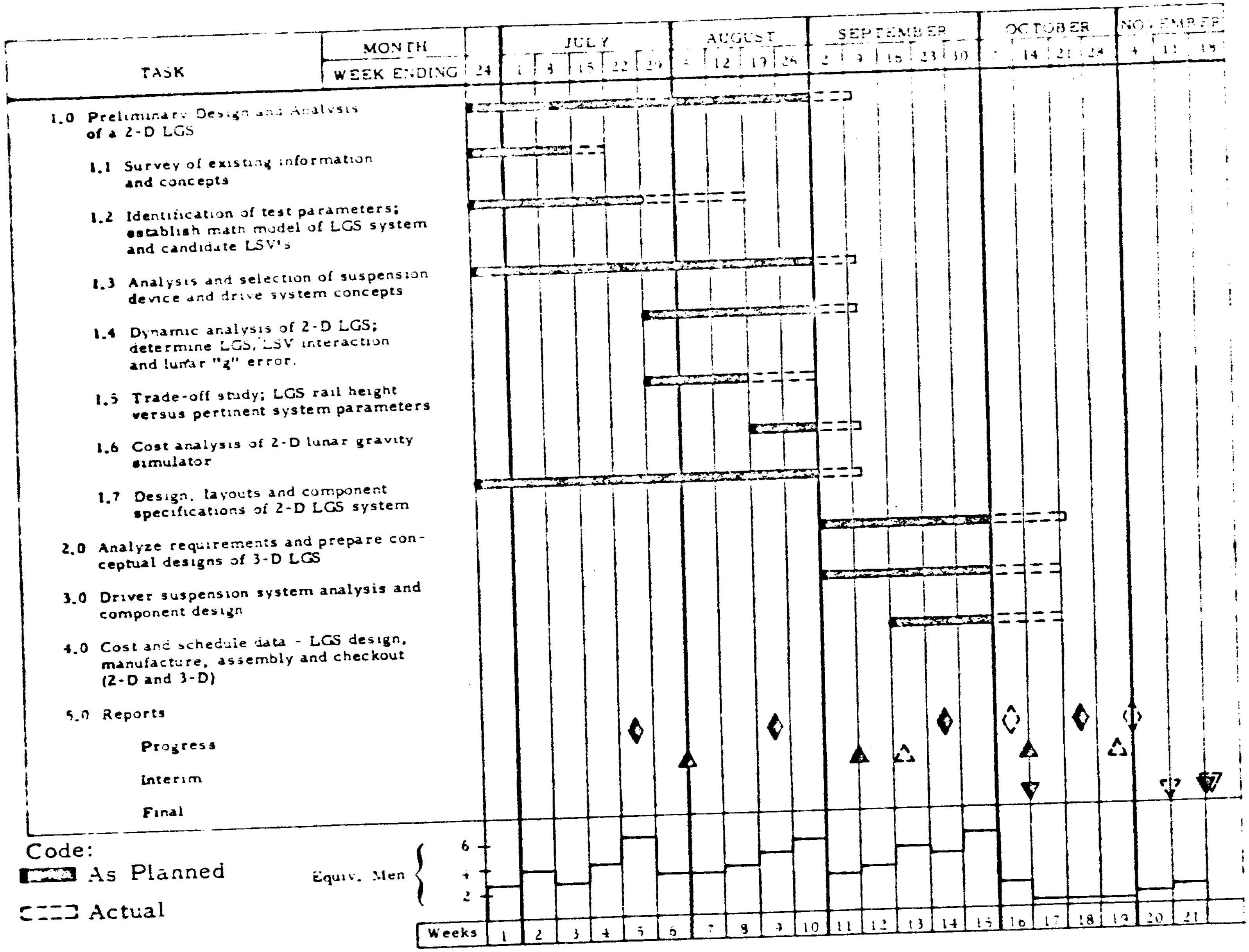


Figure 1.2 - Program Plan

Task	Study Efforts
2.0	Analyze the requirements for a three-dimensional 1/6 g simulator, based on concepts developed for the two-dimensional system, and prepare a conceptual design.
3.0	Analyze the provisions for suspending the driver of the test vehicle under simulated lunar gravity conditions and prepare a conceptual design of the necessary system adapted to the two-dimensional simulator.
4.0	Prepare cost and schedule data for the manufacture, assembly, and checkout of a two-dimensional simulator with and without a driver suspension system. Also, prepare similar data for a three-dimensional simulator system neglecting the data for the supporting steel construction.

The following sections discuss the accomplishment of these tasks and makes recommendations as to areas for future work.

Section 2
THREE DIMENSIONAL LUNAR GRAVITY SIMULATOR

2.1 SYSTEM DESCRIPTION

The basic concept for a 3-D Lunar Gravity Simulator (LGS), shown in Figure 1.1, is a suspension platform rigidly attached to an overhead trolley, with a suspension system consisting of a cable network and constant force mechanisms which support and link the Lunar Surface Vehicle (LSV) to the suspension platform. The suspension platform and trolley provide the sensor, control and drive systems necessary to enable the platform to follow directly above the LSV during dynamic tests. Further details of this system are illustrated in Figure 2.1. The only practical difference between the 2-D and 3-D systems is the inclusion of a bridge trolley to provide movement in a lateral direction (Y-axis). The 3-D system consists of the following functional elements:

- LSV Chassis and Wheel Support Frame
- Suspension System
- Suspension Platform
- Short Track (ξ), Yaw (Ψ) Bearing and Lateral (Y) Trolley Structure
- Bridge (X) Assembly

These elements will be discussed below, and any additional data not previously reported or major differences between the 2-D and 3-D systems will be noted:

2.1.1 Vehicle Chassis/Wheel Support and Frame (Same as 2-D Systems)

The vehicle chassis support frame would be constructed of a simple lightweight tubular truss frame, with the lower end attached to hard points

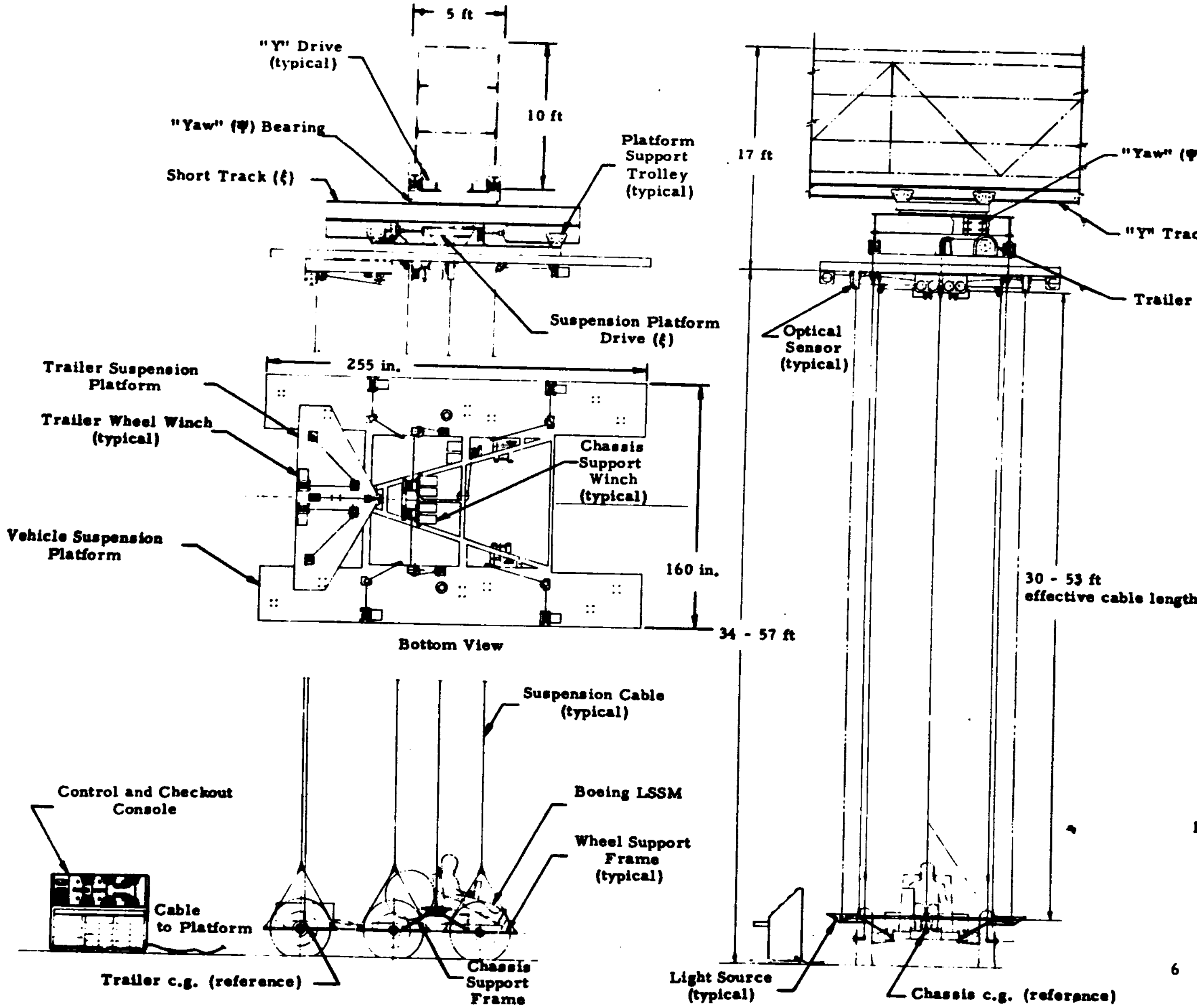


Figure 2.1

LUNAR GRAVITY SIMULATOR
3-D CONFIGURATION

on the chassis primary structure. The upper end would terminate with an omni-ball joint, located on the vehicle pitch axis and permitting complete suspension cable freedom through a 35° cone (static condition). The same type of structure would be located at the c.g. of the vehicle trailer (Boeing configuration only). The force sensing of the vehicle would be accomplished by placing a load cell between the omni-ball joint and the suspension cable. The vehicle wheel interface would consist of a support frame assembly that would attach to the outer face of the vehicle wheel hub. The assembly is composed of a mounting flange that would act as the interface for a bearing located in a housing, connected to a tubular frame yoke. The yoke ends would support ball end joints that would be located at the wheel suspension system roll axis (through c.g.). Two cables would attach to the joints and be connected together above the c.g. This arrangement would also allow for a 35° cone of cable freedom. The force sensing would be provided by instrumenting the yoke with strain gages, thereby measuring frame deflection.

2.1.2 Suspension System

- Suspension Cables and Winch Subassemblies

The 1/6 g condition of the LSV will be maintained by supporting 5/6 of the weight of the vehicle chassis and wheels by the use of constant force suspension cables, attached to the interface and that are reeved by hydraulic powered, servo actuated drum winches, mounted on the suspension platform. The length of the cables will be about 60 feet. The cables will have an electrical conductor core for signal transmission.

The only significant modification to the existing 2-D system is the addition of two cable spools on the vehicle winches. This suspension system could be used on either the 2-D or 3-D simulator configuration to support the arms of the driver-astronaut.

- Driver-Astronaut Suspension System

Since the lunar surface vehicle driver-astronaut will wear a hard suit that will be securely attached to the vehicle, only his arms need be suspended for the lunar gravity simulation. This arm suspension force to be maintained is approximately 20 pounds, assuming only one-sixth (1/6) of its earth weight acting at the c.g. The force would be provided by a constant force spring motor that is attached to a cable and driven by the vehicle chassis winch. This system is suitable for either the 2-D or 3-D systems.

2.1.3 Vehicle Suspension Platform

The platform description for the 2-D configuration is nominally the same for the 3-D. The vehicle, or main platform structure has been strategically beefed-up to increase natural frequency from 10 to 16 Hertz (Hz). If it is determined that the LSV trailers can pivot, the suspension system will include a trailer suspension platform that will also pivot, and will be actuated by a hydraulic cylinder drive (Ψ_T). The suspension platform travels along the overhead track on four (4) support trolley wheels. The platform is propelled fore and aft (with respect to the support track) by a hydraulic cylinder.

- Optical Tracking System

To minimize error in the simulation, the center of the suspension platform must be aligned over the LSV c.g. In order to sense the error that does exist and to correct for that error, an optical tracking system will be used to provide feedback to the suspension platform drive controls. This system will consist of two (2) light sources mounted on the LSV support frames outboard of the cable attachment.. The lights are aligned on the pitch axis and would provide a beam that would be perceived by two (2) optical sensors, mounted under the suspension platform. Each sensor would consist of a light-focusing lens system and a photomultiplier tube that is divided into four (4) equal quadrants. An error signal would be produced when the light beam was received unequally on the quadrants.

- Suspension System Control and Checkout Console

The suspension system control and checkout console will consist of a modified component rack cabinet with the representative vehicle schematic on an inclined panel. Each wheel and chassis attachment point will be shown with a digital voltmeter and adjusting potentiometer located adjacent. A checkout panel would also be included with a functional array of illuminated push-button switches, that would check a system and readout to a common digital voltmeter. The console may be used to calibrate the system by the following procedure:

- Adjust wheel suspension control potentiometers until the proper weight is reached (the wheel suspension weight would have already been established).
- Adjust vehicle chassis control potentiometer until the vehicle is lifted off the ground. (Align c.g. if vehicle does not rise in a level fashion.)
- Calculate 5/6 of vehicle chassis and wheel suspension system weight and adjust potentiometers to read this weight.

The console would contain up to 40 servo amplifiers for the force control system and trolley drive system. Additionally, appropriate power supplies will be required for the servo amplififers and other electrical elements. In all probability, temperature conditioning of the console will be required for the critical electronic elements. This system could be located in an observation and control room as illustrated in Figure 1.1.

2.1.4 Short Track (ξ), "Yaw" (Ψ) Bearing and Trolley Structure

The 3-D short track/main suspension platform interface will be the same as the rail support structure (Section 5.1.5 of Reference 2), which this structure replaces in the 3-D configuration. The short track is used for short, quick response transient conditions. This will compensate for the relatively slow reaction time of the supporting bridge structure. The short track is structurally supported by the outer race of the "yaw" (ψ) bearing. The inner race of the bearing is geared to an electronically controlled (SCR) electric motor. This permits the entire platform structure below it to rotate, maintaining the proper angular orientation relative to the LSV.

The inner race is supported by the four "Y" trolley assemblies. Two (2) SCR-controlled electric motors will be used to drive the system along the "Y"-axis track and bridge assembly.

2.1.5 "Y" Track and Bridge Assembly

This assembly is a modified Warren truss bridge having a five (5) foot base, a height of ten (10) feet, and a length of one hundred (100) feet. The lower corner structures include a rail to accommodate the "Y" trolley drive system. The bridge is supported at each end on wheels that are driven by the four "X" drive motors (one SCR-controlled electric motor for each wheel). The bridge would be driven for coarse movement of the suspension platform along the "X" axis.

2.1.6 Safety Provisions

The three linear trolley systems will be fitted with limit switches and impact absorbers at their ends to prevent trolley overtravel. The trailer platform drive is designed for limited excursion such that internal hydraulic buffers are sufficient to prevent damage. The yaw drive system will rely on a manually operated emergency shutoff, since the system is free to pivot 360° and has a relatively low rate of rotation.

Emergency shutdown switches will be located at the control console, the vehicle driver's seat and at other strategic locations.

2.1.7 Simulator Facility Building

The above sections comprise the Lunar Gravity Simulator.

These functioning elements will require an enclosure which has the following minimum internal dimensions: height, 74 feet; width, 100 feet; and length, 200 feet. It would need a rail along each side of the 200-foot length, at a minimum height of 63 feet, capable of supporting the ends of the bridge structure. A building of this size would permit sufficient ridges and troughs (patterned after the lunar terrain) to permit driving the LSV in an oval or figure eight pattern.

2.1.8 LGS System Weight Summary

A weight summary of the functioning elements of the Lunar Gravity Simulator is presented in Table 2.1. The weight is divided into the three major structural groupings as a convenient means of presentation.

Table 2.1

VEHICLE SUSPENSION PLATFORM WEIGHT SUMMARY
THREE-DIMENSIONAL SYSTEM

Component	Weight (lb)	
Suspension Platform		
Main, or Vehicle		
Structure	941	
Honeycomb panels	222	
Wheel winches and pulleys (4 sets)	128	
Chassis winches and pulleys (3 sets)	285	
Trolley assembly (4 sets)	220	
Hydraulic power supply, electric drive motor and Hydraulic Actuator (ξ)	520	
	2,316	
Trailer		
Structure	124	
Wheel winches and pulleys (2 sets)	64	
Honeycomb panels	54	
Actuator	24	
	266	
Sub-Total		2,582
Short Track and Yaw Bearing Structure		
Structure	1,060	
Trolley Assembly (4 sets)	284	
Y-axis drive motors (2)	576	
Platform yaw motor	24	
Sub-Total		1,944
Bridge Assembly		
Bridge structure	30,600	
X-axis drive motor (4)	374	
Misc. structure and accessories	2,000	
Sub-Total		32,974
Total		37,500

2.2 STRUCTURAL CONSIDERATIONS

The major consideration in the structural design of any dynamic system is the need to optimize between individual system requirements. For example, the required power for the various drive systems is related to the physical properties such as mass, moment of inertia, etc., of the moving elements of the system. Therefore, light weight and low inertia are required to minimize the drive system. However, this must be tempered with another requirement; to ensure accuracy of response, the simulator's natural frequency must be well above the highest forcing frequency of the system. As a result of these incompatible requirements, the structural design must be just stiff enough to avoid vibration problems and yet be as light as possible. This involves still another decision - of the various materials available; which of the structural materials would be the lightest, cost the least, be the easiest to fabricate and have the best delivery date? In order to evaluate these factors, calculations were made of stress, deflection and frequency of a basic framework. The natural frequency requirements were found to present the limiting factor in the design. Therefore, the frequency analysis is presented, with supplementary stress and deflection calculations as necessary.

2.2.1 Frequency Analysis

An analog computer program simulating the LGS/LSV combination was used to determine maximum forces, velocities and accelerations of specific points of interest. This program and its results are discussed in Reference 2. The largest imposed force on the suspension platform occurred during a 2 g deceleration of the lunar surface vehicle. Using a 50-foot cable length, the trolley drive force (F_d) reached a value of 5,400 pounds (negative x direction). The observed frequency was approximately 5 Hz. Additional test cases at lower deceleration rates gave similar but lower forces. The trolley force data are presented in Figure 2.2. The

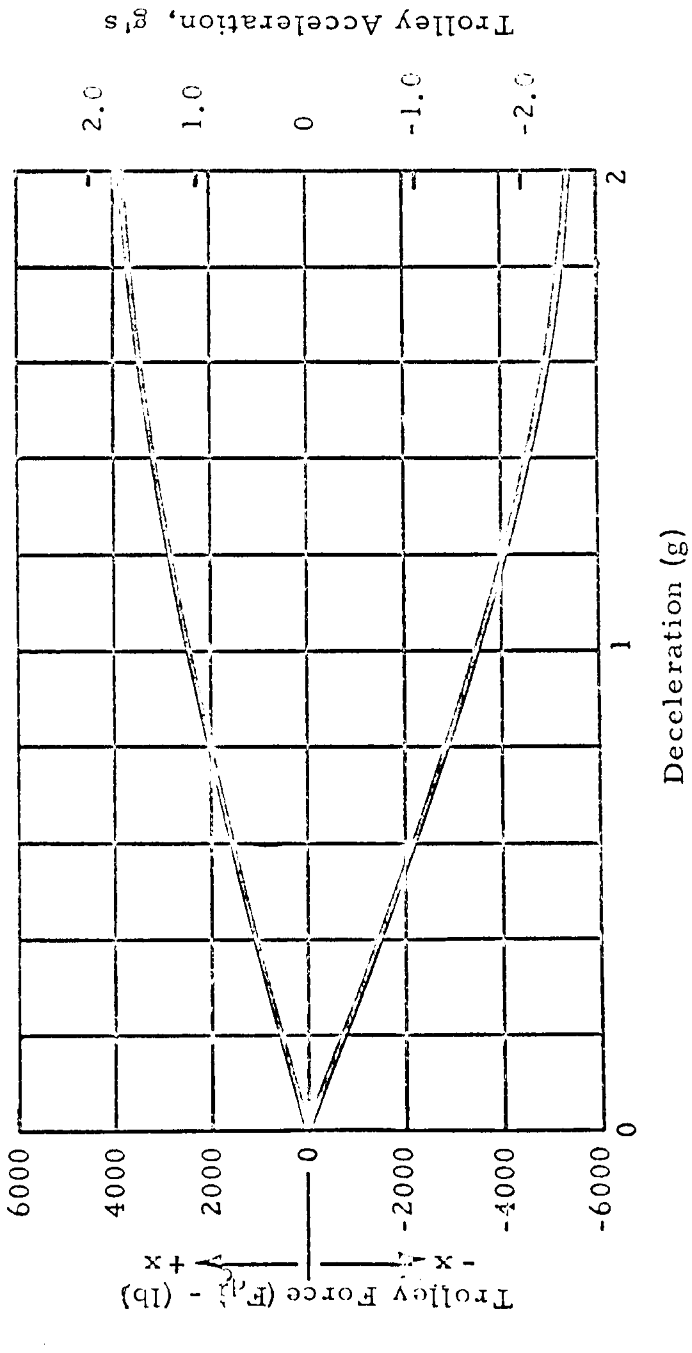
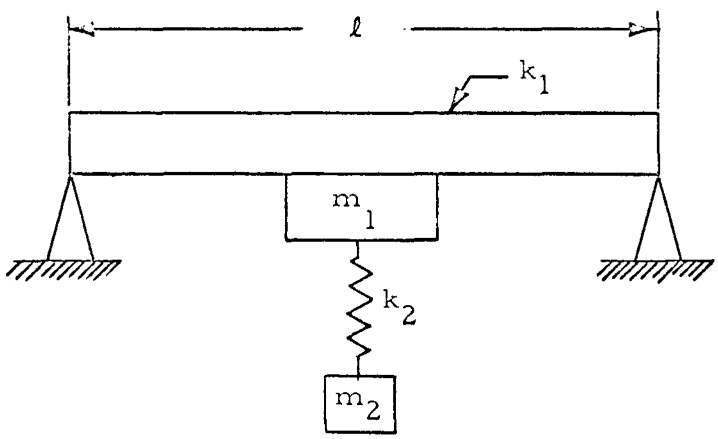


Figure 2.2 - Peak Horizontal Trolley Force vs LSV Deceleration

negative forces are larger than the positive forces at each deceleration level. In addition, the positive forces during an acceleration is also smaller because the vehicles are designed for an .8 g acceleration limit, while the deceleration rate is 2 g.

To assure that the natural frequency of the support structure avoids the 3 - 5 Hz range, each section should be considerably above this value. Since the structure consists of two major parts, this system is usually analyzed as a simple supported beam with two degrees of freedom. The general configuration is shown below.



In the figure above, two spring-mass systems are connected in a series system where the rigid masses m_1 and m_2 are both assumed to move along the same vertical line. The massless springs have linear force-deflection relationships (as long as the deflections in beam are kept small) denoted by spring constants k_1 and k_2.

When the mass m_2 is small relative to the primary structure, it is unable to influence the motion of mass m_1. The response of m_2 to a shock force may be evaluated by determining the response of the primary structure (k_1) and using this response as the excitation for the secondary structure (k_2). The continuing periodic vibration of the primary structure appears to the secondary structure as steady-state vibration. A condition of resonance would occur if the natural frequency of the secondary structure were equal to the primary structure; therefore, the frequencies should be different. Preferably, the natural frequency of the secondary structure should be higher than the primary; otherwise, the deflection of spring k_2 may equal or exceed the displacement of m_1. A secondary structure with a natural frequency twice the primary natural frequency experiences forces and amplitudes 1.35 times as great, as indicated in Figure 2.3. This is a tolerable amount of amplification; it can be achieved practically and constitutes a reasonable design objective. If this criterion is met, damping requirements for the secondary structure need not be stringent. This approach has had wide application to equipment that involves combinations of elastic structures.

The natural frequency for an elastic system with two degrees of freedom may be determined from

$$\omega_f^2 = \frac{1}{2}\left[\left(\frac{k_1 + k_2}{m_1}\right) + \frac{k_2}{m_2} \pm \sqrt{\left(\frac{k_1 + k_2}{m_1}\right)^2 + \frac{2k_2(k_2 - k_1)}{m_1 m_2} + \left(\frac{k_2}{m_2}\right)^2}\right] \qquad (1)$$

By using the substitutions

$$C = \frac{k_2/m_2}{k_1/m_1} = \left(\frac{f_2}{f_1}\right)^2 \quad \text{and} \quad \omega = 2\pi f$$

$$\frac{f}{f_1} = \left\{\frac{1}{2}\left[1 + C + C\frac{m_2}{m_1} \pm \sqrt{(C - 1)^2 + C\frac{m_2}{m_1}\left(C\frac{m_2}{m_1} + 2C + 2\right)}\right]\right\}^{\frac{1}{2}} \qquad (2)$$

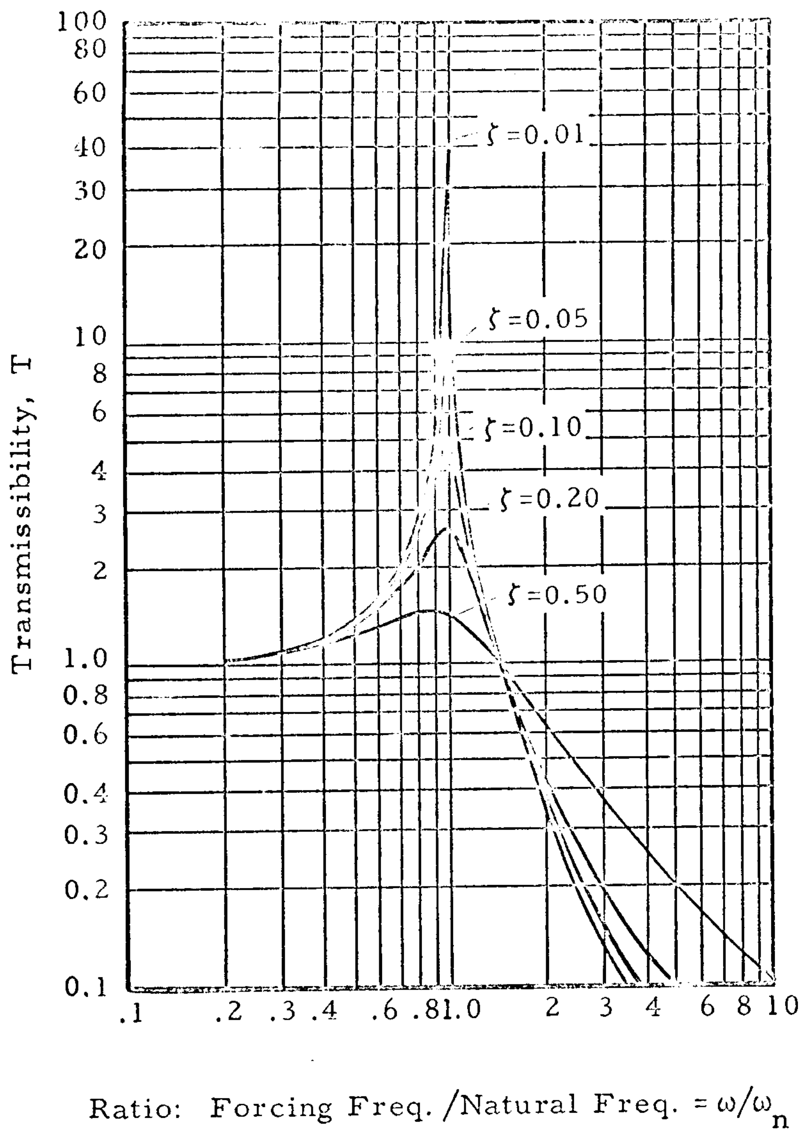


Figure 2.3 - Transmissibility of a Viscous-Damped System (Force transmissibility and motion transmissibility are identical numerically. The fraction of critical damping is denoted by ζ)

This equation has two rational roots, thus implying two system natural frequencies for each particular bridge and suspension platform natural frequency ratio. However, usually only the lower frequency is the frequency of interest. This equation is plotted in Figure 2.4 in terms of dimensionless parameters. The figure is best described by a practical example. If the natural frequency of the suspension platform (f_2) is twice that of the bridge (f_1), then $C = 4$. Following the lower $C = 4$ line, it may be seen that the system frequency (f) approaches the bridge frequency (f_1) as the mass ratio decreases. The purpose of this figure is to help select the additional frequency ratio factor needed to assure that the total system natural frequency is out of the forcing frequency range. Assuming $C = 4$ for the reasons stated above and based on preliminary weight estimates, then if the mass ratio is .15 or less, then $f/f_1 = .92$. This is commonly called the system degradation ratio.

Figure 2.3 illustrates the transmissibility, or multiplication factor, for a single-degree-of-freedom viscous-damped system. Every structure has some inherent damping, and assuming a small nominal value such as 1% of critical damping, it is seen that if the frequency ratio is kept to .7 or less, the transmissibility factor is 2 or less. Coupled with the maximum deflection anticipated, this appears an acceptable criteria.

Using these factors in combination, then the design frequency

$$f_1 = \frac{\text{System Forcing Frequency}}{(\text{Transmissibility Ratio}) \times (\text{System Degradation Ratio})} \tag{3}$$

$$= \frac{5}{.7(.92)} = 7.8 \text{ Hz}$$

Therefore, adding a safety margin, the design frequency of the bridge is established as 8 Hz and the suspension platform twice that, or 16 Hz. In addition, the frequency ratio ($f/f_1 = 1.6$) is also below the upper $C = 4$ bound and eliminates any possible secondary amplification. As a summary of the

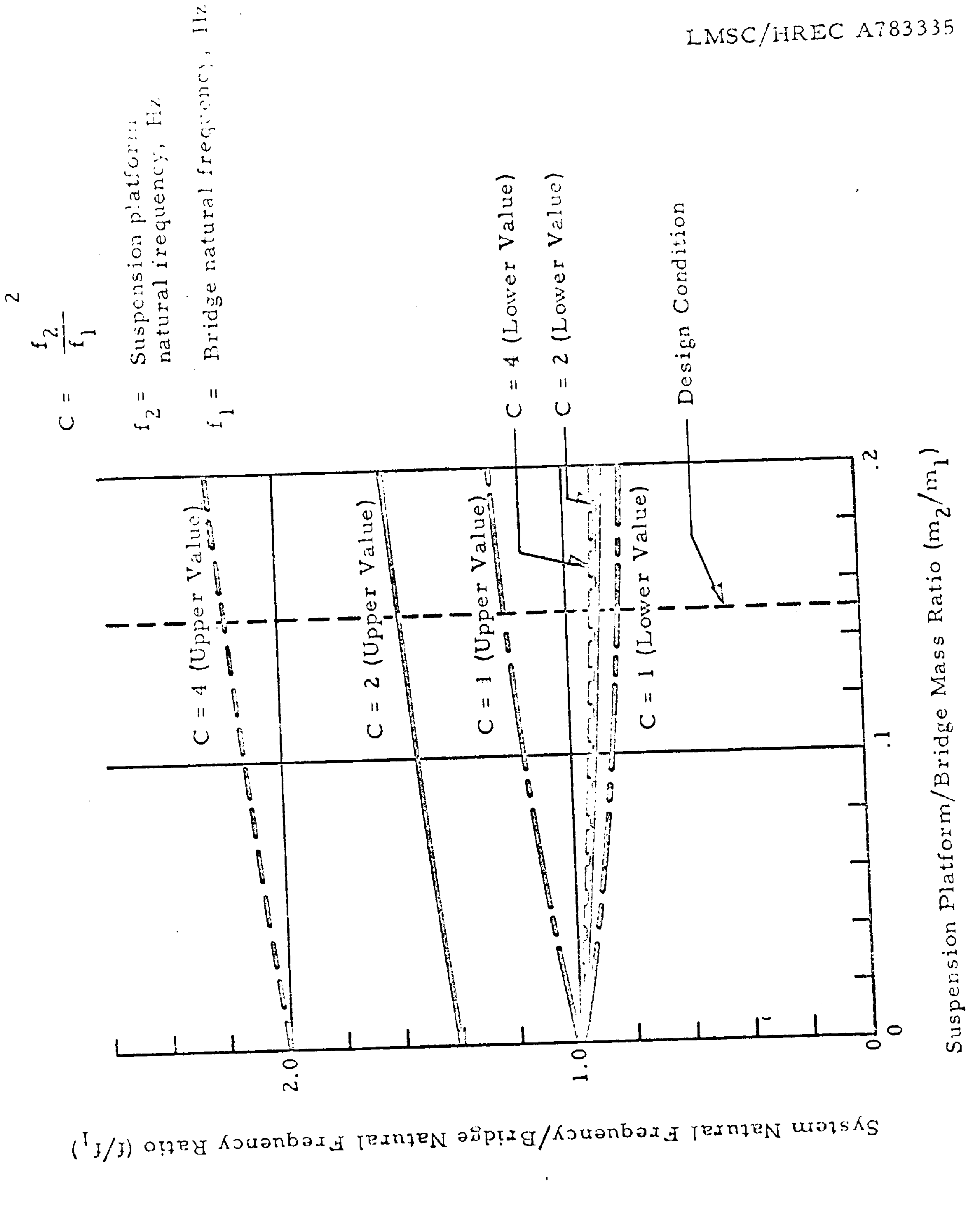


Figure 2.4 - Frequency Ratios for Mass and Deflection Ratios of a Two-Degree-of-Freedom System

foregoing calculations, the established design criteria are as follows:

Suspension platform design frequency (16 Hz)
Bridge design frequency (8 Hz)
Bridge - platform natural frequencies (7.4 and 17.4 Hz)
System forcing frequency (5 Hz).

It has been assumed, to this point, that the suspension platform mass is small relative to the bridge weight. The weights of each shall now be determined to prove this assumption was correct.

2.2.2 Suspension Platform Weight

The previous weight summary in Reference 2 for the vehicle suspension platform was based on the two-dimensional LGS and had a minimum natural frequency of 10 Hz. However, as shown in the previous section, the suspension platform should have a natural frequency of 16 Hz. Larger structural members and cross members were added, increasing the stiffness, to meet the frequency requirement.

An additional section of the platform was added to provide yaw capability and lateral translation. This configuration is illustrated in Figure 1.1. The structure consists of welded aluminum channels and beams attached to a large-diameter yaw bearing structure. This structure is then attached to the lateral translation trolleys to give a total suspended weight (m_2) of approximately 4500 pounds.

2.2.3 Bridge Design

Considering the bridge to be a uniform beam with a concentrated load, the effects on the natural frequency may be determined by the following equation from Reference 3, Section 42-23;

$$\omega^2_n = \frac{\frac{d}{\ell^2}\sqrt{\frac{E}{u}}}{\delta} \quad \text{radians per second} \tag{4}$$

where

ω'_n = natural frequency, rad/sec (= $2\pi f$)

d = radius of gyration of cross section, in. (= $\sqrt{I/A}$)

E = Young's modulus, lb/in.^{2}

u = mass density of beam material, lb-sec^2/in.^{4} (= ρ/g)

ℓ = length of beam, in.

δ = dimensionless factor which accounts for beam end fixity and mass ratios. For a simply supported beam with a mass in the center,

$$\delta = \sqrt{0.0130 + 0.0208\, m_1/m_2} \tag{5}$$

m_2 = mass of platform, lb-sec^2/in.

m_1 = total mass of bridge, lb-sec^2/in.

Substitution and recombining the values, the following equation is derived:

$$I_{reqd} = \frac{\omega^2 \ell^3}{Eg} (0.013\, W_{beam} + .0208\, W_{load}) \tag{6}$$

A number of decisions must be made at this point; (1) the type of material, (2) the unsupported length of the bridge, (3) the natural frequency, and (4) the cross-sectional area of the structure. Each of these items is discussed below.

- Type of Material: Conventional fabricated steel girder construction was chosen for the bridge rather than welded aluminum trusses because (1) the engineering and drafting costs would be much less for a conventional design, (2) costs of materials and fabrication would be less since a large aluminum structure must undergo a development program, (3) the factor $\sqrt{E/u}$ is virtually the same for steel and aluminum; therefore, the use of aluminum would not result in an appreciable weight savings, and (4) the need for engineering and development of the aluminum structure would require an extremely long delivery schedule.

- Length of Bridge: The bridge ideally should be long enough to allow all the various lunar surface vehicles room enough for a full 180° turn. The turn radius and width of each vehicle will be used as a guide for the minimum bridge span.

The Bendix MOLAB concept has an effective turn radius of 40 feet in both hard and soft soil on a 0° slope at 5 mph. (These data are from Figures B-21 and B-22 of Reference 4). This radius should be adequate. The vehicle is reported as 14 feet, 8 inches wide. This would require a minimum bridge width of 95 feet.

The Bendix LSSM Concept turn radius is dependent upon the final design steering system. Quoting from Reference 5, "The turn radius of the present design is 16.2 m, based on single axle Ackermann steering and it may be decreased to 8.5 m if double Ackermann steering is used; it is anticipated that increase of wheel steering angle above 15° is possible which will shorten the turn radius of single axle steering to 12 m." Using the vehicle width of 7 feet, 8 inches in combination with the above turn radii, we get 112 feet, 8 inches for the present design, 64 feet for the double Ackermann steering and 86 feet, 6 inches for the improved steering angle.

The Boeing MOLAB Model 944-004 performance summary gave a 7.2 m (23.5 ft) minimum turn radius at a speed of 10.8 km/hr (6.7 mph). This gives a total width of 48 feet, 5 inches.

The Boeing LSSM six-wheel vehicle minimum turn radius was calculated from the maximum wheel turn angles and wheelbase. This is approximately a 19-foot radius with a vehicle width of 8 feet, 10 inches, or the needed turn width is 47 feet.

In summary, the bridge span should be at least 95 feet to accommodate all the present vehicle concepts (except for the Bendix LSSM present design). To allow some tolerance for turns, we will use 100 feet as the design length for the bridge.

- Natural Frequency: The design frequency is 8 Hz, as determined in a previous section of this report.
- Cross-Sectional Area: In Equation (6), if $\rho A_t \ell$ is substituted for W_{beam}, where A_t = bridge cross-sectional area and ρ is density;

$$I_{reqd} = \frac{\omega^2 \ell^3}{Eg} (.0103\ \rho A \ell + .0208\ W_{load})$$

$$\omega = 2\pi f = 2\pi(8) = 50.24 \text{ rad/sec}$$

$$\ell = 100 \text{ ft} = 1200 \text{ in}$$

$$E = 29 \times 10^6 \text{ lb/in}^2$$

$$\rho = .283 \text{ lb/in}^3$$

$$W_{load} = 4550 \text{ lbs}$$

$$g = 386 \text{ in/sec}^2$$

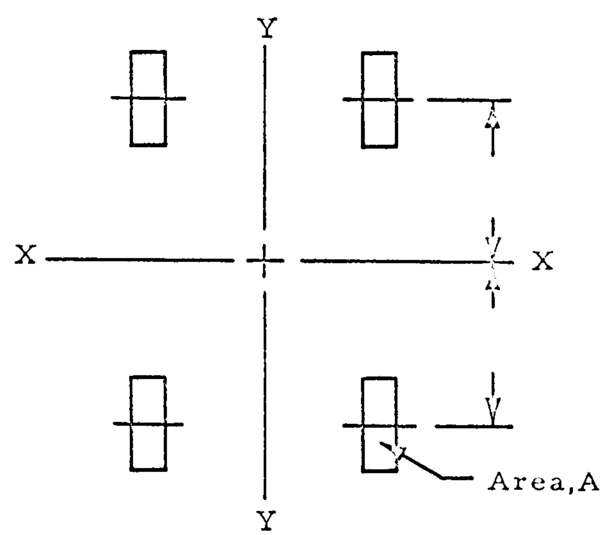
$$I_{reqd} = 36{,}890 + 1{,}364\, A_t \tag{7}$$

At this point, a distinction must be made between the cross-sectional areas which contribute to the bridge weight and that which has effective lateral stiffness. Calculations based on a number of open web and closed web beams show that from 30 to 60% of the structure does not contribute to the structural stiffness. Therefore, we will assume that the gross area is 1.5 times that of the net area, or $A_t = 1.5\, A_{net}$.

Now $$I_{reqd} = \sum_{i=o}^{i} (I_{o_i} + A_i d_i^2) = I_{o_i} + \sum_{i=o}^{i} A_i d_i^2$$

Since we know that the structure must be quite deep, we may assume that ΣI_o is small. A further assumption is that the effective masses are concentrated at the far corners, as shown below.

$$\therefore \Sigma A_i d_i^2 = 4 A_i d^2 = A_{net} d^2 = A_{net} d^2 = I_{reqd}$$



Finally, $I_{reqd} = A_{net}d^2 = 36{,}890 + 1{,}364\ (1.5)\ A_{net}$ (8)

$$A_{net}d^2 = 36{,}890 + 2{,}046\ A_{net}$$

$$d^2 = 2{,}046 + \frac{36{,}890}{A_{net}} \tag{9}$$

It can be seen from Equation (9) that the net area and the depth of beam are interrelated. Table 2.2 tabulates the values for various cross-sectional areas.

Table 2.2

BRIDGE WEIGHT HEIGHT AND MOMENT OF INERTIA

A_t ($in.^2$)	A_{net} ($in.^2$)	d (in.)	W_{beam} (lb)	$I_{x\ reqd}$ ($in.^4$)
30	20	62.4	10,188	77,810
60	40	54.5	20,376	118,730
90	60	51.6	30,564	159,650
120	80	50.0	40,752	200,570
150	100	49.1	50,940	241,490
180	120	48.5	61,128	282,410

The increase in net cross-sectional area above 40 square inches is of diminishing value. However, we will select the value for $A_{net} = 60\ in.^2$ for a bridge weight of 30,564 pounds in order to meet the initial assumption that $W_{load}/W_{beam} = 4{,}550/30{,}560 = .15$, and to provide an adequate safety factor.

A 12 x 15 standard channel gives a cross-sectional area of $10.20\ in.^2$ and was selected as the basic corner structure. This gives a maximum bridge beam height of h = 2d + 12 or 115 inches.

2.2.4 Bridge Structural Analysis

The bridge is subjected to a torsional moment caused by the horizontal trolley force (F_d) as described in Section 2.1.1. The angle of twist is calculated from $\theta = M\ell/GI_p$. The value of I_p, the polar moment of inertia, was determined by the normal assumption that the beam be designed for a lateral force equal to 1/4 of the vertical force. This would require a section modulus of $I_z = At^2 = 1/4\ Ad^2$ (consequently $t = 1/2\,d$) in order to maintain the same natural frequency. Now $I_p = I_x + I_z = 1.25\ I_x$ where $I_x = 159{,}650\ in.^4$ from Table 2.2. The moment $M = F_d h$ where h is the distance from the short track trolley to the centroid of the beam. The value of h is 90 inches, as determined in the foregoing section. The value of torsional modulus (G) for steel is $12 \times 10^6\ lb/in.^2$. Therefore,

$$\theta = \frac{F_d h \ell}{G I_p} = \frac{(5400\ lb)(90\ in.)(1200\ in.)}{12(10^6)\ lb/in.^2\ (159{,}650\ in.^4)}$$

$$\theta = .00030\ \text{radians, (or } .02^\circ).$$

Therefore, there is very little torsional distortion of the bridge.

Checking the shear stress,

$$S_s = \frac{T_r}{I_p}$$ where $T = F_d h$, and r is the distance to the outermost fiber. In this case, this is a corner of the beam and

$$r = \sqrt{(d + 1/2\,a)^2 + (t + 1/2\,b)^2}$$

where a and b are the height and width respectively of the corner structural member. Using the 15x3-3/8 channel mentioned previously.

$$r = \sqrt{(51.6 + 7.5)^2 + (25.8 + 1.68)^2} = 73.4\ in.,$$

$$S_s = \frac{(5400)(90)(73.4)}{159{,}650} = 2{,}200\ psi$$

Finally, the bending stress must be checked

$$S_b = \frac{Md}{I} = \frac{(1/2\ W_{load} + 1/4\ W_{beam})d}{4 I_{channel}} = \frac{9860(51.6)}{4(10.20)} = 12{,}450\ psi$$

Therefore, the beam is adequate for all principal stresses.

2.2.5 Recommendations

The following recommendations are made as a result of the structural analysis:

1. If the suspension platform mass is small in relation to the bridge mass, the continuing periodic vibration of the bridge may be considered a steady-state vibration for the suspension platform.
2. The bridge natural frequency should be kept well above the highest forcing frequency of the system to avoid deleterious force and deflection magnification.
3. If the suspension platform frequency is twice that of the bridge, internal damping of the system may be assumed to be adequate.
4. It must be pointed out that the calculations made above are based on theoretical systems. A detailed analysis cannot be made until a particular building is selected and the bridge crane existing in it has been thoroughly examined. The results of such an examination is the only means of determining the full extent of a structural redesign or the necessity for replacing the bridge.

2.3 CONTROL SYSTEM ANALYSIS

2.3.1 Control Requirements

The control tasks encountered in the 3-D configuration to keep arbitrarily moving LSV's under lunar gravity conditions at all times can be generally stated as follows (See Figure 2.5):

1. Keep the center of the suspension platform vertically aligned with the LSV center of gravity (LSV - c.g.). and accomplish this without requiring excessive short-track displacements, ξ.
2. Keep the short track and the LSV longitudinal axis aligned.
3. Keep the logitudinal axis of the trailer-platform aligned with the logitudinal axis of the trailer.
4. Provide constant 5/6 g vertical support of each of the LSV primary masses (chassis, wheels, etc.).

The 3-D configuration will use a drive to obtain the same fast response in the direction of the LSV's longitudinal axis, as required in the 2-D configuration. However, since maximum displacement (ξ) along this short-track (Figure 2.5) is only ± 2 feet in the 3-D configuration, a piston-cylinder drive is selected for its simplicity over the hydraulic motor selected for the 2-D configuration. The 3-D LSV suspension system and accompanying force control system will be identical to that described for the 2-D system (Sections 2 and 5 of Reference 2).

The short-track and trolley can pivot with respect to the bridge structure. This pivotal motion, Ψ, has to be controlled by a trolley yaw drive system (Ψ-drive). The lateral motion of the short-track center along the bridge (Y-coordinate) must also be controlled by a servo system, denoted as the lateral trolley drive (or Y-drive) system. Bridge position, X, is controlled by drive systems at each bridge end (X-drive). An additional control system is required to keep the trailer-platform aligned with the LSV trailer and will be described in a later section.

Six different types of subsystems, as listed below, are thus required to control the 3-D version:

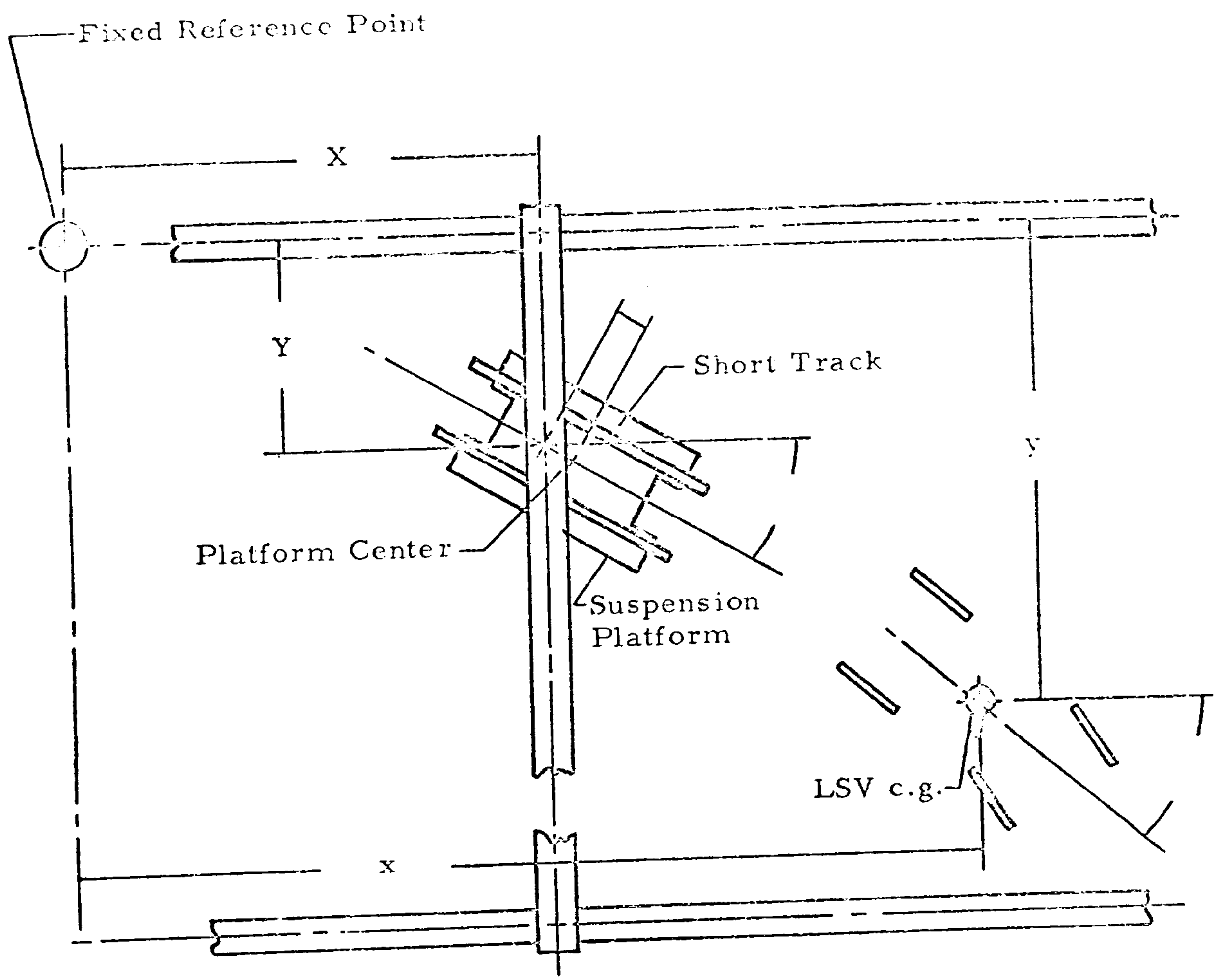


Figure 2.5 - Three-Dimensional System Coordinate Notations

- Cable force control system (one per cable) (same as in 2-D system)
- Short-track trolley (ξ-) drive system
- Short-track - yaw (ψ-) drive system
- Lateral trolley (Y-) drive system
- Bridge (X-) drive system
- Trailer - Platform (ψ_T-) drive system

Their conceptual designs will be outlined in subsequent sections.

2.3.2 Recommended Control Schemes

In an effort to minimize sensor requirements and interference problems for the various coupled control systems, the following control strategy is recommended for positioning the suspension platform. In the general case shown in Figure 2.6 where the LSV-c.g. (point L) is out of vertical alignment with both short-track pivot (point P) and the suspension platform center 0, the control goals for each subsystem are:

ξ-drive: $$e_\xi = e_X \cos\Psi + e_Y \sin\Psi - \xi \longrightarrow 0 \tag{10}$$

(Move platform center to $0'$)

Ψ-drive: $$e_\Psi = \psi - \Psi \longrightarrow 0 \tag{11}$$

(Align short-track with LSV-longitudinal axis, see Figure 2.5)

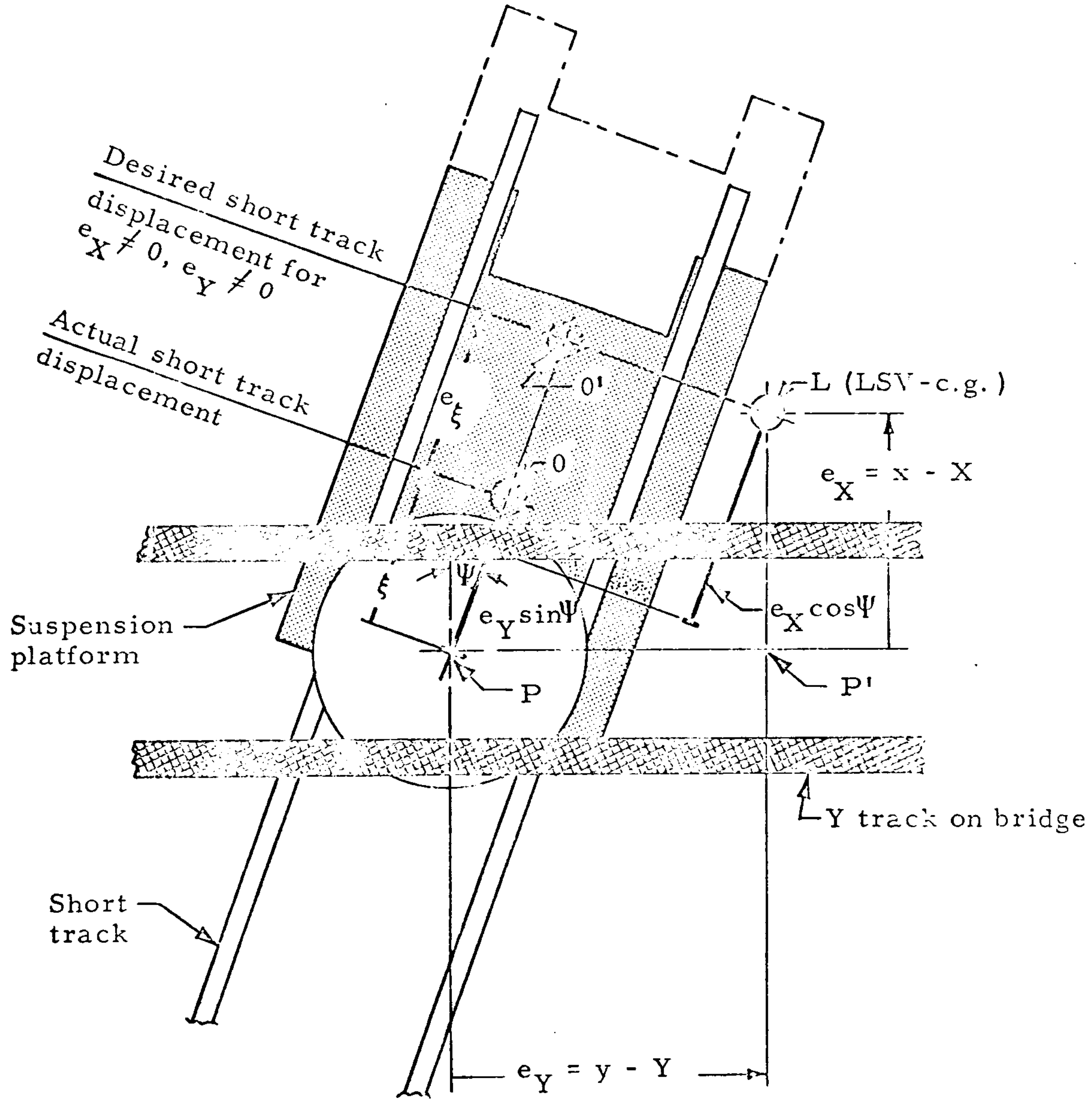
X-drive: $$e_X = x - X \longrightarrow 0 \tag{12}$$

(Move pivot point P to P')

Y-drive: $$e_Y = y - Y \longrightarrow 0 \tag{13}$$

(Align P' with L)

where: e = displacement error term



Plan View of General Platform Position

Figure 2.6 - Sketch Showing Relationship of Control Errors

This strategy ensures that the platform center is driven toward the LSV-c.g. as closely as possible by the fast-response ξ-drive during transients, while under steady-state conditions (V_{LSV} = const) short-track displacements ξ are zeroed.

The control scheme for alignment of trailer and trailer-platform is based upon

$$\psi_T - \Psi_T \rightarrow 0.$$

The most straightforward method of control is to sense directly the error angle (ψ_T) between LSV-main chassis and LSV-trailer by a potentiometer and use the signal to control the trailer platform angle Ψ_T by a hydraulic actuator as sketched in Figure 2.7. The corresponding block diagram is shown in Figure 2.8.

2.3.3 Minimum Sensor and Resolver Requirements

While two potentiometers are sufficient to control the trailer platform yaw motion, a total of 5 variable signals is required to control bridge, short-track, and trolley positions. From Equations (10) to (13) it is seen that signals proportional to ξ, Ψ, $\psi - \Psi$, x - X and y - Y are necessary to compute the four error signals e_ξ, e_Ψ, e_X and e_Y. Two optical sensors mounted along the LSV pitch axis (Figure 2.9) provide signals proportional to $\psi - \Psi$, x - X, and y - Y. Each of the two optical sensors mounted under the suspension platform senses the relative horizontal x- and y- displacement between the sensor and its light source which is mounted on the LSV pitch axis. The optical sensing system is described in detail in 2.3.4.

The following notations are used:

Right sensor signal for x-direction: u_R

Right sensor signal for y-direction: v_R

Left sensor signal for x-direction: u_L

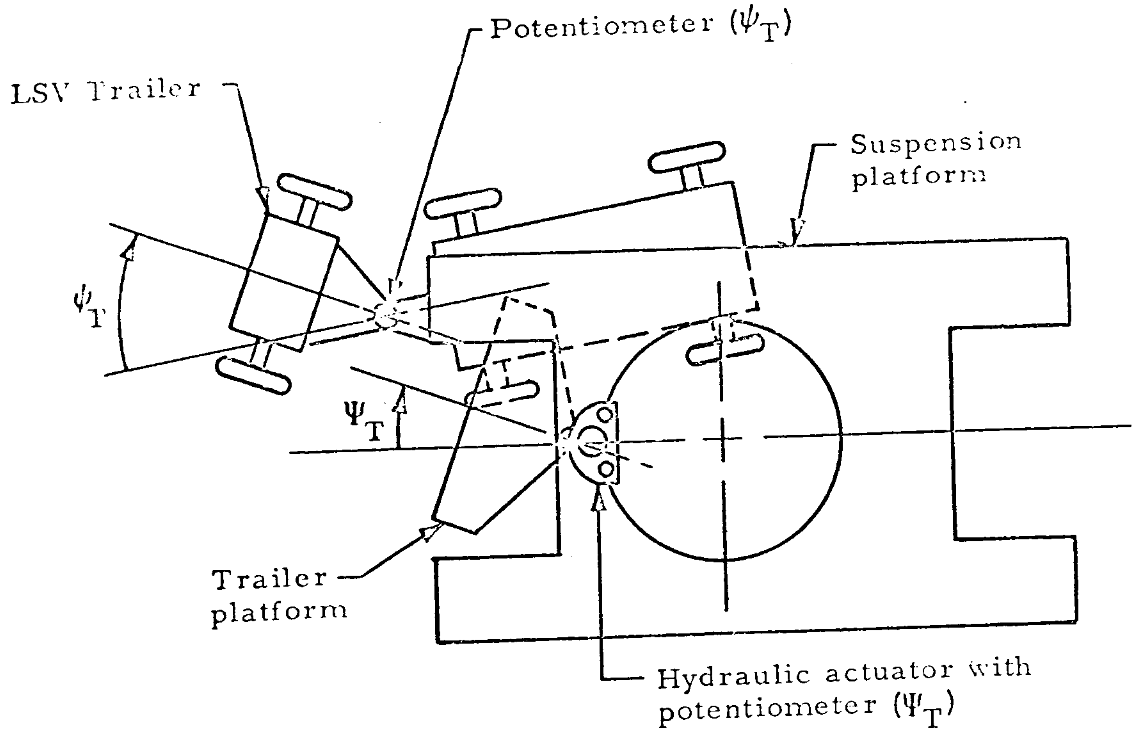


Figure 2.7 - Notations for Trailer Platform Yaw Control (top view)

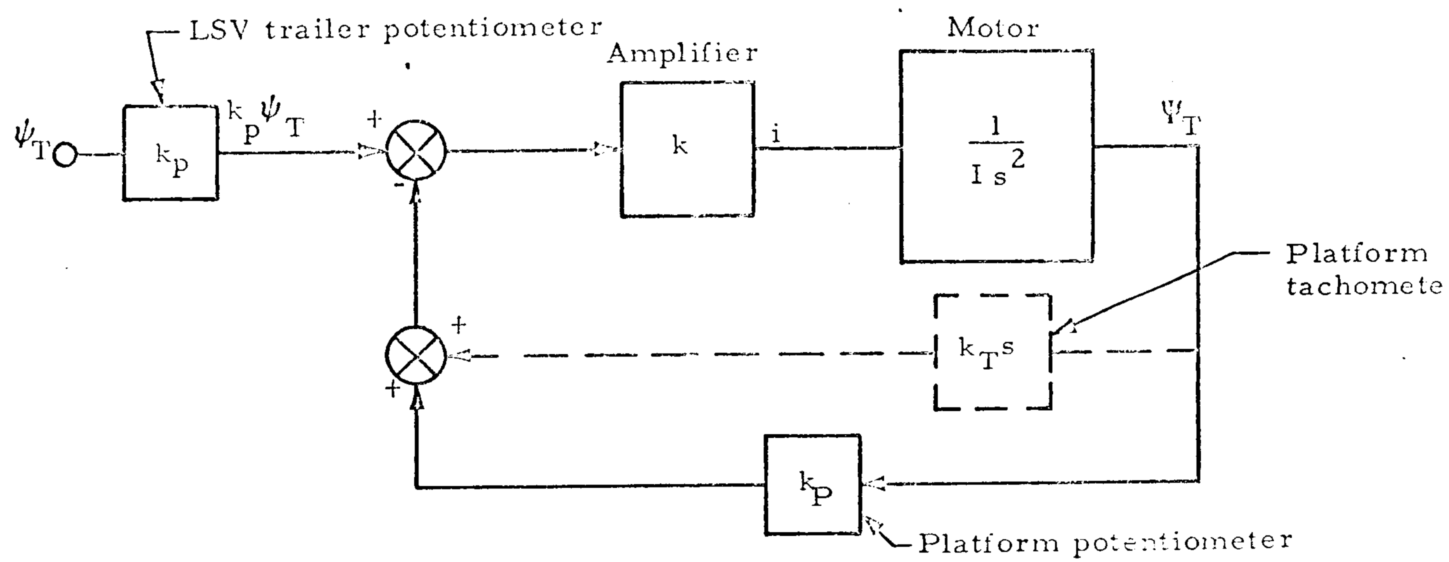


Figure 2.8 - Block Diagram for Trailer Platform Yaw Control

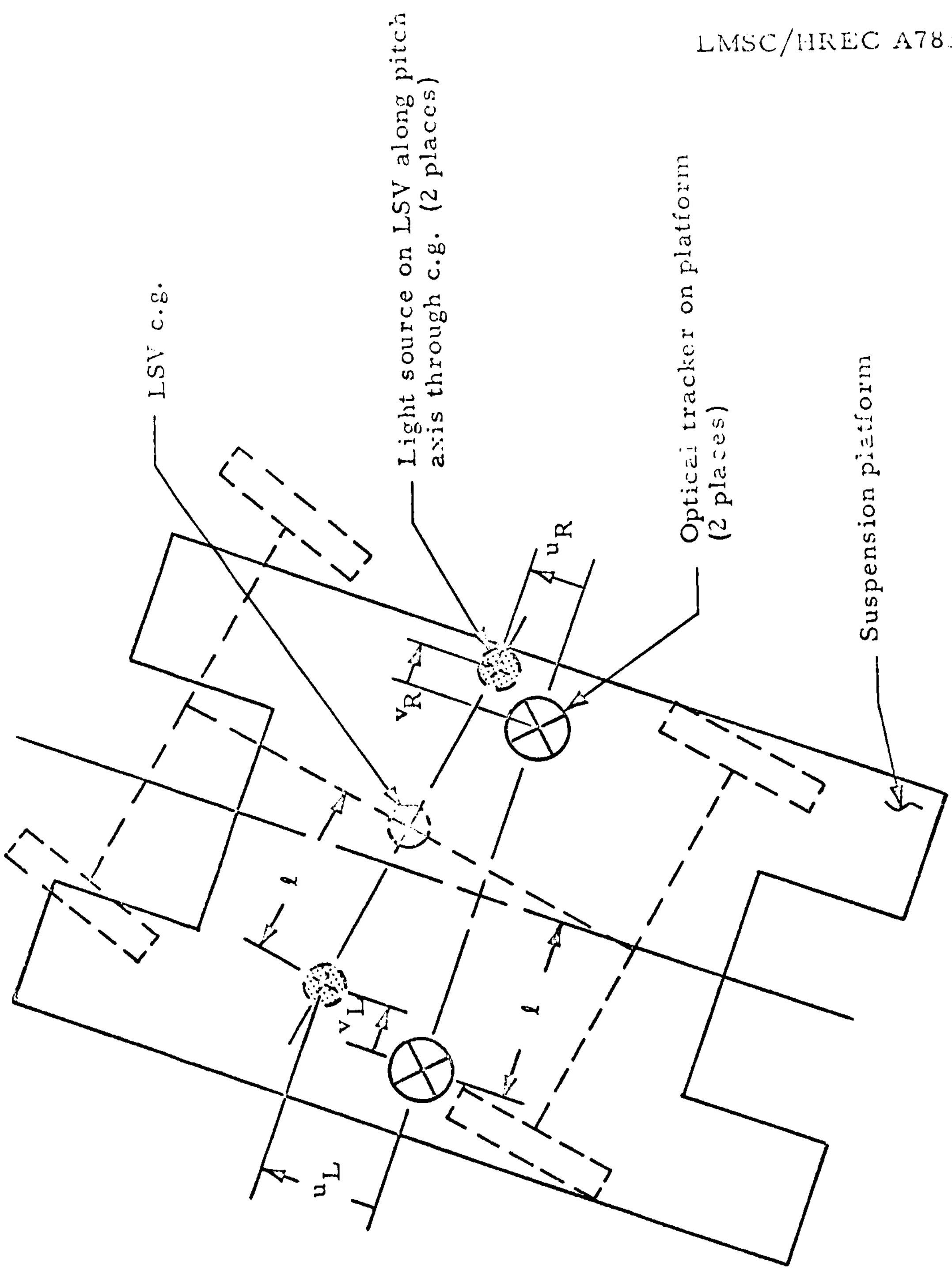


Figure 2.9 - Top View of Sensor Configuration

Left sensor signal for y-direction: v_L

Assuming that the sensors are linear, the outputs are

$$u = k' \cdot \frac{\text{displacement}}{\text{cable length}} , \qquad \text{where } k' \text{ is in volts.}$$

or $\quad u = k \cdot \text{displacement}, \text{ where } k = \frac{k'}{L}, \text{ in volts per foot.}$

Referring to Figures 2.5 and 2.6, the sensor outputs may be expressed in terms of the desired variables. In the 3-D case with roll (ϕ), yaw (ψ), and translational (x and y) motions of the LSV these expressions are:

$$u_R = k\left[x - X - \xi \cos\Psi - \ell \cos\phi \sin(\psi - \Psi)\right] \tag{14}$$

$$u_L = k\left[x - X - \xi \cos\Psi + \ell \cos\phi \sin(\psi - \Psi)\right] \tag{15}$$

$$v_R = k\left[y - Y - \xi \sin\Psi - 2\ell(\sin^2\frac{\phi}{2} + \cos\phi \sin^2\frac{\psi - \Psi}{2})\right] \tag{16}$$

$$v_L = k\left[y - Y - \xi \sin\Psi + 2\ell(\sin^2\frac{\phi}{2} + \cos\phi \sin^2\frac{\psi - \Psi}{2})\right] \tag{17}$$

where:

k is a proportional voltage constant and ℓ is the half span between the optical sensors (see Figure 2.9).

Adding or subtracting the corresponding signals yields the desired error signals:

$$2ke_X \equiv 2k(x - X) = u_R + u_L + 2k\xi \cos\Psi \tag{18}$$

$$2ke_Y \equiv 2k(y - Y) = v_R + v_L + 2k\xi \sin\Psi \tag{19}$$

$$2k\ell \cos\phi e_\psi \equiv 2k\ell(\cos\phi)(\psi - \Psi) \approx u_L - u_R \tag{20}$$

Equation (20) holds for small angles $\psi - \Psi$ where $\sin(\psi - \Psi) \approx \psi - \Psi$. The $\cos\phi$-term in Equation (20) results in slight gain changes to -18% for roll angles up to $\phi = \pm 35^\circ$ which can be tolerated in the trolley yaw feedback system without extra compensation. The same is true for gain changes $k = k'/L$ due to changes in cable length $\overline{L}$. In order to obtain the desired error signals e_X, e_Y, (Equations 18 and 19) $\xi\cos\Psi$ and $\xi\sin\Psi$ have to be generated and added to the sums $u_R + u_L$ and $v_R + v_L$, respectively. The most straightforward and least expensive way to obtain these products and all additional data processing required for computing the error signal and the velocity-feedback signals

$$\dot{e}_\xi = \dot{e}_X \cos\Psi - e_X \sin\Psi + \dot{e}_Y \sin\Psi + e_Y \cos\Psi - \dot{\xi}, \tag{21}$$

$$2k\dot{e}_X = \dot{u}_R + \dot{u}_L + 2k\dot{\xi} \cos\Psi - 2k\xi \sin\Psi, \tag{22}$$

$$2k\dot{e}_Y = \dot{v}_R + \dot{v}_L + 2k\dot{\xi} \sin\Psi + 2k\xi \cos\Psi, \tag{23}$$

and

$$2k\ell \cos\phi \dot{e}_\Psi \approx \dot{u}_L - \dot{u}_R, \tag{24}$$

is to mount a package of six sine-cosine-potentiometers on the short-track pivot shaft. The required resolver circuit is shown in Figure 2.10 where a linear pot is added to provide a signal proportional to the platform yaw angle (Ψ), for display at the control console. The input signals $2k\xi$ and $2k\dot{\xi}$ can be generated by a potentiometer and tachometer driven by the short-track platform motion. The rate signals $\dot{u}_R$, $\dot{u}_L$, $\dot{v}_R$, $\dot{v}_L$ are obtained by differentiating the optical sensor outputs.

This results in the overall sensor and resolver requirements for 3-D position control of the suspension platform as listed in Figure 2.11.

A block diagram of the combined subsystems and their main disturbances is shown in Figure 2.12.

In a detailed analysis, careful consideration will be given to keep bridge vibrations at a minimum by selecting the closed-loop control frequencies far enough from the bridge's first bending mode which is predicted to be around 8 Hz.

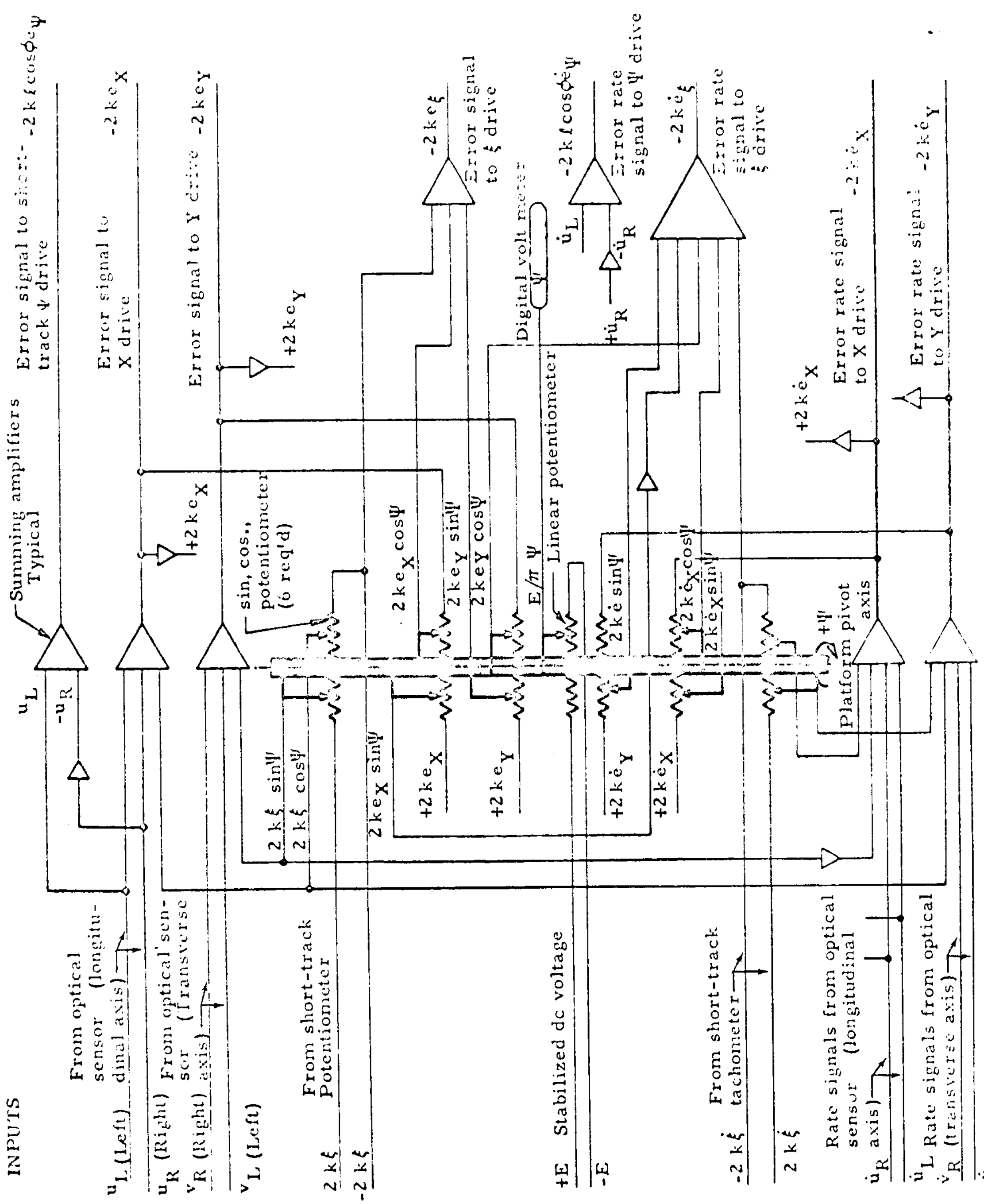


Figure 2.10 - 3-D LGS Analog Resolver Circuit

Number	Sensor (Resolver Component)	Mounted on	Input	Output	Required for
1	Right optical sensor	Suspension platform	$(x-X)/L$ $(y-Y)/L$ ψ/L	u_R, v_R	Bridge, short track, trolley position and yaw control
2	Left optical sensor	Suspension platform	$(x-X)/L$ $(y-Y)/L$ ψ/L	u_L, v_L	Bridge, short track, trolley position and yaw control
3	ξ-potentiometer	Short track	ξ	$k\ell\xi$	Bridge, short track and trolley position control
4	$\dot{\xi}$-tachometer	Short track	$\dot{\xi}$	$k\ell\dot{\xi}$	Bridge, short track and trolley position control
5...10	Sine-cosine potentiometers	Short track Pivot shaft	ψ, u, $\dot{u}$, v, $\dot{v}$, $k\xi$, $k\dot{\xi}$	e_X, e_Y, e_ξ, e_ψ $\dot{e}_X$, $\dot{e}_Y$, $\dot{e}_\xi$, $\dot{e}_\psi$	Bridge, short track, trolley position and yaw control
11	ψ-potentiometer	Short track Pivot shaft	ψ, E	$\frac{E}{\pi}\psi$	Checkout and display
12	ψ_T-potentiometer	LSV trailer Pivot shaft	ψ_T, E	$\frac{E}{\pi}\psi_T$	Trailer platform yaw control
13	ψ_T-potentiometer	Trailer suspension platform pivot shaft	ψ_T, E	$\frac{E}{\pi}\psi_T$	Trailer platform yaw control
14...30	Operational amplifiers	Resolver chassis		error signals e, $\dot{e}$	

Figure 2.11 - Sensor and Resolver Requirements for 3-D Position Control of Suspension Platforms for Main LSV and Trailer

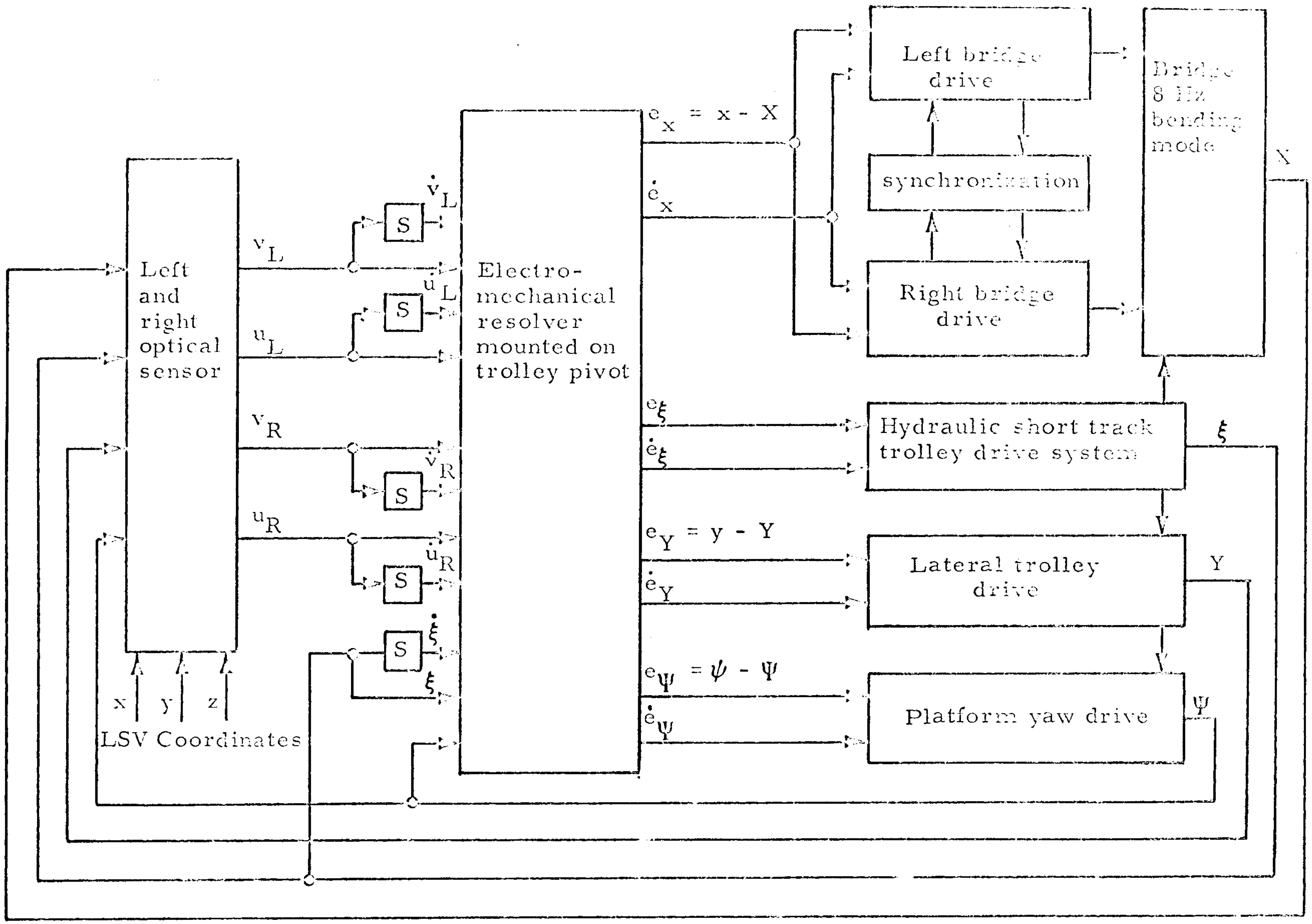


Figure 2.12 - Overall Block Diagram for Trolley Positioning Control Using Two Optical Sensors

2.3.4 Optical Tracking System

Two optical trackers mounted on the LGS trolley will provide LSV displacement and velocity information to the trolley control system.

Trackable light sources are mounted on each end of the LSV pitch axis equidistant from the roll axis (Figure 2.13a). Each light source will emit constant intensity illumination of approximately 100 lumens per steradian over its upper hemisphere. (A common 150-watt frosted light bulb would be excellent in this application.) A 20-inch diameter background of dark diffuse material (velvet or black felt) and conical lamp shade will prevent extraneous light reflections from interfering with the perception of the overhead trackers.

The optical trackers are mounted on the LGS trolley directly over the LSV light sources. Each of the trackers will provide displacement and velocity information along two axes.

When the platform and LSV are perfectly aligned (zero error condition); (1) the common axis of the trackers is parallel to the LSV pitch axis and (2) each of the two tracker axes, (perpendicular to the common axis) is parallel to the LSV roll axis. Displacement from this reference position is shown in Figure 2.9 which illustrates relative positions of the light sources and trackers, and corresponding tracker-sensed errors (u_L, v_L, u_R, v_R).

Information from these two trackers is resolved into $(x - X)$, $(y - Y)$, and $(\psi - \Psi)$ displacement and rate of displacement information which is fed into the trolley control system.

The image of each light source is focused on the photomultiplier tube of its tracker. The cathode photomultiplier tube is divided into four equal quadrants. As the LSV light source image moves from the center of the cathode to a typical position as indicated in Figure 2.13b, the corresponding voltage outputs of the tube change as shown. Enlarging the image reduces

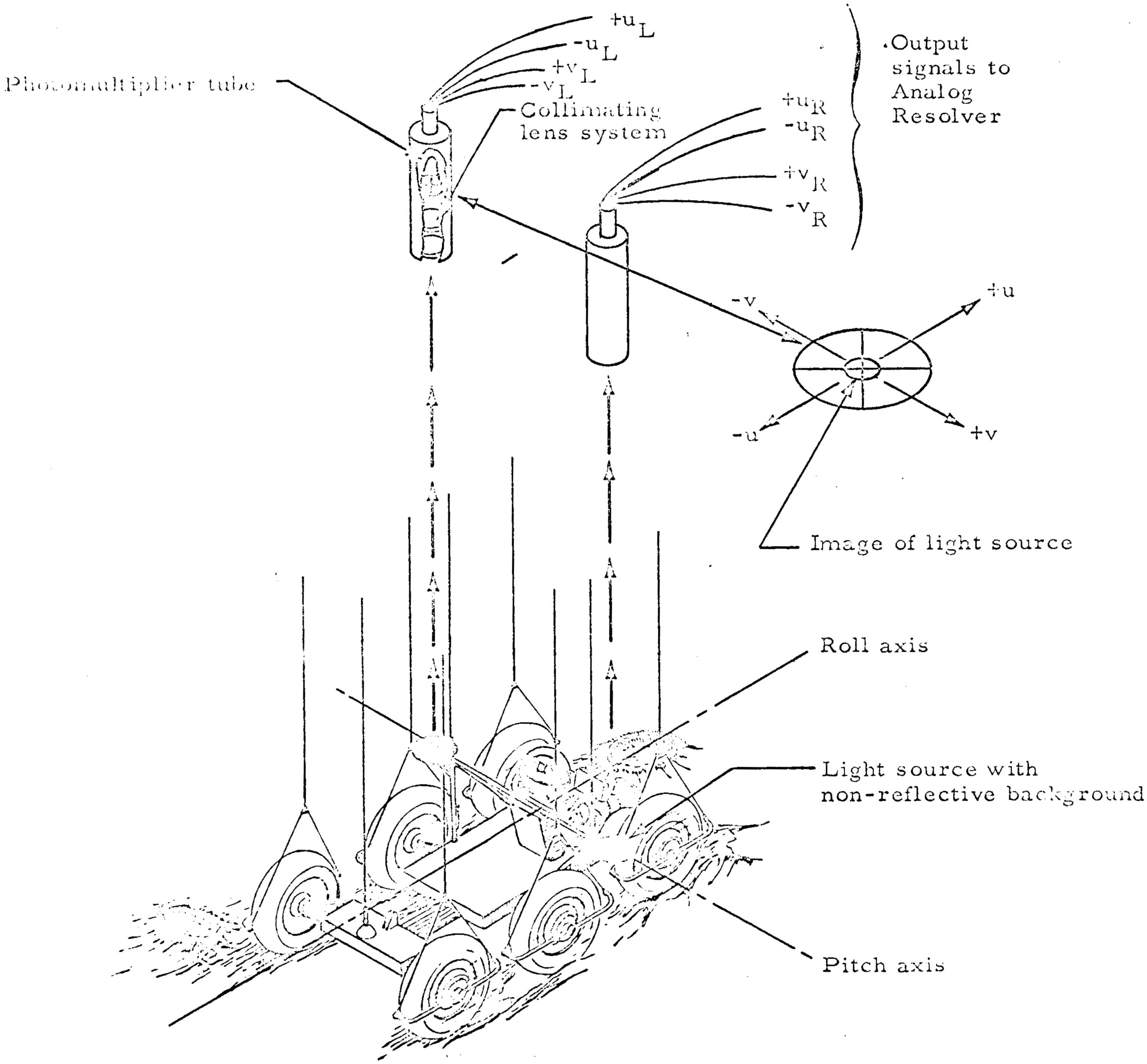


Figure 2.13a - Optical Tracking System

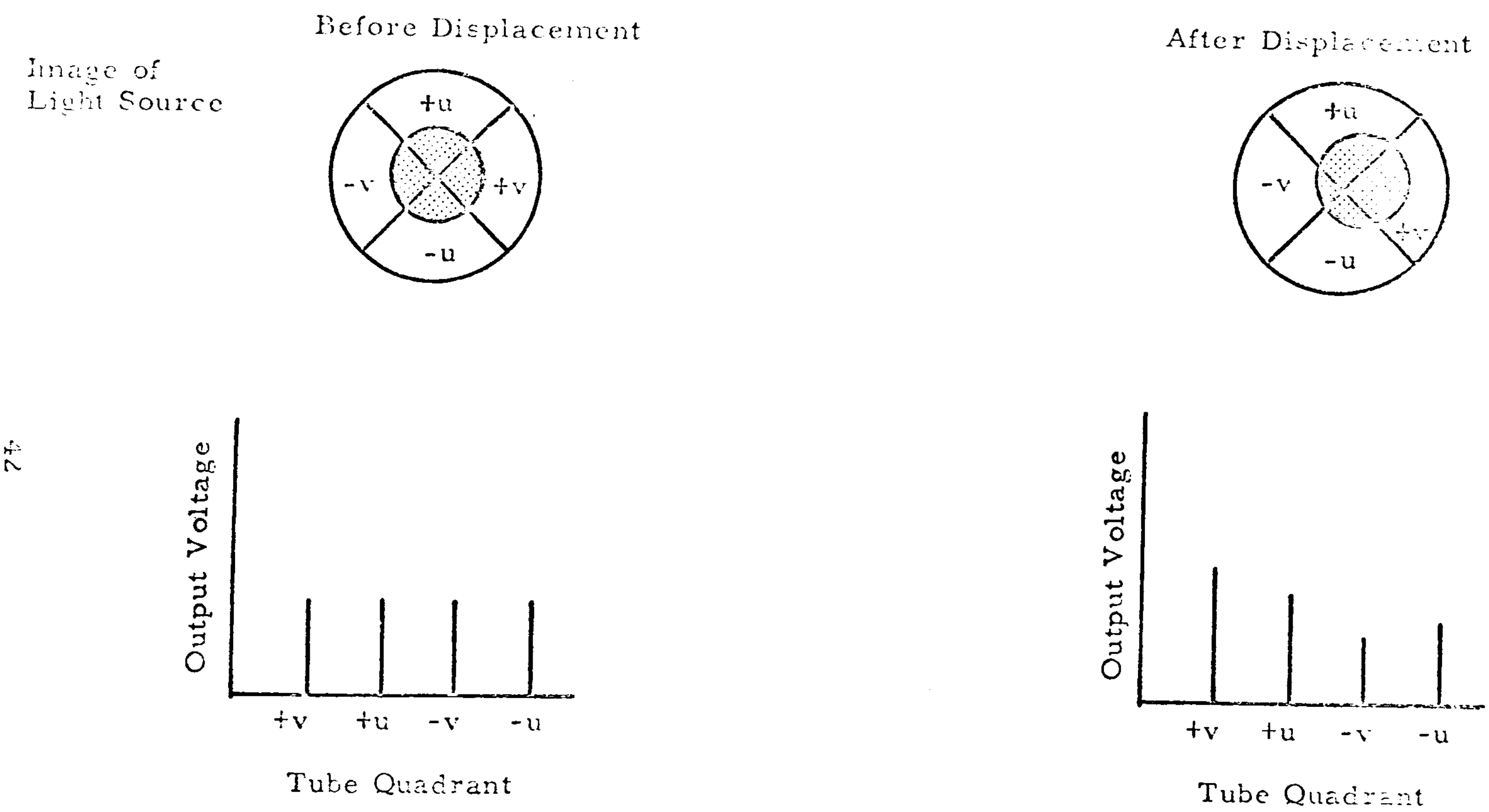


Figure 2.135 - Quadrant Photomultiplier Image and Voltage Output

the tracker field of view and increases the precision of the tracker output voltage.

The image trackers are provided with an automatic sampling system to sample voltages from each of the four quadrants. Automatic gain control compensates for variations in gain due to differences in light image intensity as the vertical distance between LSV and trolley changes.

2.4 DRIVE SYSTEM ANALYSIS

Drive systems for five separate platform movements must be provided in order to properly maintain the LSV suspension points over a moving vehicle. Appropriate combinations of these systems shall be capable of meeting the displacement, velocity and acceleration characteristics of the vehicle. Hydraulic and SCR-controlled dc motors were evaluated for use in each system. These drive systems were:

- Bridge (X)
- Lateral (Y) Trolley
- Platform Yaw (ψ)
- Platform Short-Track (ξ)
- Trailer Yaw (ψ_T)

The performance requirements, description of system and motor selection for each system is discussed in the following paragraphs.

2.4.1 Bridge (X) Drive System

The bridge drive requires the largest drive system since it carries all the LGS structure, system controls and all other drive systems. In order to minimize these drive power requirements, an auxiliary short-track (ξ) drive system is used to provide high-transient acceleration capabilities. This reduces the high accelerative forces which otherwise would be necessary by the bridge drive system.

To calculate the power requirements, the evaluation is divided into two parts; normal operation and a deceleration mode.

- Normal Operation

The maximum normal power is required when the LSV is traveling at maximum speed and is also accelerating at the maximum rate:

$$\dot{X}_{max} = 18.25 \text{ ft/sec}^2 \text{ (Reference 6)}$$

$$\ddot{X}_{max} = 0.1 \text{ g} \quad \text{(Reference 6)}$$

$$W = \text{weight of bridge} = 37{,}500 \text{ pounds}$$

$$\therefore F = 0.1\ (37{,}500) = 3{,}750 \text{ pounds}$$

$$\text{Power Required} = F\,\dot{X}_{max}/550 = 125 \text{ hp maximum}$$

Applying a 50% performance margin,

$$P_{max} = 187 \text{ hp (140 kW)}.$$

The maximum normal bridge acceleration force requirement occurs when the LSV is traveling at maximum speed in a circle of minimum radius, r. Then its full normal acceleration, a_n is directed along the bridge (X) direction of travel.

Let

$$V = \text{maximum vehicle velocity} = 15 \text{ fps (Reference 4)}$$

$$r = \text{minimum radius of turn} = 50 \text{ feet (Ibid)}$$

Then

$$a_n = V^2/r = (15)^2/50 = 4.5 \text{ ft/sec}^2 \text{ or } 0.14 \text{ g}$$

Thus, the 0.14 g is the maximum acceleration requirement of the bridge ($\dot{X}$).

Let

$$W = \text{bridge weight} = 37{,}500 \text{ pounds}$$

$$F = \text{acceleration force, pounds}$$

Then

$$F = W\,\ddot{X}/g = 37{,}500\ (0.14) = 5250 \text{ pounds}$$

Since this maximum bridge requirement is reached only at zero bridge velocity, this 5250 pound force appears as stall torque requirement on the motors. The bridge motion in this case is simple harmonic, so the large stall torque is required only transiently. However, it will be a decisive factor in final motor design selection.

- Deceleration Mode:

An important additional requirement is imposed when the following deceleration sequence is performed by the LSV:

(1) 2.0 g deceleration until velocity drops 4 fps.*
(2) 0.1 g deceleration until the vehicle stops.

Except for error transients for less than 1 second, the suspension platform trolley follows the LSV through these excursions. The required platform trolley track length aft of the trolley "rest" position is a function of the deceleration rate of the bridge. This track length requirement is greatest when the deceleration is from maximum speed (18.25 fps).

The bridge deceleration is related to required aft track length as follows: for preliminary design purposes, small transients are neglected and the suspension platform trolley simply decelerates at a 2 g rate from 18.25 fps to 14.25 fps, and then continues to decelerate at 0.1 g until the LSV stops. Meanwhile, the bridge decelerates at its maximum rate. The velocities of suspension trolley and bridge as they progress through this maximum deceleration are shown in Figure 2.14. The required aft track

*The 4 fps is set for preliminary design purposes; the 2.0 g and 0.1 g values are per Reference 6.

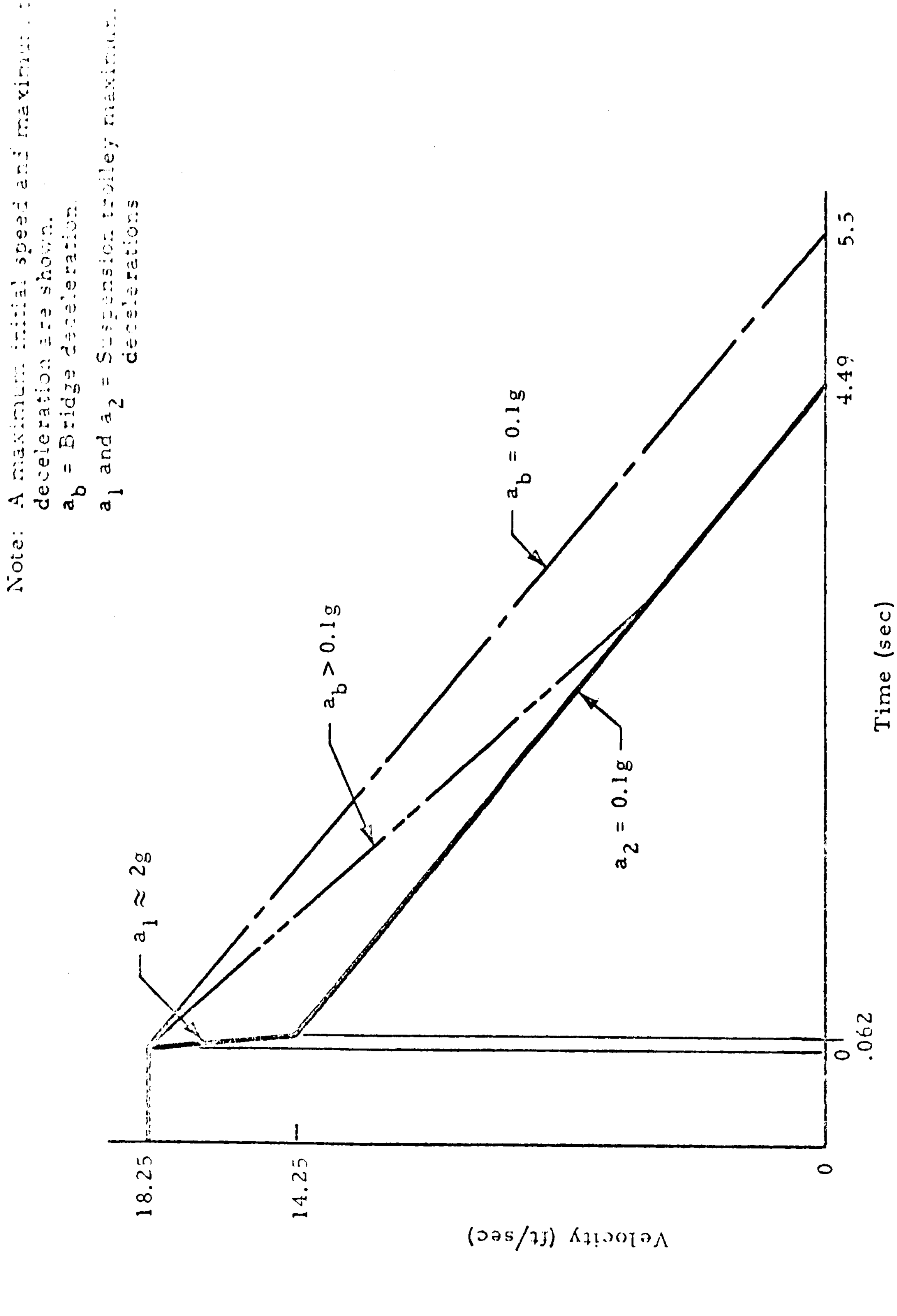


Figure 2.14 - Bridge and Suspension Trolley Velocities vs Time

is derived as a function of bridge accelerations as follows:

Let S_b = Bridge displacement, feet

S_T = Trolley displacement, feet

V_1 = Bridge and trolley initial velocity, fps

V_2 = Trolley velocity at $t = \Delta t$, fps

V_b = Bridge Velocity, fps

V_t = Trolley velocity, fps

a_1 = Trolley surge deceleration, fps^2

a_2 = Trolley steady deceleration, fps^2

a_b = Bridge acceleration, fps^2

t = Time after deceleration begins, seconds

Δt = Time duration of trolley surge deceleration, seconds

$$S_b = V_1 t - \tfrac{1}{2} a_b t^2 \tag{25}$$

$$S_t = V_1 \Delta t - \tfrac{1}{2} a_1 \Delta t^2 + V_2 (t - \Delta t) - \tfrac{1}{2} a_2 (t - \Delta t)^2 \tag{26}$$

$$V_2 = V_1 - a_1 \Delta t \tag{27}$$

Combining,

$$S_b - S_t = \tfrac{1}{2}(a_2 - a_b) t^2 + (a_1 - a_2)\Delta t(t) - \tfrac{1}{2}(a_1 - a_2)\Delta t^2 \tag{28}$$

Substituting values,

$$a_1 = 2g = 64.4 \text{ ft/sec}^2, \quad a_2 = 0.1g = 3.22 \text{ ft/sec}^2$$

$$\Delta t = \frac{V_1 - V_2}{a_1} = \frac{18.25 - 14.25}{2(32.2)} = .062 \text{ sec}$$

Equation (28) becomes

$$S_b - S_t = \tfrac{1}{2}(3.22 - a_b)t^2 + 3.80t - .118 \tag{29}$$

when

$$V_t = V_b,\ S_b - S_t \text{ is at a maximum}$$

$$t = \frac{3.80}{a_b - 3.22} \tag{30}$$

Equation (30) is valid only for time Δt or later and before either bridge or trolley has stopped. Since the trolley will stop first (or at the same time as the bridge), the time for the trolley to stop will be the upper limit of time for validity of Equation (30) (See Figure 2.14).

then

$$V_t = (V_1 - a_1 \Delta t) - a_2(t - \Delta t) = 0$$

Substituting and solving for t,

$$t = 4.49 \text{ sec}$$

From Equation (30); the limits of a_b can be calculated

$$4.06 < a_b \leq 64.4 \ \text{fps}^2$$

A final expression for $S_b - S_t$ can now be written with limits. Substituting Equation (30) into (29) and applying limits for a_b;

$$S_b - S_t = \frac{7.22}{a_b - 3.22} - 0.118 \qquad 4.06 < a_b \leq 64.4 \tag{31}$$

Converting to g's for convenience,

$$\text{Let } a = 32.2\, a_{bg}$$

where

$$a_{bg} = \text{bridge acceleration, g's}$$

Equation (31) becomes

$$S_b - S_t = \frac{7.22}{32.2\, a_{bg} - 3.22} - 0.118 \qquad 0.126 < a_{bg} \leq 2 \qquad (32)$$

Figure 2.15 was plotted from this expression. It can be seen from this figure that if the bridge deceleration is only 0.1g, an aft track length of 19 feet is required. Now, if a bridge deceleration rate twice this, or .2 g can be applied for 2 seconds, the aft track length can be reduced to 2 feet. This value was chosen as the design value.

The suspension platform translation track total length must be determined. The length aft of the trolley rest position has been established at approximately two feet. In practice, it is not expected that the vehicle will be run backward at its full forward speed; however, until this criterion is modified, a two-foot forward track will be provided in the design also. Total track length of four feet is thereby required, with the piston centered at rest position.

- Description of Drive System

Four electric motors, two on each end of the bridge, were selected to power the bridge. Each motor will drive a wheel through a speed reducer. The full load carried by the wheels must either supply adequate friction or an alternate system must be selected.

Let μ = coefficient of friction
= .15 for steel on steel

w = total weight on wheels (excluding the LSV)
= 37,500 pounds

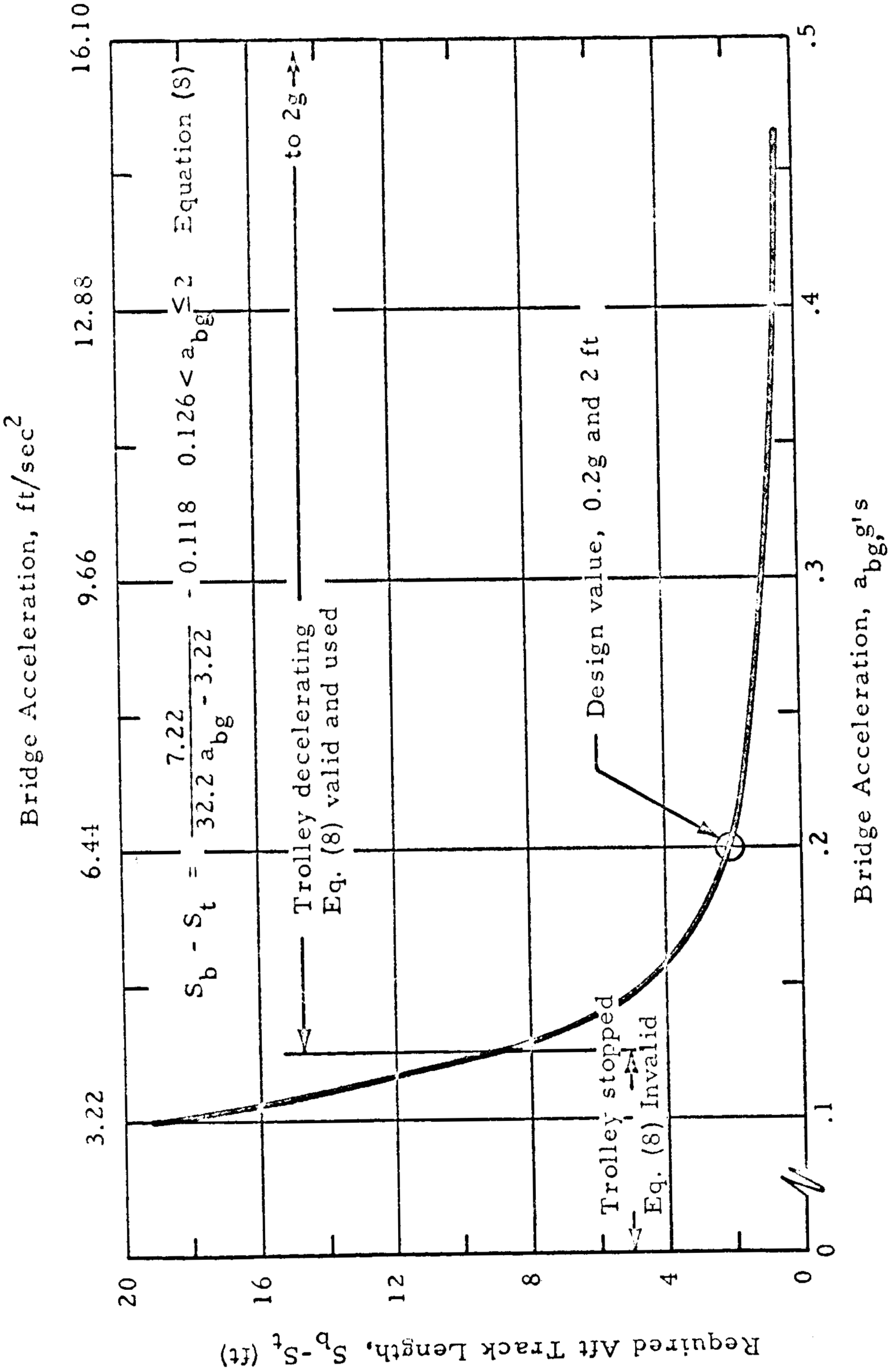


Figure 2.15 - Required Aft Track Length vs Bridge Acceleration Capability

$$F_f = \text{friction force available} = \mu w$$

$$F_f = 5{,}625 \text{ pounds}$$

However, the driving force required, F_d,

$$F_d = Ma$$

$$= \frac{37{,}500}{g} \ (0.2\ g)$$

$$F_d = 7{,}500 \text{ pounds (maximum for deceleration mode)}$$

Addition of an LSV would increase the friction force available, but it would still be inadequate. Alternate methods such as a capstan and cable, or rack and pinion drives for providing the required drive force should be evaluated.

- Motor Selection

Direct current motors will be used to power each of the four drive wheels, but the system will use a single silicon-controlled rectifier (SCR) control unit. All four motors will have armatures in series to equalize driving torque. The motor system specification would be approximately as follows:

- 4 each - Dripproof motor. 440 V, 60 Hz power. 1150 rpm, 25 hp with thermal protection. Stall torque $\gtrsim$ 1.4 times maximum running torque.
- 1 each - SCR-dc control unit for 100 hp motor. Stall current must be $\gtrsim$ 1.4 times running current. Has reversing and special dynamic braking which actuates on command deceleration hardover.
- 1 each - Resistive bank for motors in dynamic braking mode.

The basic motors will provide the short-term power required for maximum bridge acceleration, which is twice nominal rating for up to 5 seconds. The special dynamic braking will switch the armatures to a resistive bank for the critical one or two seconds during maximum deceleration. The rest of deceleration will be through regenerative braking (0.1g). The motors must therefore take about 3.5 times their nominal current during this very short period.

2.4.2 Lateral (Y) Trolley Drive System

The lateral trolley is driven along the bridge to provide an additional degree of freedom for the suspension system (see Figure 2.1). This system like the X-drive system, must provide the capability for both the normal operation and deceleration modes.

- Normal Operation

This mode of operation is identical with the bridge drive system with the exception of the accelerated mass.

$$\dot{x}_{max} = 18.25 \text{ ft/sec}$$

$$\ddot{x}_{max} = 0.1 \text{ g}$$

$W = 4425$ lb (weight of LGS less bridge weight, see Table 2.1)

$$F = 0.1\,(4425) = 442 \text{ lb}$$

$$\text{Power Required} = \frac{442\,(18.25)}{550} = 14.7 \text{ hp}$$

P_{max}, assuming a 50% performance margin = 20 hp (15 kW)

As in the bridge (X) drive, at stall condition a force of 0.14 g is required.

- Deceleration Mode

As in the bridge (X) drive, a 0.2 g trolley deceleration is required since the vehicle can travel along either axis. However, this does not change the maximum power requirement.

- Description of Drive System

Two electric motors will be used to power the transverse trolley. Each motor will drive through a speed reducer, one pair of wheels rigidly connected by an axle. One pair will be at each end of the trolley. The full load carried by the trolley will be borne on these driving wheels to assure adequate driving friction. Since the LSV weight is sizable compared to the transverse trolley weight, friction on the trolley drive wheels will be adequate.

- Motor Selection

Direct current motors will be used, one for each axle. A single silicon-controlled rectifier (SCR) unit will be used. The two motors will have armatures in series to equalize driving torque while requiring only one control. The motor system specification would be approximately as follows:

Quantity	Description
2 ea	Dripproof Motor, 440 V, 60 Hz power, 1150 rpm, 10 hp, thermal protection, stall torque ≥ 1.4 times maximum running torque.
1 ea	SCR-dc control unit for 20 hp motors. Stall current must be at least 1.4 times maximum running current. Has reversing and special dynamic braking which actuates on command deceleration hardover.
1 ea	Resistive bank for motors in dynamic braking mode.

The basic motors will provide the short-term power required for maximum lateral trolley acceleration which is twice normal rating for up to 5 seconds. The special dynamic braking will switch the armatures to a resistive bank for a critical one or two seconds during maximum deceleration. The rest of deceleration will be through regenerative braking (0.1 g). The motors must therefore take up to about 3.5 times their nominal current during this very short period.

2.4.3 Suspension Platform Rotational Drive System

This drive system (Ψ) rotates the suspension platform tracks to align with the LSV. Since LSV turning radius is large and speed of the LSV during a minimum radius turn is small, this drive system has a very low accelerational requirement. Indeed, friction is probably its primary load.

- Performance Requirements

Based on an LSV maximum speed of 15 fps at its minimum turning radius of 50 feet, and its ability to establish this turn from straight line travel, angular acceleration, α, = 0.30 rad/sec^2.

Let

I_t = total moment of inertia, lb-ft-sec^2

I_p = moment of inertia of mass which rotates under Ψ bearing (except translating parts on bottom suspension platform)

k = radius of gyration of translating parts on bottom suspension platform, ft

M_p = mass of parts on bottom suspension platform, lb-ft/sec^2

$$I_p = 2000$$

$$k = 2.5$$

$$M_p = \frac{2582}{32.2} = 80$$

$$I_t = I_p + M_p k^2$$

$$= 2000 + 80\,(2.5)^2 = 2500 \text{ lb-ft-sec}^2$$

Torque for acceleration then must be

$$T = I_t \alpha = 2500 \times .300 = 750 \text{ lb-ft}$$

Power is

$$N_{max} = \frac{T\omega}{550} = \frac{750\,(.330)}{550} = .408 \text{ hp } (.55 \text{ kW})$$

Adding 0.6 hp for bearing friction and performance margin, a value of 1.0 hp (1.34 kW) is required.

- Description of Drive System

An electric motor is geared to an internal spur gear which rotates about the Ψ-axis with the lower suspension tracks and platforms. No special overload or braking problems exist on this drive.

- Selection of Motor

A dc motor controlled by an SCR control unit will be used. The motor specification would be approximately as follows:

Quantity	Description
1 ea	Dripproof Motor. 440 V, 60 Hz power, 1150 rpm. Frame NEMA 215. 1 hp. Thermal Protection.
1 ea	SCR-dc Control Unit for 1 hp motor. Has reversing and regenerative braking.

2.4.4 Suspension Platform Short Track (ξ) Drive System

Since the suspension platform is directionally oriented with the LSV as described in Section 2.1.3, a translational drive system operable always in the direction of travel of the LSV must be provided. Such a translational system is used to follow the rapid transient accelerations of the vehicle. By specification, the average of these transient accelerations is small. Therefore, this high-performance drive system, while accelerating rapidly, never reaches near maximum vehicle velocity and moves only a short distance. Hence, it requires only a short track which simplifies its drive system as compared to the 2-D design requirements. Except for transient excursions (ξ) along this track, the suspension platform remains in a single preset position with respect to the track and support framework.

- Performance Requirements

The following performance requirements must be established:

1. Driving force and its relation to speed,
2. Maximum velocity, and
3. Maximum excursion.

F_{max} = maximum driving force on suspension platform

= 5400 pounds (See Figure 2.2.)

V_{max} = maximum velocity of suspension platform with respect to its track base, in direction of LSV path

= 6 fps (Reference 2, Figure 4-30)

From the referenced document, it was found that V_{max} and F_{max} did not occur simultaneously in the 2-D system. Likely, they will not coincide in the 3-D short track system and the transient horsepower requirement is not clearly definable. System requirements will be based upon force and speed which do not occur simultaneously.

- Description of Drive System

Two double-acting double-rod hydraulic cylinders with pistons directly driving the suspension platform direct, along its track base, were chosen. The decision was based upon the requirement for a high-force, low-inertia, relatively short stroke actuator to rapidly accelerate the 2,500-pound platform. The cylinders will be controlled by a servo valve. Hydraulic power will be derived from the power unit used by the suspension force control system (described in Section 5.6 of Reference 2). Accumulator requirements for the 2-D configuration will be modified as required to deliver the transient requirements of the short track (ξ) drive system.

- Selection of Drive Cylinder

Using a 3,000 psig system, each of the two pistons must deliver 2,700-pounds peak force.

Let

A = piston area, in.^{2}
P = total pressure, lb
S = unit pressure, psig
D = piston diameter, in.

Then

$$A = \frac{P}{S} = \frac{2{,}700}{3{,}000} = .900 \text{ in.}^2$$

$$A = \frac{\pi D^2}{4}$$

or

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times .900}{\pi}} = 1.07 \text{ in.}$$

Since the connecting rods will slightly reduce the effective area, and to provide some performance margin, a 1-1/8" diameter cylinder will be used. Piston travel as defined by the required trolley excursion length (Section 2.4.1) is 24" in each direction from rest position. A control system must be devised to provide instantaneous full power to the cylinder, yet not keep the power unit under an unreasonable strain continuously. The 3,000 psi supply was chosen to minimize system size and weight. Available flow of 5 gpm will fully cover the .275 gallons per stroke and a very roughly assumed maximum duty cycle.

- Alternate Drive System

An alternate ξ-drive system is the all-electric Unlimited Stroke Actuator offered by V. B. Actuators of Las Vegas, Nevada. Its operation is similar to a linear induction motor, but apparently delivers more force with a lighter armature. Its simplicity is especially appealing compared to the hydraulic systems which require relatively large, heavy power units and fluid reservoirs. However, this electrical actuator must remain an alternate until hardware and electrical details of this uncommon device are better defined. Control and electrical filter requirements are the primary details lacking. Conventional geared electric motors are categorically rejected due to their high inertia-to-power ratio.

2.4.5 Trailer Rotational Drive System

A single hydraulic cylinder will be used to drive the trailer suspension platform. The cylinder-type actuator will be attached to a fulcrum point on the trailer platform to drive it relative to the main vehicle suspension platform. Its performance requirement will be small compared to the cylinder system of Section 2.4.4. A servo-valve will be required to control it. The worst case performance requirement will occur on the maximum deceleration situation discussed in Section 2.4.1 causing the trailer to tend to "jack-knife." The cylinder will be similar to those selected in Section 2.4.4 but having a single rod and a 6 to 12-inch stroke.

The primary reason for selection of a hydraulic cylinder instead of electric motor is the readily available hydraulic supply on the suspension trolley.

2.5 FACILITY CONSIDERATIONS

The lunar gravity simulator could be designed for use in an outside environment, but unless a facility was located in a dry, sunny climate, there would be a large loss in operating time due to rain, cold and other types of inclement weather. Thus, it appears likely that the lunar gravity simulator will be installed inside a building, preferably an existing building which would either contain or could be easily modified to the desired requirements. This would minimize the capital outlay for the program and the length of time before a test program could be run.

2.5.1 Building Requirements

- Building Height: The total required internal height is the summation of two requirements. First, the design requirement for testing of the lunar surface vehicle on a maximum slope requires a total elevation change of 23 feet (7 m) (Scope of Work, Reference 6). This is consistent with the cable length limitations of 30-53 feet (9.1 to 16.1 m) established by a compromise of the LGS static longitudinal acceleration errors and the dynamic lunar gravity errors imposed on the LSV. (See Sections 3 and 4 of Reference 2.) Secondly, the height needed for the physical structure required includes such items as the height from the ground to the highest point on chassis or wheel support frames. This distance is 5 feet for the MOLAB. It also includes the depth of the suspension platform (6 feet) and the horizontal bridge structure (10 feet). These give a minimum building height of 74 feet (22.5 m).

- Building Width: A minimum length of 100 feet was selected for the bridge crane in Section 2.2. However, if this length is increased, the weight of the bridge must also be increased as the square of the length. For instance, if the length is increased by 25 feet or 25%, the weight must be increased by $(125/100)^2$ or 59% to maintain the 8 Hz natural frequency. This increase in bridge span would require larger drive motors, also. A span of 125 feet is suggested as an upper limit.

- Building Length: The maximum length for a system of this type is almost unlimited. The minimum length is naturally the smallest diameter in which the vehicle can be turned. However, this gives no room at all for a straight high-speed run.

Therefore, the distance needed is at least twice the vehicle turn diameter. This would allow either a "Figure Eight" or oval path. This gives a minimum length of 200 feet, assuming the MOLAB turn radius at 5 mph on a flat floor. If the turn were banked, the turn radius could be shortened, although the actual distance was not determined. The longest run in a rectangular area would be a diagonal and would give the longest straight line operating time.

- Additional Requirements: An additional requirement is a bridge crane or a bridge crane rail structure adequate for the imposed loads. The crane ideally should be capable of speeds to 18.25 ft/sec, with an acceleration or deceleration rate of .2 g in all directions in the horizontal plane (x and y axes). In addition, the natural frequency of the bridge when carrying a load of 5,000 pounds must be 8 Hz or higher. Such a crane is not likely to exist. Therefore, the crane must be amenable to structural and mechanical changes. Preferably, a new structure would be built for existing crane tracks or rails.

2.5.2 Existing Facilities

A survey of data on facilities for lunar gravity simulation showed that the only existing facility large enough for surface vehicles is the Langley Lunar Landing Research Facility in Virginia. Complete details of this facility were not readily available. However, the major design data were reported in Reference 7 and will be quoted extensively in the following section.

- Langley LLRF: This facility is designed for research in piloting problems for a lunar approach and touchdown. The facility is designed to support a LEM with full fuel load (30,000 pounds) but the present tests use a half-scale prototype weighing 10,000 pounds. The facility itself consists of a gantry structure with an overhead crane as shown in Figure 2.16. Nominal dimensions of the structure are 240 feet high, 300 feet wide and 400 feet long. However, the usable interior dimensions covered by the bridge crane are 175 feet high, 50 feet wide and 400 feet long.

  A comparison of the LGS requirements and the LLRF capabilities is shown in Table 2.3. This comparison is based on the following modifications of the LLRF: (1) a suspension platform, with separate winch systems for the LGS wheels and chassis is installed on the LLRF bridge and dolly structure; (2) a short-stroke (± 2 ft) high-acceleration longitudinal track framework is required to provide ± 2.5 g transient acceleration capability; (3) a yaw bearing is required

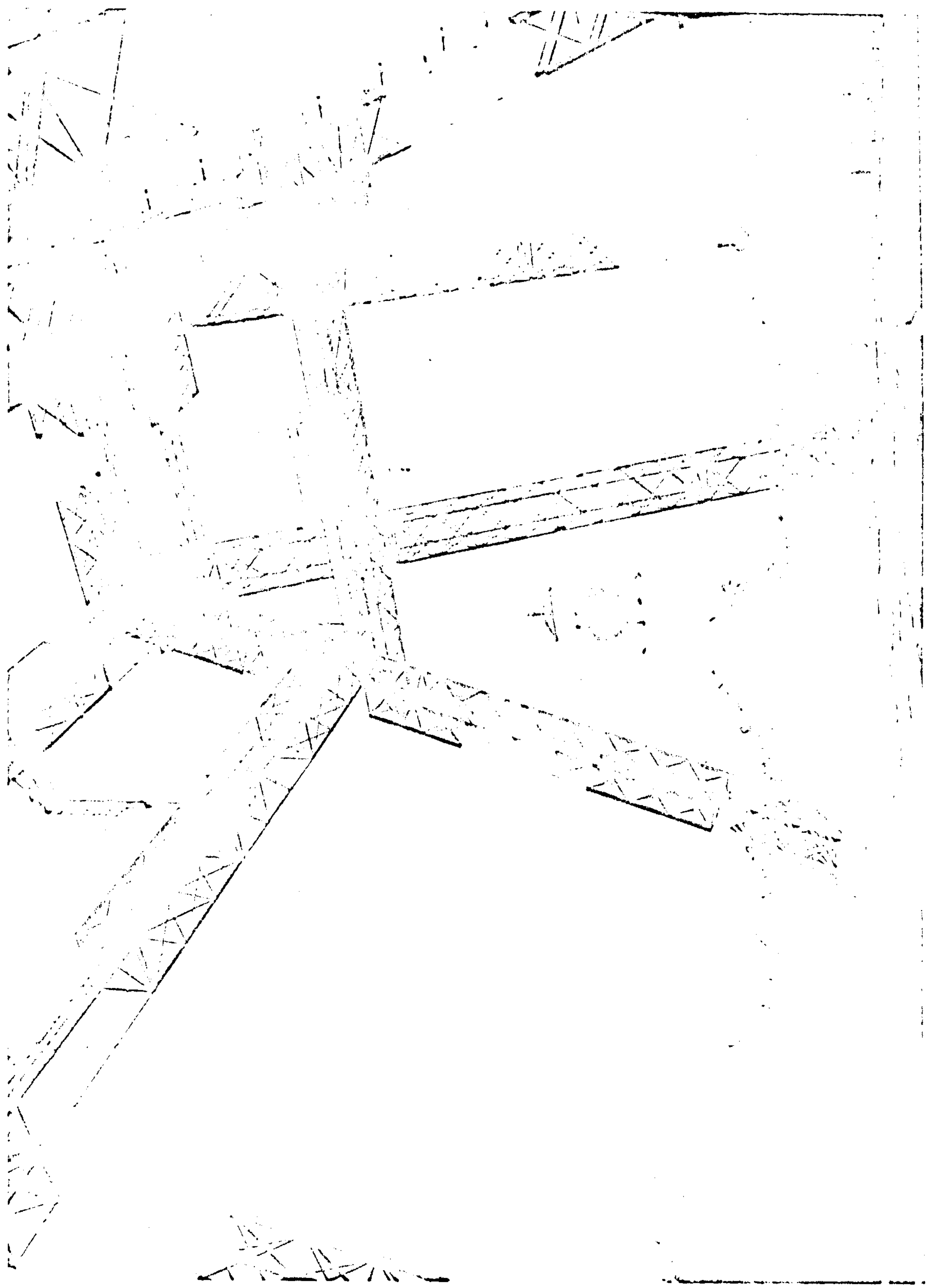


Figure 2.16 - Langley Lunar Landing Research Facility

Table 2.3

COMPARISON OF LGS REQUIREMENTS WITH LLRF CAPABILITY

	LGS Requirements	LLRF Capability	3-D LGS
Height	74 ft	175 ft	-
Width	100 to 125 ft	50 ft	+
Length	200 ft min.	400 ft	+
Vertical (Z-axis)	See Note 1		
Travel	23 ft	LLRF Modification required. See Note 1	
Velocity	1		
Acceleration	1		
Lateral (Y-axis)			
Travel	100 ft min.	50 ft 2	-
Velocity	15 ft/sec 2	10 ft/sec	Marginal
Acceleration	$\pm$0.2 g 2 3	$\pm$0.1 g	Marginal
Longitudinal (X-axis)			
Travel	200 ft min.	400 ft	+
Velocity	18.25 ft/sec	49.7 ft/sec	+
Acceleration	$\pm$0.2 g 3	+.17 g, -.39 g	Marginal

Notes:

1 A suspension platform and associated suspension device are required to provide independent suspension for each wheel and chassis. See Table 2.3 for separate wheel and chassis requirements.

2 The LLRF width imposes a vehicle turn angle limitation of about $\pm 45^{\circ}$ from the longitudinal axis. When this limitation is used, the 3-D LGS requirements are: velocity = 10.5 ft/sec and acceleration = $\pm$.14 g.

3 This value is for the bridge drive systems only. A short track, or drive mounted with yaw capability, is required to meet transient ($\pm$2.5 g) acceleration requirements.

between the short-track frame and the suspension platform; and (4) alignment of the suspension platform with the LSV would be accomplished with the optical sensing system described in Section 2.3 of this report.

The LLRF has several deficiencies as compared to desired LGS requirements. The lateral width is much less than desired. The test course would be limited to using 45° turns rather than 90° turns (as recommended in Reference 7) because of the limited width of 50 ft. Also, the lateral velocity and acceleration capabilities are marginal with the LSV at 15 mph on a 45° turn. Therefore, the LSV speed must be reduced to about 12 mph to match the LLRF lateral acceleration capability. Of more importance is the 175 ft bridge height which would result in lunar gravity errors of as much as 100% (Figure 2.17) and would create doubt as to the validity of the resulting test data. The most optimistic reduction of height to 140 ft would still result in a 75% error. An additional problem may exist due to the natural frequency of the LLRF structure. Calculations based on photographic observations indicate that the natural frequency may be on the order of 4 to 6 Hz. This is the natural frequency range of the short track (ξ) drive system. In summary, it appears the the LLRF offers only a high-speed bridge crane and crane structure which would require extensive modification to be applicable to LSV testing.

- MSFC Buildings: A survey was conducted of the high-bay buildings located at MSFC. A summary of the building dimensions and crane characteristics is presented in Table 2.4.

  Only two of the buildings would meet the usable dimensions criteria. These were (1) the Components and Subassembly Acceptance Building (Bldg. 4752) and (2) the Multipurpose Vehicle Technology Facility (Bldg. 4755). Each of the other buildings, with two exceptions, did not have the required height. These two - Buildings 4619 and 4649 - were not wide enough to accommodate a 180° vehicle turn. These buildings would, however, be the prime candidates for the installation of a 2-D system since less space would be committed for this installation.

  None of the cranes in any of the buildings would meet the performance criteria or the natural frequency requirements. Only the cranes in Buildings 4752 and 4755 will be examined in detail for the 3-D configuration since the other buildings were not high enough.

  Building 4752 has a 20-ton crane with a 53-foot hook height. The building is 101 feet, 8 inches wide. The building plan is shown in Figure 2.18. The crane section was visually estimated as 5 feet high. The usual method of designing bridge cranes is to assume the rated load is applied at the center of the span and

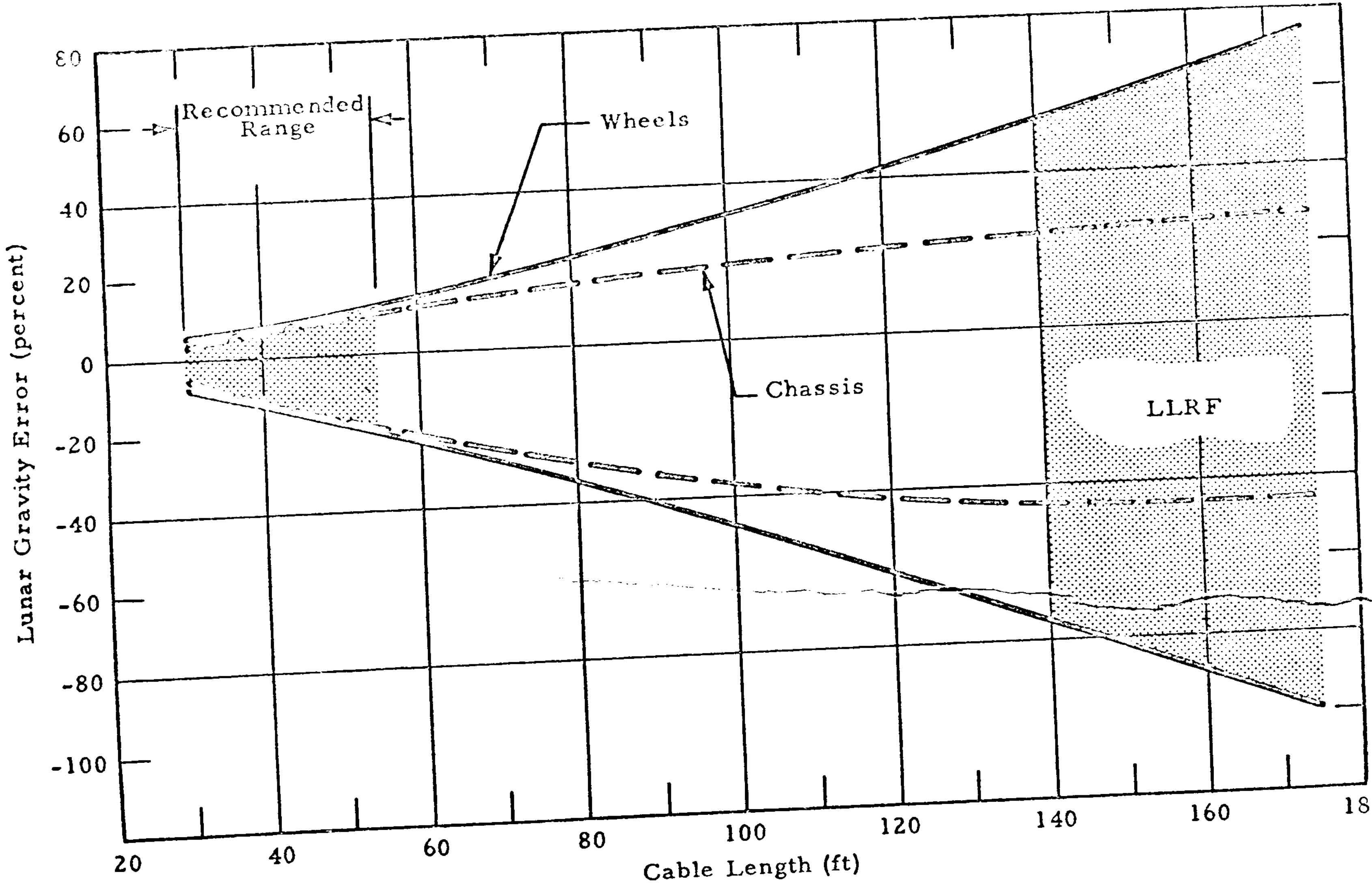


Figure 2.17 - Lunar Gravity Error vs Cable Length for LSSM in Pitch Configuration

Table 2.4

SUMMARY OF MSFC HIGH-BAY BUILDING FACILITIES

Bldg. No.	Name	Internal Dimensions			Crane Data				Comments*
		Height	Width	Length	No.	Capacity	Hook Ht.	Max. Speed	
4705	Fab. & Assy. Bldg.	48'-9"	103'-6"	432'-0"	2	10 tons	35'-3"	<4-½ ft/sec	Height, hook height both too low, unacceptable
4707	Fab., Assy. & Hydrostatic Test Facility	34'-0"	98'-0"	395'-0"	2	20 tons	25'-0"	<4-½ ft/sec	Height, hook height too slow, not wide enough, unacceptable
4708	Quality Assurance Lab	43'-9"	103'-6"	500'-0"	2	10 tons	33'-9"	<4-½ ft/sec	Height, hook height too low, unacceptable
4752	Components and Sub-Assembly Acceptance Bldg.	65'-0"	101'-8"	213'-6"	1	20 tons	53'-0"	<4-½ ft/sec	Height, slightly lower than optimum, first choice
4755	Multipurpose Vehicle Technology Facility	86'-0"	126'-0"	190'-0"	2	90 tons	70'-0"	<4-½ ft/sec	Length slightly shorter than minimum, second choice
4619	Extension to Load Test Annex	90'-0"	93'-4"	199'-0"	2	20 tons	80'-0"	<4 ft/sec	Not wide enough, would be adequate for 2-D configuration
4649	Transportation Hangar Bldg.	63'-0"	75'-0"	200'-0"	1	25 tons	50'-0"	<4-½ ft/sec	Not wide enough, height is slightly lower than optimum, second choice for 2-D system
4656	Assembly Bldg.	38'-0"	52'-0"	202'-0"	2	30 tons	30'-0"	<4 ft/sec	Height, hook height too low, not wide enough, unacceptable

*In all cases, the crane speed was too slow.

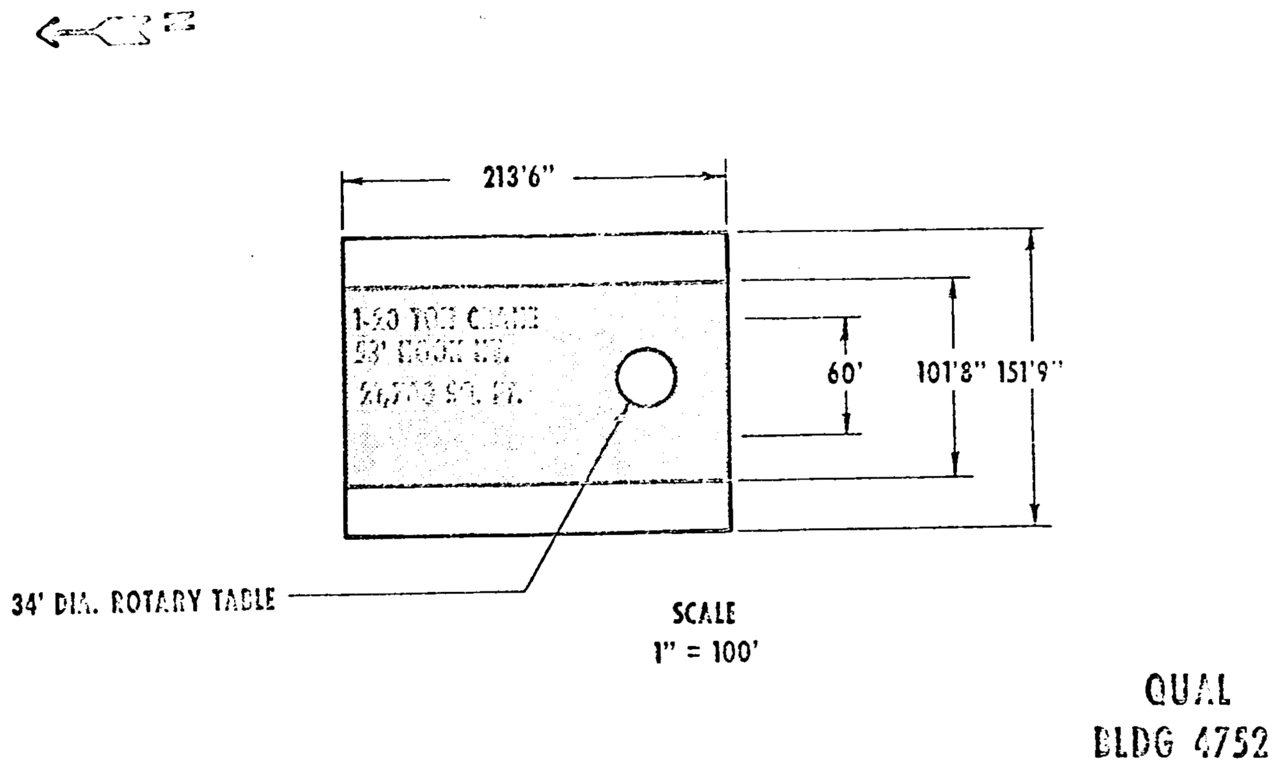


Figure 2.18 - Components and Sub-Assembly Acceptance Building

allow a maximum deflection equal to 1/360 of the span. Therefore,

$$\delta = \frac{\ell}{360} = \frac{P\ell^3}{48EI} + \frac{5W_{bridge}\ell^3}{384EI}$$

where

ℓ = bridge span = 1200 in.

P = rated load = 40,000 lb

E = Young's modulus = 29×10^6 lb/in.^{2}

I = section modulus, in.^{4}

Solving for I,

$$I = 14{,}900 + .0031\ W_{bridge}$$

Substituting $A_{net}d^2 = I$ and $W_{bridge} = A_{gross}\ell$ and using the assumption $A_{gross} = 1.5\ A_{net}$ as described in a previous section, we find the net cross-sectional area $A_n = 1{,}490/d^2 - 1.6$.

The value of d is approximately 27 inches

$$\therefore \quad A_n = 20.5 \text{ in.}^2$$

$$W_{bridge} = 10{,}450 \text{ lb}$$

$$I = 14{,}923 \text{ in.}^4$$

$$f_n = 1.97 \text{ Hz}$$

It can be seen that the natural frequency of the crane is too low. Either the crane would have to be rebuilt or another crane built. It would seem easier to build and install a new crane with the required drive characteristics than to try to rebuild the present one.

This building comes the closest to meeting all the requirements and may be a prime candidate for the installation of the three-dimensional concept of the lunar gravity simulator.

A special protective covering would need to be installed to protect the 34-foot rotary table. The table would not present any problems in the use of the building, since the test surfaces could be placed to avoid the area.

Building 4755 is a multi-bay structure as shown in the plan view in Figure 2.19. Either of the two high-bay areas would be acceptable. Only the length of 190 feet is less than the minimum dimensions. The cranes in these bays are rated at 90 tons but have a 126-foot span. Using the same method of calculation as before, the natural frequency was found to be approximately 4.2 Hz. This is still too low, but the crane could more readily be reworked to achieve the required frequency of 8 Hz.

However, in addition to the structural rework, the drive motors for the X and Y drives would have to be replaced.

2.5.3 Recommendations

Building 4752 is recommended for further study as a prime candidate for the installation of the Lunar Gravity Simulator, with Building 4755 as an acceptable alternate.

It should be emphasized that the examination of the buildings at MSFC was quite cursory. The elimination of the various buildings was based upon scale plans and visual examination. The calculations of the natural frequencies of the cranes is based on rough approximations and theoretical vibration analyses. Additional information and study would be required to determine whether either or both cranes could be strengthened to meet the requirements, or whether it would be more economical to install a new crane. Therefore, more study is recommended in this area.

2.5.4 LGS Facility Power Requirements

Power requirements were estimated for the total Lunar Gravity Simulator facility. The bridge drive motors consume most of the power. As a result, their maximum power and duty cycle predominates in the facility requirements. A tabulation of major power consumers is shown on the following page. From this, maximum and average values of power are estimated. Other items, such as electronics consoles are negligible as power consumers.

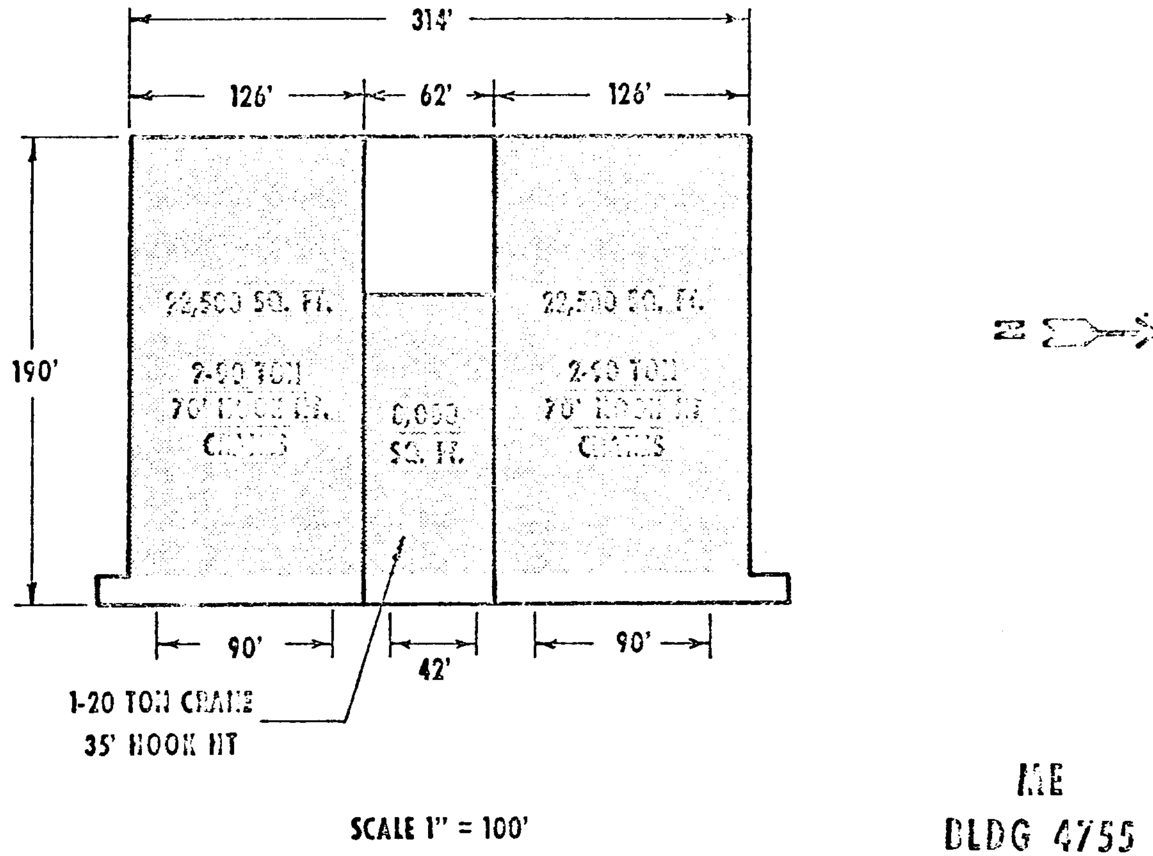


Figure 2.19 - Multipurpose Vehicle Technology Facility

Item	Output (hp)	Efficiency	Input (hp)
Bridge drive motors (X) (maximum acceleration power)	187	80%	234
Hydraulic power for vertical winches (z)*	50	70%	71
Lateral trolley drive motors (Y)	20	80%	25
Yaw drive motor (ψ)	1	80%	1
Hydraulic power for suspension trolley (ξ) and trailer platform angle (Ψ_T)	10	70%	15

The critical power requirement is established during the bridge acceleration and is determined by the maximum LSV velocity and nominal acceleration. Thus, 234 hp is required for $18.25/0.1 \times 32.2 \simeq 5.7$ seconds. For this maximum acceleration condition, the vehicle would not be climbing near its maximum rate, nor would the lateral trolley. A 50% power is estimated for these latter two items. Maximum facility power becomes

Bridge drive motors, full power	234
Vertical winches, half power	35
Lateral trolley, half power	12
Total power maximum for 4.49 sec	281 hp

Long-term power must be based upon a rather complex duty cycle. However, for a first approximation, the maximum long-term duty cycle is estimated at 50% total power maximum or

$$50\% \times 281 = 140 \text{ hp } (188 \text{ kW})$$

*400 Hz power, all other input power is 60 Hz.

2.5.5 Simulated Lunar Terrain Configuration

In order to evaluate the LSV's locomotion performance to compare it with dynamic objectives, it will be necessary to contour the floor of the facility to conform to the characteristics of the lunar terrain. The model should be patterned after the terrain described in NASA documents, i.e., Engineering Lunar Model Surface (ELMS, TR-83-D), Engineering Lunar Model Obstacles (ELMO, TR-145-D), and the findings of the latest studies based on returned photos taken by Surveyor spacecraft.

Figure 2.20 illustrates a representative configuration that would include a hill, level plane and a bank. It would include a sampling of soft and hard soil, rocks, crevices and craters. This configuration would permit evaluation of the conditions required to establish vehicle, steady-state soil resistance, steady-state slope negotiations, vehicle internal losses, acceleration and deceleration, steering, and obstacle negotiation.

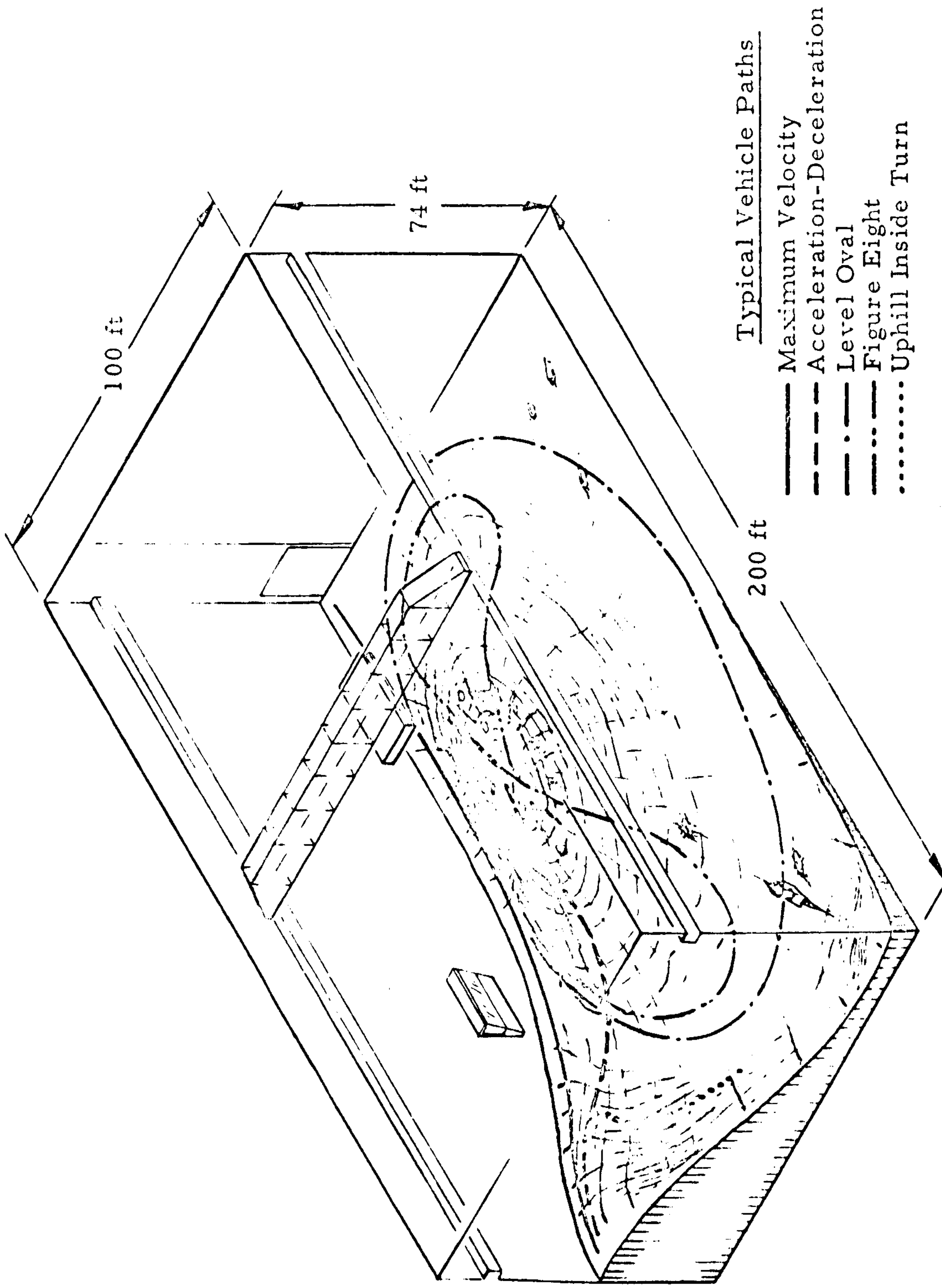


Figure 2.20 - Simulated Lunar Terrain Configuration

Section 3
DRIVER SUSPENSION SYSTEM

Task 3 of the Work Statement requires the preliminary design and analysis of a system for suspending the driver of a test vehicle under simulated gravity conditions and preparation of a conceptual design adapted to the two-dimensional simulator. This section presents the results of studies under this task.

The data used indicate that the driver would use only a hard suit and would be fastened securely to the vehicle, except for his arms. Thus, only the arms must be suspended for lunar gravity simulation. This suspension force will be maintained at a nominal 5/6 g (which leaves only equivalent lunar weight at the c.g. of each arm) by a constant force spring motor on each arm. The spring motor force (estimated at approximately 20 pounds) is ultimately borne by the overhead trolley through a cable-winch system. This study considered the LSSM as representative of the LSV driver configuration.

Analysis within this system considers the suspension system in perfect alignment with the LSV, which is practically the actual case for steady-state operation. Even the small transient platform errors, when they occur, have a negligible effect insofar as this analysis goes.

3.1 PERFORMANCE REQUIREMENTS

Five-sixths (5/6) of the weight of the astronaut must be supported while he performs all functions required of him in the LSSM driver's seat. The hard suit which will be used is fastened firmly to the seat by means of seat belts, one around the lap, another across the thighs of the astronaut. This leaves only his arms free to move; therefore, only they must be supported external to the vehicle. The remainder of the weight of the suited astronaut will be borne by the LSV.

Performance of each arm will be limited horizontally from outstretched on side to outstretched forward, and vertically from a horizontal plane through the shoulders down 45°. They must also bend to reach the seat belts, which fasten at the top. Twisting of the arms $\pm 30^{\circ}$ is required. No impairment of these operations shall occur as a result of the external arm suspension.

3.2 RECOMMENDED APPROACH

Each of the astronaut's arms will be partially supported by a vertical cable reeved from the suspension platform and attached to the arm through a constant force spring. The suspended cable length will vary from 30 to 53 feet depending upon the LSV elevation and the "rough" cable length adjustment will be accomplished by reeving the cable to an extra drum on each of the chassis support winches (Figure 3.1). Each winch will accomodate the arm on its side of the vehicle. Since each winch is servo driven to vertically follow the vehicle at a point close to the astronaut's arm, the constant force device (spring) will be required to compensate for the relative displacement between the arm and the LSV. The constant force device will be a constant torque motor spring (Negátor or equivalent) with the motor chassis hooked to the c.g. of the suited astronaut's arm (Figure 3.2). The motor spring will maintain a constant tension in the cable. This tension will be preselected to equal 5/6 of the weight of the suited arm plus the motor spring weight, thereby leaving only lunar weight to be supported by the astronaut himself.

Since the astronaut's arm must twist, the motor spring chassis itself must be free to swivel and turn about the arm to prevent undesirable restoring torques on the arm. A simple hook permits the swivel; a ball bearing permits twisting of the arm (Figure 3.2). When operation without a driver is required, the arm suspension device can be simply hooked to any convenient point on the vehicle chassis near its normal operating position. The motor spring will maintain tension to prevent cable fouling.

Note: Figure adapted from Reference 2, Figure 5.6

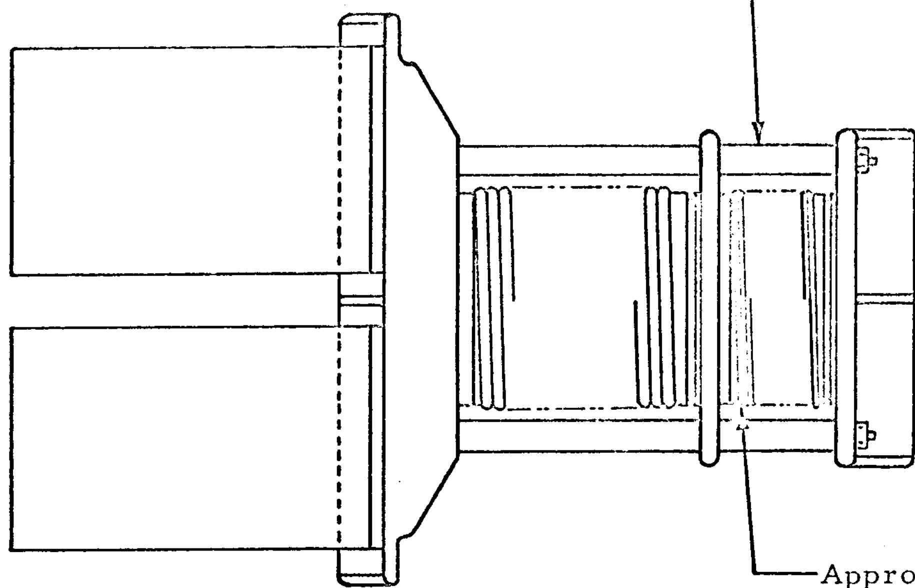


Figure 3.1 - Winch Assembly Modified to Support Astronaut's Arms

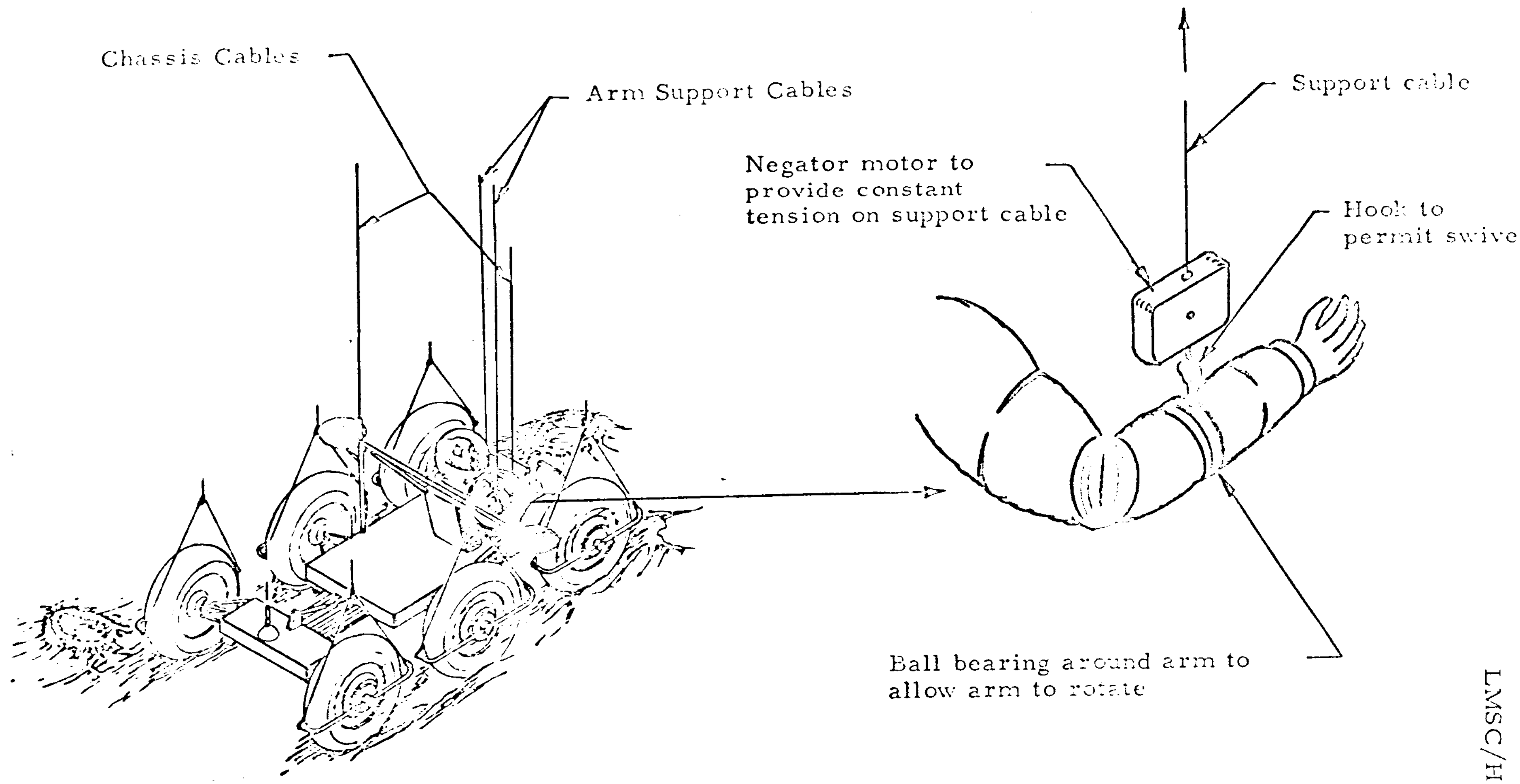


Figure 3.2 - Support of Man's Arm for Lunar Simulation

3.3 PROBLEM AREAS

There were no significant problem areas; however, certain limitations to true lunar simulation do exist as discussed below.

3.3.1 Constant Force on Arm

The astronaut will physically feel no less than earth gravity, regardless of how well the suit is supported to reduce effective weight. However, the suit/astronaut interface near the wrist will support the astronaut's arm in such a way as to simulate his muscle-supported loads; an undesirable load equal to the unsuited arm weight (nominally 10 pounds) must be borne at this wrist interface. It may be desirable to provide additional support points for the simulation.

3.3.2 Inertia Forces

Since frictional forces are quite low in the hard suit, relatively rapid jerks of the arm may be performed by the astronaut. The Negátor motor box (and cable to a lesser extent) will change the sensation of the motion by adding undesirable inertia to the arm. Total mass will be approximately 25 pounds instead of 20 pounds (nominal values). The effects of both horizontal and vertical components of the added inertia are discussed below.

- Horizontal Sensation Error

  A horizontal acceleration of 2 g for approximately .06 seconds is the maximum anticipated for the vehicle and its astronaut-driver. The motor spring chassis ($\approx$5 pounds) inertia adds 10 pounds or 25% to the arm horizontal force during this transient. This should not have a significant effect on the simulation because of the short duration. The primary effect will be the perturbation to the suspended motor spring assembly and the subsequent cable lateral vibration. For this reason, the motor spring assembly should be designed for minimum weight, packaged volume and distance from the arm hook point to the motor spring c.g.

- Vertical Sensation Error

  The maximum vertical acceleration anticipated for the LSV chassis is approximately 10 ft/sec^2 at the chassis c.g. plus a 6 rad/sec^2 pitch acceleration. (See Figures 4.11 and 4.15 of Reference 2.) Using a maximum arm c.g.-to-chassis c.g. value of 2 feet, the total vertical acceleration at the arm c.g. is approximately 22 ft/sec^2 or $\simeq 0.7$ g. The corresponding vertical sensation error force due to the motor spring assembly is 0.7 x 5 or 3.5 pounds. This peak error force is the same magnitude as the 3-1/3 pounds arm gravitational force for the 1/6 g environment. It emphasizes the importance of minimizing the motor spring weight. The error force will tend to increase the total vertical displacement excursion for the arm. Also, the error force occurs when the astronaut is driving the LSV at well above the normal riding comfort limits (approximately 1/4 g and 1.6 rad/sec^2 pitch acceleration).

3.3.3 Arm Side Restoring Forces

In normal operation, the astronaut's arm will move the point from directly beneath the central hook point. As a result, the cable pull angle will vary from vertical to produce up to .57 pound in the worst case; viz., when the vehicle is at its highest point and the astronaut's arm is extended full length and swung completely to a side or forward position.

This is a comparatively small side force which occurs at the two extreme arm positions and at minimum cable length. This value should be minimized by locating the cable suspension point to favor the majority of the arm positions. Also, some of the arm exercises involving extreme positions may be planned for the lower LSV elevations (longer cable lengths). Derivation of the mechanics of the side load forces is presented in the following paragraphs.

- General Algebraic Solution

  For this analysis, assume a pivot point above and stationary with respect to the vehicle. The arm is supported from that point by a constant tension cable. Figure 3.3 illustrates a general configuration and force diagram.

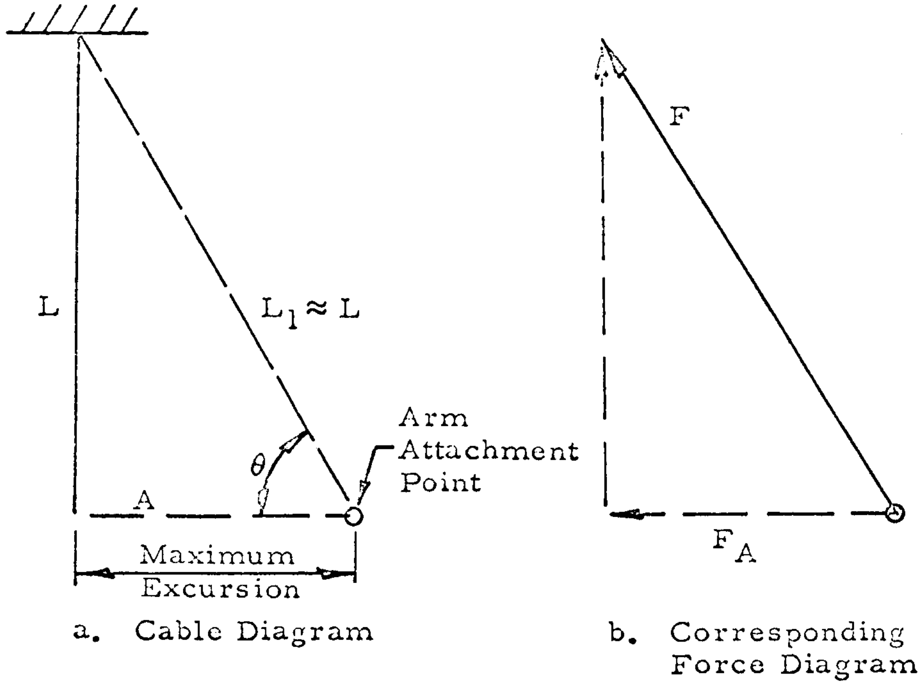


Figure 3.3.- Horizontal View of Arm Cable and Forces

Let

- L = cable length when vertical, feet
- L_1 = cable length when arm is off to maximum distance, feet
- A = horizontal distance when arm is off to maximum distance, feet
- F = cable force, constant, pounds
- F_A = side load, pounds
- θ = cable pull-off angle, degrees

Since $L = L_1$ for small values of θ, a proportion immediately establishes F_A

$$\frac{F_A}{F} = \frac{A}{L} \quad \text{or} \quad F_A = F\frac{A}{L}$$

A is related to the angle swept by the extreme motions of the outstretched arm as shown in Figure 3.4.

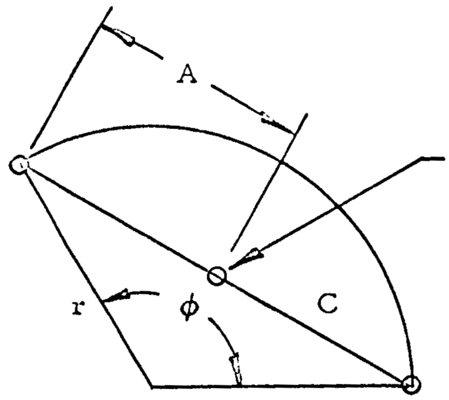


Figure 3.4 - Plan View of Astronaut Arm Sweep

Let ϕ = angle swept by extreme motion of outstretched arm, degrees

C = chord subtending ϕ, feet

Placing the pivot point at the midpoint of C will minimize the distance the arm attachment point must travel from vertical.

$$\therefore A = \frac{C}{2}$$

Since

$$C = 2\, r \sin(\tfrac{1}{2}\phi)$$

$$A = r \sin(\tfrac{1}{2}\phi)$$

Thus,

$$F_A = F\frac{A}{L} = F\frac{r \sin\frac{1}{2}\phi}{L}$$

- Numerical Solution

At the c.g. of an arm of the suited astronaut, the support cable will be attached. The force, F, used above must equal 5/6 of the weight to be supported at the c.g.

$$F = \frac{5}{6}\text{(Astronaut's unsuited arm wt. + suit arm wt.} \\ \text{+ Negator box wt.)}$$

$$= \frac{5}{6}(10 + 10 + 5)$$

$$\approx 20 \text{ lb}$$

For the worst expected case, let

$\phi = 90^{\circ}$ (max.)

$r = 1.20$ feet (max.) (Reference 8)

$L = 30$ feet (min.)

$$F_A = F\frac{r \sin\frac{1}{2}\phi}{L} = 20\frac{1.20 \sin\frac{1}{2}(90^{\circ})}{30} = .57 \text{ pounds}$$

Section 4
THREE-DIMENSIONAL SIMULATOR COST ANALYSIS

The costs of a three-dimensional LGS are presented in Table 4.1, and a design and fabrication schedule are given in Figure 4.1. The total cost of $681,400 is estimated for a 57-week program. A fee of $51,100 ($7\frac{1}{2}$%) is estimated for a total of $732,500. This estimate is based on the following assumptions:

1. A suitable building will be available for the installation of the simulator. It is further assumed that a bridge crane rail of sufficient strength is installed in the building, but the crane itself cannot be economically adapted to the simulator's requirements. Further study and analysis of the bridge will be necessary when a particular building is made available.
2. The costs of a new bridge crane, with installation and checkout, are included in this estimate. Other costs for modification of the building are not included.
3. The estimate includes all analysis, design, fabrication, installation and qualification costs necessary for delivery of the completed system.
4. Labor costs are based on $12 per hour for project and research and design specialists; $10 per hour for analysis, engineering and testing; and $8 per hour for shop labor and drafting time.
5. The projected schedule is dependent on Vendor's quoted long lead times for certain procured items, notably the chassis and wheel winches. Reduction of this time could reduce the overall schedule by as much as 4 weeks; conversely, any slippage of delivery would lengthen the schedule.
6. Documentation of the design shall meet Military Specification MIL-D-1000, Category E, Form 3.
7. Although modification of the LSV is required, the total costs cannot be determined at this time and were not included.
8. Analog and digital computer costs were estimated at $100 and $450 per hour, respectively.

Additional analysis of the various systems may eliminate a number of specially designed items and could result in a reduction of cost and schedule time.

Table 4.1

COST ANALYSIS SUMMARY

Initial Design Phase	-	16 Weeks	-	$123,800
Stress Analysis		1290 Hours		
Design and Analysis		3730		
Drafting		1750		
Programmer		920		
Analog Computer		200		
Digital Computer		20		
Detail Design Phase	-	15 Weeks	-	$125,700
Stress Analysis		1340 Hours		
Design and Analysis		4590		
Drafting		3120		
Programmer		320		
Analog Computer		140		
Digital Computer		14		
Hardware Fabrication Phase	-	26 Weeks	-	$137,600
Fabrication		8060 Hours		
Installation		3050		
Testing and Qualification		2370		
Liasion		890		
Materials Cost				$294,300
		Cost		$681,400
Fee ($7\frac{1}{2}$%)				51,100
			Total Cost	$732,500

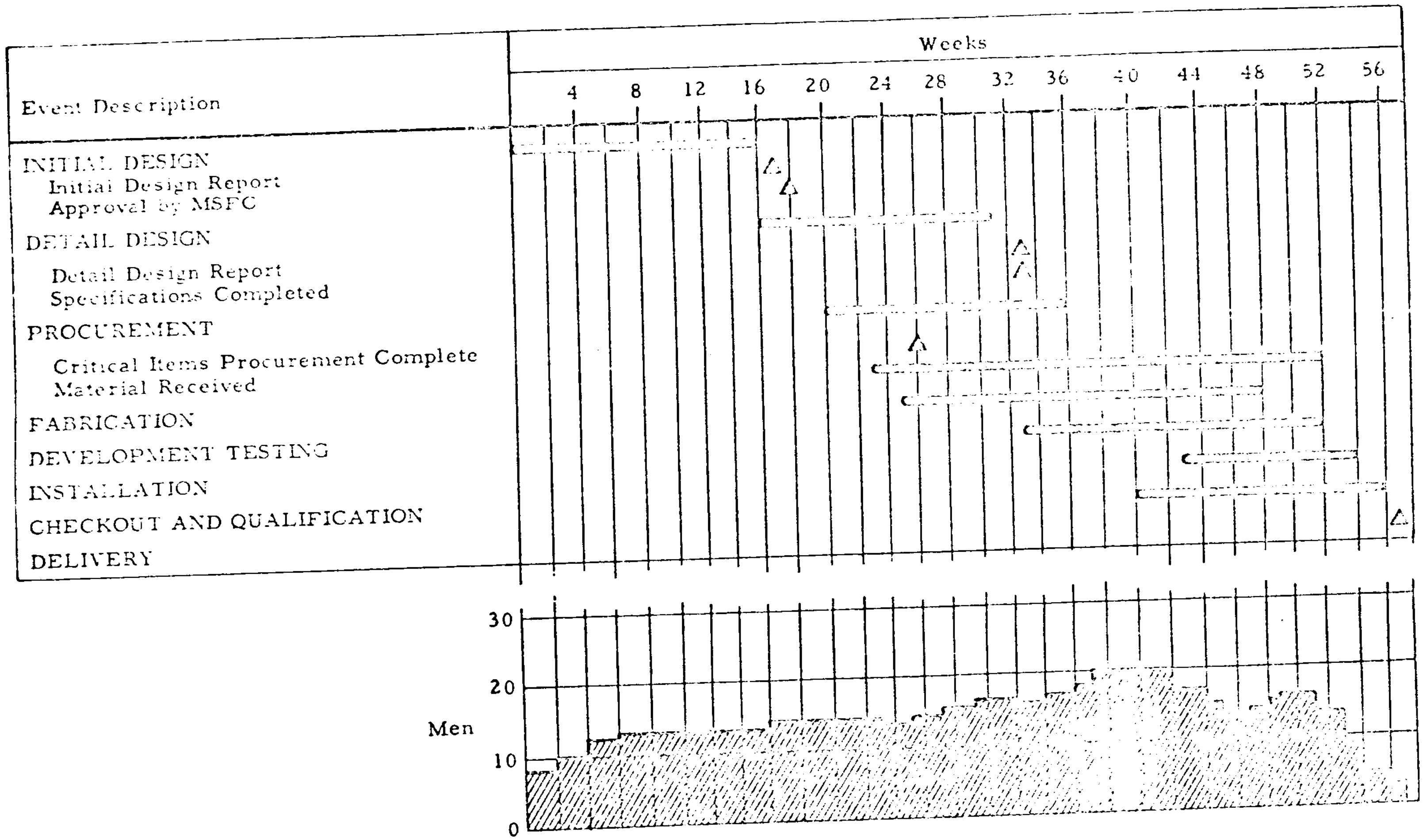


Figure 4.1 - Design and Fabrication Schedule for Two-Dimensional Capability Lunar Gravity Simulator

Section 5
CONCLUSIONS AND RECOMMENDATIONS

The results of the tasks performed during this reporting period suggest the following conclusions and recommendations:

1. The 3-D configuration described is recommended for further design and development efforts. Further efforts should include analysis of the 3-D system using analog computer methods to further delineate drive system and control requirements.

2. The bridge structure for the 3-D system should be designed to a stiffness criteria with a natural frequency in the vertical bending mode of approximately 8 Hz. Also, the suspension platform structure should have increased stiffness for a design natural frequency of 16 Hz. A Warren truss structural configuration is recommended.

3. An analog resolver network using optical type displacement sensing is recommended for controlling the 3-D drive system.

4. Silicon-controlled-rectifier electric motor drive systems are recommended for the bridge (X), lateral (Y), and suspension platform yaw (ψ) drive systems. Hydraulic cylinder actuators are recommended for the short track (ξ) drive system.

5. Use of the Langley Lunar Landing Research Facility for the LGS requirements would involve considerable modification to the facility, would be restrictive in the lateral dimension and would involve considerable (75 to 100%) lunar gravity error at maximum dynamic conditions.

6. A suitable building for the 3-D LGS would have the minimum internal dimensions of 74 feet in height, 100 feet in width and 200 feet in length. Also, the facility would contain a bridge crane track capable of supporting 20 tons minimum.

7. Further study of facilities for the LGS and subsequent selection of a facility is recommended prior to initiating further LGS design and development efforts.

8. A driver suspension concept based on supporting 5/6 of the weight of the astronaut's arms is recommended.

9. The total cost for the design, fabrication, assembly and checkout of a 3-D LGS is estimated at $732,500. The corresponding schedule is 57 weeks.

21-1052
3-P

8 JUN 1967

Section 6
REFERENCES

1. Wysor, R. B., "First Interim Report, Preliminary Design Study of a Lunar Gravity Simulator," LMSC/HREC A783082, 29 July 1966.

2. Wysor, R. B., "Second Interim Report, Preliminary Design Study of a Lunar Gravity Simulator," LMSC/HREC A783245, 9 September 1966.

3. Harris, Cyril M. and Charles E. Crede, Shock and Vibration Handbook, Volume III, McGraw-Hill Book Company, New York, N. Y., 1961.

4. Bendix Systems Division, The Bendix Corporation, "ALSS Payloads Final Report, Volume II, Book 2," ALSS-TR-013, BSR-1119, BSC-45336, June 1965.

5. Bendix Systems Division, The Bendix Corporation, "LSSM Preliminary Design Study, Local Scientific Survey Module for Apollo Extension System," First Interim Report, BSR-1180, September 1965.

6. Scope of Work, Exhibit "A" of NASA/Marshall Space Flight Center Procurement Request No. DCN 1-5-21-00032, 26 October 1965.

7. Miles, E. R., "Suitability of Langley Lunar Landing Research Facility (LLRF) for Lunar Vehicle Testing," Apollo Extension Systems Studies, Northrop Space Laboratories, NSL E 30-80, 6 February 1966, Revised 11 March 1966.

8. Wood, K. D., Spacecraft Design, Volume II, Johnson Publishing Company, Boulder, Colorado, 1964.