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DESIGN OF A SIMULATED CRUISE SCENE VISUAL ATTACHMENT

VOLUME I

DESIGN REPORT

D3-8464-1

DECEMBER 1970

THE BOEING COMPANY
WICHITA DIVISION - WICHITA, KANSAS, 67210

DESIGN OF A

SIMULATED CRUISE SCENE VISUAL ATTACHMENT

By Robert J. Rue and Chester W. Kielar

VOLUME I

DESIGN REPORT

Prepared under Contract No. NAS2-5524

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Ames Research Center Moffett Field, California

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ABSTRACT

This report presents the results of a study to design a three degree-of-freedom servosystem to provide a visual attachment for manned aircraft simulator television displays. A set of design drawings and critical component specifications are a part of the study results. A literature search was conducted and a list of publications is included. Several design approaches were developed and a comparison matrix was established to systematically evaluate these approaches. A specific approach was selected and a final design was completed and analyzed.

PREFACE

This report summarizes the work conducted by the Wichita Division of the Boeing Company under Task III of Contract NAS2-5524, "Visual Attachment for Simulated Cruise Scene." The National Aeronautics and Space Administration Technical Monitor was Mr. John C. Dusterberry of the Simulation Science Division. The Boeing Company Project Leader was Mr. C. Rodney Hanke of the Stability, Control and Flying Qualities Organization, Wichita Division.

TABLE OF CONTENTS

TABLE OF CONTENTS (Contd.)

SUMMARY

This report consists of three volumes:

D3-8464-1	"Design of a Simulated Cruise Scene Visual Attachment." Design Report
D3-8464-2	"Design of a Simulated Cruise Scene Visual Attachment." Major and Critical Component Specifications
D3-8464-3	"Design of a Simulated Cruise Scene Visual Attachment." Assembly and Detail Drawings.

Volume I presents the study and design of the Cruise Scene Servosystem. Some of the problems encountered in similar visual display systems are discussed and the various concepts considered in the initial investigation of the problem are outlined. Servo responses are specified for the three axes, (roll, pitch, and heading) and system performance requirements are defined.

The section on design development reports the studies of five concepts in more detail. Preliminary hardware designs are also developed. Consideration of the orientation of rotational axes provides a common base for further discussion. It serves to show the impact of the requirement for computer program input command compatibility.

Three areas of design are determined to be critical and are investigated in some detail. These areas are associated with the television cameras, fiber-optics, and linear actuators.

Practical limitations and general problem areas of the heading servo and scene are discussed. The pitch and roll axes are combined and five candidate systems are developed and evaluated. A comparison matrix is established.

Evaluation of the design concepts is accomplished using a set of eight criteria with a weighting factor that reflects overall system requirements.

A detailed investigation into the major design areas of the selected system is accomplished prior to starting the final system design. Each axis is studied to determine the practical limitations imposed as a result of the specified approach chosen. Final system specifications are established. The system design of the selected approach deals with component selection and detail design of fabricated parts.

The servosystem controls are developed using the data of the specified hardware components. The electronic/electrical section outlines the control system components and establishes the diagrams necessary to implement the system. A system block diagram and description is given. The analysis of each subsystem provides definition of the parameters, signal flow diagrams, and system equations. A simulation of one of the servo systems is described. Sample response verification plots of the simulated heading servo are shown. The system parameters are optimized in a computer simulation.

The cost is estimated for implementing the Servoscene Generation and a list of references and contacts from the results of the literature survey conducted early in the project is included.

Volume II presents the major and critical component specifications.

Volume III presents assembly and detail drawings. Detailed parts list and assembly and fabrication notes are included on the drawings.

INTRODUCTION

Background

General purpose television out-the-window display systems for manned aircraft simulators are complex, expensive devices requiring not only large initial financial outlays, but high maintenance and operational costs. Linear motions are not visually perceived by the pilot when high altitude cruise flight conditions are being simulated. Only three degrees of angular motions are needed to simulate the outside world. Such a simulation device can be built more simply and economically than a general purpose six degree-of-freedom facility. The high altitude visual display system presented in this report will perform reliably and effectively thereby releasing the elaborate visual display system for more sophisticated tasks.

Design Requirements

The servosystem design will provide a simulated view of the horizon from an altitude of at least 20,000 feet. This sky-horizon scene will extend a full 360° in heading. The ground portion will contain only nondescriptive features. Maximum coverage of the scene will be consistent with the display requirements in Table I. The servosystem will provide an optical image suitable for pickup by a standard vidicon television camera system. The output of the visual scene generator will be simulated color display. The use of a black and white camera with external coding will be considered. +

TABLE I
Display Performance Requirements

Axis	Range	Rate	Acceleration	Accuracy
Roll	±180°	1.5 Rad/Sec	5.8 Rad/Sec^2	±0.33°
Pitch	+40° -60°*	3.0 Rad/Sec	16 Rad/Sec^2	±0.33°
Heading	Continuous	1.5 Rad/Sec	5.0 Rad/Sec^2	±1.0°/Sec

The scene will be comprised of three windows which will subtend at least 46 degrees horizontally by 34 degrees vertically. EIA standard RS-170 will be the criteria for picture quality.

\+ Responsibility of the contractor will be limited to design for appropriate video levels to be externally coded as required by NASA. Reference Coordination meeting, JDusterberry, DDust, JSmith, and RJRue 27 July 1970.

* Maximum practicable downlook angle.

The display generated will provide realistic motion. The servosystem will be designed such that the response of the display will be smooth and free from objectionable oscillations and overshoot for normal aircraft inputs.

The design will be consistent with good electronic and electromechanical design practices (e.g. the A.C. grounding will be isolated from D. C. returns.

NOMENCLATURE

Symbol	Description	Units
B	brightness of sphere scene	Lumens/ft^2
C	current	amps
C_D	delta current	amps
C_O	controlled output in servosystems	– –
D	diagonal of television raster on vidicon tube	mm
D_S	diameter of sphere	inches
E_R	ripple voltage of tachometer	%
E_S	illumination on sphere scene	ft. candles
E_V	illumination on face of vidicon tube	ft. candles
e	efficiency of ball screw	%
e_i	input voltage	volts
e_o	output voltage	volts
F_A	force applied to platform in roll	lbs
F_D	viscous damping constant	lb-ft/rad/sec
F_H	force in heading axis drive belt	lbs
F_L	force in the phase of the platform	lbs
F_P	force in pitch axis drive ball screw	lbs
F_R	force in roll axis	lbs
G	servosystem forward transfer loop	– –
H	servosystem feedback transfer loops	– –
I_H	moment of inertia in heading axis	lb-ft-sec^2
I_I	intensity of light source	candle power

NOMENCLATURE CONTINUED:

Symbol	Description	Units
I_P	moment of inertia in pitch axis	lb-ft-sec^2
I_R	moment of inertia in roll axis	lb-ft-sec^2
i_a	motor armature current	amps
i_L	load current	amps
J	inertia load of motor	lb-ft-sec^2
K_B	back E.M.F. constant of motor	volts/rad/sec
K_M	motor constant	oz-in/ $\sqrt{\text{watts}}$
K_T	torque constant of motor	lb-ft/amp
K_a	gain of linear amplifier	— —
K_g	gain of tachometer	volts/rad/sec
K_θ	pitch position scale factor	volts/degree
K_Φ	roll position scale factor	volts/degree
K_Ψ	heading position scale factor	volts/degree
L_A	lever arm of pitch platform drive	inches
L_I	linearity	%
L_R	pitch radius of heading drive belt gear	inches
L_a	inductance of armature winding	henries
l	lead of ball screw	inches/turn
M	aircraft axis of rotation for side-looking pitch	— —
m	mass	lb-sec^2/ft
N_H	gear ratio of heading drive	— —

NOMENCLATURE CONTINUED:

Symbol	Description	Units
N_P	gear ratio of pitch drive	– –
P	aircraft axis of rotation for forward-looking roll	– –
P_P	input power to produce peak torque	lb-ft/sec
P_S	output power at motor shaft	lb-ft/sec
R_a	resistance of armature winding	ohms (Ω)
R_I	inner radius of sphere	inches
R_L	minimum load resistance	ohms (Ω)
R_i	reference input to servosystems	– –
r	radius arm	ft
S	LaPlace variable	1/sec
T_F	total breakway torque of motor	oz/in
T_H	torque in heading axis	lb-ft
T_N	ripple torque of motor	%
T_P	torque in pitch axis	lb-ft
T_R	torque in roll axis	lb-ft
T_T	temperature rise constant	^oC/watt
T_e	temperature	^oC
T_p	peak torque of motor	lb-ft
V_D	delta voltage	volts
V_P	voltage at peak torque	volts
V_a	voltage across armature winding	volts
W_D	power dissipated in motor	watts

NOMENCLATURE CONTINUED:

Symbol	Description	Units
W_P	input power to produce peak torque	watts
W_S	output power at motor shaft	watts
W_W	weight of sphere	lbs
X	aircraft referenced longitudinal axis	– –
Y	aircraft referenced lateral axis	– –
Z	aircraft referenced vertical axis	– –
Θ	aircraft pitch angle (Euler)	degrees
Φ	aircraft roll angle (Euler)	degrees
Ψ	aircraft heading (Euler)	degrees
α_H	heading axis angular acceleration	rad/sec^2
α_P	pitch axis angular acceleration	rad/sec^2
α_R	roll axis angular acceleration	rad/sec^2
β_A	actual horizontal field-of-view	degrees
β_E	effective horizontal field-of-view	degrees
δ	system damping ratio	– –
θ	servomotor shaft position	degrees

LIST OF ILLUSTRATIONS

FIGURES: PAGE

LIST OF ILLUSTRATIONS CONTINUED:

FIGURES PAGE

DESIGN RESULTS

The servo cruise scene generator designed meets all requirements of the proposal with changes discussed in the design review meetings. The device initially will have three out-the-window scenes (3 cameras) with provisions for two more as shown in Figure 1.

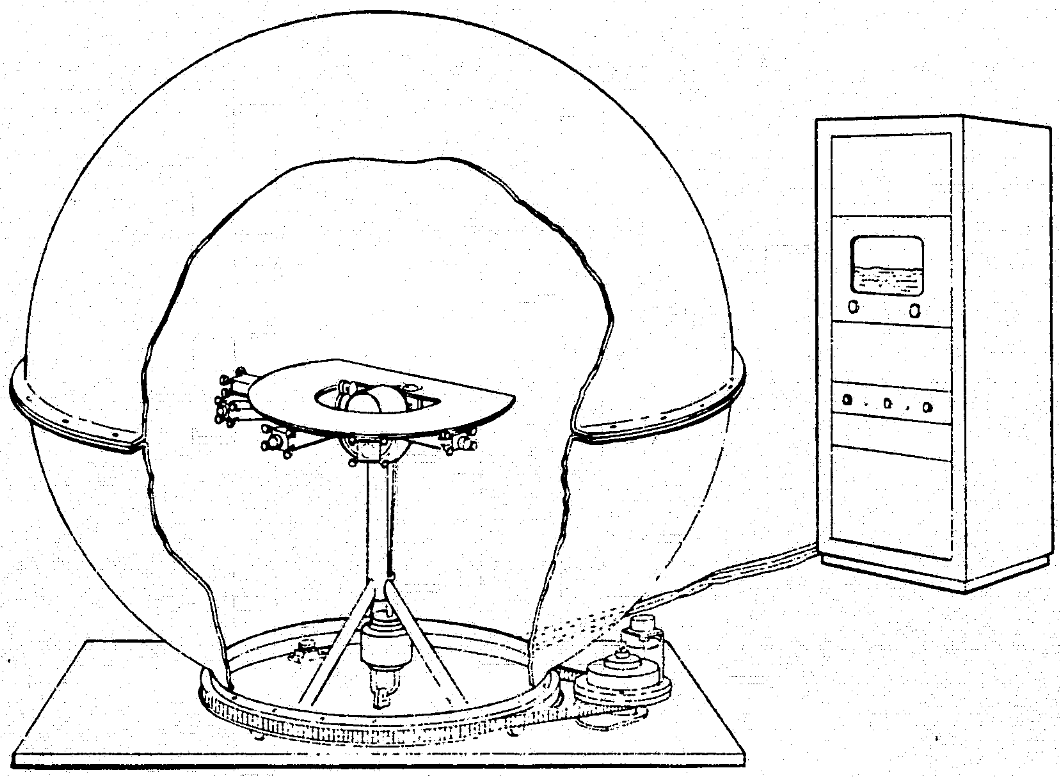
The rotation of the scenes will be provided by three servosystems, in heading, roll, and pitch. The heading servosystem will have continuous motion capability in velocity with a maximum rate of 1.5 radians per second. This servo will rotate an opaque sphere with a sky-horizon painted internally. A specific heading position will be commanded as an initial condition by mode selection switches under control of the operator or computer.

The pitch axis servo will provide control of the camera in position throughout a $\pm 70^{o}$ range in the vertical plane with an angular acceleration of 16 radians per $sec.^2$

The roll axis servo will also have the capability of $\pm 70^{o}$ travel. It is also a position servo with an angular acceleration potential of 5.8 radians per $sec.^2$ This axis will carry the scene illumination lamps that are clustered about the cameras to provide even illumination of each view.

The control interface cabinet will provide mounting for the camera control units, television monitor, servoamplifiers, control panels, and input/output terminations.

The cost of fabrication, assembly, and checkout of the designed system is estimated to be $58,125. This would not include installation and checkout at the NASA Facilities.



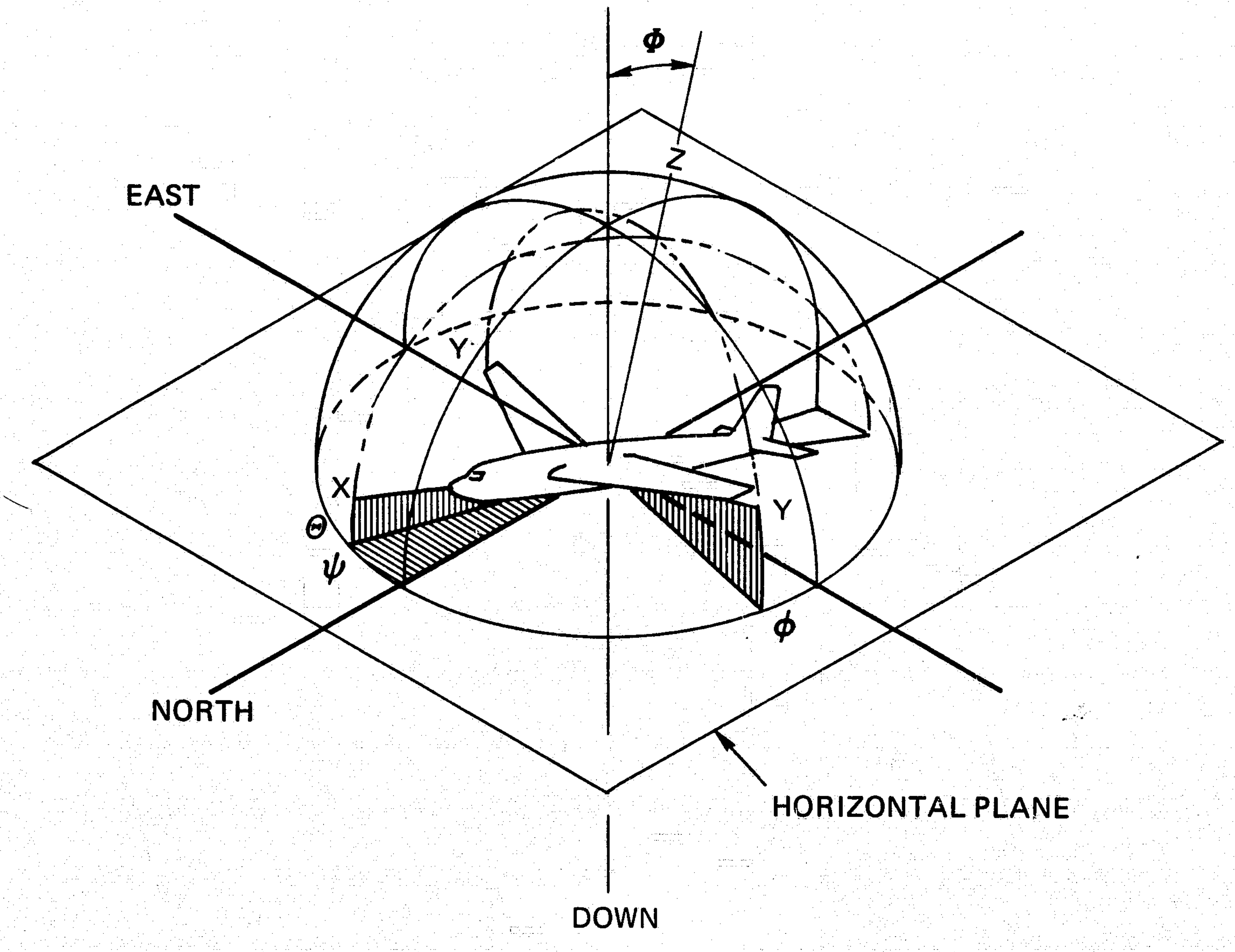
SERVO SCENE GENERATOR CONCEPT
FIGURE 1

DEVELOPMENTAL DESIGN APPROACHES

Axis Orientation

The standard Euler Angle Axes System is used to relate the aircraft body axes to space fixed axes. North, East, and down are mutually perpendicular coordinates in the earth frame reference and ψ, Θ, and Φ are the standard Euler angular rotations.

Three window directions were selected as the initial requirement for development of the potential design approaches. These directions are centered about the X, Y, and Y′ shown in Figure 2.



ANGLE DEFINITION
FIGURE 2

The pilot's view through these windows in the aircraft will be rotated through the specified Euler angles with respect to the earth. A pitch rotation, Θ, causes an apparent translation of the view through the forward window (along the aircraft X axis), and produces a rotation of the view through the Y axis window.

Critical Design Areas

The specifications of three types of components were found to be particularly critical to the design. These were the Television Sensor (Camera) specification, fiber-optics bundle characteristics, and the ball screw capabilities.

Television Camera

The characteristics of the vidicon type camera were deemed sufficient to satisfy the proposed requirements. A cost factor of at least a magnitude is incurred for other type cameras. The MTI Model VC-20, Figure 3, is typical of small, light weight, vidicon cameras which fulfill the requirements of EIA standard RS-170. Appendix A compares five commercially available vidicon cameras.

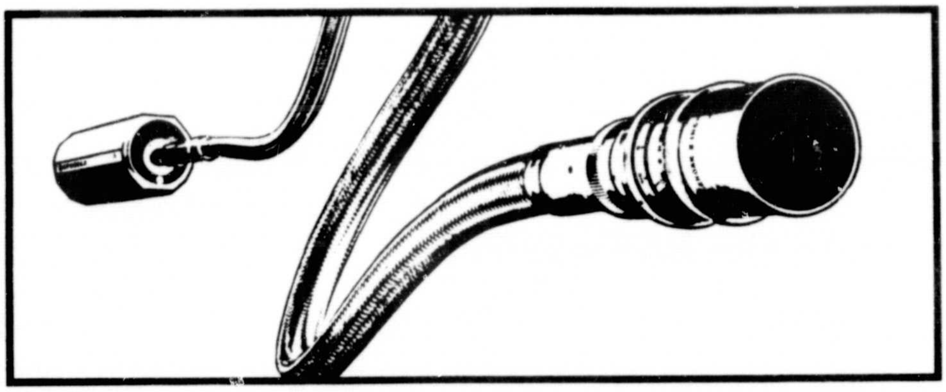
SMALL VIDICON CAMERA
FIGURE 3

Fiber-Optic Bundles

The characteristics of the fiber-optic bundles for transmitting the television image from the viewing lens to the vidicon faceplate should include the following.

- 500 line static resolution (coherent)
- Adequate flexibility
- High reliability under multiple flexing

The fiber bundle capable of fulfilling these requirements weighs approximately five pounds per foot, measures 1.3 inches in outside diameter, and has a bending radius of about three inches. No increase in efficiency over direct viewing is expected in this application. Figure 4 illustrates a typical fiber bundle.

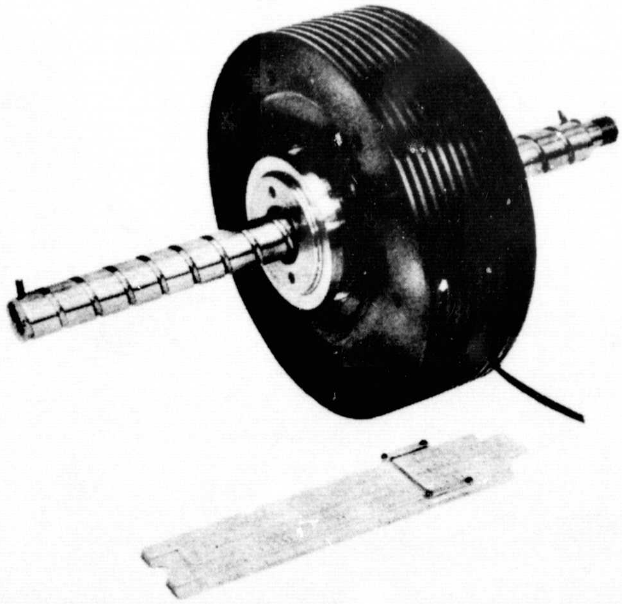


TYPICAL COHERENT FIBER-OPTICS BUNDLE
FIGURE 4

Ball Screw Drive Element

A simple method of converting rotary to linear motion is through the use of a ball screw where the nut is contained as part of the hub of a torque motor. By using a preloaded double nut, backlash in the drive can be eliminated at the expense of a small increase in friction.

Calculations show that for a standard ball screw, presently available on the market, typical characteristics of a system such as that shown in Figure 5, for a force level of approximately 500 pounds would be as follows.



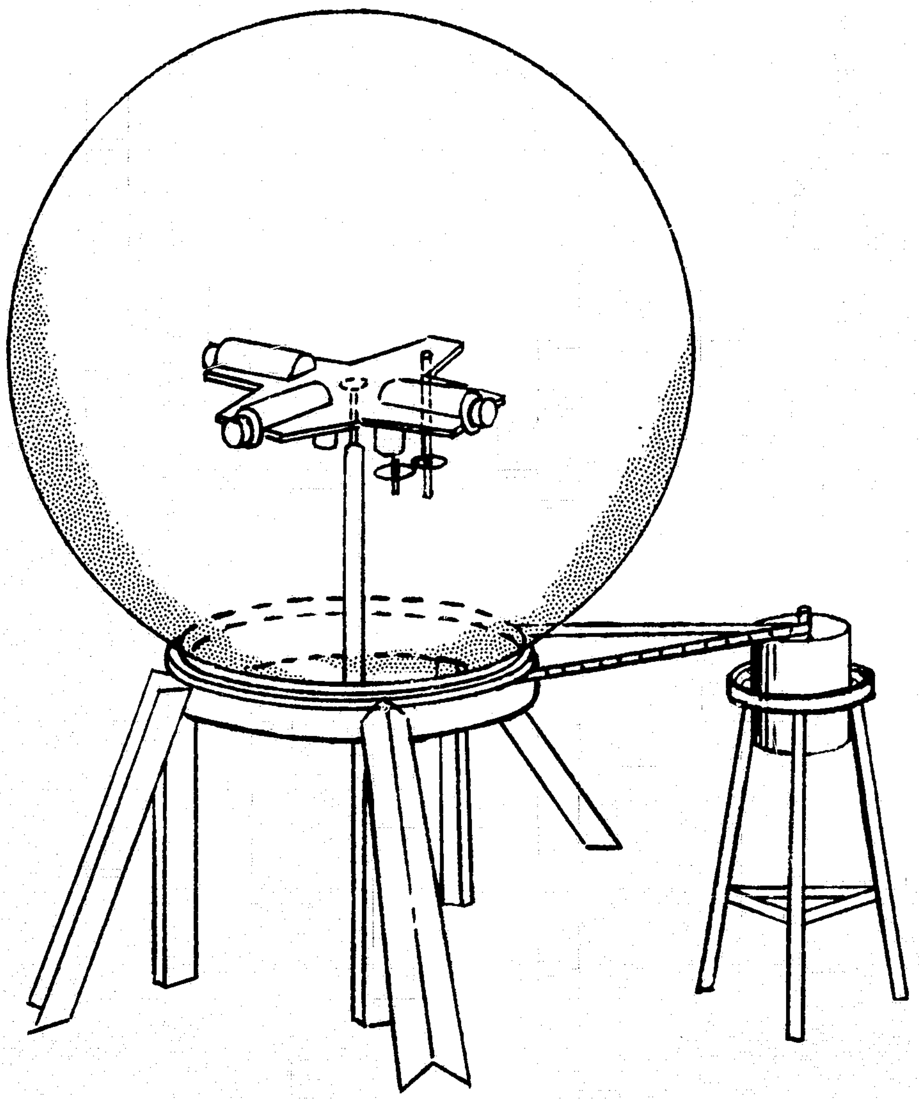
BALL SCREW AND MOTOR DRIVE
FIGURE 5

Size of screw	0.5 in. (dia.)
Lead	.590
Max. Velocity	4.4 in/sec
Motor Housing	5.625" dia. x 5.4" long
Approx. Weight	1 lb.

Heading Servomechanism

Sphere

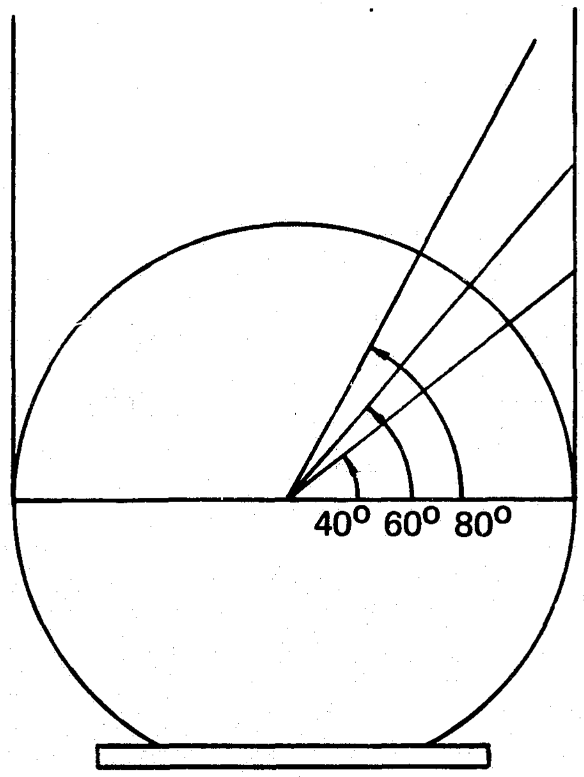
The concept of the incorporation of the heading into a servo driven scene is basic to all the design approaches. This design concept of Figure 6 shows the main design influence of the internal pitch and roll devices on the bowl or cylinder in size.



SCENE ENCLOSURE
FIGURE 6

The choice of a cylinder can result in a rather large cumbersome system requiring significant power to drive, and an extra high ceiling. A spherical shape will minimize this height requirement and will simplify the fabrication as shown in Figure 7. For angles greater than 45°, a cylindrical display rapidly increases height requirements.

The size of the area set aside for the overall system was approximately a 10-foot cube. Allowing some clearance at both the ceiling and the floor, an eight foot sphere would seem to be about the practical limit in size. The inertia load presented to the heading drive system by the sphere is seen to be a predominant factor in the servopower requirements so that minimization of the sphere diameter is important. Materials of low density such as polyurethane are questionable because of durability, especially if a translucent or opal optical characteristics are needed.



CYLINDRICAL VS. SPHERICAL SCENE
FIGURE 7

Scene

The generation of an image depicting either high altitude flight or flight above a uniform cloud layer can be accomplished by painting a scene inside of the sphere. Different types of paint and decorating techniques must be used depending on the decision of whether to back or front light the scene. Since air brush techniques associated with transparent dyes require more time to develop than opaque pigment techniques, internal illumination is considered best from the standpoint of decoration. Unwanted gloss effects and possible double reflections can be eliminated by painting the scene on the inside of the transparent sphere.

An advantage of backlighting a translucent scene is that a potential exists for back projecting a moving cloud scene to simulate penetration of the aircraft into a cloud layer.

Servodrive

The main heading drive servo should be a smooth operating system of unlimited rotation. A velocity servodrive with heading rate as the input command is an appropriate choice for a servomotor to meet this requirement. With attention to gimbal function, the high gains normally possible with velocity feedback will provide a minimum breakway level consistent with the ability of the pilot to perceive this motion. The initial position of the servo can be set by a gain and feedback selection controlled by.the computer initial condition mode control system.

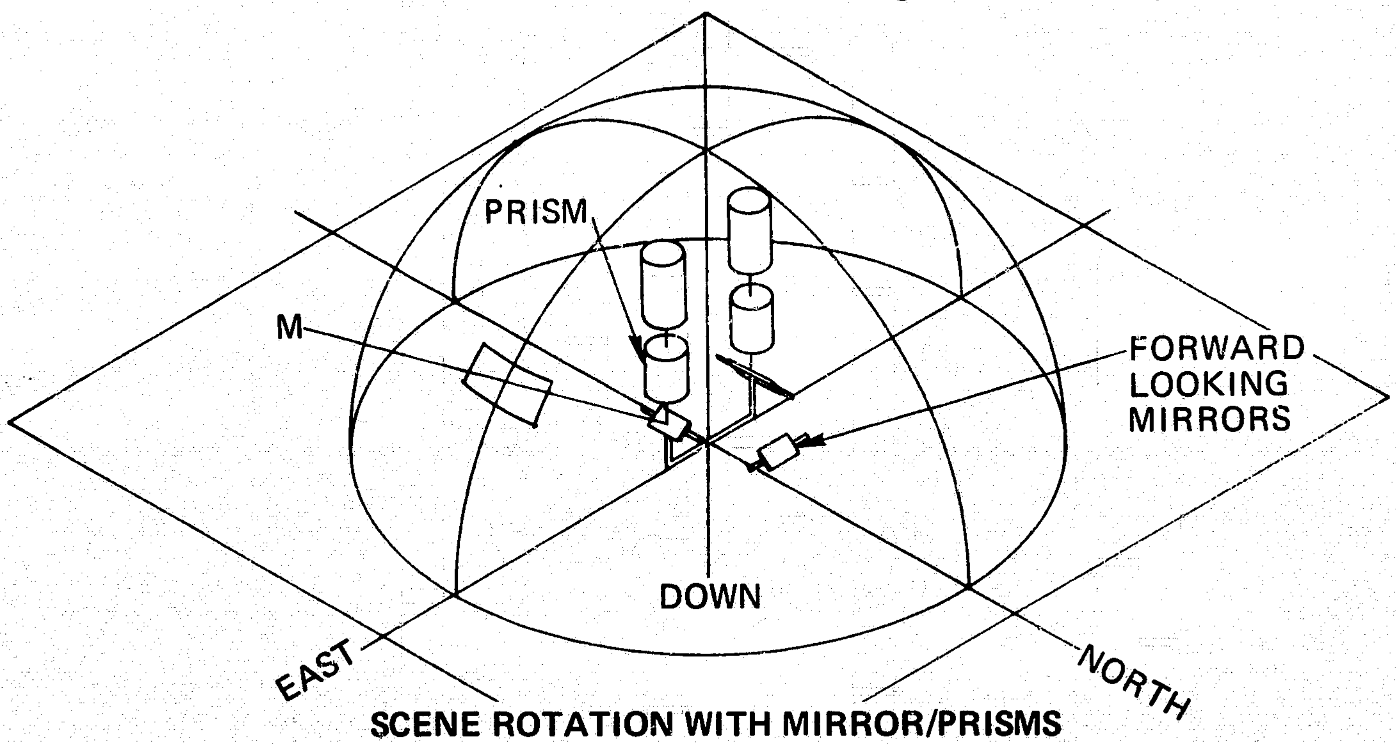
Pitch and Roll Mechanisms

Two basic design approaches for providing pitch and roll motion(s) were considered. One approach is to view the scene through a set of optical elements which incorporate mirrors, prisms, etc. to obtain the visual angles appropriately. The optical element method is widely accepted for single view systems where close approach to a physical model is needed. It is characterized by high response at low power. A second approach is to view the scene directly and move the sensor. Although direct, the sensor motion approach is relatively unique because most applications have fairly restrictive space requirements. Where more than one scene is required, the simplicity of the direct approach is immediately appealing.

Because of the multiple scene requirement, several combinations of the two basic approaches were considered as potential motion system designs. These concepts are outlined and discussed in the following paragraphs.

Stationary Cameras With Mirrors

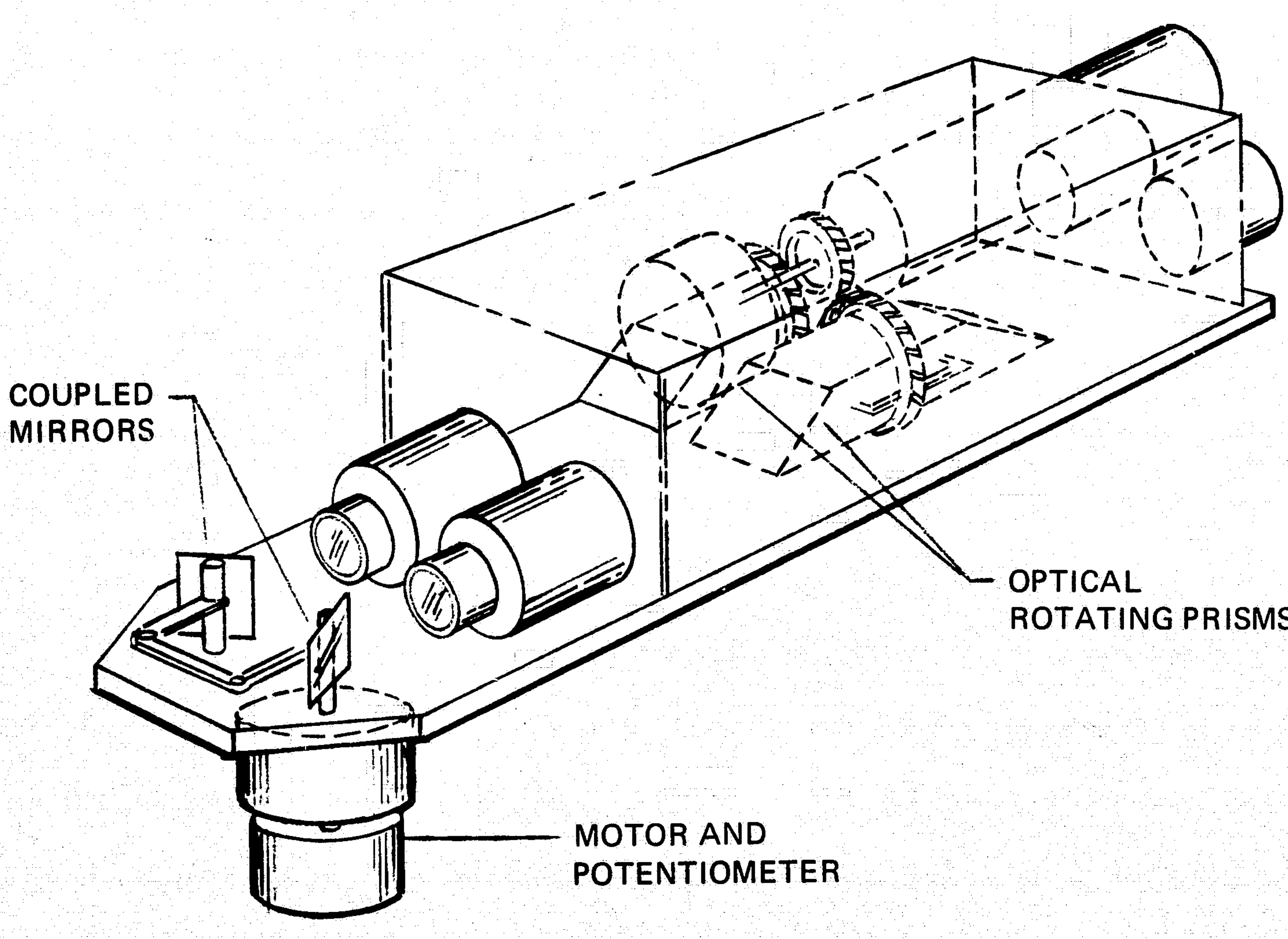
The type of mechanism shown in the conceptual sketch, Figure 8, is capable of providing rotational motion for two side looking scenes. The mirrors are coupled together and are driven directly by the motor/potentiometer package to produce a simulated rotation about one of the horizontal axes, North, in Figure 8 at twice the mirror deflection. The rotation of the side-looking mirrors are complementary.



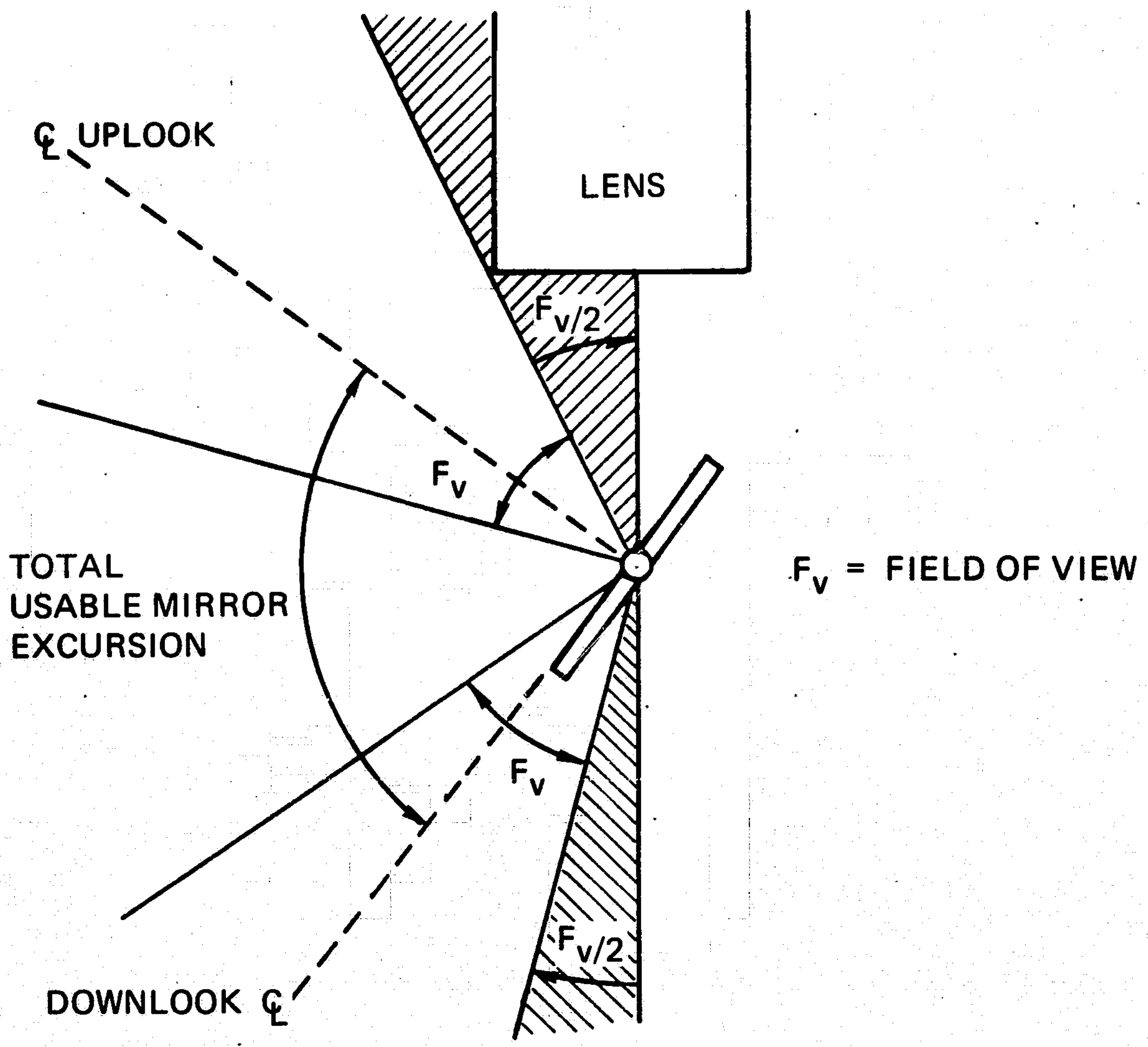
SCENE ROTATION WITH MIRROR/PRISMS
FIGURE 8

By rotating the prisms the sensor "window" will then be rotated about the axis M. A similar diagram could be drawn for a forward view by a single mirror system. In simulating a rotation of the Euler angle, Θ, while a displacement exists in the angle Φ, the axis M must trace a cone about East. See Figure 8. This establishes a need for a coordination rotation of the prisms and mirrors in addition to rotation of the scene coordinates about Down. It can be seen that this would not be compatible with the requirement for the forward looking sensor, consequently, a separate system for simulating the heading angle, Ψ, must be used.

An intrinsic limitation of the simple mirror rotation system depicted in Figure 9 can be seen by referring to the sketch in Figure 10. Note that practical limitations would restrict this excursion to somewhat less than indicated due to physical relationship of lens and mirror.



STATIONARY CAMERAS WITH MIRRORS
FIGURE 9



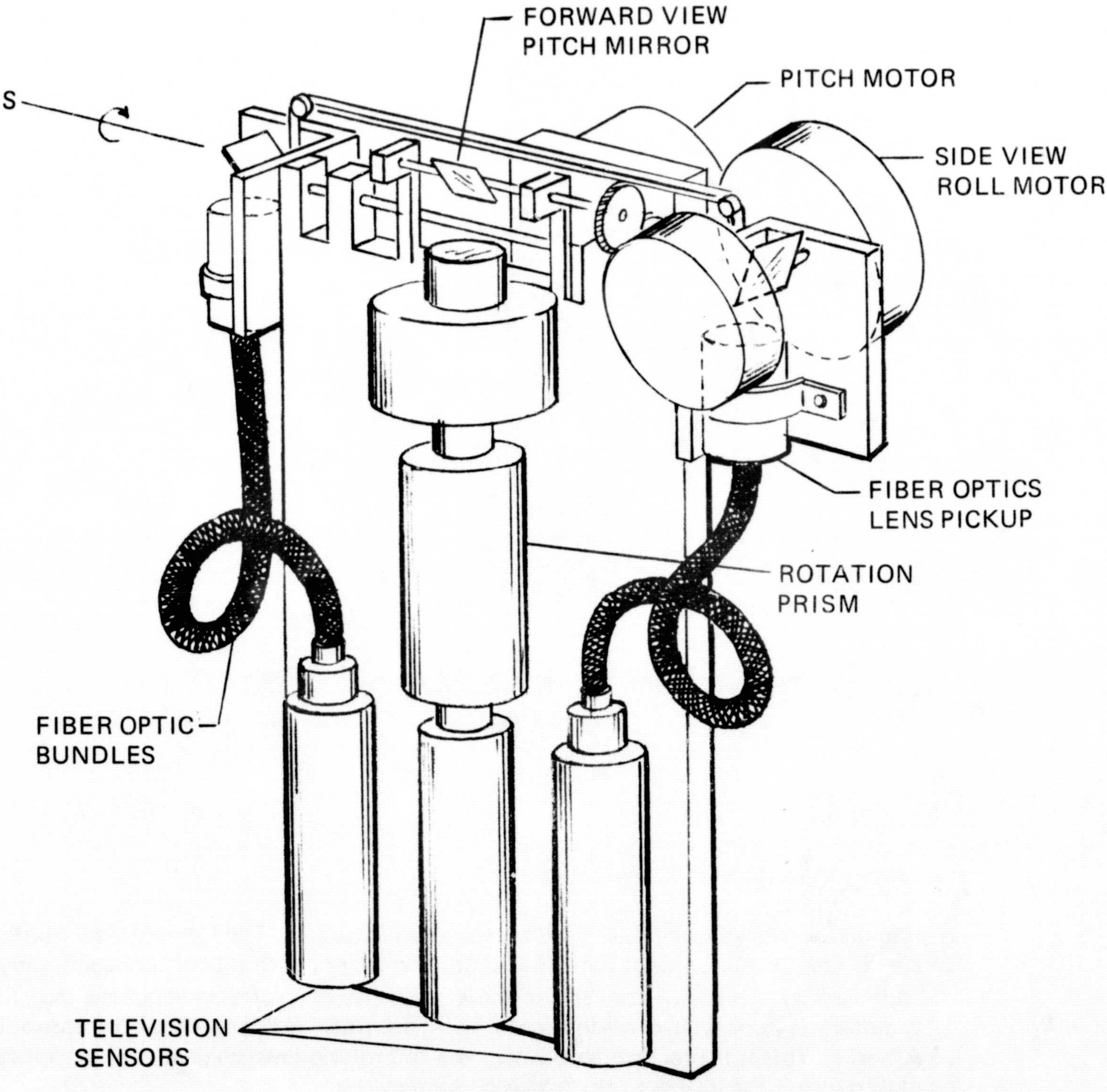
THEORETICAL MIRROR ANGLE LIMITATIONS
FIGURE 10

Stationary Cameras With Fiber-Optics

A potential approach using fiber-optics is shown in Figure 11. The forward view mirror system is identical in concept with the stationary camera with mirror approach, only the side view systems have the additional compatibility of a rotation about the axis, S. This motion is coordinated mechanically with the pitch motion, Θ, of the forward view system. It is not practical to connect the roll rotation mechanically. Fiber-optics bundles transmit the images to the television sensors.

The inertia of the fiber-optics lens pickup and holder would be larger than the inertia of a simple mirror but less than the television sensor. Large pitch angles would require enough fiber bundle length to prevent the bending of the bundles to less than minimum radius. The additional torque required to bend the bundles would be appreciable with respect to the dynamic forces involved.

One advantage of the additional side view rotation mechanism is that heading motion, Ψ, can now be simulated for both views by a single heading servosystem.



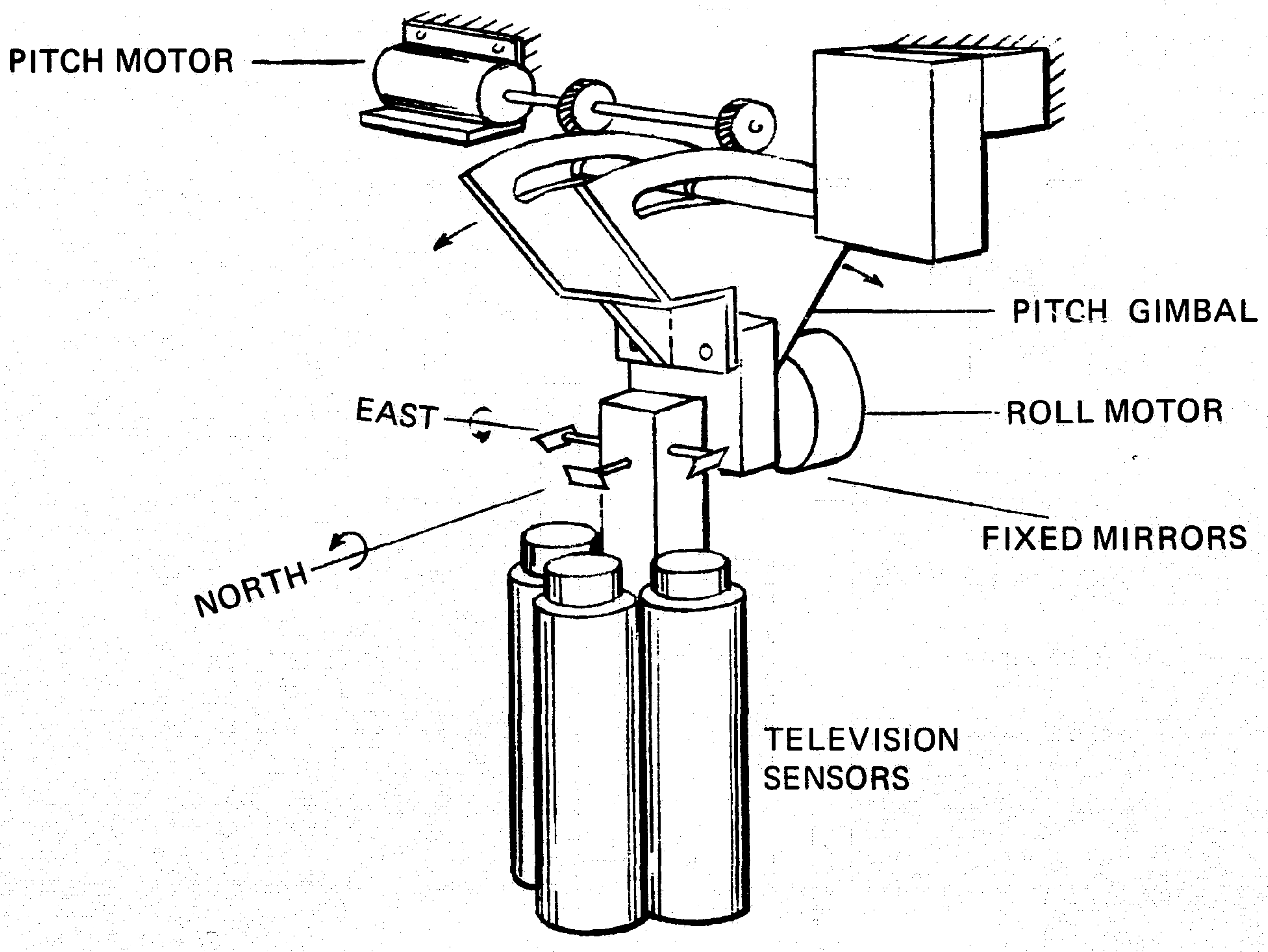
STATIONARY CAMERAS WITH MIRRORS AND FIBER-OPTICS
FIGURE 11

Gimballed Platform - Rotary Actuators

The practical matter of introducing large pitch angles and heading angles into the sensor platform without mechanical interference is the primary problem of rotating the sensors directly.

Figure 12 shows a concept providing roll motion about the axis, North, that is carried on a hubless pitch gimbal. The fixed mirrors allow the television sensors to be grouped more compactly to enhance the total possible roll displacement.

For efficient operation, a method would have to be devised to counteract the gravity component acting on the television sensors in the roll axis.



ROTARY ACTUATORS WITH CAMERAS
FIGURE 12

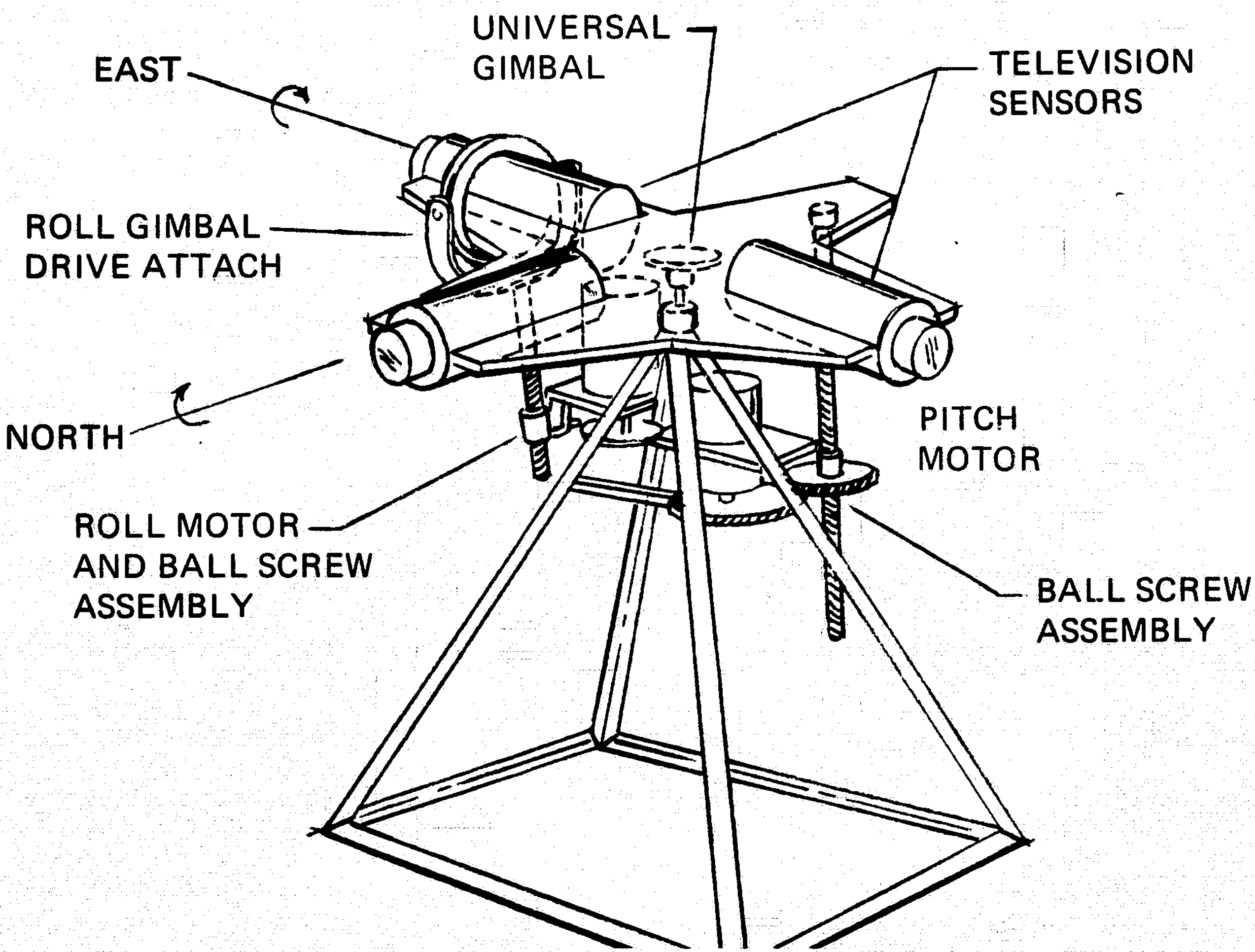
Gimballed Platform - Linear Actuators

A camera platform mounted on a universal gimbal is shown in Figure 13. This concept uses a ball screw assembly to rotate the platform in Θ about the horizontal axis, East. The universal gimbal will restrict motion to a vertical plane passing through the ball screw attach point. The drive motors and ball screw nuts will pivot about their attach point. The roll motion, Φ, is controlled by a roll motor and ball screw assembly. The attach point for the ball screw requires a two degree of freedom universal joint pivoting about the sensor centerline. The sensors should be located as close to the center gimbal as possible to minimize the platform and sensor inertia.

Allowing a more remote placing of the motors results in more available platform space. The roll thrust at large pitch angles will cause significant forces in the gimbal mechanism that restricts pitch to a vertical plane.

Gimballed Platform - Fiber-Optics

The television sensors in Figure 13 can be remotely located on the base of the support stand by transmitting the optical information through fiber-optic bundles similar to the method described in Figure 11. Some rotational inertia savings can be made with this approach; however, added static force will be required to bend the fiber bundle.



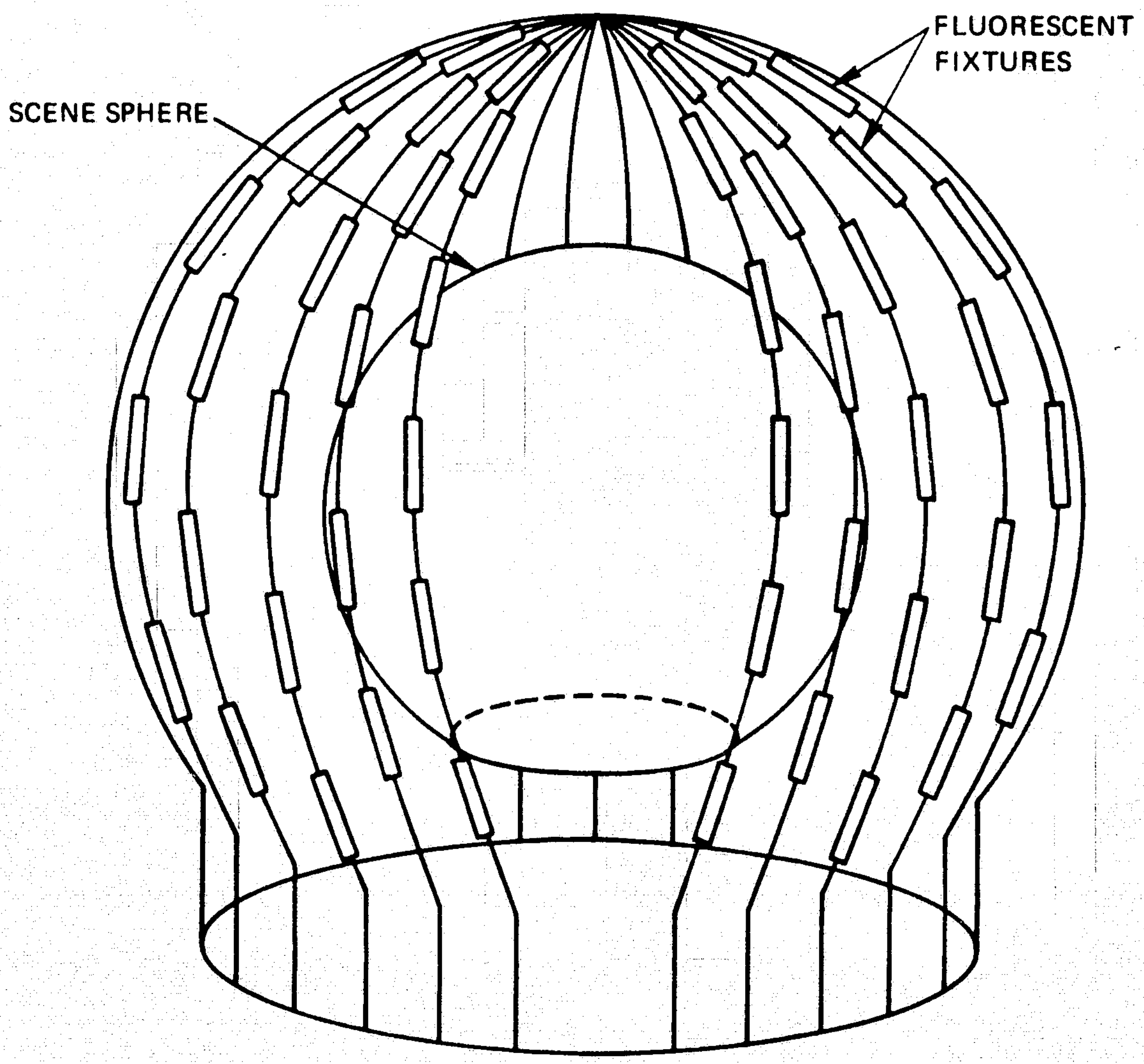
LINEAR ACTUATED GIMBAL WITH CAMERAS
FIGURE 13

Scene Illumination Considerations

A major consideration is the question of internal vs external scene illumination. This factor is greatly affected by the type of pitch and roll mechanisms used. For those systems which carry the sensors or lenses on a gimbal platform, the most direct approach is to mount light sources on the gimbal and illuminate the scene in front of each sensor individually. The scene would consist of paint or other opaque decorative materials on the inside of the sphere. Where stationary cameras are used it is impractical to mount the lamps on the pitch or roll mechanism because of space and mechanical complexity considerations.

Illumination of the total potential coverage of the servoscene system from fixed lamps inside the sphere would require many sources and could produce shadows and other unevenly illuminated areas.

A concept using external light sources is portrayed in Figure 14. Such a system would require at least 50% more space horizontally and 25% more ceiling height. This approach is deemed relatively cumbersome and inconvenient to maintain and ventilate. Appendix B contains calculations of the physical requirements.



EXTERNAL SCENE ILLUMINATION
FIGURE 14

DESIGN APPROACH EVALUATION

Evaluation Criteria Definition

A set of criteria was constructed to establish a basis for selection of a particular design approach. These criteria cover the areas of specific concern in the design, fabrication, and operation of a "Simulated Cruise Scene Visual Attachment." They provide the means for conveying the interpretation of the system requirements. The areas are listed below in order of relative importance along with a short description of each category.

1. Cost – Reflects the ease of fabrication and the type and quantity of materials required.
2. Reliability – The relative reliability rating is based on an estimate of projected maintenance and operational characteristics.
3. Resolution and Accuracy – An evaluation of the intrinsic accuracy and resolution of the implementation implied by the particular design approach.
4. Development – The degree to which state-of-the-art techniques are used. (A measure of development risk.)
5. Performance – An evaluation of the overall system design objectives.
6. Integration – Comparison of the ease of installation, and initial operational checkout, and the computer interface requirements.
7. Size/Shape – Criteria concerning the bulkiness and general shape outline. (Establishes whether the mechanism will efficiently utilize available space.)
8. Versatility – Comparison of potential future applications adaptation.

Comparison Matrix

The matrix shown in Figure 15 was constructed to compare the five design approaches. It provides a systematic method of viewing the total system. The eight evaluation criteria categories were assigned weighting factors on the basis of relative importance to the overall system design. These weighting factors were agreed upon in the design review with NASA-Ames. The sum of the columns under each concept provide the basis for selection of the design approach.

The cost of fabrication and the reliability of operation were judged to be major considerations of selection, since one of the main usages of the device will be to free a more elaborate facility for more sophisticated tasks. The resolution accuracy of the system was rated third in importance. The proper balance between proven methods and advanced techniques must be retained so that the resulting system will reflect state-of-the-art design without incurring large developmental risks. Designs requiring higher developmental risks were given lower scores.

MAX. POSSIBLE POINTS	EVALUATION CRITERIA / DESIGN CONCEPT	STATIONARY CAMERA		GIMBALLED PLATFORM			COMMENTS
		MIRRORS	FIBER-OPTICS	ROTARY ACTUATORS	LINEAR ACTUATORS	FIBER-OPTICS	
100	1. COST	55	60	60	60	40	1
100	2. RELIABILITY	45	40	70	80	80	2
80	3. RESOLUTION ACCURACY	60	60	60	75	70	3
70	4. DEVELOPMENT	65	55	60	60	55	4
50	5. PERFORMANCE	40	30	30	35	40	
40	6. INTEGRATION	30	30	35	35	35	
30	7. SIZE/SHAPE	10	30	30	30	30	7
30	8. VERSATILITY	15	15	20	25	30	8
500	TOTAL	320	320	365	400	380	

1. Fiber-Optic bundles are a high cost item.

2. The stationary systems must have view correlation mechanisms.

3. The stationary systems must have view correlation mechanisms and external illumination. A hubless sector drive is required for "rotary actuators".

4. The application of Fiber-Optics results in higher development risks.

7. The stationary/mirror approach requires two heading scenes.

8. Extra views are very difficult with the stationary/mirror approach.

COMPARISON MATRIX
FOR SELECTION OF DESIGN APPROACH

FIGURE 15

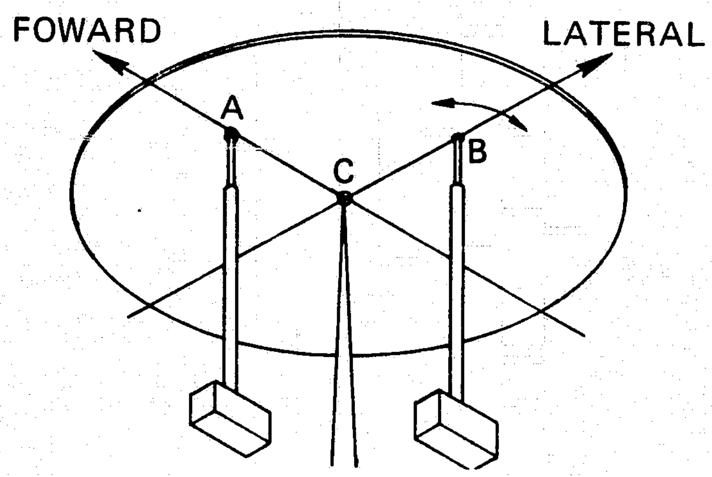
Versatility or adaptability to potential future applications, although difficult to assess, gives a measure of the relative ease that such factors as extra views, larger angles, and full color can be added.

Evaluation Results

The evaluation matrix shows that the gimballed platform system with linear actuation is rated highest. There is not a wide point spread between any of the design concepts, however the cost is the main difference between the two higher rated concepts. The most significant comparison considerations of each row of the matrix are listed under comments.

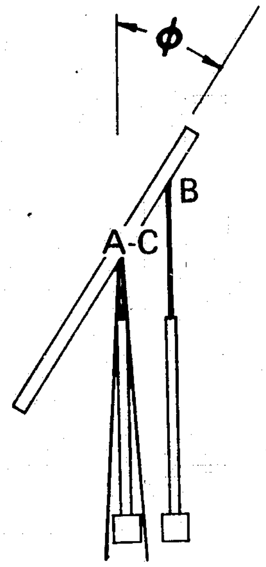
Refinement of Selected System

The vidicon cameras can be located near the center of the platform rotation when each servoactuator is mounted remotely and independently drives the platform relative to the ground reference plane. However, consideration must also be given to the gimbal restraint structural problems that this configuration will cause when large angular displacements are commanded concurrently. An example of a gimballed platform with independent linear actuators is shown in Figure 16.



TWO LINEAR ACTUATOR ARRANGEMENT
FIGURE 16

The platform is restrained from rotating about the central pivot point, C, in the plane ABC. At 65^{o} displacement from horizontal about the axis A-C (See Figure 17), a command for rotation about the axis B-C will produce a torque in the plane ABC of over twice the torque about B-C.



ANGLE RELATIONSHIP WITH TWO LINEAR ACTUATORS
FIGURE 17

The implementation of a device to restrain this motion destroys the space advantage. As a result, a decision was made to use a rotary actuator for roll. To minimize the inertia contribution of the motor/tachometer these are placed at the center of rotation of both axes. The adoption of this approach has the added advantage that the pitch actuator need have but one axis of freedom. This is accomplished by design of an outer gimbal so that the pitch actuator drives the roll motor case.

Final System Specifications

The initial system specifications as defined in the proposal document are the minimum guidelines used to establish the design of the Cruise Scene Visual Attachment. As agreed in the design review at NASA-Ames, the requirement for ± 180^o travel in roll is not consistent with the chosen design approach. With this one exception, the intent of the original requirements is reflected in the following final design characteristics.

Roll Axis

Range	± 70 degrees
Rate	1.5 Rad/Sec
Acceleration	5.8 Rad/Sec^2
Accuracy	± 0.33 degrees

A change from the initial ± 180 degrees is required for the chosen approach because roll is a rotation of the cameras. The side-looking cameras cause interference with the main platform support at angles in excess of 70°.

Pitch Axis

Range	± 70 degrees
Rate	3.0 Rad/Sec
Acceleration	16 Rad/Sec^2
Accuracy	0.33 degrees

The final pitch specifications exceed the initial system characteristics as defined in the proposal document. The pitch drive system is operated as a position servo. Since the ball screw mechanism introduces nonlinear motion (linear to rotary) the position potentiometer is mounted between the camera platform and the stationary support so that the potentiometer is sensing actual platform angle in pitch. This eliminates the need to use nonlinear elements in the computer program at the expense of nonlinear loop gain. The system acceleration specification is met in this axis at the extreme angles (± 70 degrees) which is the design limit case.

Heading Axis

Range	Continuous
Rate	1.5 Rad/Sec
Acceleration	5.0 Rad/Sec^2
Accuracy	± 1.0 deg/sec

The heading axis design will meet or exceed all of the required specifications. Continuous heading angles are obtained with a velocity servosystem. A position potentiometer is switched into the system when an initial reference position is desired.

Video

Scanning Rates	
Horizontal	525 lines
Vertical	60 hertz
Sensor Tube Type	Vidicon
Video Output Impedance	75 ohms
Size	2.75" x 2.75" x 10.3"

Weight	3.5 pounds (less lens)
Synch Gen	EIA Standard (RS-170)
Illumination	
Source intensity	183 ft. candles
Scene illumination	81 ft. candles
Design Volts	13 V
Watts	20 W
Max Candle Power	225 c.p.
Illum. cone per lamp	44^{o} (50% power points)
Rated life	300 hrs.

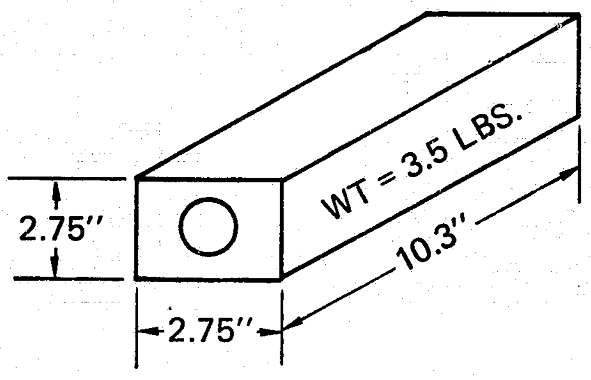
SYSTEM DESIGN FOR SELECTED APPROACH

Roll Platform

The platform size and shape is determined mainly by the number and the physical size of the cameras, the size of the roll motion actuator, and the illumination sources carried on the platform. The platform and component size must be kept small to minimize the inertia of the system.

Camera Selection

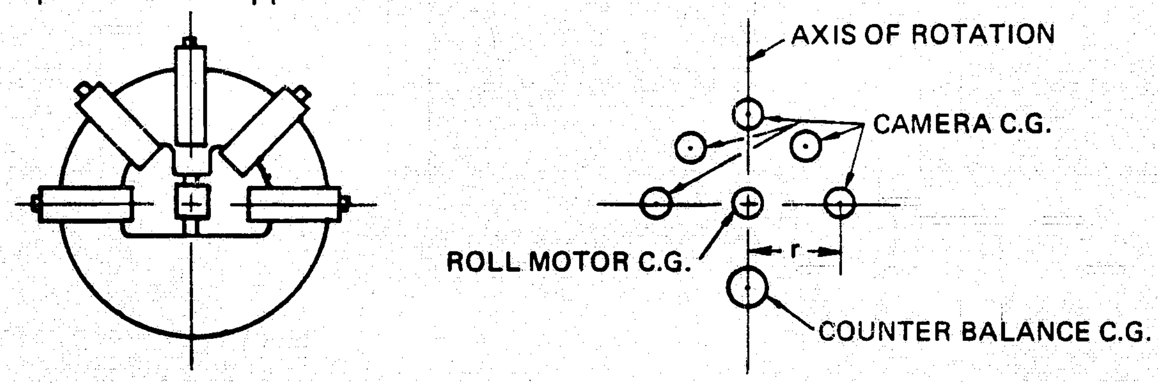
The MTI Model VC-20 remote head vidicon camera was selected. The physical size and dimensions are shown in Figure 18. The field-of-view and inertia problems dictated that the smallest available camera be used. This camera is available with external EIA synchronization and will meet RS-170, "Electrical Performance Standards, Monochrome Television Studio Facilities."



TELEVISION CAMERA PHYSICAL CHARACTERISTICS
FIGURE 18

Servomotor

A Magnetic Technology torque motor, Model 5125-220-023, will be used at a one-to-one gear ratio to provide the ± 70 degree roll motion. The moment of inertia of the platform assembly was determined by using a point mass model shown in Figure 19. Moment of inertia, speed, power, heat dissipation, and response calculations are presented in Appendix C.



ROLL PLATFORM MASS DISTRIBUTION
FIGURE 19

The maximum roll motor torque required to meet a roll acceleration of 5.8 rad/sec^2 with a platform moment of inertia of 0.577 lb-ft-sec^2 is 3.3 lb-ft.

Tachometer and Potentiometer

An Aeroflex Laboratories Tachometer, Model TG 52W-5, was selected for a design guide for the roll system angular velocity sensor. This unit is brushless and was chosen for high signal-to-noise ratio and sensitivity over the required velocity range. A CIC Model 205, Standard Linear, 50 KΩ, 0.1% linearity, 4 watt potentiometer was selected. This potentiometer is mounted on the motor/generator housing to provide angular position feedback. The potentiometer is an infinite resolution Cermet type with a linearity of 0.1%. This provides an accuracy of 0.18° in position. The shaft is geared at a 2:1 ratio with the roll platform. This gear ratio permits a large portion of the potentiometer to be used for improved accuracy.

Pitch Mechanism

Servomotor

A Magnetic Technology, Model 5125-220-008, motor will meet the pitch axes requirements. The maximum rate demanded of the motor occurs at zero degrees pitch. The maximum pitch motor torque required to meet a pitch acceleration of 16 rad/sec^2 with a platform inertia of 0.032 lb-ft-sec^2 is 2.5 lb-ft. Calculations of motor performance requirements are found in Appendix C. An overspeed condition is required to achieve the specified maximum pitch rate of 3 rad/sec. The overspeed situation is carefully defined in Appendix C. The motor and amplifier can operate reliably in this manner without damage.

Tachometer and Potentiometer

A Magnetic Technology Tachometer, Model 5125-B-058 was chosen as a design guide. This tachometer will fit in the same diameter case as the motor. The tachometer is selected for its sensitivity, moment of inertia, physical size and maximum allowable speed. At a maximum rate of 43 rad/sec, the tachometer output is 54.2 volts or approximately two-thirds of maximum rated voltage.

A CIC Model 78 potentiometer with a value of 50 K Ω , a linearity of 0.15%, and an infinite resolution Cermet element is selected. The pitch angle accuracy of 0.33 degrees requires a potentiometer linearity of 0.236%. The pitch servodrive is nonlinear between the motor and the platform. The potentiometer is mounted to the roll motor support yoke and the shaft is geared at a 2:1 ratio.

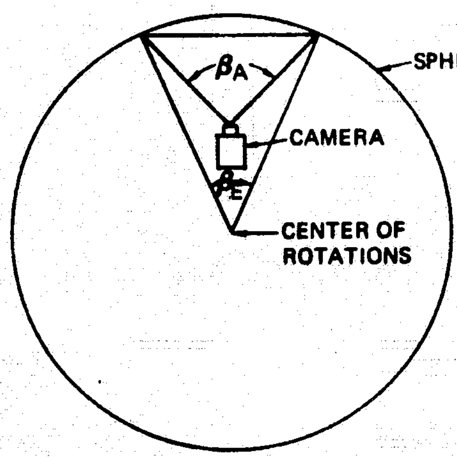
Heading System

Sphere

The sphere surrounds the pitch/roll mechanisms, has an inside diameter of 74.5 inches, and weighs approximately 80 pounds. A 30 inch diameter opening provides access to the pitch/roll mechanisms and clearance for their rigid fixed base support.

The positive drive belt method of coupling to the drive motor is selected because of simplicity, quietness, and efficiency. A Worthington belt with a one-half inch pitch and one inch width couples the 36" gear on the sphere to the 3" pulleys on the drive motors.

The sphere will be made of an opaque plastic which is required for internally illuminating the scene to be painted on the inside. Lamps will be mounted on the gimballed platform with the cameras. The camera lens position will be approximately 18 inches from the platform center as determined by a physical layout. A one inch vidicon tube with a lens focal length of 5.3 millimeters will provide a horizontal field-of-view of 84°, which will result in a 46° scene window when the camera is located in its established position. Appendix E provides additional calculations for an alternate lens. Figure 20 shows the relationship between the field-of-view, β_E, relative to the platform center and the camera lens field-of-view, β_A. Calculation are given in Appendix C.



CAMERA POSITION IN SPHERE
FIGURE 20

Servomotor

An Inland Motor Corporation, D. C. torque motor, Model 5730, is selected for the heading drive system. This motor is rated at 7.0 lbs-ft. The motor capability to provide required system response, power, rate, and temperature limitation is discussed in Appendix C.

Tachometer and Potentiometer

An Inland Motor Corporation tachometer, Model TG-2801 is chosen for the heading drive system. The tachometer is selected for its sensitivity, moment of inertia, physical size and maximum allowable speed. The tachometer output is 0.635 volts/radian/sec.

A CIC potentiometer, Model 205, rated at 50 KΩ, 0.1% linearity, and 4 watt is selected for the heading system. It is mounted on the motor/tachometer housing and driven by the motor shaft at a 12:1 ratio which provides a 1 to 1 ratio between potentiometer and sphere. When the potentiometer is switched into the servosystem, it will provide position feedback. This establishes a reference point when switched in initially.

Scene Illumination

The illumination source used for a design guide is a General Electric No. 1383 miniature reflective lamp with a frosted envelope. This lamp has a 225 candle power rating. Four lamps per camera are necessary to provide a uniform coverage of light. The lamps will be mounted around and near the video television camera lens. The candle power distribution, Figure 21, is down 3.4 dB of maximum at 100 degrees of coverage with the lamps at an angle of 50 degrees with respect to each other. The life of these lamps is approximately 300 hours which is extended by a factor of three when powered by an 11.5 volt source. This reduction in voltage lowers the power requirement of the lamps by 10% for a total of 300 watts while reducing the illumination by approximately 25%. This increase in operational reliability will cause an insignificant loss of picture quality.

A cooling fan mounted at the base of the sphere extracts the heat dissipated by the illumination lamps and the motors. A total of 400 watts is the maximum power which will be dissipated within the sphere.

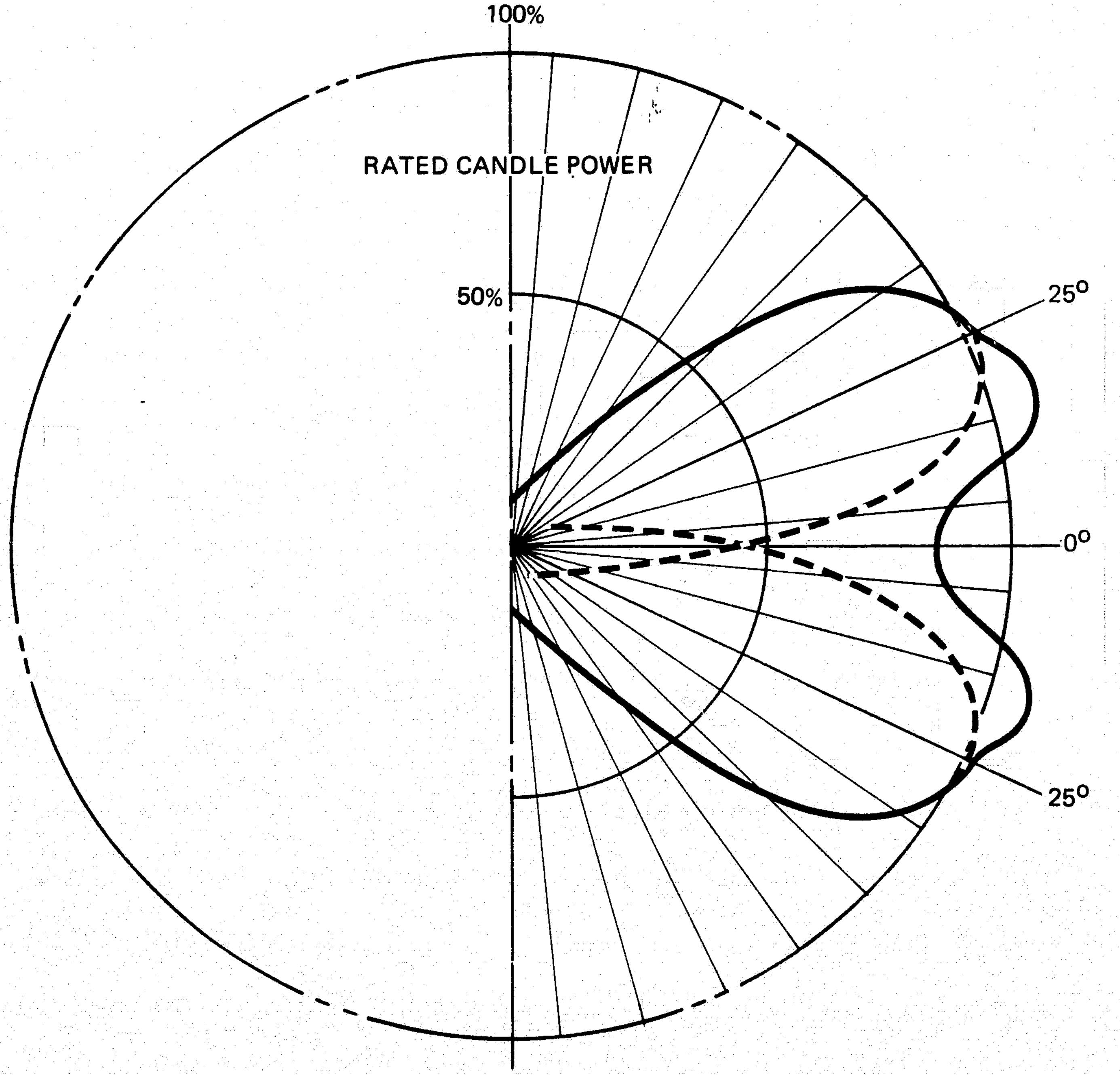
Figure 22 shows the geometry of the illumination angles and the required depth of field. The lighting source must adequately illuminate a solid angle of at least 100° to ensure coverage of the television raster scene. At an object distance of 18 inches, a lens setting of f/4 is adequate to provide the desired 4.2 inch depth of field. An illumination level at the scene of 56 foot candles is sufficient to ensure high picture quality from the vidicon camera. Appendix B provides calculations of the illumination values.

Electronic Design

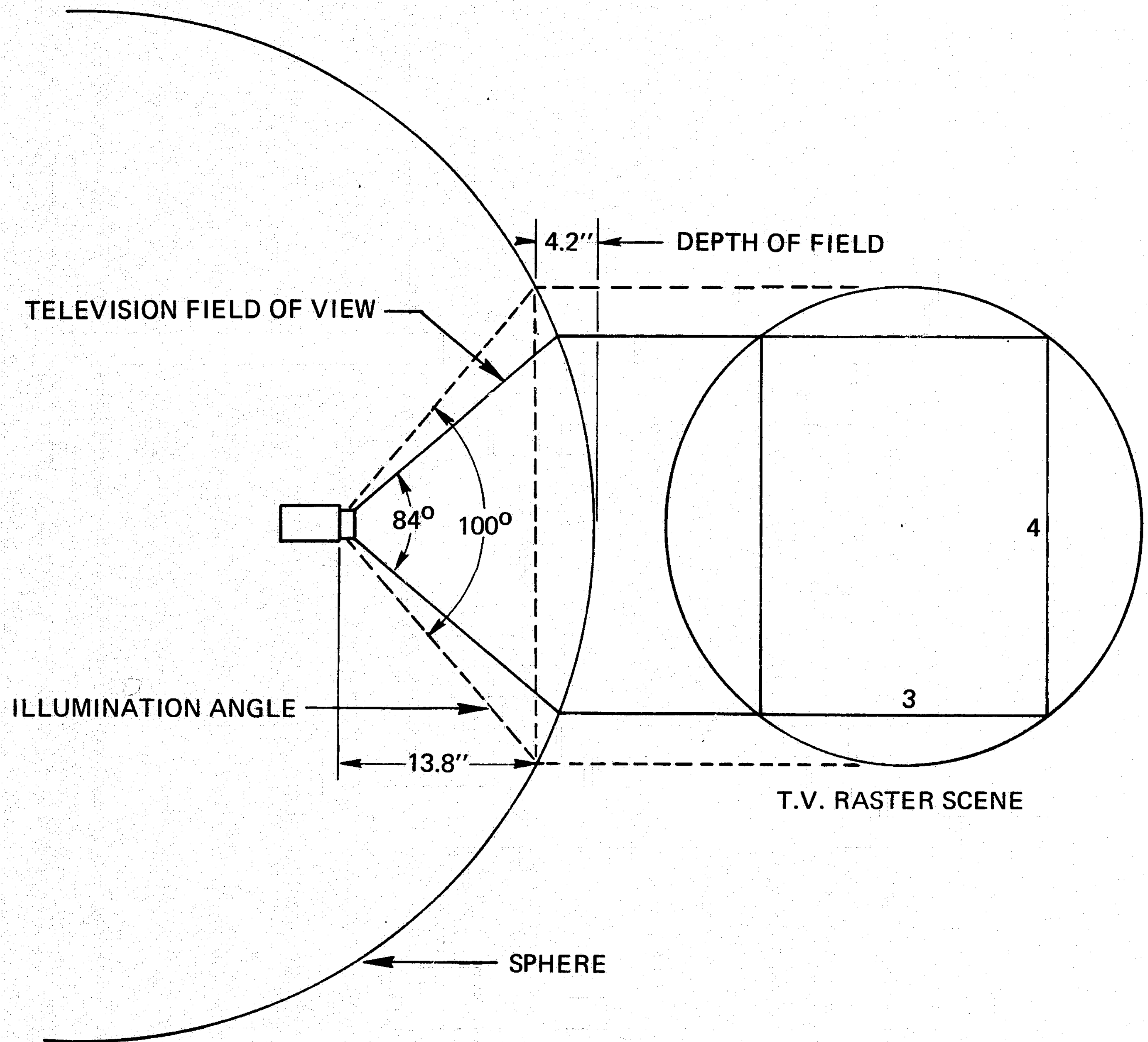
Analytical models of the Visual Display System were used to determine system gain requirements. An analog simulation was performed to verify system response. A specific type of feedback control was selected to give the required response characteristics for each system.

Heading

A Control Systems Research amplifier Model 500 PRA, is selected to drive the heading servomotor. The primary control variable of the heading servo is velocity. The closed loop system gain of 10 is established to provide smooth motion of the servo at the lower end of the velocity range.

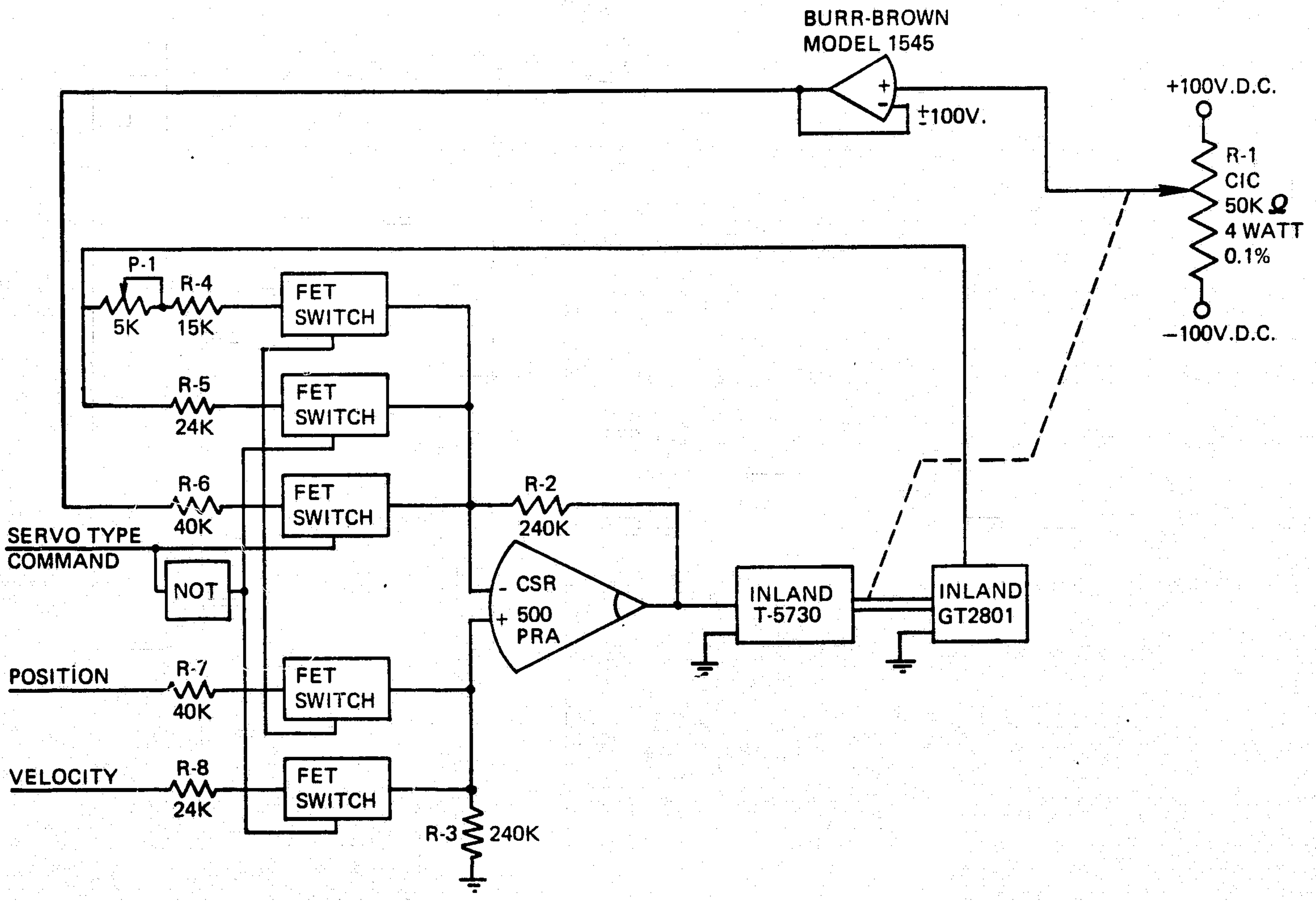


ILLUMINATION SOURCE PATTERN
FIGURE 21



ILLUMINATION ANGLE REQUIREMENTS
FIGURE 22

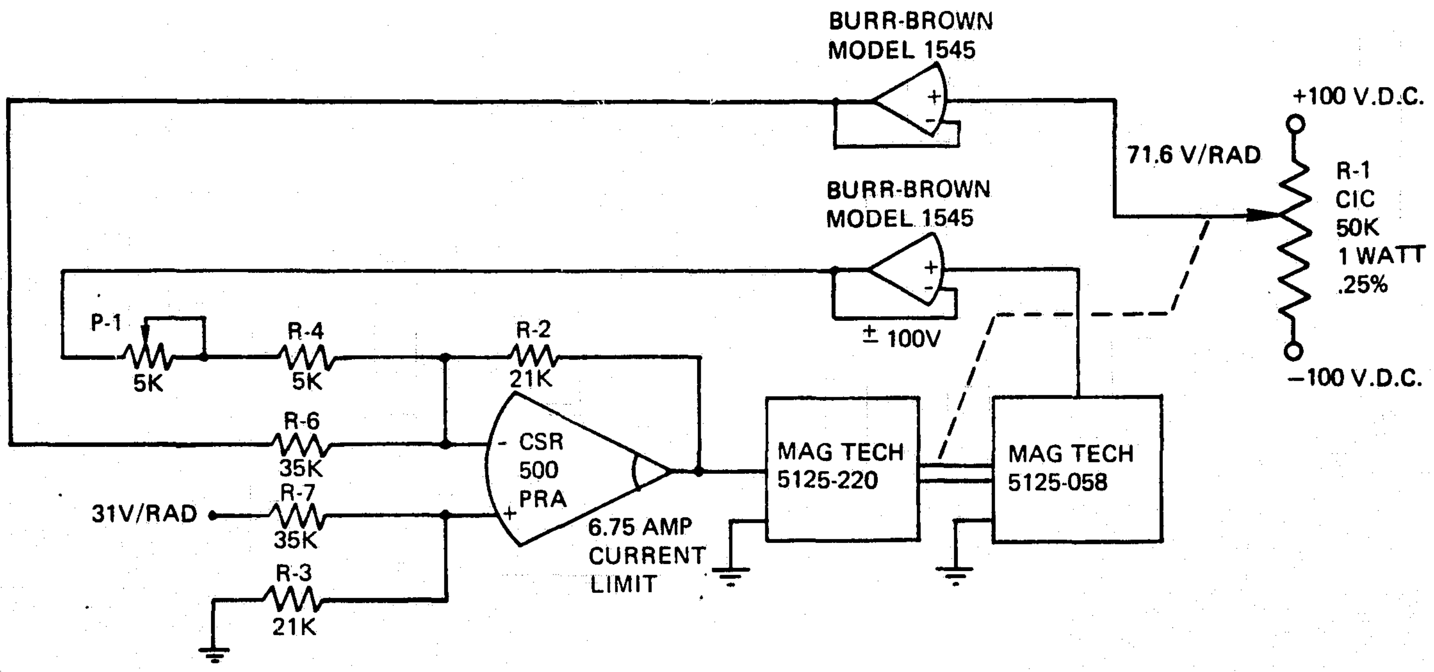
The gain of the tachometer is altered and potentiometer feedback is added to convert the servo to a position system when in the initial condition mode. The servo has a tachometer feedback gain in this mode of 2.55 and a closed loop system gain of 190. A system wiring diagram is given in Figure 23. The simulation output responses verify the calculated position and velocity feedback gains and response curves for step inputs are shown in Appendix D.



HEADING SERVO WIRING DIAGRAM
FIGURE 23

Pitch

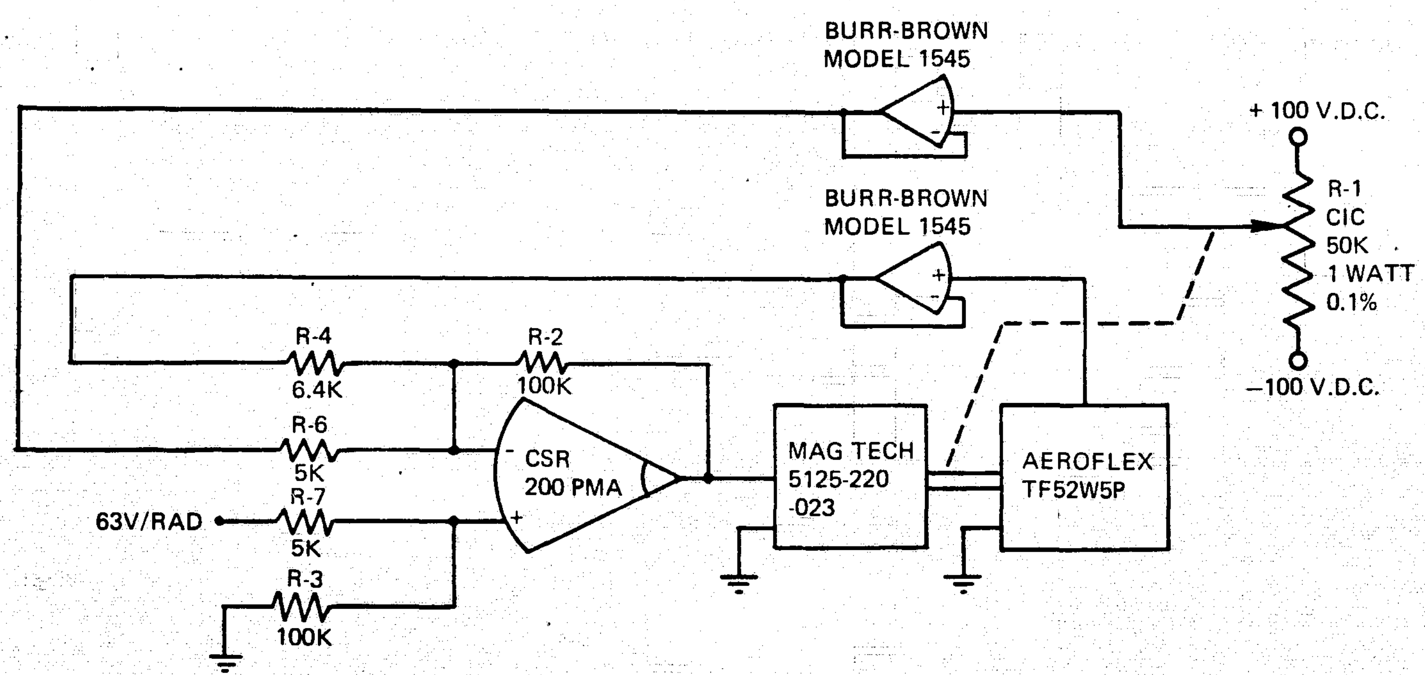
A Control Systems Research amplifier, Model 500 PRA, is selected to drive the pitch servomotor. Adjustable voltage and current limits of 25 volts and 6.75 amps respectively, are provided in the amplifier to guard against overdriving the motor. The primary control variable for the pitch servosystem is position. Velocity is used for system damping. A nonlinear motion exists between the ball screw actuator and the gimballed platform. Gain for the velocity feedback is 2.48 for the system with a damping ratio of 0.7 and a closed loop gain of 43. The pitch system wiring diagram is shown in Figure 24. Calculations of pitch system parameter values may be found in Appendix D.



PITCH SERVO BLOCK DIAGRAM
FIGURE 24

Roll

A Control Systems Research amplifier, Model 200 PMA, is selected to drive the roll servomotor. The primary control variable for the roll servosystem is position. Velocity is used for system damping. The velocity feedback gain is 0.78 for the system with a damping ratio of 0.7 and a closed loop gain of 1140. The roll system wiring diagram is shown in Figure 25. Calculations of roll system parameter values may be found in Appendix D.



ROLL SERVO WIRING DIAGRAM
FIGURE 25

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COST ESTIMATE

This is a budgetary cost estimate. The costs of components, materials, and labor are submitted by major assemblies. Commercially available parts costs are based on catalog prices and specially fabricated items were costed from unofficial quotes. The estimate is based on a three camera system.

Roll Platform

Item Description	Cost
Television Cameras & Controls	$ 9,000
Sync Generator	900
Motor & Tachometer	1,000
Potentiometer	100
Lenses	1,800
Materials	250
Labor	6,650
	$19,700

Servodrive Electronics

Item Description	Cost
Servoamplifiers	$ 3,000
Cabinet	300
Electronic Components	600
D.C. Power Supplies	400
Materials	1,000
Labor	7,600
	$12,900

Pitch Stand

Item Description	Cost
Linear Actuator	$ 6,500
Potentiometer	100
Materials	250
Labor	3,230
	$10,080

Heading & Scene Sphere

Item Description	Cost
Sphere	$ 1,500
Motor & Tachometer	675
Potentiometer	100
Materials	250
Labor	3,800
	$ 6,325

Total System Documentation

Item Description	Cost
Assembly, Checkout & Response	$ 6,080
Operation & Maint. Manual Preparation	3,040
	$ 9,120

Total Estimated Cost – Three Camera System

Equipments	$25,975
Materials	1,750
Labor	30,400
	$58,125

Estimate – Five Camera Additional Cost

Television Cameras & Controls	$ 6,000
Sync Generator	900
Lenses	1,200
Materials	75
Labor	3,800
	$11,975

MANUFACTURER	MODEL	CAMERA HEAD		RESOLUTION	RS-170 SYNC. GEN.
		SIZE	WEIGHT		
MTI	VC-20	H-2.75" W-2.75" L-9.75"	3.5 LB.	H-800 V-375	NOT INCL.
COHU	2006-011 3951-511	D-3" L-11.5"	5.3 LB.	H-700 V-400	INCL.
FAIRCHILD	TCS-950B	D - 2.88" L - 14.25"	5.0 LB.	H-800 V-350	INCL.
DIAMOND POWER	SF3	D - 4" L - 14.25"	9.0 LB.	H-1200 V-350	NOT INCL.
GENERAL ELECTRIC	4TE26BIC 4PX76AI	H-6" W - 4" L - 12.5"	11.0 LB.	H-800 V-350	GRASS VALLEY RS-170 MONOCHROME 900 SERIES $1105

TELEVISION CAMERA COMPARISON CHART

FIGURE 26

Fiber-Optics

Typical specifications and characteristics for present day fiber-optics are listed as follows:

- Resolution of up to 50 line pairs/mm.
- Fiber-optic light acceptance angle ± 30° maximum.
- Minimum bending radius of 4 - 6 inches for 1/2 inch diameter bundle at a distance of at least 2 inches from the mounting hardware.
- Available with reliable service up to 7 million S-bends and .25 million rotations ± 170° with no failure. (The maximum limits are dependent upon the cable construction and housing connections.)
- The weight of an 8 x 10 mm fiber-optic bundle in a stainless steel sheath will average greater than 4 pounds per foot.
- Approximately 10% light loss per foot length.

To transmit a high quality image through a coherent fiber-optics bundle requires that:

- The static image resolution exceed that required by the television system requirements.
- The flexing of the fiber-optics bundle have minimal fiber breakage and high reliability.
- The weight of the required fiber-optic bundle and its associated mounting hardware be minimal.
- The minimum bending radius of the fiber-optics bundle be much less than the bundle lengths.
- The viewing angles of light acceptance be greater than the desired lens field-of-view.
- The method of light coupling from the fiber-optics bundle to the vidicon provides a high quality image. The state-of-the-art in quality coherent fiber-optics can provide a static resolution of about 50 line pairs/mm using 10 micron fibers. The scanned faceplate of a vidicon is 12.7 mm by 9.5 mm. A horizontal resolution of 500 lines requires a fiber-optics bundle size of

$$D \frac{4}{5} \times 100 \text{ lines/mm} = 500 \text{ lines}$$

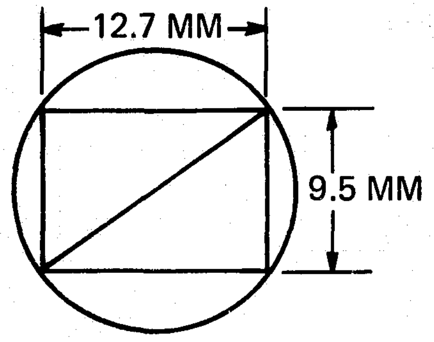
or

$$D \frac{25}{4} \text{ mm} = 6.25 \text{ mm minimum}$$

or

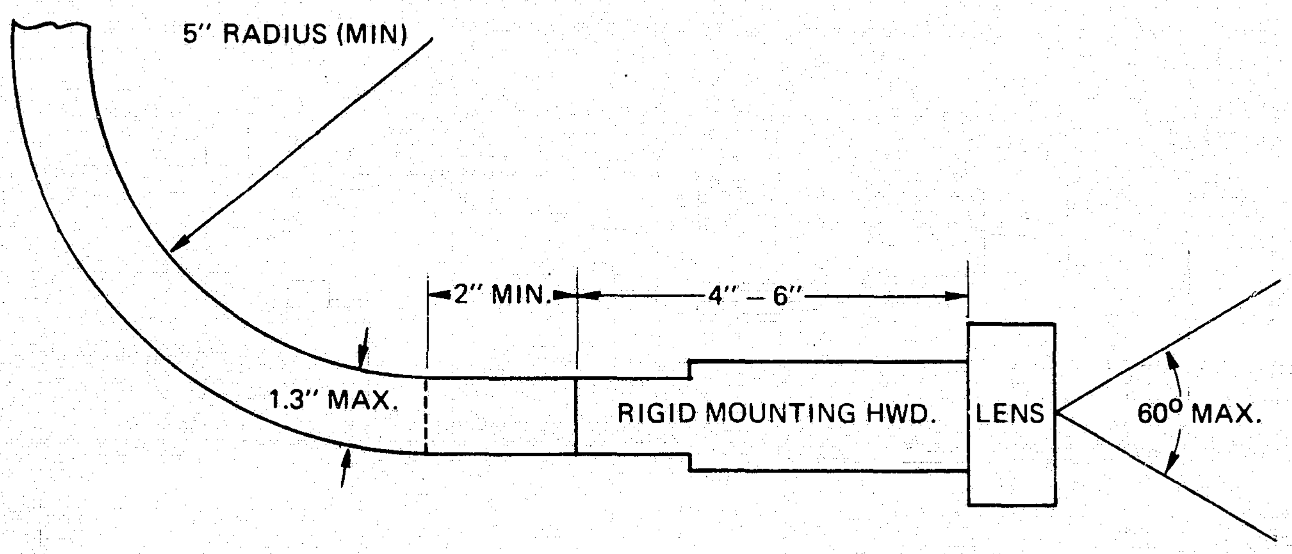
$$\text{Approximately } \frac{1}{4} \text{ inch diameter bundle}$$

Since the faceplate of the vidicon (See Figure 27) is greater than 6.25 mm, optical coupling is needed to project the image from the fiber-optic face plate to the vidicon camera. Fiber-optic faceplates could be used to expand the 6.25 mm bundle to the required size for the vidicon faceplate but additional losses of 30% in the resolution and 25% in light will occur due to the misalignment of the fibers at the fiber-optic bundle interface. A maximum resolution of 500 lines or greater requires fiber-optics with a 1:1 faceplate bonding to minimize resolution and light loss. Weight and flexibility become limiting factors.



VIDICON RASTER LAYOUT
FIGURE 27

Figure 28 shows a typical fiber bundle termination. Excessive light losses require the field-of-view be restricted to less than 60°. Lens coupling and mounting hardware should be 4 inches or larger with a minimum of 2 inches allowed for cushioning before any bending is imposed, (See Figure 28). The bending radius can be shortened but this will result in increased fiber breakage.



BENDING RESTRICTIONS OF A TYPICAL 8 X 10 MM FIBER-OPTICS BUNDLE
FIGURE 28

APPENDIX B - ILLUMINATION CALCULATIONS

The illumination (E_V) on the face of the vidicon must be .1 $< E_V <$10 foot candles for adequate scene illumination. (An illumination of 2 foot candles was chosen). The light reflected from the sphere scene is assumed to be 50% of the incident illumination.

The solid angle of illumination must be greater than, or equal to, the television camera lens field-of-view (At least 100^o).

The angle between the television camera and the lighting as viewed from the scene is approximately zero. The illumination on the surface of the vidicon tube behind the lens is computed, by

$$E_V = \frac{B\pi}{4(\text{f-number})^2}$$

where E_V is required illumination in foot candles on the vidicon faceplate, B is the brightness of the sphere scene in lumens and f-number is the F-stop setting of the television camera lens. The value of B is given as,

$$B = .5\, E_s$$

where E_s is the illumination of the inner sphere scene. The illumination E_s in foot candles is given by

$$E_s = \frac{I_l \operatorname{Cos} \sigma}{R_l^2}$$

where I_l is the intensity of the source lighting in candle power and R_l is the distance from the source to the inner sphere surface. The intensity requirement of a point source lamp at a distance R where Cos σ = I is

$$I_l = E R_l^2$$

or

$$I_l = 2\, B R_l^2$$

$$I_l = \frac{2R_l^2 \; E_V \; 4(\text{f-number})^2}{\pi}$$

Substituting the values for E_V, R_l^2 , and f-number,

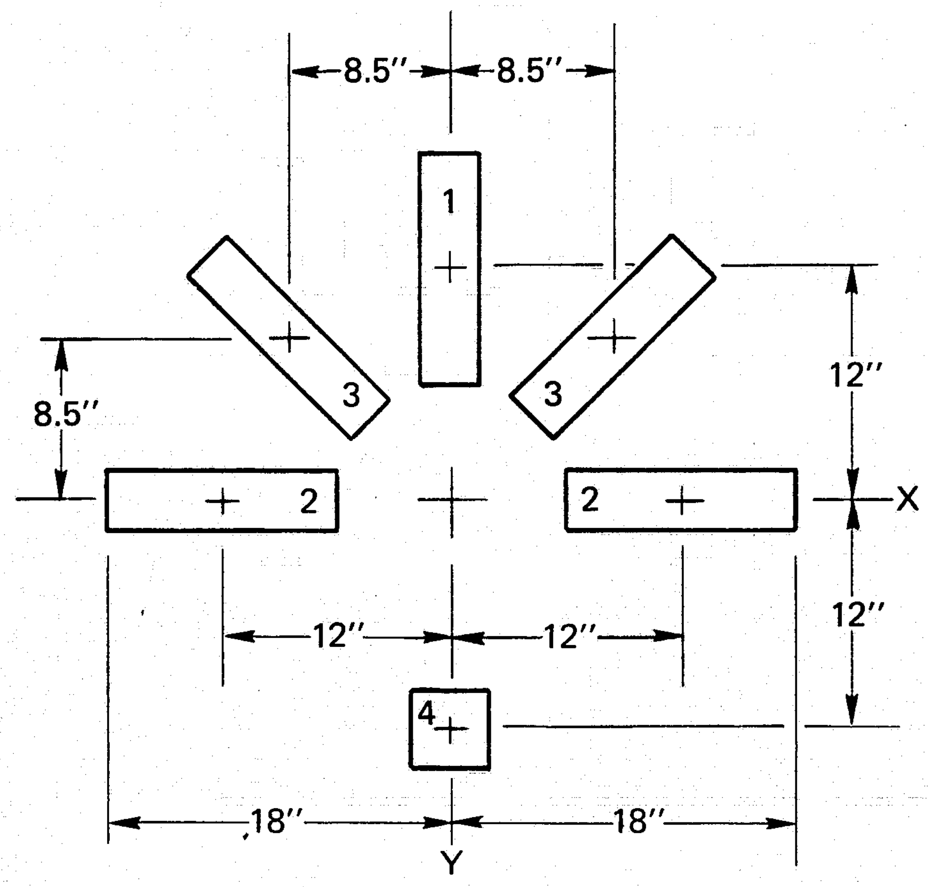
$$I_l = \frac{2\,(1.5)^2\;(2)\;(4)\;(4)^2}{\pi} = 183 \text{ candle power.}$$

The General Electric No. 1383 lamp mounted in groups of four lamps per camera with an angle of 50 degrees separation between the lamps will provide an average illumination intensity of approximately 95% of rated candle power (or 210 candle power) over an illumination angle of 93 degrees. When the light of this group of lamps is combined with the adjacent lighting groups the angle of illumination will be uniform within 2.5 dB over the entire viewing area at an illumination level greater than or equal to the minimum value required.

APPENDIX C - DESIGN CALCULATIONS

Counterbalance

The platform moments of inertia for the roll and pitch axes will consist of the sum of the moments of the major components of the platform plus a counterbalance. The counterbalance weight is calculated by balancing the moments about the axis as shown in Figure 29.



PHYSICAL LAYOUT OF CAMERAS AND COUNTERBALANCE
FIGURE 29

$$M_1 = (1\ \text{ft})\ (5\ \text{lb}) = 5\ \text{lb-ft}$$

$$M_3 = (.707\ \text{ft})\ (5\ \text{lb}) = 3.5\ \text{lb-ft}$$

$$M_4 = (5\ \text{lb-ft}) + (7.0\ \text{lb-ft}) = 12\ \text{lb-ft}$$

The lever arm of the counterbalance = 1 ft.

The weight required is 12 pounds.

Roll Axis

The roll axis moment of inertia is calculated as follows.

$$I_{R(1)} = \left[\frac{1}{12}\right] m\ (a^2 + b^2)$$

where m = 5/32.2 lb-sec^2/ft, a = 2.75, b = 2.75.

$$I_{R(1)} = \left[\frac{1}{12}\right]\left[\frac{5}{32.2}\right] (7.56 + 7.56)$$

$$= \frac{(.01292\ \text{lb-sec}^2/\text{ft})\ (15.12)\ \text{in}^2)}{1.44 \times 10^2\ \ \text{in}^2/\text{ft}^2} = 1.36 \times 10^{-3}\ \text{lb-ft-sec}^2$$

$$I_{R(2)} = m\,r^2$$

where m = 5/32.2 lb-sec^2/ft, r = 1 ft,

$$I_{R(2)} = \frac{5}{32.2}(1^2) = 0.15\ \text{lb-ft-sec}^2$$

$$I_R(3) = m\ r^2{}_3$$

where m = 5/32/2 lb-sec^2/ft, r = 8.5/12 ft.

$$I_{R(3)} = (0.15)\ (0.708) = 0.106\ \text{lb-ft-sec}^2$$

$$I_R(4) = \frac{1}{2} m\ r^2$$

where m = 12/32.2 lb-sec^2/ft, r = 2/12 ft.

$$I_{R(4)} = \frac{(1)\quad (.373)\quad (.167)}{(2)}$$

$$I_{R(4)} = 0.0312\ \text{lb-ft-sec}^2$$

$$I_{R(P)} = \frac{1}{2} M b^2$$

where m = 12/32.2 lb-sec^2/ft, b = 1ft.

$$I_{R(P)} = \frac{1\ (1)^2\ \text{lb-sec}^2\ \text{-ft}^2}{(32.2)\ \text{ft}}$$

$$I_{R(P)} = 0.031\ \text{lb-ft-sec}^2$$

The total moment of inertia is,

$$I_R(\text{total}) = I_1 + 2\ I_2 + 2\ I_3 + I_4 + I_{R(P)}$$

$$= 0.00136 + 0.30000 + 0.21200 + 0.03120 + 0.0310$$

$$I_R = 0.577 \text{ lb-ft-sec}^2$$

The motor maximum torque (T_R) is computed by

$$T_R = I_R \alpha R$$

Substituting we have

$$T_R = 3.35 \text{ lb-ft.}$$

The maximum power output P is

$$P_R = (T_R)(\omega_R)$$

or

$$P_R = 5.03 \text{ lb-ft/sec or } 6.82 \text{ watts.}$$

A Magnetic Technology Motor, Model 5125-220-023, will meet the required responses with the calculated load. This torque motor has a peak torque output of 5.2 lb-ft. and a rated shaft output of 35 watts. The no-load speed of the motor is 19 rad/sec which exceeds the specified rate of 1.5 rad/sec.

The power dissipated (P_{RMS}) in motor is

$$P_{(RMS)} = \left[T_{RMS}/K_M\right]^2$$

where $T_{(RMS)}$ = RMS torque required (.707 T_p)

and

$$K_m = \text{motor constant.}$$

The power is 29 watts.

The motor temperature rise constant (T_T) is

$$T_T = 0.8\ ^oC/\text{watt}$$

and the maximum allowable temperature of the motor is

$$T_R = 130\ ^oC.$$

The final temperature, $T_{e\ (final)}$, above ambient, $T_{e\ (amb)}$, is

$$T_{e(Final)} = T_{e\,(amb)} + T_T\ P_{(RMS)}$$

or

$$T_{e(Final)} = 49.2^oC.$$

Temperature is not a critical design problem in the roll servomotor.

Pitch Axis

The pitch axis moment of inertia consists of the sum of the moments of the major components of the platform plus a counterbalance.

In Figure 29

$$I_p(1) = M\ r^2$$

Where m = (5/32.2) lb-sec^2/ft, r = 1 ft.

$$I_{P(1)} = \left[\frac{5}{32.2}\right] (1)^2 = 0.15 \text{ lb-ft-sec}^2$$

$$I_{P(2)} = \left[\frac{1}{12}\right] (a^2 + b^2)\ m$$

Where m = 5/32.2 lb-sec^2/ft, $(a^2 + b^2) = 0.106$ ft^2

$$I_{P(2)} = \frac{(5.0)\ (1.06 \times 10^{-1})}{(1.2 \times 10)\ (3.22 \times 10)} = 1.37 \times 10^{-3}$$

$$I_{P(2)} = 0.00137 \text{ lb-ft-sec}^2$$

$$I_{P(3)} = m\,r^2$$

Where m = 0.15 lb-sec^2/ ft, r = (8.5/12) ft.

$$I_{P(3)} = (0.15)\ (0.708) = 0.106 \text{ lb-ft-sec}^2$$

$$I_{P(4)} = m\ r^2$$

Where m = 12/32.2 lb-sec^2/ft, r = 1 ft.

$$I_{P(4)} = 0.373 \text{ lb-ft-sec}^2$$

$$I_{P(P)} = \left[\frac{1}{12}\right] m\ b^2$$

Where m = 12/32.2 lb-sec^2/ft, b = 1 ft.

$$I_{P\,(P)} = 0.031 \text{ lb-ft-sec}^2$$

The total moment becomes,

$$I_{P(total)} = I_{1x} + 2\,(I_{2x}) + 2\,(I_{3x}) + I_{4x} + I_{P(P)}$$

$$= 0.15000 + 0.00274 + 0.21200 + 0.37300 + 0.03100$$

$$I_P = 0.769 \text{ lb-ft-sec}^2$$

The reflected load at the motor due to the platform is determined by

$$I_{P(motor)} = I_{P(Load)} \left[\frac{\ell}{(L_A)\,(2\,\pi\;\; e)} \right]^2$$

Where

$I_{P(motor)}$ = reflected load at motor

$I_{P(Load)}$ = load at platform

ℓ = lead of the ballscrew

L_A = lever arm of platform drive

e = efficiency (90%)

$\alpha_{P(Load)}$ = angular acceleration of platform

$\alpha_{P(motor)}$ = angular acceleration of motor

F_P = force on ballscrew

At the platform

$$F_P = \frac{I_{P(Load)} \;\; \alpha_{P(Load)}}{L_A}$$

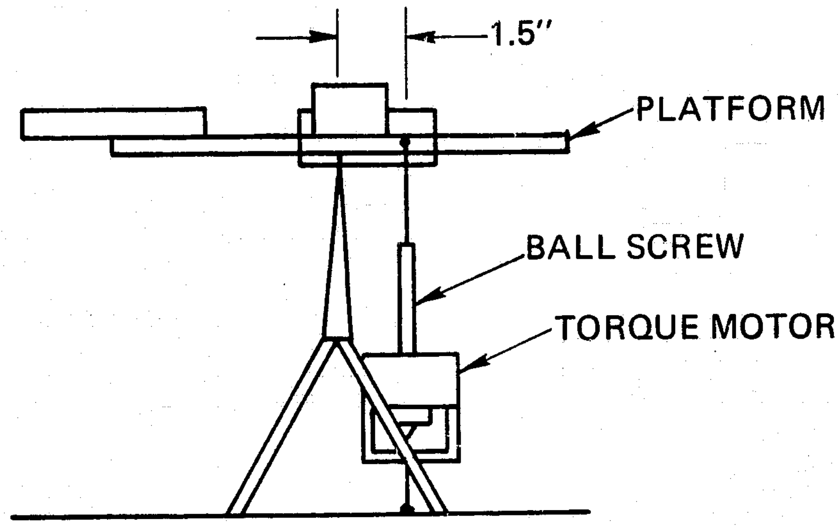
$$T_{P(motor)} = \frac{F_P\,\ell}{2\,\pi\;\, e} = \frac{I_{P(Load)} \;\; \alpha_{P(Load)}\,\ell}{2\,\pi\, e\, L_A}$$

$$\alpha_{P(motor)} = \alpha_{P\,(Load)} \left[\frac{2\,\pi\, e\, L_A}{\ell} \right]$$

$$I_{P(motor)} = \frac{T_{P(motor)}}{\alpha_{P(motor)}} = I_P\,(Load) \left[\frac{\ell^2}{4\,\pi^2\;\, e^2\;\, L_A^2} \right] = I_{P(Load)}\;\; N^2{}_P$$

where $N_P = \dfrac{\ell}{2\pi e L_A}$

For the ballscrew chosen and the relationship from Figure 30,



PITCH DRIVE MECHANISM LAYOUT
FIGURE 30

$$\ell = 0.59 \text{ in/rev.}$$

$$e = .90$$

$$L_A = 1.5 \text{ in.}$$

$$I_{P(motor)} = N_P^2 \; I_{P(Load)} = \left[\frac{(0.59)}{(6.28)\;(0.9)\;(1.5)}\right]^2 I_{P(Load)}$$

$$I_{P(motor)} = (0.00485) \quad I_{P(Load)}$$

$$I_{P(motor)} = 37.3 \times 10^{-4} \text{ lb-ft-sec}^2$$

The reflected inertia is also a function of the pitch angle, θ.

$$I_{P(motor)} = I_P \text{ (Load)} \left[\frac{1}{\cos\;\theta}\right]^2$$

At 0^o, $I_{P(motor)} = 0.00373 \text{ lb-ft-sec}^2$

At 70^o where the $\cos^2 70^o$ equals 1.17×10^{-1},

$$I_{P(motor)} = 3.73 \times 10^{-3} \text{ lb-ft-sec}^2 \left[\frac{1}{1.17 \times 10^{-1}}\right] = 0.0319 \text{ lb-ft-sec}^2$$

$$\alpha_{P(motor)} = \alpha_{P(Load)} \left[\frac{(6.28)\;(.9)\;(1.5)}{(0.59)}\right] = 230 \text{ rad/sec}^2$$

At 0^o,

$$\alpha_{P(motor)} = 230 \text{ rad/sec}^2$$

At 70^o,

$$\alpha_{P(motor)} = (230 \text{ rad/sec}^2)\ (\cos 70^o)$$

$$\alpha_{P(motor)} = 78.8 \text{ rad/sec}^2$$

$$\omega_{P(motor)} = W_L\ (14.35) = (3.0 \text{ rad/sec})\ (1.435 \times 10)$$

$$\omega_{P(motor)} = 43.05 \text{ rad/sec.}$$

$$\alpha_P(0^o) = 230 \text{ rad/sec}^2$$

and

$$\alpha_P(70^o) = 78.8 \text{ rad/sec}^2$$

The torque, T_P, required at the motor to produce these accelerations is given by,

$$T_{P(motor)} = I_{P(motor)}\ \alpha_{P(motor)}$$

At a pitch angle of 70 degrees,

$$T_{P(motor)} = 480 \text{ oz-in.}$$

At zero degrees of pitch,

$$T_{P(\theta^o)} = 165 \text{ oz-in}$$

The motor input voltage V_P, for the maximum required motor shaft rate is

$$V_P = K_V\ \omega_{P(motor)}$$

where

$$K_V = 0.53 \text{ volts/rad/sec,}$$

$$\omega_{P(motor)} = 43 \text{ rad/sec}$$

or

$$V_P = 22.8 \text{ volts.}$$

An additional voltage must be applied to produce torque at the required shaft velocity. The maximum allowable voltage is specified by the practical limitation that the motor shaft power output, W_s, be no more than 50% of the power input, W_P, for peak torque, T_P, at stall

or

$$W_s(\text{max}) = (0.5)\ (W_P).$$

In this case

$$W_s(\text{max}) = 70 \text{ watts.}$$

But

$$W_s = \omega_{P(\text{motor})}\ T_{P(\text{motor})}$$

Solving for $T_{(\text{max})}$

$$T_{P(\text{motor})} = 1.2 \text{ lb-ft}$$

or

$$T_{P(\text{motor})} = 230 \text{ in-oz.}$$

The excess voltage required to produce the above torque can be derived from the motor winding constants.

$$K_T = 74.5 \text{ oz-in/amp.}$$

The current needed to produce this torque is

$$C_D = T_{P(\text{motor})}/K_T = 3.1 \text{ amps.}$$

The voltage, V_D, to produce this current is

$$V_D = R_A\ C_D$$

Where

V_D = input voltage increment

R_A = Resistance = 0.80

C_D = current = 3.1 amps.

or

$$V_D = 2.48 \text{ volts.}$$

The total voltage required to produce the maximum shaft power is

$$V_{(total)} = 25.2 \text{ volts.}$$

A nominal voltage maximum of 25.5 volts is the amplifier output voltage limit. The maximum allowable power applied to the input of the motor was investigated since current overload can cause demagnetization of the rotor.

The input current is limited to a value which will produce the required maximum torque at the motor shaft to accelerate the inertia load of the system.

The required peak torque $T_{P(motor)}$ from previous calculations is

$$T_{P(motor)} = 480 \text{ in-oz.}$$

A nominal maximum torque of 500 in-oz is used to establish the input current limit by the following calculation,

$$C_{(max)} = \frac{T_{P(motor)} \ (70^{o})}{K_T} = 6.72 \text{ amps}$$

A nominal current limit to the motor of 6.75 amps is thus established for the pitch motor. The amplifier output is limited as follows,

$$V_{(Limit)} = \pm 25.5 \text{ volts D.C.}$$

$$C_{(Limit)} = \pm \ 6.75 \text{ amps D.C.}$$

The power dissipated in the motor at the maximum load condition was evaluated for thermal considerations.

Maximum voltage, $V_{(max)}$, and maximum current, $C_{(max)}$, will be applied to the motor winding during maximum acceleration. Values for these parameters are

$$V_{(max)} = 25.5 \text{ volts}$$

$$C_{(max)} = 6.75 \text{ amps}$$

$$R_a = 0.8 \text{ ohms}$$

$$W_D = \text{Power dissipated } (C^2R).$$

or

$$W_D = 36.4 \text{ watts.}$$

The temperature rise constant for this motor is $0.8^{o}C$ watt and the maximum allowable winding temperature is $130^{o}C$.

This gives,

$$\Delta T_e = 29^oC.$$

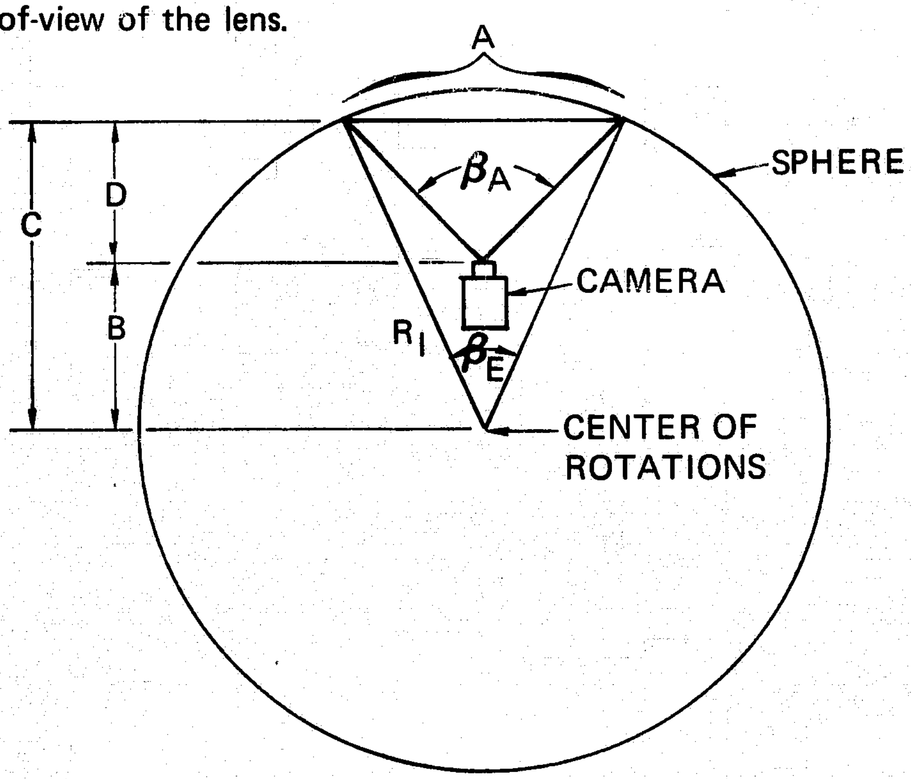
The final temperature of the motor winding during the highest power condition will be

$$T_{e(ambient)} + \Delta T_e = 55^oC.$$

this final temperature is based on a 100% duty cycle.

Heading Axis

The lens used will provide a horizontal field-of-view of 84^o and will be located at a distance of 18 inches from the center of the pitch/roll system. The size of the sphere will depend on the position and field-of-view of the lens.



CAMERA - SPHERE FIELD-OF-VIEW RELATIONSHIP
FIGURE 31

From the geometric relationships in Figure 31

$$A = 2\ (D)\ \tan \frac{\beta_A}{2}$$

Where

= field of view of the lens (84^o)

β_E = field of view of the system (specified)

C = distance of focal plane

A = horizontal coverage on sphere surface

From the field-of-view requirement defined in the specifications,

$$A = 2(C) \tan \frac{\beta_E}{2}$$

Where

$$\beta_E = 46^\circ$$

Substituting

$$C \tan \left(\frac{46^\circ}{2}\right) = D \tan \left(\frac{84^\circ}{2}\right)$$

Since $D = C\text{-}B,$

$B = 18$ inches

$\tan 23^\circ = 0.424$

$\tan 42^\circ = 0.9$

the relation becomes

$$C(0.424) = (C\text{-}B)\ (0.9)$$

$$.9B = 0.9C - (0.424)C$$

$$16.2'' = (0.474)C$$

$$C = \frac{16.2''}{0.474} = 34.2$$

The sphere radius from the above relations is,

$$R_I = C/(\text{Cos } 23^\circ)$$

where

R_I = sphere radius

$\text{Cos } 23^\circ = 0.92$

$$R_I = \frac{34.2 \text{ inches}}{0.92} = 37.25 \text{ inches}$$

The overall diameter of the sphere with a nominal wall thickness of 1/4 inches will be,

$$D_s = 2\ (R_1 + .25) \text{ inches} = 75.0 \text{ inches}$$

The hollow sphere moment of intertia is

$$I_s = \frac{2M_s \ R_I^2}{3}$$

where

$I_{H(Load)}$ = moment of inertia of the sphere

m = mass of the sphere

R_I = radius of the sphere (37.5 in)

W_W = weight of the sphere = (80 lb)

m = (80 lb)/(32.2 ft/sec^2) = 2.48 lb-sec^2/ft.

Substituting these values into the moment of inertia equation, we have

$I_{H(Load)}$ = 2 (2.48 lb-sec^2/ft) (3.75 x 10 in)^2 / 3 (12 in/ft)^2

$I_{H(Load)}$ = 1.61 lb-ft-sec^2

The torque and forces in the belt are considered in selecting the belt type and size. For the loads in this case,

$$T_{H(load)} = I_{H(Load)} \ \alpha_{H(Load)}$$

Where

$T_{H(Load)}$ = torque required at the load to produce $\alpha_{H(Load)}$

$I_{H(Load)}$ = moment of inertia of the sphere

$\alpha_{H(Load)}$ = acceleration required by specifications.

For this case,

$I_{H(Load)}$ = 16.1 lb-ft-sec^2

$\alpha_{H(Load)}$ = 5.0 rad/sec^2

Substituting these values into the torque equation, we get

$T_{H(Load)}$ = (16.1 lb-ft-sec^2) (5 rad/sec^2)

$T_{H(Load)}$ = 81 lb-ft

This torque is used to find the maximum force which the belt must apply to the load

$$F_H = T_{H(Load)}/L_R$$

Where

F_H = force in the belt to produce maximum torque

L_R = pitch radius of the belt fear.

or

$$F_H = (81 \text{ lb-ft}) / (1.5 \text{ ft})$$

$$F_H = 54 \text{ lb.}$$

The reflected moment of inertia at the motor is

$$I_{H(motor)} = I_{H(Load)} / N_H{}^2$$

$$= (16.1 \text{ lb-ft-sec}^2) / (12)^2$$

$$I_{H(motor)} = 0.112 \text{ lb-ft-sec}^2$$

The total inertia load at the motor shaft is,

$$I_H = I_{H(motor)} + I_{H\,(rotor)} + I_{H(tach)}$$

Where

$I_{H(rotor)}$ = 0.005 lb-ft-sec^2

$I_{H(tach)}$ = 0.000146 lb-ft-sec^2

$$I_H = (.112 + .005 + .000146) \text{ lb-ft-sec}^2$$

$$I_H = 0.117 \text{ lb-ft-sec}^2$$

The inertia load of the tachometer is negiibible.

The torque requirement for the motor is

$$T_{H(motor)} = T_{H(Load)/N}$$

$$T_{H(motor)} = 6.75 \text{ lb-ft}$$

The power required at the motor to drive the load is

$$P_H = T_{H\,(motor)}\ \omega_{H(motor)}$$

where

$$T_{H(motor)} = 6.76 \text{ lb-ft}$$

and

$$\omega_{H(motor)} = N \ \omega_{H(Load)}$$

Since

$$\omega_{H(Load)} = 1.5 \text{ rad/sec,}$$

then

$$\omega_{H(motor)} = 18 \text{ rad/sec.}$$

The motor no-load speed is 27 rad/sec which is more than adequate.

The shaft power is

$$P_{H(motor)} = T_{H(motor)} \ \omega_{H(motor}$$

$$P_{H(motor)} = 122 \text{ lb-ft-sec.} = (165 \text{ watts})$$

The average power loss is

$$W_{(RMS)} = \left[\frac{T_{(RMS)}}{K_M}\right]^2$$

$$W_{(RMS)} = 132 \text{ watts.}$$

The duty cycle of the heading servo is nominally 30% or less.

The RMS power will be approximately 43 watts. The temperature rise due to the power loss is

$$T_{e(Final)} = T_{e(ambient)} + W_H T_T$$

where

$$W_H = 43 \text{ watts}$$

$$T_T = 2^{o}\text{C/watt}$$

$$T_{e(Final)} = 26^{o}\text{C} + (43 \text{ watts}) \ (2^{o}\text{C/watt})$$

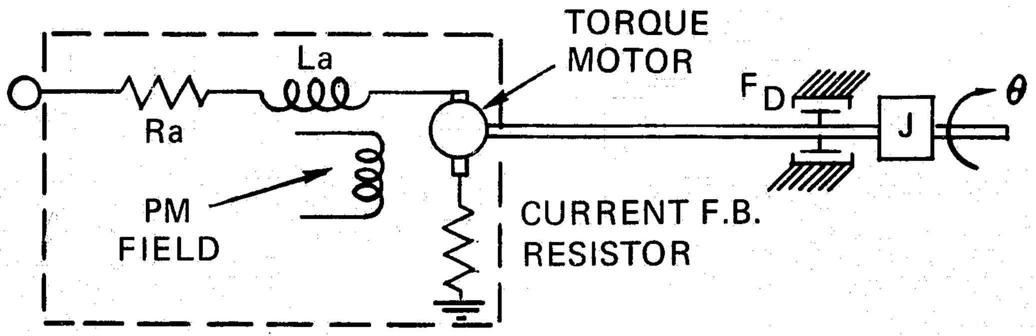
$$T_{e(Final)} = 112^{o}\text{C.}$$

A cooling fan will be required for the heading motor during periods of unusual heading commands.

APPENDIX D - SERVOANALYSIS CALCULATIONS

Servo Parameter Definition

Figure 32 depicts an open loop D.C. servosystem used for the pitch, roll, and heading servomotors. Representations of the viscous damping and load inertia are included.



MOTOR/LOAD REPRESENTATION
FIGURE 32

The equations which represent the dynamics of the system are

$$J\ddot{\Theta} + F_D\dot{\Theta} = K_T\, i_a$$

and

$$L_a\, \dot{i}_a + R_a\, i_a = V_a - K_B\dot{\Theta}$$

From these equations

$$\ddot{\Theta} = \frac{K_T\, i_a - F_D\dot{\Theta}}{J}$$

and $$\dot{i}_a = \frac{V_a - K_B\,\dot{\Theta} - R_a\, i_a}{L_a}$$

where the system parameters and variables are identified as

Θ = output position angle in radius

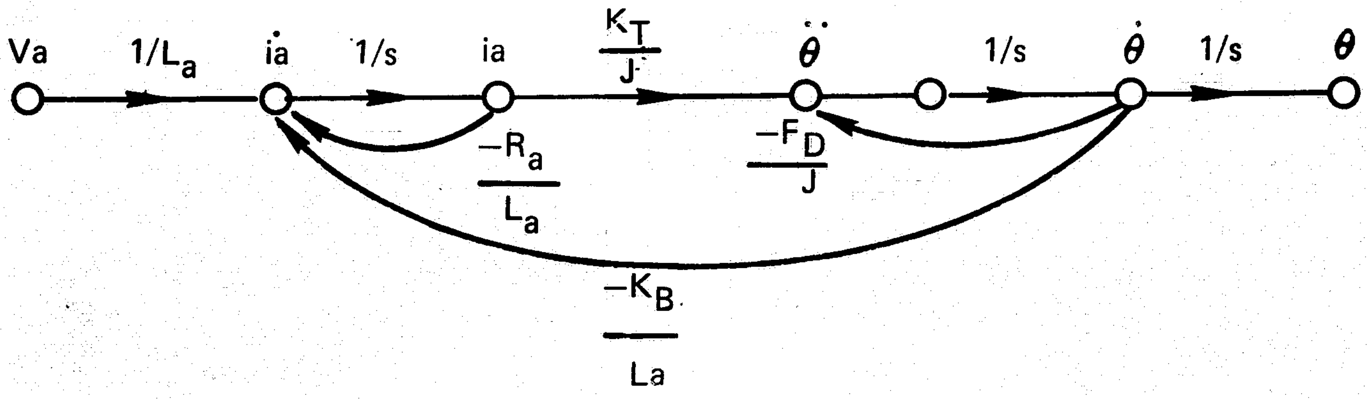
V_a = input voltage in volts

i_a = motor armature current in amps

K_a = gain of linear amplifier

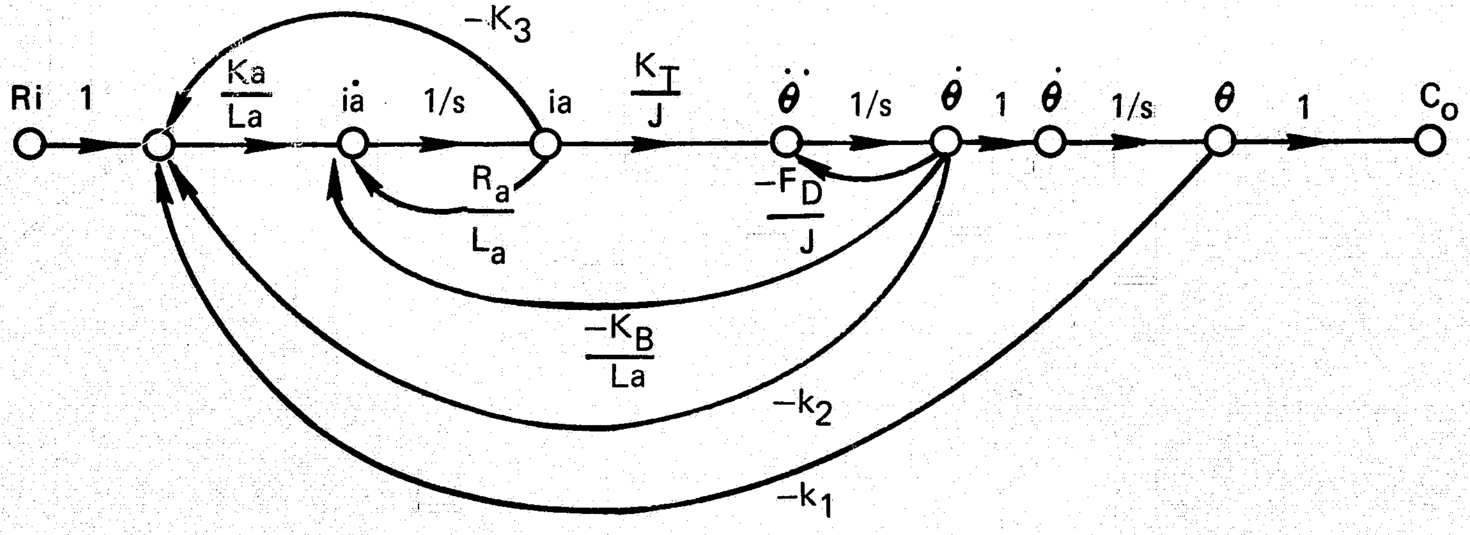
R_a	=	resistance of armature winding in ohms
L_a	=	inductance of armature winding in henries
J	=	inertial load, lb-ft-sec^2
F_D	=	viscous damping constant lb-ft/rad/sec
K_T	=	motor torque constant in ft-lb/amp
K_B	=	motor back EMF constant in V/rad/sec

A flow diagram of the open loop system equations is shown in Figure 33 where s is the LaPlace variable.



OPEN LOOP SIGNAL FLOW DIAGRAM
FIGURE 33

A computer program was constructed to simulate the specific loads and motor tachometer characteristics for verification of the calculated feedback gains. Figure 34 shows the total system flow diagram.

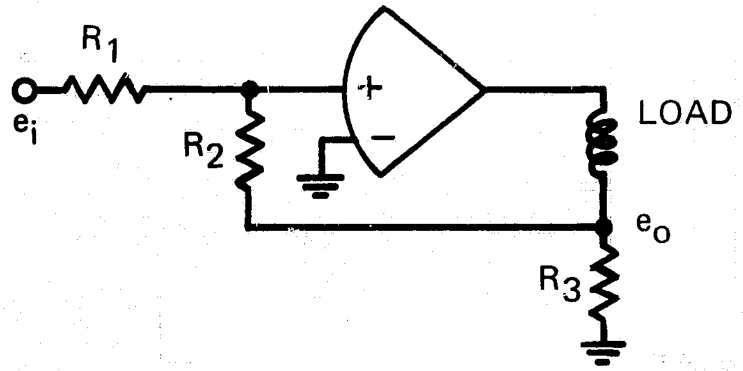


CLOSED LOOP SIGNAL FLOW DIAGRAM
FIGURE 34

Both current and voltage sources were considered in the electric design of the servomechanisms.

Current Feedback

The current source concept uses armature current feedback as shown in Figure 35.



CURRENT FEEDBACK SCHEMATIC
FIGURE 35

No direct voltage feedback is used on the operational type power amplifier. The current gain function can be determined from

$$e_o = \frac{R_2}{R_1} e_i \quad \text{where} \quad e_o = i_L R_3.$$

Then

$$i_L R_3 = -\frac{R_2 e_i}{R_1}$$

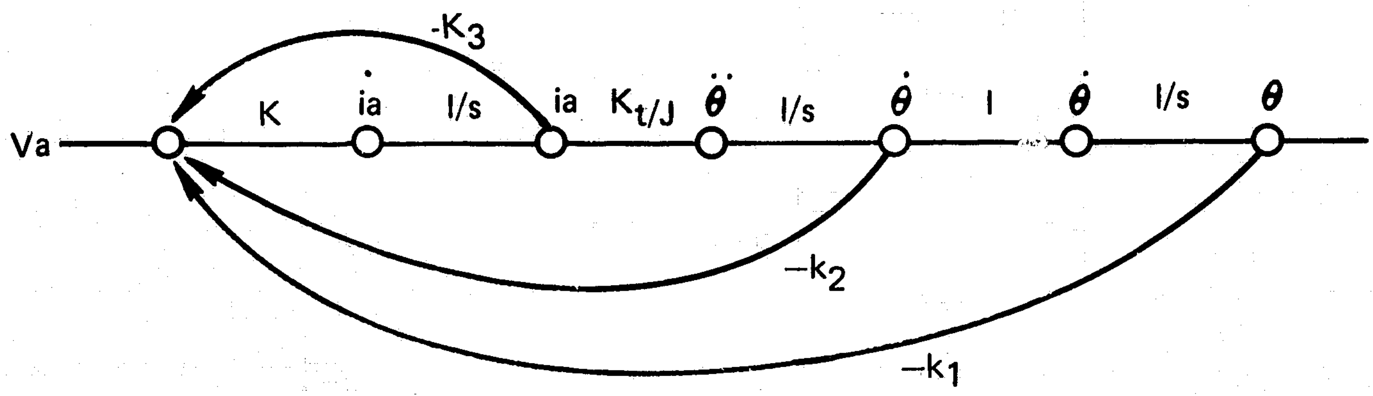
and

$$i_L = \left[\frac{R_2}{R_1 \; R_3}\right] e_i$$

$\dfrac{R_2}{R_1 \; R_3}$ is the gain function of the amplifier in amps/volt.

The high output impedance of the amplifier decreases the system viscous damping. The Inland D.C. torque motor, T-5730, possesses a viscous damping coefficient of .26 lb-ft/rad/sec for zero source impedance and .003 lb-ft/rad/sec for an infinite impedance source. Most of the damping must be supplied externally for a current source drive. One advantage of the current source amplifier is that it effectively reduces the time constant.

The use of current source allows the omission of several system parameters from Figure 34. This simplification does not affect the simulation accuracy when the summing amplifier is operated open loop. A simplified signal flow is shown in Figure 36.



SIMPLIFIED CURRENT AMPLIFIER FLOW DIAGRAM
FIGURE 36

In a computer simulation of this system, k_3, k_2, and k_1 , were varied to obtain the values which would provide suitable operation. The system in general tends to be unstable.

Voltage Source

The voltage source flow diagram of the system is shown in Figure 37. It is derived by noting that the stall torque of the motor, T_p, is proportional to the voltage at the motor from the amplifier. This torque is used to accelerate the inertia of the motor and the load. If we set the load torques equal to the motor torques,

$$T_p = V_a K_t = J\ddot{\Theta} + F_D \dot{\Theta} = JS^2\Theta + S F_D \Theta,$$

then

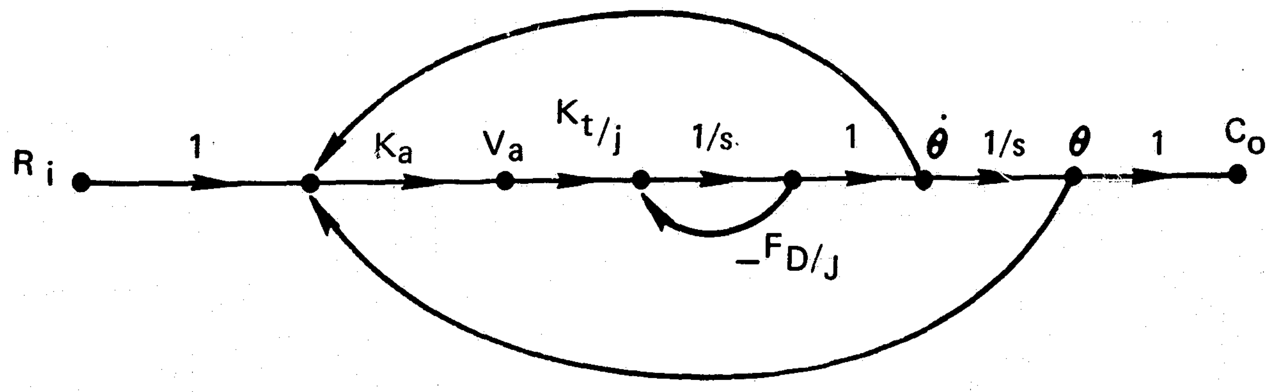
$$\frac{\Theta(S)}{V_{a(S)}} = \frac{K_t}{JS^2 + F_{DS}} = \frac{K_t/F_D}{S(J/F_D)S + 1)}$$

The time constants for the pitch, yaw, and roll servos were not considered significant and were ignored in the system analysis for gain determination of this configuration. The system transfer function is

$$\frac{C_o(S)}{R_i(S)} = \frac{\left[K_a (K_t/J\right]\left[S \quad (S + F_D/J)\right]}{1 + \left[\dfrac{K_a K_t / J}{S(S + F_D/J)}\right](K_g S + 1)}$$

where

C_o is the controlled variable and R_i is the reference input. The main advantage of the current source is the eliminating of the time constants. The voltage source amplifier is chosen for driving the servomotors.



VOLTAGE FLOW DIAGRAM
FIGURE 37

Heading Servoanalysis

The heading servospecifications and parameter definitions are summarized below:

$I_{H(Load)}$	=	inertia of rotated mass = .104 lb-ft-sec^2
$\alpha H(max)$	=	maximum acceleration = 5.0 rad/sec^2
$\omega H(max)$	=	maximum velocity = 1.5 rad/sec
δ	=	system damping ratio = .7
ωN	=	system frequency bandwidth = 25 rad/sec
ψ	=	angle of rotation = 360^o (continuous)
K_t	=	.355 lb-ft/volt
J	=	.104 lb-ft/sec^2
F_D	=	1.26 lb-ft/sec
T_e	=	L/R = .355 x 10^{-3} sec
L_a	=	5 millihenries
R_a	=	1.5 Ω
K_g	=	gain of tachometer in volts/rad/sec.

With the change from current to voltage sources the units or K_t changed from torque/amps to torque/volts and the units on K changed from amps/volt to volt/volt.

The transfer function of the motor and load is

$$\frac{\theta_o(s)}{V_a(s)} = \frac{K_t/F_D}{s\left[\frac{J}{F_D}s + 1\right]}$$

Substituting the specific values for the heading servo,

$$\frac{\theta_o(s)}{V_a(s)} = \frac{.355/.26}{s\left[\frac{.104}{.26}s + 1\right]} = \frac{3.35}{s(s+2.44)}$$

This expression will be used to develop the system closed loop transfer function (See Figure 38).

Amplifier Selection

A Constols System Research Company amplifier, Model 500 PRA, is recommended for the pitch and heading servomotors. This amplifier is capable of supplying 25 amps at 25 volts. Adjustable limits are provided on the amplifier to limit the maximum current and voltage to the specified 6.75 amps and 25.5 volts. (See motor selection). This amplifier is suitable as either a current or voltage source by simply choosing the proper feedback elements.

The Model 200 PMA amplifier is recommended to drive the roll servo and can supply 23 volts at 8.5 amps.

With tachometer feedback (switch SW-1, Figure 38 closed), the closed loop transfer function is

$$\frac{C_o(s)}{R_i(s)} = \frac{KG}{1 + KG\,(1 + K_g s)} = \frac{\dfrac{K_a K_t/J}{s\,(s + F_D/J)}}{1 + \dfrac{K_a K_t/J}{s\,(s + F_D/J)}(K_g s + 1)}$$

or

$$\frac{C_o(s)}{R_i(s)} = \frac{625}{s^2 + (625 K_g + 2.44)\,s + 625.}$$

Equating the coefficients of the charactertistic equation,

$$s^2 + (625 K_g + 2.44)s + 625 = s^2 + 2\delta\omega_N s + \omega_N^2.$$

Solving for K_g with $\delta = .7$, $K_g = \frac{32.56}{625} = .052$ V/rad/sec.

In practice the tachometer voltage feeds directly into the amplifier which effectively increased the tachometer output by the position transducer scale factor of 31.5, so that,

$K_g = .052$ V/rad/sec. $(31.5) = 1.63$ V/rad/sec

The Inland Motor Company tachometer, TG 2801, has a sensitivity, Kv, of .635 V/rad/sec. The Ratio,

$$K_g/K_v = 2.55$$

is included in the tachometer buffer amplifier.

$$\frac{C_o(s)}{R_i(s)} = \frac{105\ K_a}{s\ (s + 2.44)\ + 105\ K_a}$$

The characteristic equation of a second order system is

$$\frac{\omega_N{}^2}{s^2 + 2\ \delta\ \omega_N + \omega_N{}^2}$$

which allows the calculation of K_a needed to design the required bandwidth. By equating equal powers of s

$$\omega_N^2 = (25)^2 = 105\ K_a,$$

or

$$K_a = 6$$

Also

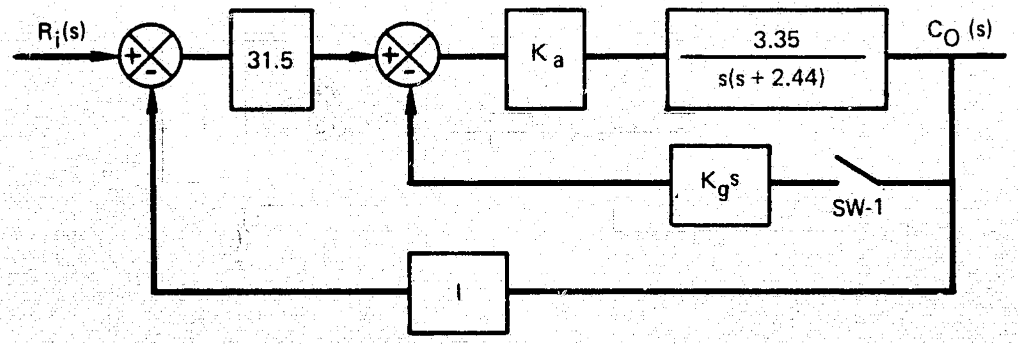
$$2\ \delta\ \omega_N = 2.44$$

giving

$$\delta = \frac{2.44}{2\ (5)} = .244$$

Such a low damping factor results in unsatisfactory system operation and does not meet the desired .7 damping factor.

The block diagram of the heading system is shown in Figure 38.



HEADING SYSTEM BLOCK DIAGRAM
FIGURE 38

The closed loop transfer function of this system with SW-1 open is

$$\frac{C_o\ (s)}{R_i\ (s)} = \frac{31.5\ (K_a) \left[\dfrac{3.355}{s\ (s + 2.44)}\right]}{1 + \left[\dfrac{(31.5)\ (K_a)\ 3.35}{s\ (s + 2.44)}\right]}$$

where the feedback potentiometer scale factor is 31.5 volts/rad.

Simulation

The block diagram of Figure 38 for the heading system was programmed on the analog computer to verify the performance of the system. The simulation results presented in Figures 39 and 40 show the heading position and position rate for a step input. The damping of the system without tachometer feedback was too low and caused considerable overshoot. The plots of Θ and $\dot{\Theta}$ from the simulation diagram Figure 39 verify the overshoot in the system without external damping. Figure 40 shows Θ and $\dot{\Theta}$ with the position and forward loop gains set as previously calculated plus tachometer feedback.

For set input, the maximum velocity of the motor was 14.85 rad/sec, well below the 27 rad/sec motor limit. The velocity feedback gain was calculated for a position step input. The system reaches 90% commanded velocity in two hundred milliseconds. Figure 40 shows the response of the velocity system for a step input. The heading servomotor will be operated with velocity as the commanded variable. The closed loop Bode plot of Figure 41 indicates the 3 dB point of the velocity system to be at 18.5 rad/sec. Position feedback is employed when the initial conditions of the scene are to be set.

Pitch Servomotor Analysis

The pitch systems gains are calculated using the transfer functions developed in the general analysis sections.

$$\frac{\theta}{V_a} = \frac{K_t/F_D}{s\ (J/F_D\ s + 1)\ \ (s \quad {}_e + 1)}$$

where

$$K_t = 0.487 \text{ lb-ft/volt}$$

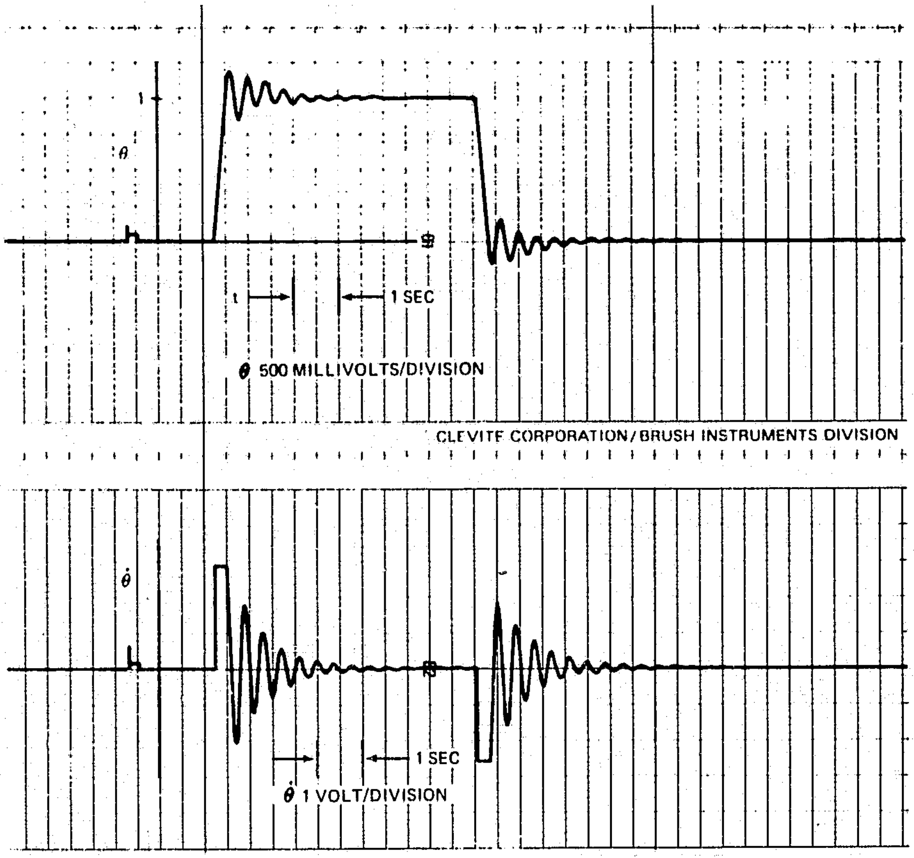
$$J = 0.0319 \text{ lb-ft-sec}^2$$

$$F_D = 0.274 \text{ lb-ft/rad/sec}$$

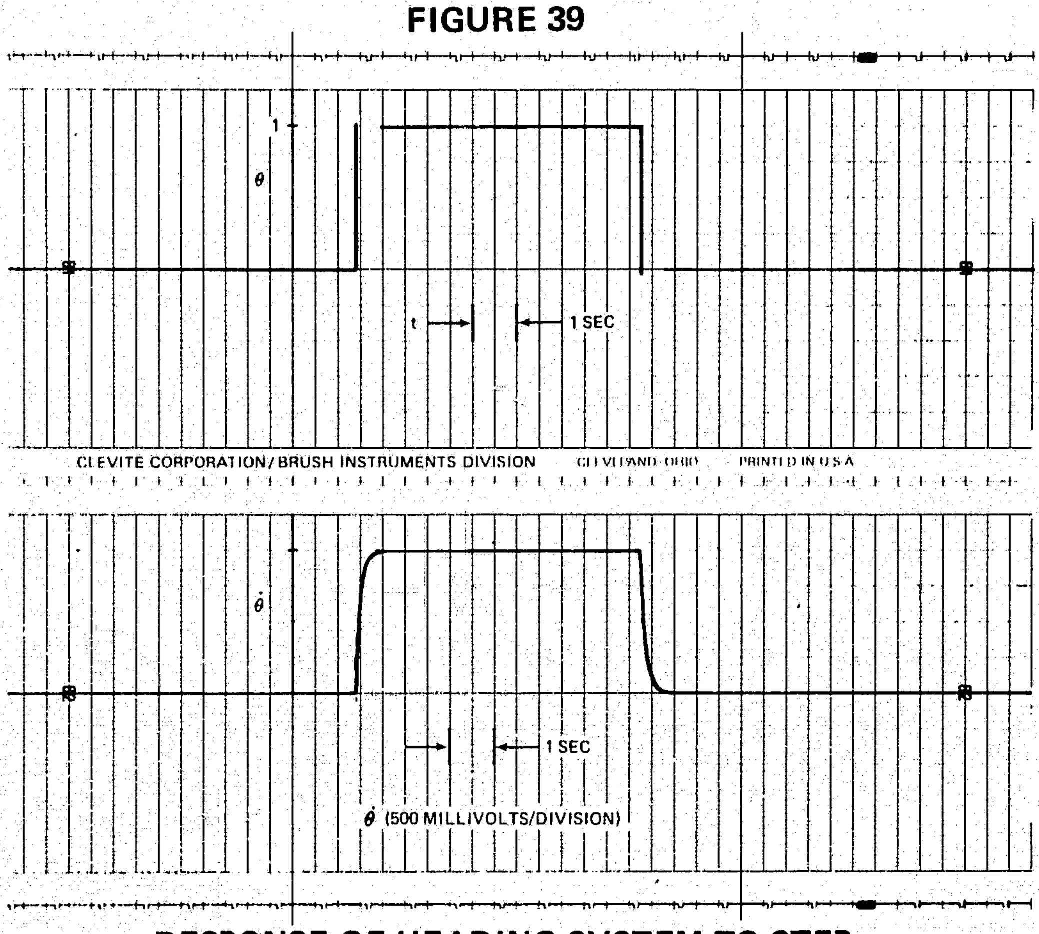
$$J/F_D = 0.116 \text{ rad-sec}$$

$$K_t/F_D = 1.78 \text{ rad/sec/volt}$$

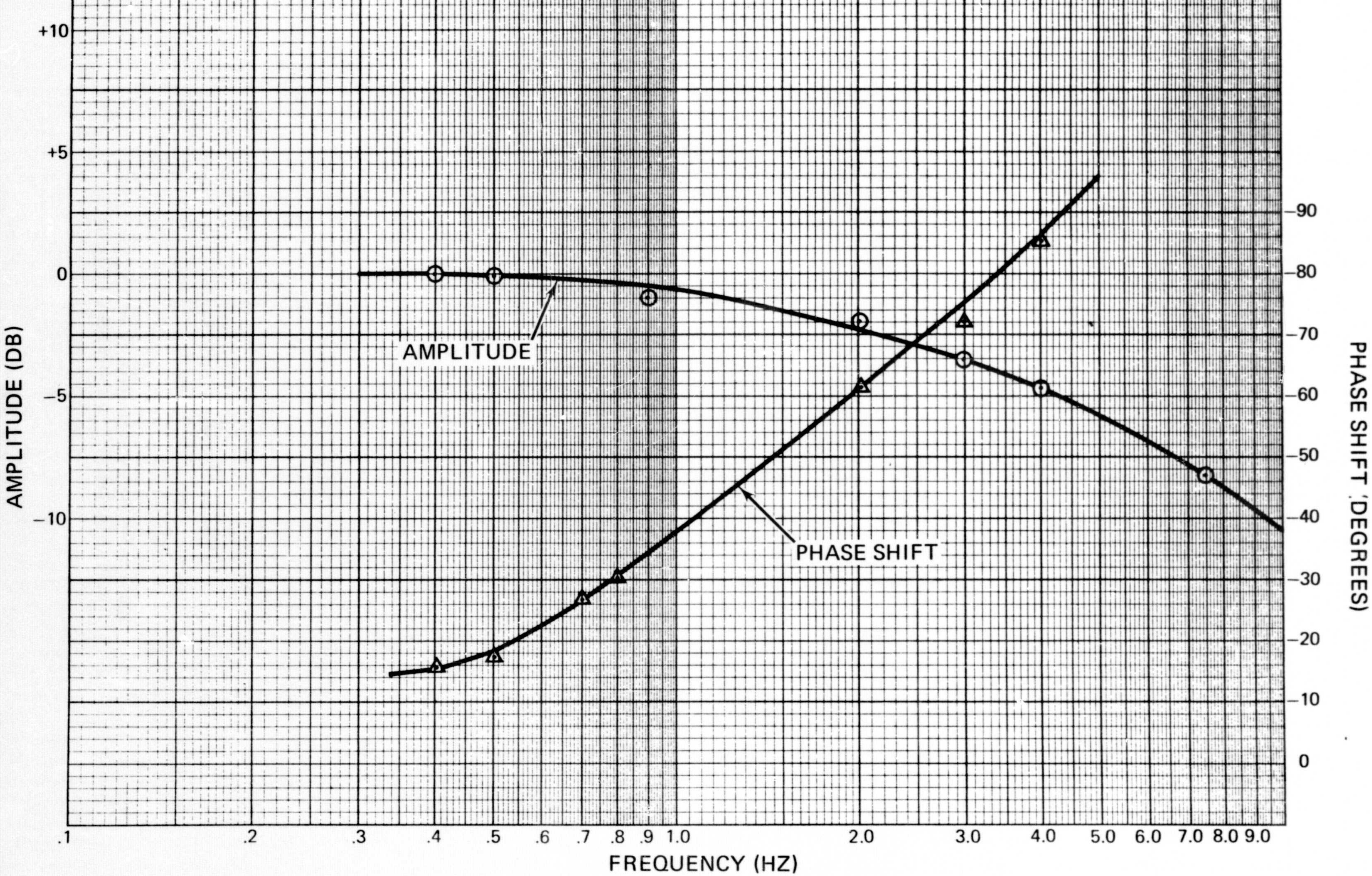
$$\frac{\sigma}{V_a} \quad \frac{1.78}{s\ \ (0.116\ \ s\ \ + 1)} = \frac{15.4}{s\ \ (s\ \ + 8.8)}$$



RESPONSE OF HEADING SYSTEM TO STEP INPUT WITH NO EXTERNAL DAMPING
FIGURE 39

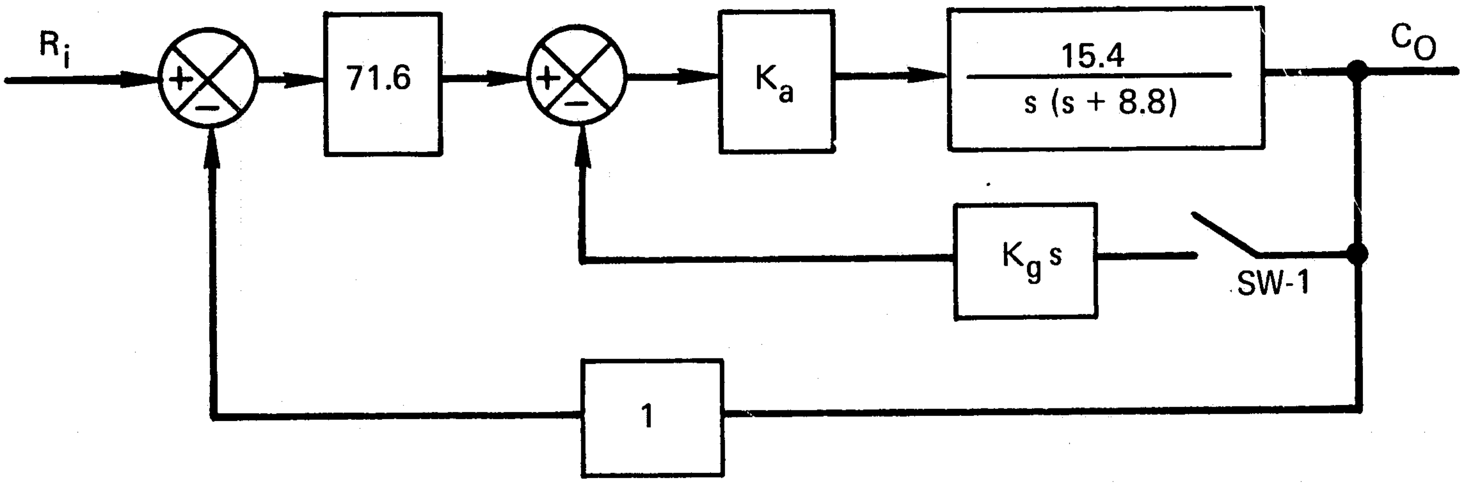


RESPONSE OF HEADING SYSTEM TO STEP INPUT WITH TACHOMETER FEEDBACK
FIGURE 40



BODE PLOT FOR HEADING VELOCITY SERVO
FIGURE 41

The inclusion of the motor and load transfer functions in the total system is shown in the block diagram of Figure 42.



PITCH SYSTEM BLOCK DIAGRAM
FIGURE 42

With SW-1 open the closed loop transfer function is

$$\frac{C_o}{R_i} = \frac{(71.6\ K_a)\ (15.4)/\ s\ (\ s\ +\ 8.8)}{1\ +\ (71.6\ K_a)\ (15.4)\ /\ s\ (\ s\ +\ 8.8)}$$

$$\frac{C_o}{R_i} = \frac{71.6\ K_a\ (15.4)}{s\ (\ s\ +\ 8.8)\ +\ 71.6\ K_a\ (15.4)}$$

$$s^2 + 8.8\,s + 71.6\ K_a\ (15.4) = s^2 + 2\delta\omega_N + \omega_N{}^2$$

By equating equal powers of s,

$$\omega_N{}^2 = 625$$

$$K_a = (625/71.6)\ (15.4) = 0.6$$

and

$$2\,\delta\omega_N = 8.8$$

$$\delta = \frac{8.8}{2\ (25)} = 0.174$$

This damping factor is too low and the system will require external damping. With SW-1 closed the closed loop transfer function is,

$$\frac{C_o}{R_i} = \frac{625/\ s\ (\ s + 8.8)}{1\ +\ 625\,(K_g\,s\ +\ 1\ )\ /\ s\ (\ s + 8.8)}$$

$$\frac{C_o}{R_i} = \frac{625}{s^2 + (625\,K_g + 8.8)\,s + 625}$$

Again, equating coefficients of s to find K_g we have,

$$625\,K_g + 8.8 = 2\delta\omega_N.$$

Set $\delta = 0.7$

$$625\,K_g = 35 - 8.8 = 26.2$$

$$K_g = \frac{26.2}{625} = 0.042 \text{ volts/rad/sec}$$

The tachometer voltage is increased by the same position transducer gain of 71.6.

$$K_g = (71.6)\ (0.042) = 3 \text{ volts/rad/sec.}$$

Roll Servoanalysis

The transfer function for the motor and load is

$$\frac{\Theta}{V_a} = \frac{K_t/F_D}{s\left[(J/F_D)\,s + 1\right]}$$

where,

$K_t = 0.307$ lb-ft/volt

$J = 0.577$ lb-ft-sec^2

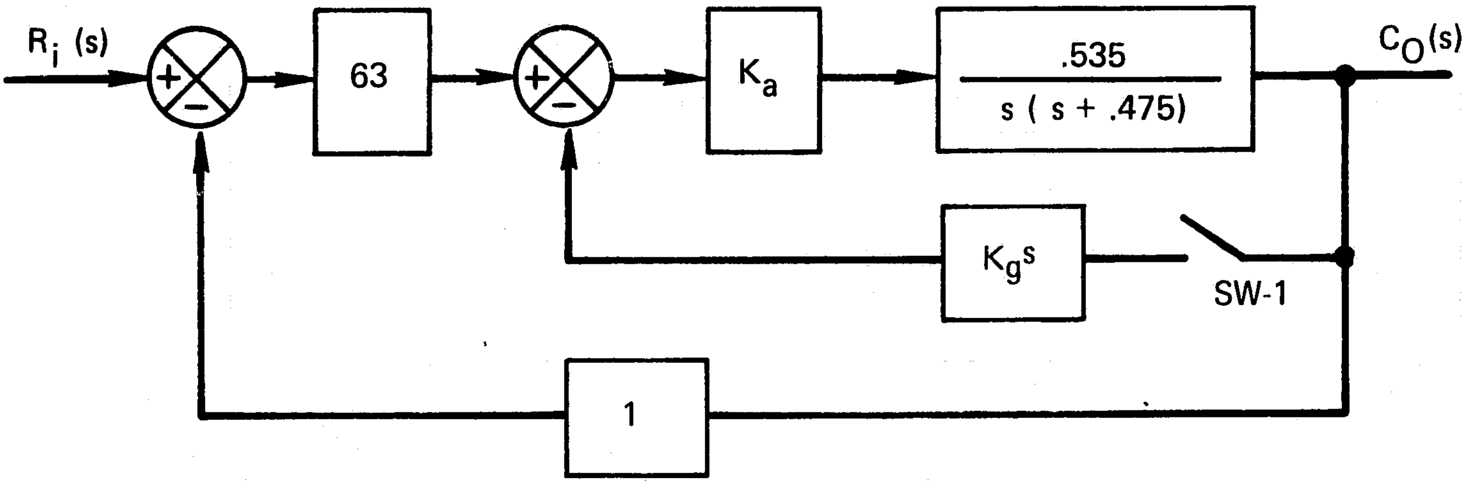
$F_D = 0.274$ lb-ft/rad/sec

$K_t/F_D = 1.12$ rad/sec/volt

$J/F_D = 2.1$ rad-sec

$$\frac{\Theta}{V_a} = \frac{1.12}{s\ (2.1\,s + 1)} = \frac{0.535}{s\ (s + 0.475)}$$

The inclusion of the motor and load transfer function in the total system is shown in the block diagram in Figure 43.



ROLL SYSTEM BLOCK DIAGRAM
FIGURE 43

The closed loop transfer function of the total system is

$$\frac{C_o(s)}{R_i\,(s)} = \frac{(63\ K_a)\ (0.535)\ /\ s\ (s\ +\ 0.475)}{1\ +\left[(63\ K_a)\ (0.535)\ /\ s\ (s + 0.475)\right]}$$

$$= \frac{(63\ K_a)\ (0.525)}{s\ (s\ + 0.475)\ +\ (63\ K_a)\ (0.535}$$

$$\frac{C_o\,(s)}{R_i\,(s)} = \frac{625}{s^2\ +\ 2\delta\,\omega_N\ +\ 625}$$

By equating equal powers of s,

$$\omega_N 2 = (25)^2 = 625$$

$$K_a = \frac{625}{(63)\ (0.525)} = 18.6$$

and similarly,

$$2\delta\omega_N = 0.475$$

$$\delta = \frac{0.475}{2\ (25)} = 0.0095$$

This damping factor is inadequate for proper system response and indicates the need for tachometer feedback.

With SW-1 closed the closed-up response is,

$$\frac{C_o(s)}{R_i\,(s)} = \frac{625/\ s\,(s + 0.475)}{1 + 625\ (K_g\ s + 1)\,/\,s\ (s + 0.475)}$$

$$\frac{C_o(s)}{R_i(s)} = \frac{625}{s^2 + s\ (625\ K_g + 0.475) + 625}$$

If we let $\delta = 0.7$

then

$$625\ K_g + 0.475 = 2\,\delta\,\omega_N = 35$$

$$K_g = \frac{34.525}{625} = 0.055 \text{ volts/rad/sec}$$

The tachometer voltage will feed directly into the amplifier and will be multiplied by the gain of the position transducer.

$$K_g = (0.055)\ (63) = 3.5 \text{ volts/rad/sec}$$

A summary of resistor values and component data is shown in Table 2 and 3. Block diagrams in the LaPlace domain for all three systems are shown in Figure 44.

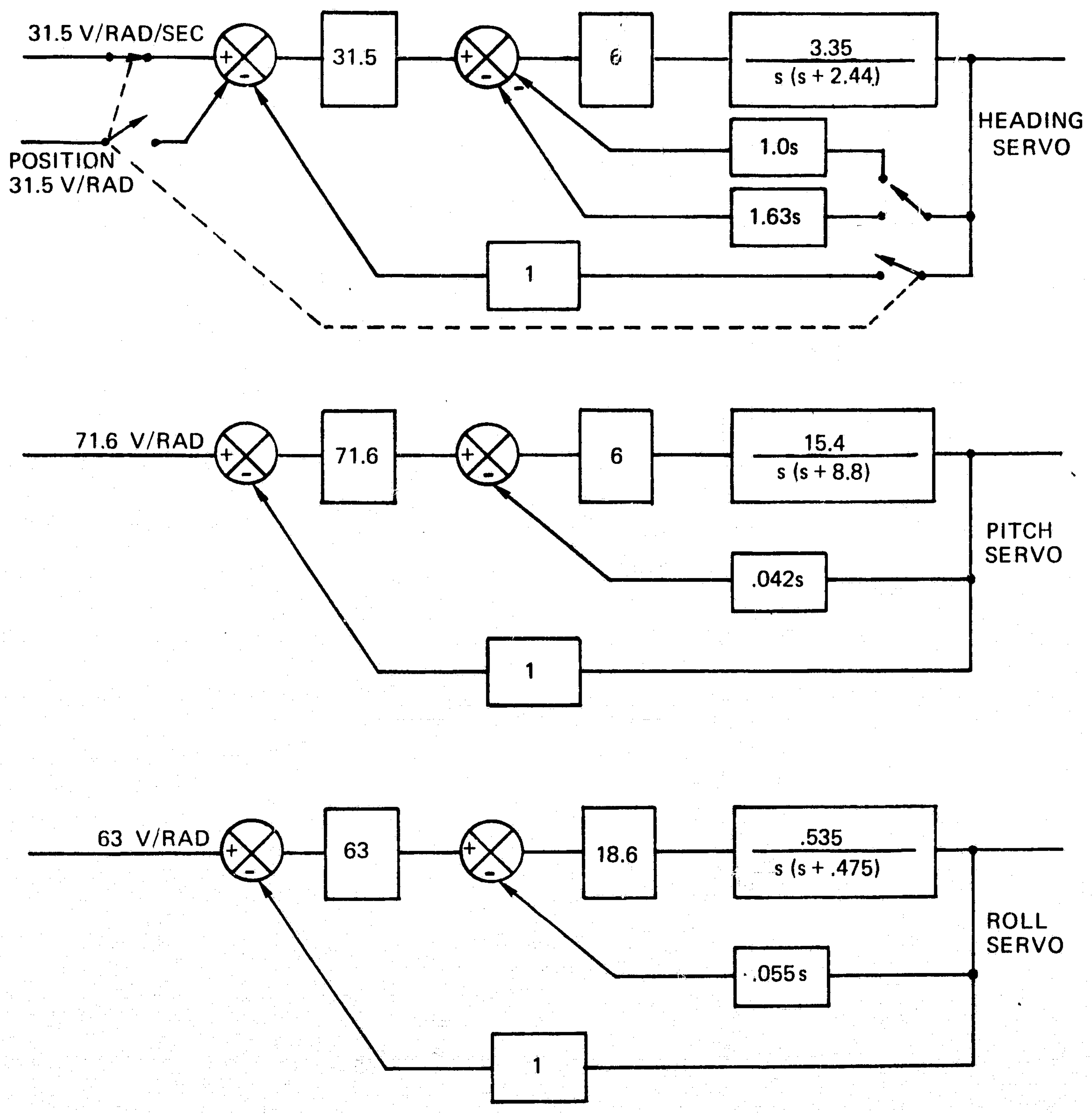
SERVO RESISTOR VALUE SUMMARY
TABLE II

RESISTORS VALUES		SERVO MECHANISM					
FUNCTION TYPE		ROLL		PITCH		HEADING	
		Ω	WATTS	Ω	WATTS	Ω	WATTS
R1	POSITION TRANSDUCER	50K	1	50K	1	50K	1
R2	DIFF. AMP – FEEDBACK OUTPUT TO INVERTING INPUT	100K	1/2	21K	1/2	240K	1/2
R3	DIFF. AMP FEEDBACK + INPUT TO GROUND	100K	1/2	21K	1/2	240K	1/2
R4	VELOCITY FEEDBACK INPUT RESISTANCE VELOCITY MODE	6.4K	2	9K	2	15.7K	1
R5	VELOCITY FEEDBACK INPUT RESISTANCE VELOCITY MODE	NA	NA	NA	NA	24K	1/2
R6	POSITION FEEDBACK INPUT RESISTANCE ON CONTROL OUTPUT	5K	2	35K	1/2	40K	1/2
R7	POSITION INPUT RESISTANCE ON REFERENCE INPUT	5K	2	35K	1/2	40K	1/2
R8	VELOCITY INPUT RESISTANCE ON REFERENCE INPUT	NA	NA	NA	NA	24K	1/2

NA = NOT APPLICABLE

SERVO COMPONENT DATA SUMMARY
TABLE III

SERVO MECHANISM				
	MOTOR	TACH	POWER AMP	BUFFER AMPL.
PITCH	MAGNETIC TECH 5125-220-008	MAGNETIC TECH 5125B-058	CONTROL SYSTEMS RES. 500 PRA	BURR BROWN 1545
YAW	INLAND T-5730	INLAND TG-2801	CONTROL SYSTEMS RES. 500 PRA	BURR BROWN 1545
ROLL	MAGNETIC TECH 5125-220-023	AEROFLEX TG52W-5	CONTROL SYSTEMS RES. 200 PMA	BURR BROWN 1545



SERVO SYSTEMS BLOCK DIAGRAM
FIGURE 44

APPENDIX E - CAMERA LENS SELECTION

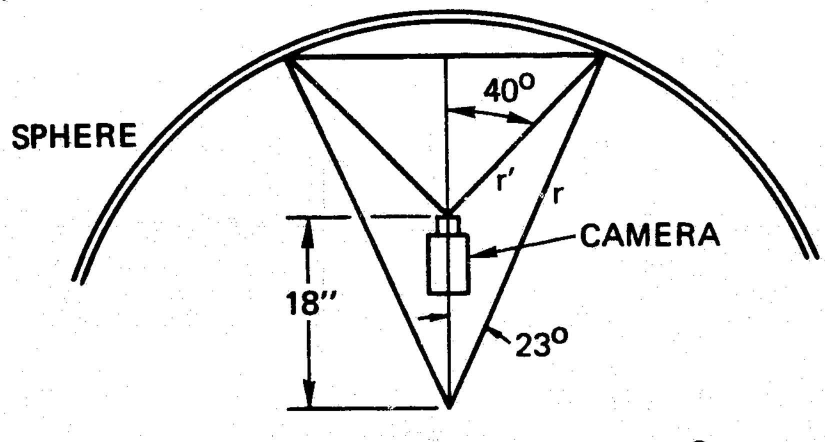
The requirements of the Cruise Scene Visual Attachment developed in the design section describe a wide-angle, low distortion lens system. These requirements result in the recommendation of a lens of approximately 84^{o} horizontal field-of-view (for a standard TV roster), a transmission characteristic of 70%, and a weight less than 1 pound. Parameters that also impact system performance are optical path distortion and eveness of illumination over the field of the lens. Nominal distortion for television systems is about 5 percent of the intended value. Illumination fall-off is not generally objectionable if it results in less than 2:1 variation from the center of the edge of a monitor.

Investigation into the availability of a suitable lens for this application indicates that there is a limited number of compensated wide-angle lenses available from stock. Angenieux Corporation has a lens in stock which comes close to meeting all of the requirements. Their Model R-7 is available for either 16 mm film or vidicon application. The R-7 has been distortion compensated to less than 5 percent over the entire field. The type of lens being considered has an inherent light fall-off as a function of angle from the optical axis. The Angenieux R-7 lens has a light transmission of 60% at the extreme edge of the field relative to the transmission at the lens axis. To correct for this, the scene illumination can be distributed to compensate for this fall-off. Since the illumination sources of adjacent cameras overlap, the edges of the field on the scene will be brighter than the center by approximately 60%. A summary of the Model R-7 characteristics is:

Field-of-view	=	80^{o}
Light Transmission	=	79%
Distortion	=	<5%
Weight	=	14 oz.

The field-of-view of this lens is nearly sufficient and there are several potential ways to incorporate it into the design. Angenieux suggests that it may be possible to obtain the 84^{o} with a simple low power negative meniscus lens added to the front. A second possibility is that the field-of-view required could be reduced to 80^{o} by either making the sphere larger, shifting the camera toward the center of the sphere, or a combination of both.

The increase in the sphere size required to provide an 84^{o} field-of-view with standard Angenieux R-7 lens position at 18" from the center of the system is calculated as follows using the geometry shown in Figure 45.



GEOMETRY OF CAMERA AND SPHERE FOR 80^{O} FIELD OF VIEW
FIGURE 45

$$r \sin 23^{o} = r' \sin 40^{o}$$

$$r \cos 23^{o} = r' \cos 40^{o} + 18$$

$$r \cos 23^{o} = \frac{r \sin 23^{o} \cos 40^{o} + 18 \sin 40^{o}}{\sin 40^{o}}$$

$$r \left[(\sin 40)(\cos 23) - (\sin 23)(\cos 40)\right] = 18 \sin 40$$

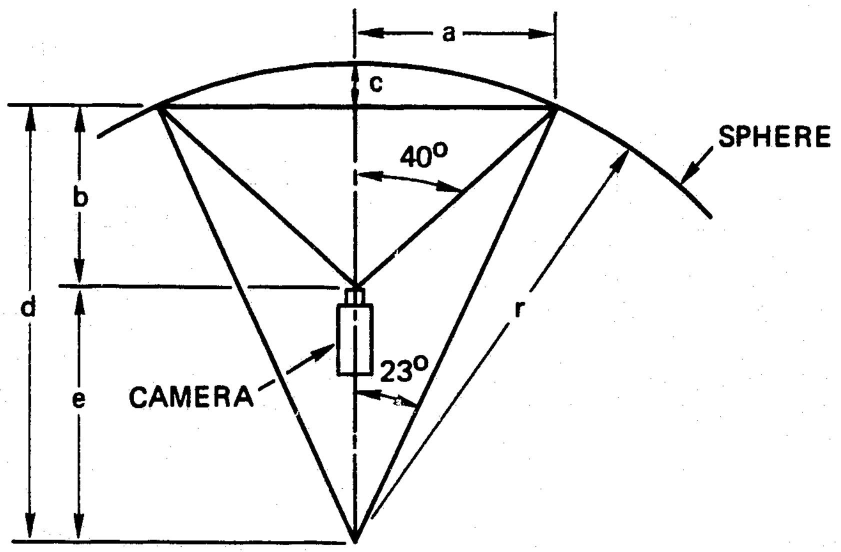
$$r = \frac{18 \sin 40}{\sin 40 \cos 23 - \sin 23 \cos 40}$$

$$r = \frac{11.6}{0.592 - 0.299} = \frac{11.6}{.293}$$

$$r = 39.6''$$

This is an increase of 3.6 inches in the sphere radius.

The amount of displacement required in the camera position to obtain the 84^{o} with a 36" diameter sphere is calculated as follows from Figure 46.



GEOMETRY OF THE CAMERA AND SPHERE FOR 36" RADIUS
FIGURE 46

$$a = r \sin 23^{o}$$

where $r = 36''$,

$$a = 36\,(0.391) = 14.1''$$

$$b = \frac{a}{\tan 40^{o}} = \frac{14.1}{0.839} = 16.8''$$

$$c = r - d$$

$$d = 36'' \cos 23^{o} = 36'' (0.921)$$

$$d = 33.2''$$

$$c = 36.0 - 33.2 = 2.8''$$

The camera lens position, e, is given,

$$e = r - b - c = 36'' - 16.8'' - 2.8''$$

$$e = 16.4''$$

The camera displacement toward the center from the present 18" using the 36" radius sphere is then,

$$D = 18 - 16.4 = 1.6''$$

A satisfactory solution is possible using the R-7 lens modification or a combination of the two above methods.

APPENDIX F - VENDOR REFERENCES

Amplifiers

Aeroflex Laboratories, Inc.
South Service Road
Plainview, Long Island, New York 11803
(516) 224-6417

Control Systems Research, Inc.
1811 Main Street
Pittsburgh, Pa. 15215
(412) 781-1887

Inland Controls, Inc. - Kollmorgen Corporation
Alpha Drive
Pittsburgh, Pa. 15238
(412) 781-6011

Fiber-Optics

American Optical Corporation
14 Mechanic Street
Southbridge, Mass. 01550
(617) 674-3211

Bausch & Lamb, Inc.
61470 Bausch Street
Rochester, New York 14602
(716) 232-6000

Bendix Mosaic Fabrications Division
Galileo Park
Sturbridge, Mass. 01518
(617) 347-9191

Electro Fiber Optics Corporation
45 Water Street
Worchester, Mass. 01604
(617) 835-6082

General Electric Company
Electronics Park, Bldg. No. 6
Syracuse, New York 13201
(315) 456-2584

Optics Technology, Inc.
901 California Avenue
Palo Alto, California 94303
(415) 327-6660

Lamps

Chicago Miniature Lamp Works
4433 North Ravenswood Avenue
Chicago, Illinois 60640
(312) 784-1020

General Electric Company
Miniature Lamp Department
Nela Park
Cleveland, Ohio 44112
(216) 266-2121

Sylvania Lighting Center
100 Endicott Street
Danvers, Mass. 01923
(617) 777-1900

Motors & Tachometers

Servo

Aeroflex Laboratories, Inc.
South Service Road
Plainview, Long Island, New York 11803
(516) 224-6417

Bodine Electric Company
2500 West Bradley Place
Chicago, Illinois 60618
(312) 478-3515

Bowman Instrument Corporation
8000 Bluffton Road
Fort Wayne, Indiana 46809
(219) 747-3121

Cedar Division - Control Data Corporation
5806 West 36th Street
Minneapolis, Minnesota 55416
(612) 929-1681

Clifton Division - Litton Industries - Litton Precision Products Division
5050 State Road
Drexel Hill, Pa. 19026
(215) 622-1000

Inland Motor Corporation
501 First Street
Radford, Va. 24141
(703) 629-3972

Electro-Mechanical Division - Indiana General Corporation
517 West Walnut Street
Oglesby, Illinois 61348
(815) 883-8453

Kearfott Division - Singer - General Precision, Inc.
1150 McBride Avenue
Little Falls, New Jersey 07424
(201) 356-4080

Killsman Instrument Corporation
575 Underhill Boulevard
Syosset, New York 11791
(516) WA1-4300

Kollsman Motor Corporation
Mill Street
Dublin, Pa. 18917
(215) 249-3561

Magnetic Technology
21001 Kittridge Street
Canoga Park, California 91303
(213) 887-7700

Linear

Cedar Division - Control Data Corporation
5806 West 36th Street
Minneapolis, Minnesota 55416
(612) 929-1681

EEMCO Division - Instrumentation Motors - Electronics Specialty Company
4612 West Jefferson Boulevard
Los Angeles, California 90016
(213) 733-0151

Optics

Eastman Kodak Company
343 State Street
Rochester, New York 14650
(716) 325-2000

Mati Elgeet Division
4225 West Henrietta Road
Rochester, New York 14623
(716) 334-6880

National Cine Equipment, Inc.
37 West 65th Street
New York, New York 10023
(212) 799-4602

Optical Products Division - Teledyne Company
1725 Peck Road
Monrovia, California 91016
(213) 357-2216

Perkin-Elmer - Optical Group
Main Avenue
Norwalk, Conn. 06852
(203) 762-1000

Precision Optics - Division Penna Optical Company
234 South 8th Street
Reading, Pa. 19603
(215) 376-4961

3M Company
3M Center
St. Paul, Minnesota 55101
(612) 733-1110

Angenieux Corporation of America
440 Merrick Road
Oceanside, N. Y. 11572
(516) 678-3520

Potentiometer

Beckman Instruments Inc. - Helipot Division
2500 Harbor Boulevard
Fullerton, California 92634
(714) 871-4848

Computer Instruments Corporation
92 Madison Avenue
Hempstead, New York 11550
(516) 483-8200

CTS Electronics Inc.
Box 1278
Lafayette, Indiana 47902
(317) 463-2565

IRC Division - TRW, Inc.
401 North Broad Street
Philadelphia, Pa. 19108
(215) 922-8900

Sphere

Rohm & Haas
1920 South Tubeway Avenue
Los Angeles, California 90022
(213) 685-5060

Ray Products
703 South Palm Avenue
Alhambra, California 91803
(213) 283-8877

Galigher Company
554 West 8th Street
Salt Lake City, Utah 84110
(801) 359-8731

Vidicon Cameras

Ampex Corporation
401 Broadway
Redwood City, California 94063
(415) 367-2011

Cohen Electronics Inc.
Box 623
San Diego, California 92112
(714) 277-6700

Concord Electronics Corporation
1935 Armacost Avenue
Los Angeles, California 90025
(213) 478-2541

ECSO - General Electric Company
1 River Road
Schenectady, New York 12305
(518) 374-2211

GPL Television - The Singer Company
Pleasantville, New York 10570
(914) 769-5000

Maryland Telecommunications - Division KMS Indust., Inc.
57 Dodge Ave.
North Haven, Conn. 06473
(203) 239-5341