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BOOM ATTACHMENT SYSTEM

J. F. HAINES, A. T. WOODFORD, B. ABBOTT, ET AL,

SPECIAL PRODUCTS AND APPLIED RESEARCH DIVISION,
THE DE HAVILLAND AIRCRAFT OF CANADA, LIMITED

TECHNICAL REPORT AFAPL-TR-67-14

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FOREWORD

This is the final report prepared by the Aerospace STEMs group of the Special Products and Applied Research Division (SPAR) of the de Havilland Aircraft of Canada, Limited, Malton, Ontario on United States Air Force Contract AF33(615)-2724. 'Development of a Rigid Boom Attachment System', under Project 8170, Task 817008.

This work was conducted under the direction of the United States Air Force Aero Propulsion Laboratory, APFT, Wright-Patterson Air Force Base, Ohio, 45433, Mr. F.W. Oliver, Project Engineer.

This report covers work carried out between July 20th, 1965 and September 13th, 1966. Design, development and testing were conducted solely by the SPAR Division of the de Havilland Aircraft of Canada, Limited, under the direction of Mr. J.D. MacNaughton, Manager of the Aerospace STEMS Group, Special Products and Applied Research Division, The de Havilland Aircraft of Canada Limited.

Special acknowledgement is made to J.F. Haines, A.T. Woodford, B. Abbott, and J.F. Clayton of de Havilland, for their assistance during the program and in the preparation of this report. This report was submitted, in its present form, in July 1967.

This technical report has been reviewed and is approved.

James A. McMillan

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Chief, Space Technology Branch
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ABSTRACT

The low gravity field environment experienced in space presents problems to astronauts attempting to perform work outside their spacecraft. This problem can be defined in part as the difficulty to maneuver and the difficulty to keep station relative to a surface on which work is being carried out. This is mainly due to the greatly reduced restraining forces available to a space worker, particularly the lack of friction force as a result of body weight reaction. To become effective in space, man must develop means of controlling reactions to work loads.

Means have been studied to overcome this problem by restraint devices such as handholds, belts, and harnesses. Another means, and that which this report describes, is attachment to a work surface by rigid booms. A rigid boom attachment system offers a worker the advantage of being able to neglect the effect of moderate work loads.

Design and development of an evaluation model of such a system using unfurlable booms-an application of de Havilland's patented STEM (Storable Tubular Extendible Member) principle, was carried out by de Havilland along guide lines laid by the United States Air Force and within the restrictions of a limited

ABSTRACT (continued)

budget. The developed system should prove of great use in narrowing the guide lines to an optimum system.

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LIST OF ILLUSTRATIONS

LIST OF SYMBOLS

D	=	Diameter of storage drum (in.)
E	=	modulus of elasticity (lb/in^2)
F	=	axial load on boom (lb)
I	=	second moment of area of fully formed STEM (in^4)
I_o	=	second moment of area of STEM at root (in^4)
M	=	root bending moment (lb.in.)
M_{crit}	=	critical bending moment (lb.in.)
P,Q,R	=	forces reacting the torque T applied to the boom attachment system (lb.)
R_1	=	the larger of the two radii describing the STEM cross section at the root (in.)
R_2	=	The smaller of the two radii describing the STEM cross section at the root (in.)
T	=	applied torque (in.lb.)
Z	=	section modulus ($in.^3$)
b	=	width of the cross section at the root (in.)
d	=	boom nominal diameter (in)
f	=	flattening stress (lb/in^2)
h	=	depth of the cross section at the root (in.)
ℓ	=	length of extended boom (in.)
r	=	radius of the circle which passes through the three STEM booms p, q & r (in.)
t	=	thickness of the STEM element (in.)

LIST OF SYMBOLS (cont'd)

θ	=	the angle described by R_1
ϕ	=	the sum of the angles described by R_1 and R_2
ν	=	Poisson's ratio
γ	=	overlap factor
n	=	number of STEM elements per boom
ϵ	=	half of the subtended angles between booms p and q

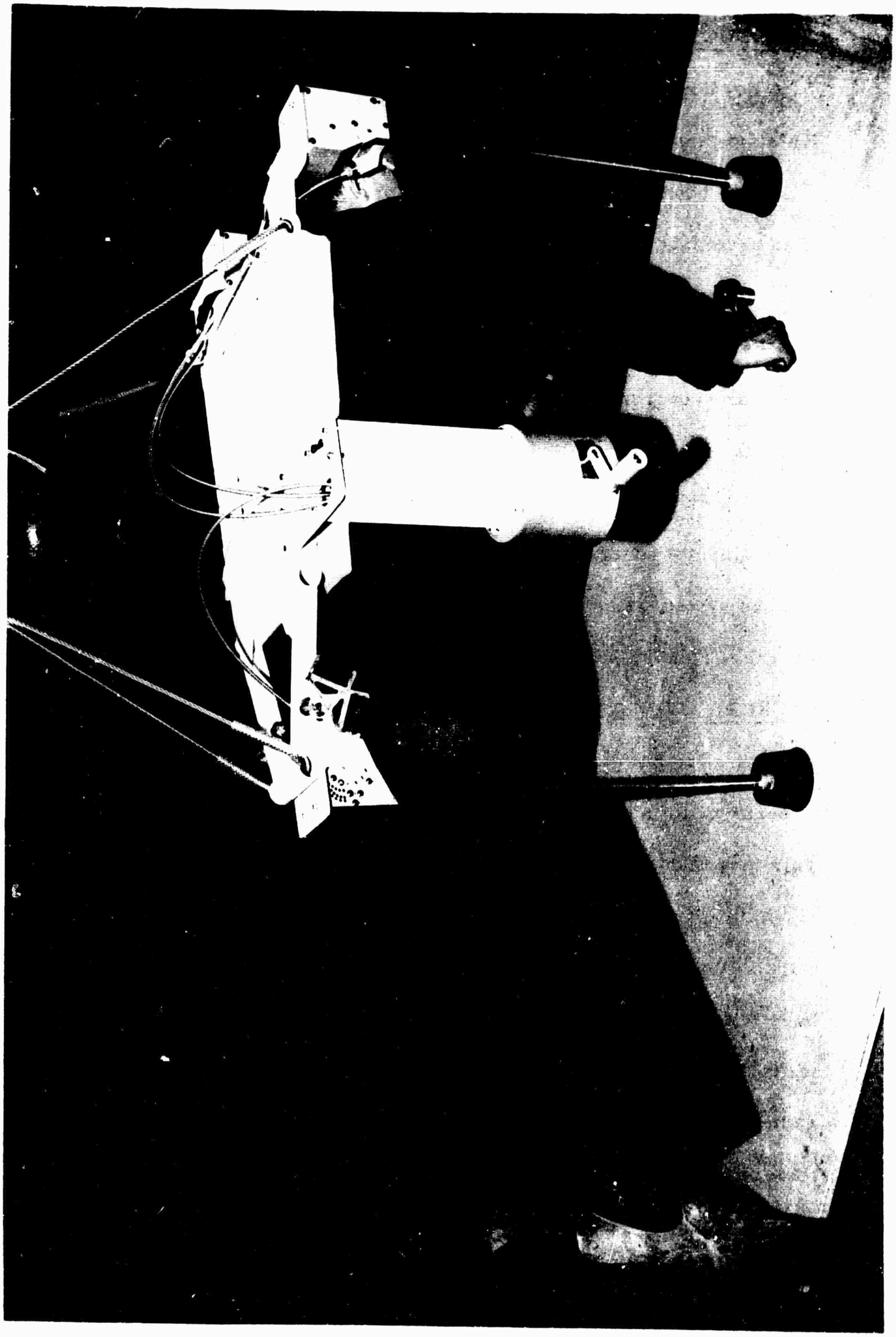


Figure 1 BOOM ATTACHMENT SYSTEM

I INTRODUCTION

A development program has been completed to deliver an evaluation model of an astronaut-boom attachment system, hereafter called the boom attachment system, which provides rigid attachment of an astronaut to a work site in a low gravity environment. This report describes the results of this program.

The system has been designed to meet the functional requirements arising from the attempts of personnel to perform extra-vehicular maintenance tasks. Consideration has also been given to the materials necessary for compatibility with the exacting environmental conditions of space.

The system is an adaption of de Havilland's patented STEM (Storable Tubular Extendible Member) principle.

II BACKGROUND

It is a logical assumption that the need will arise for astronauts to perform an increasing amount of extra-vehicular maintenance. This may take the form of inspection and repair of satellites, repair of orbital or interplanetary vehicles, and structural assembly work. However, certain problems peculiar to the space environment have to be overcome prior to successfully accomplishing these tasks. These adverse environmental conditions include vacuum, radiation and low gravity. Considerable effort has been expended to develop devices to protect the life of the space worker from these hazards. The low gravity, although not essential to life, support, is an environmental condition that will degrade the efficiency of a space worker. Low gravity becomes a serious problem when it is necessary for the astronaut to perform operations involving the application of force or torque. Low gravity conditions give rise to a lack of friction force which makes the above operations almost impossible to perform, especially if the force or torque has to be sustained. A space worker attempting to fasten a screw attachment in a low gravity environment would simply rotate about the point of application of torque with little actual work being achieved. On earth, the body weight or reaction between ground and feet provides adequate restraint to the reactive forces induced by the application of torque. In space, however, there is no restraint to these reactive forces which gives rise to rotation of the astronaut. This problem may, however, be overcome by the use of minimum reaction power tools and by physical restraints such as hand-holds, attachment belts or booms. This program has been directed towards developing a boom attachment system to provide this physical restraint by use of unfurlable structures.

A STEM is a flat strip of thin material which assumes a tubular shape of high strength when extended of unfurled. The tubular elements are formed out of strip metal heat-treated into a circular section in such a manner that the edges of the material overlap by approximately 180°. This provides the tubular elements with a bending strength almost equal to that of a seamless tube of the same diameter and wall thickness. The elements when retracted are stored in a strained flattened condition by winding them onto a drum. The STEM principle is similar to that of a carpenters steel rule.

III OBJECTIVE

The investigation and development was directed towards producing a system that would provide rigid attachment of an astronaut to a work site in a low gravity environment. The following requirements formed the basic for the technical approach.

1. An interchangeable manual or electric crank mechanism shall be investigated with the aim of driving the three boom attachment devices as a group or independently.
2. The storage container shall have a design objective of 5" x 4" x 4"; however, a package size of 8" x 4" x 4" will be acceptable by the Air Force if the design objective cannot be met.
3. Each boom attachment device shall not weigh in excess of four pounds.
4. A design study shall be made of the three point attachment system.
5. The boom attachment system shall have the capability of reacting a 45 ft. lb. torque application by the astronaut back to the structure attached to.
6. The boom attachment device shall have the capability of withstanding a minimum of 100 pounds in tension and compression.

FIGURE 2 DETAILS OF BOOM ATTACHMENT SYSTEM

7. Space qualified and compatible material shall be considered in the design; however, they will not be required in the fabrication of the boom attachment devices.

8. The boom attachment devices shall have the capability of being extended 10 feet in length although the loading conditions only apply for extended lengths of four feet.

9. The extension mechanism shall be capable of being operated by a pressurized suited astronaut in a low gravity environment.

10. A rigid attachment frame for the three boom devices shall be designed. The following configurations shall be considered for the storage containers.

    (a) Two positioned at the shoulders and one positioned on the stomach.

    (b) Two positioned at the shoulders and one positioned between the legs.

    (c) Two positioned at the hips and one positioned between the legs.

11. The extendible boom should be designed to be attached to the dispenser mechanism using a clevis or ball type connection.

A. <u>Definition of Terms</u>

It will be of assistance to explain the meaning of the terms used in describing the hardware produced under this contract.

The equipment is shown in Figure 2 and the definitions of the items are as follows:

1. End connection for attachment to adhesive pad dispenser mechanism, mechanical grasping device, or other interface.
2. Extendible boom.
3. Boom storage container.
4. Control Box (which includes the extension mechanism).
5. Rigid support frame or back frame.

' Boom attachment system' is defined as three boom containers complete with items 4 and 5.

The device providing the actual connection of the boom ends to the spacecraft surface (item 1, above) was not required under this contract. These were to be provided by the Air Force and attached by them to the boom end connections.

IV TECHNICAL APPROACH

The technical approach will be discussed under the following headings:

Analysis of optimum configuration - Sub section A.

Selection of boom element - Sub section B.

Design of boom container - Sub section C.

Design of control box - Sub section D.

Design of support frame - Sub section E.

Sub sections C - E describe the original design approach for the hardware. There has been a considerable amount of development during the course of assembly, especially to the boom containers, so that it is necessary to have a detailed hardware description in Section VI after the Development Section.

A. Analysis of Optimum Configuration

Recapping on the configurations considered for the location of the boom containers:

1) Two positioned at the shoulder and one on the stomach.
2) Two positioned at the shoulders and one positioned between the legs.
3) Two positioned at the hips and one positioned between the legs.

In discussing the relative merits from the strength point of view of the above layouts, the loads to be applied to the system have been stated as:

a) 100 lb. end load tension or compression in each boom.

b) 45 ft. lb. torque acting on the complete system.

These loads are to be applied with the booms extended to a length of 4 feet.

With regard to the end load condition the three layouts are obviously of equal merit. However, the geometry of the system has a great effect on the stresses in the booms due to the torque loading. The magnitude of the shear load applied at the tip of each unfurlable boom depends upon the radius of the circle which passes through all three STEM tips, and their relative circumferential position. For optimum design the selection of boom element is the first step to be taken, as this is governed by the boom container size limitation. Knowing the boom element parameters, the maximum acceptable shear load is known and this governs the radius of the circle passing through the boom ends. Using the boom element parameters determined in sub-section B the radius is required to be at least 17.5 inches with the containers equi-spaced on the circumference. For layouts (1) and (3) the containers are less than the optimum 17.5 inches. This means that to have a circle of 17.5 inches radius through the ends of the STEM booms, the boome must be set at an angle, the value of which changes during extension and retraction.

If the attachment system were to be designed to provide for this change of angle the attachment of the boom container to the support frame must be a pin joint. In order to resist the torque applied by the astronaut, the axis of the pins must be tangential to the circle through the boom containers. Then the bending moment, being induced by the shear loads at the tips of the STEM booms, will be applied as differential loads at the pin. It is for this reason that it is not possible to mount the boom containers on universal ball joints at the attachment frame. It is essential that this joint should resist the bending moment applied to the boom at its point of attachment, which is a result of reaction to the torque applied by the space worker.

For layout (2) the booms can be positioned to extend parallel to each other giving a boom end circle radius of 19.125 inches. This allows the boom units to be 'built in' at the attachment to the astronaut support frame. This is obviously a distinct advantage as it dispenses with the pin joints. Apart from the structural benefit this arrangement should be simpler for the space worker to use as there are no moving parts that have to be positioned correctly. It is anticipated that with layouts (1) and (3) the astronaut would have difficulty in ensuring that he had made his attachment in the correct disposition on the work surface. For these reasons layout (2) was adopted as the optimum system and the design was completed on this basis.

B. Selection of Boom Element

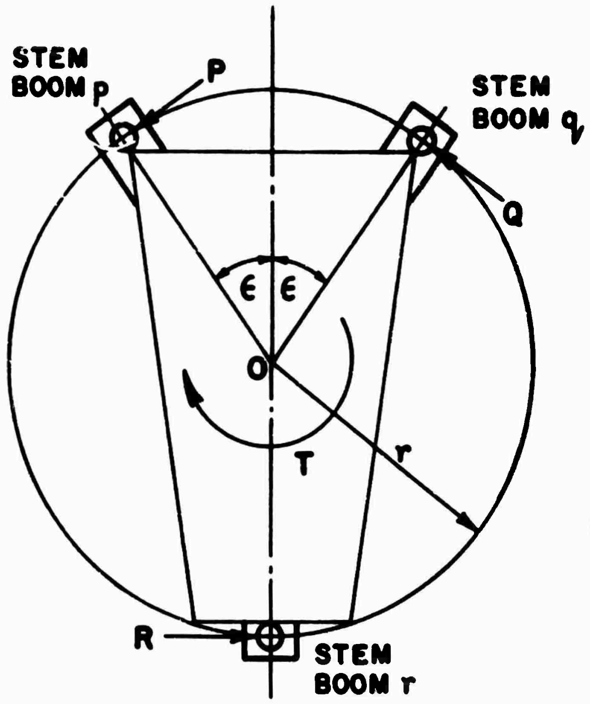
Element material	=	Stainless Steel
Ultimate tensile stress	=	230,000 lb/in^2
Modulus of elasticity 'E'	=	29×10^6 lb/in^2

Two conditions governed the choice of boom element, the container size limitation and the need for as large a boom diameter as possible. Strip width of 3 in. was selected to fit within the 4 in. wide container, and an overlap factor $\gamma = 1.27$. Therefore boom nominal diameter d is seen to be

$$d = \frac{3}{\gamma \pi} = 0.75 \text{ in.}$$

Minimum allowable $\frac{d}{t}$ ratio is given by:

$$\frac{d}{t} = \frac{E\left(\frac{D}{d} + \nu\right)}{f \frac{D}{d}(1 - \nu^2)} \qquad (1)$$



PLAN VIEW

r = 19·125"

ϵ = 25·65°

ℓ = 4 FEET FOR DESIGN LOADS

T = 45 ft lb TORQUE

P, Q, R = FORCES REACTING TORQUE

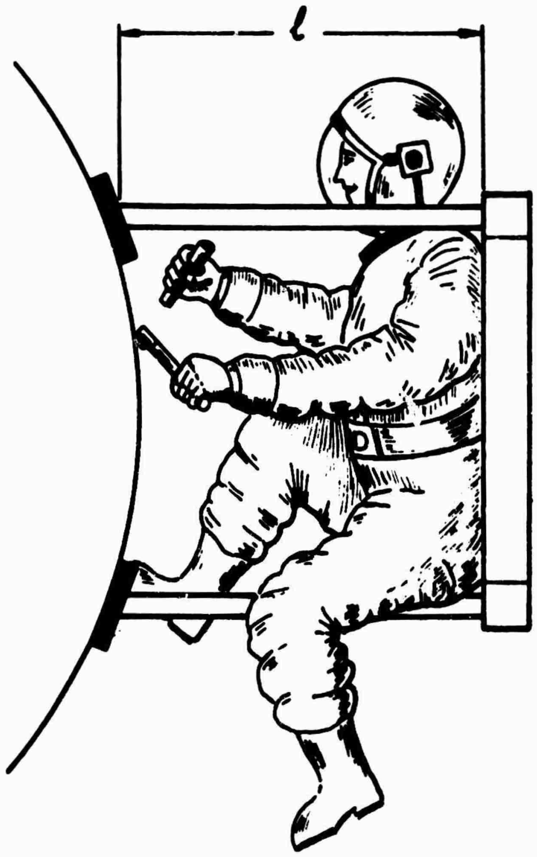


FIGURE 3 GEOMETRY OF BOOM ATTACHMENT SYSTEM

Where D $=$ Diameter of drum $=$ 1.5 in.

ν $=$ Poisson's ratio $=$ 0.3

f $=$ Flattening Stress $=$ 200,000 lb/in^2

f is chosen at a value sufficiently below UTS to provide an adequate fatigue life. Also the stress is kept at a value where relaxation of formed diameter is minimized.

$$\text{Minimum } \frac{d}{t} = 184$$

for d $=$ 0.75 in., thickness t $=$ 0.004 in.

The geometry of the system is shown in Figure 3. Using the condition of equilibrium of forces and moments, the following relationships giving the shear loads applied to each boom can be derived

$$P = Q = \frac{T}{2r(1 + \cos\epsilon)} \qquad (2)$$

and

$$R = \frac{T \cos\epsilon}{r(1+\cos\epsilon)} \qquad (3)$$

Using the values

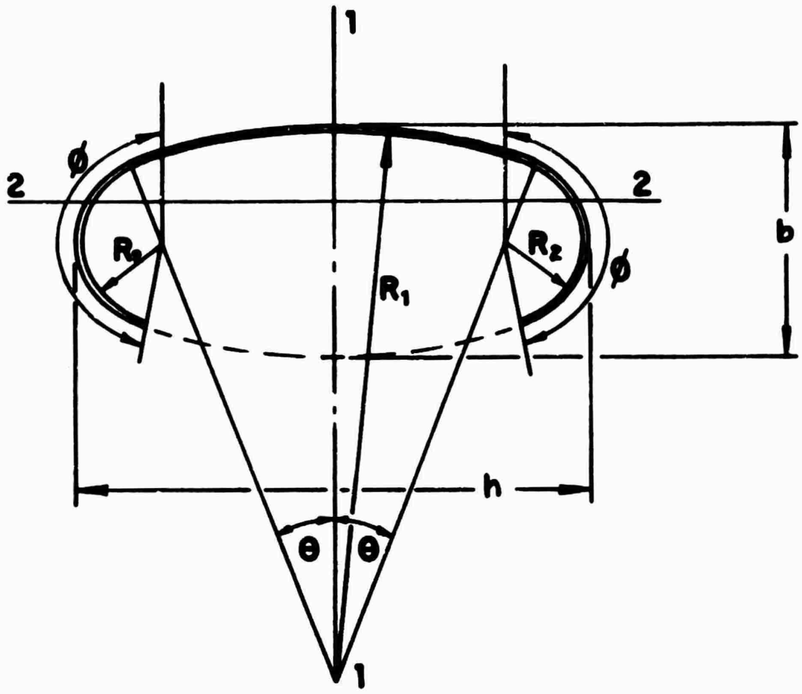
r $=$ 19.125 in.

$\cos\epsilon$ $=$ 0.9014

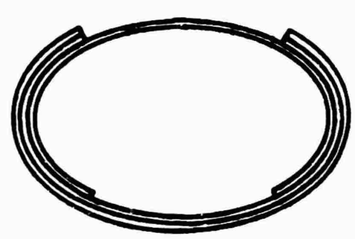
T $=$ 45 ft.lb. $=$ 540 in. lb.

then P $=$ Q $=$ 7.4 lb. and R $=$ 13.4 lb.

The choice of the number of STEMs per boom is governed by the geometry of the boom cross section at the root, which in turn is controlled by the length of the container. With a container length of 5 inches the section geometry at the root is as shown in Figure 4.



(i) VIEW SHOWING TYPICAL CROSSECTION THROUGH ONE ELEMENT AT THE ROOT



(ii) VIEW SHOWING THE ARRANGEMENT OF THE THREE ELEMENTS AT THE ROOT

FIGURE 4 STEM BOOM DETAILS

In order to determine the second moment of area of the section at the root, the values of R_1, R_2, θ and ϕ must be known. From past experience it is known that $R_2 = \frac{\gamma d}{2}$; and, given the dimensions h and b, the values of R_1, θ and ϕ are as follows:

$$R_1 = \frac{(b + h^2) - 2\gamma dh}{4(b - \gamma d)} \tag{4}$$

and hence

$$\theta = \sin^{-1}\left[\frac{h - \gamma d}{2R_1 - d}\right] \tag{5}$$

and $$\phi = \pi + \theta\left[1 - \frac{(b^2 + h^2) - 2\gamma dh}{2\gamma d(b - \gamma d)}\right] \tag{6}$$

thus given:

	h	= 1.3 in.
	b	= 1.29 in.
	d	= 0.75 in.
	γ	= 1.27
Then	R_1	= 1.33 in.
	θ	= 0.61 rad.
	ϕ	= 2.06 rad.

The section at the root has principal axes 1-1 and 2-2. Because axis 2-2 is tangential to the circle in Figure 3, bending takes place solely about axis 1-1.

It can be shown that:

$$I_{11} = t\Big[\theta(R_1^3 - R_2^3) + R_2^3\phi - R_1^3\frac{\sin 2\phi}{2} + 4R_2^2\sin\theta\cos\phi(R_2 - R_1) + \sin\theta\cos\theta(4R_2^2R_1 - 3R_2^3 - R_1^3) + 2R_2\phi\sin^2\theta(R_1^2 - 2R_1R_2 + R_2^2) + 2R_2\theta\sin^2\theta(2R_1R_2 - R_1^2 - R_2^2)\Big] \tag{7}$$

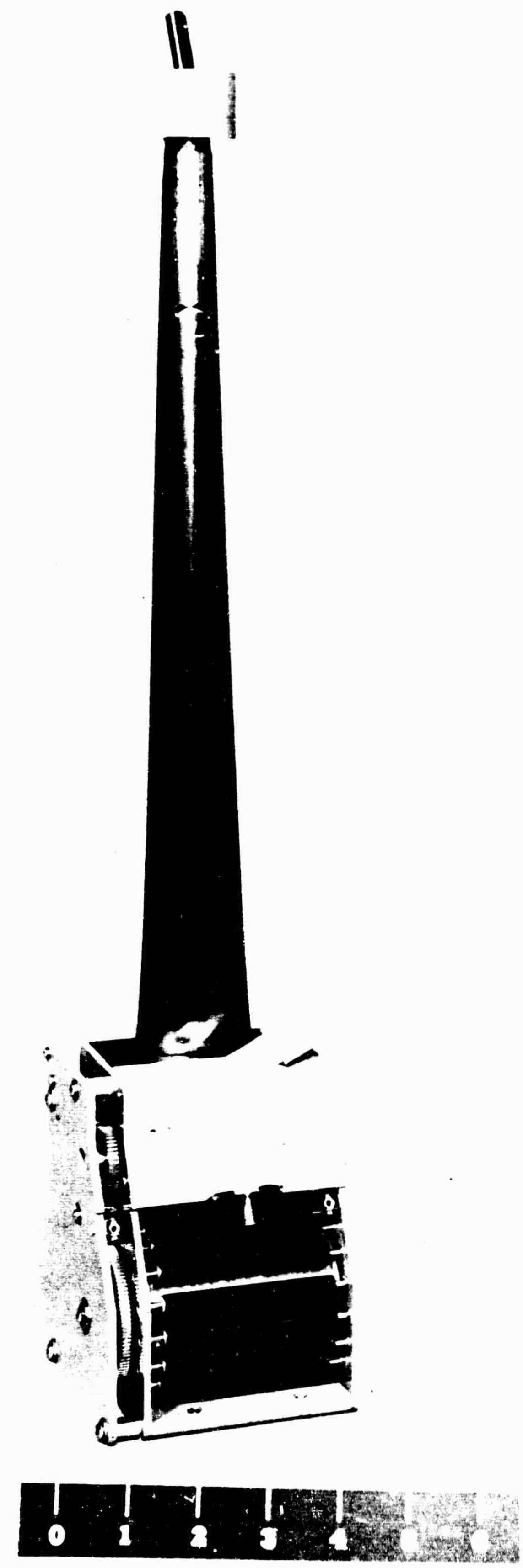


Figure 5 UNDEVELOPED BOOM CONTAINER APPEARANCE
WITH COVERS REMOVED

Which gives $I_{\shortparallel} = 0.0066 \text{ in}^4$ per element.

The critical bending moment at which instability occurs is given by:

$$M_{crit} = \frac{0.96EI_{\shortparallel}nt}{\gamma d(h+2R_1 \sin\theta)}$$

$$n = \frac{\gamma dR(h+2R_1 \sin\theta)\ell}{0.96EI_{\shortparallel}t} \qquad (8)$$

For the configuration chosen this gives a value of

$n = 3$(i.e. 3 elements are necessary per boom).

The end load at which Euler buckling occurs in STEM tubing is given by:

$$F = \frac{\pi^2 EI}{\ell^2}$$

where $\quad I = 0.525nd^3t = 2.65\times10^{-3}\text{in}^4$

$F = 330$ lb.

The end load at which local compression instability occurs is given by:

$$F = \frac{0.25nE\gamma\pi dt^2}{R_1}$$

$= 770$ lb.

Thus Euler buckling is the critical mode. It should be noted that the critical Euler buckling load of 330 lb. will cause total collapse of the boom. Before this happens other instabilities will occur. For this reason the tensile and compressive loads applied to each boom are restricted to 100 lb.

Therefore the boom element specification is as follows:

Number of booms	=	3
Element material	=	Stainless Steel
Ultimate tensile stress	=	230,000 lb/in^2
Modulus of elasticity E	=	29×10^6 lb/in^2
Element strip width	=	3 in
Element thickness	=	0.004 in
Overlap factor	=	1.27
Boom nominal diameter	$\doteq$	0.75 in
Number of STEM elements per boom	=	3.

C. Initial Design of Boom Container (see Figure 5)

To meet the size limitations on the container and in particular the length requirement of 5 inches, it was decided that each container should have two back wound elements and one forward wound element. (A STEM is back wound on the drum when the inner surface of the element 'as-formed' appears facing radially outwards on the storage drum. A forward wound element has the inner surface of the as-formed element facing inwards on the drum when stored). These were stored on the same drum and on deployment formed around each other. This theoretically gives adequate root strength with a minimum length of guidance support because the elements support each other and prevent edge buckling which is the initial instability of the partially formed tube.

The drum was contained within a cassette formed of full width rollers. It is essential that these rollers be full width rather than the conventional edge rollers since the outermost element on the drum is forward wound and its natural tendency is to bow.

This tendency has to be restrained. The forward wound and back wound elements were led out of the cassette by separate systems of rollers, so positioned that the natural ploy of elements was to form a single tube.

An engineering breadboard model was constructed to ascertain the feasibility of this arrangement and it was found to function satisfactorily.

The drive to the container for extension and retraction of the element was by means of a flexible shaft from the control box. The drive passed through worm and wheel and spur gearing to the drum in the container.

A clamp on the element at the exit from the containers was incorporated. The object of this was that, when the booms had been extended to the operating position, they might be clamped to prevent cassetting (that is, any movement of th booms 'into' or out of the container). To effect any change in extended length of the boom, the clamps must first be released.

D. Initial Design of Control Box

The control box (see Figures 6 and 12) was designed to be located at the astronaut's hip. It was felt that this position was satisfactory for the physical function of winding the crank handle. In addition it was less restrictive on the astronaut's access to his work area and the flexible drive shafts could conveniently be

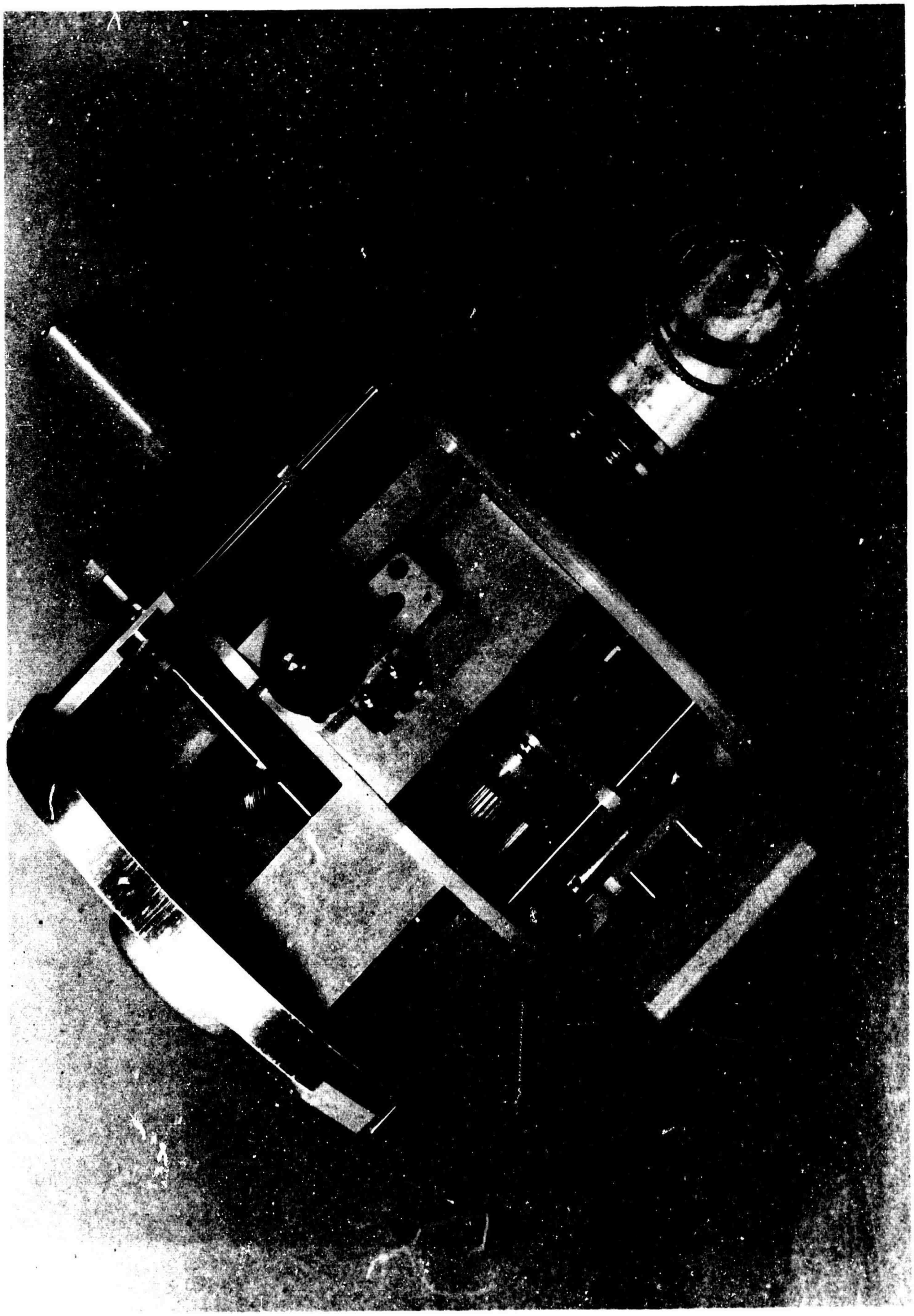


Figure 6 CONTROL BOX WITH COVER REMOVED

carried round the back of the astronaut.

The requirements laid out under 3.0 Objective, have resulted in a fairly complex design for the control box. Power drive was supplied by electric motor or by hand crank and the design was such that either of these supplies may be used, there being a rapid changeover mechanism. The drive was transmitted by spur gearing to the three flexible drive shafts connected to each boom container. The drive to each or any of these shafts may also be disconnected or restored by depressing a lever. In order to safeguard the system, slipping clutches were incorporated in both the manual and electric drives and these clutches were set to slip before any damage occured in the drive mechanism.

The boom controls were located on the forward face of the control box with a central push button and three levers equally spaced around the periphery of the container in the same sense as the location of the boom containers. By depressing the lever the drive to the appropriate boom container was disconnected. When depressed the levers were flush with the front face of the control box. This was in order that the astronaut may be able to identify which levers have been depressed by feel through his gloves. The drives to the boom containers were reconnected by depressing the central push button which restored any depressed levers to their position proud of the front cover.

E. Initial Design of the Back Frame

From the start of the project it was anticipated that the three

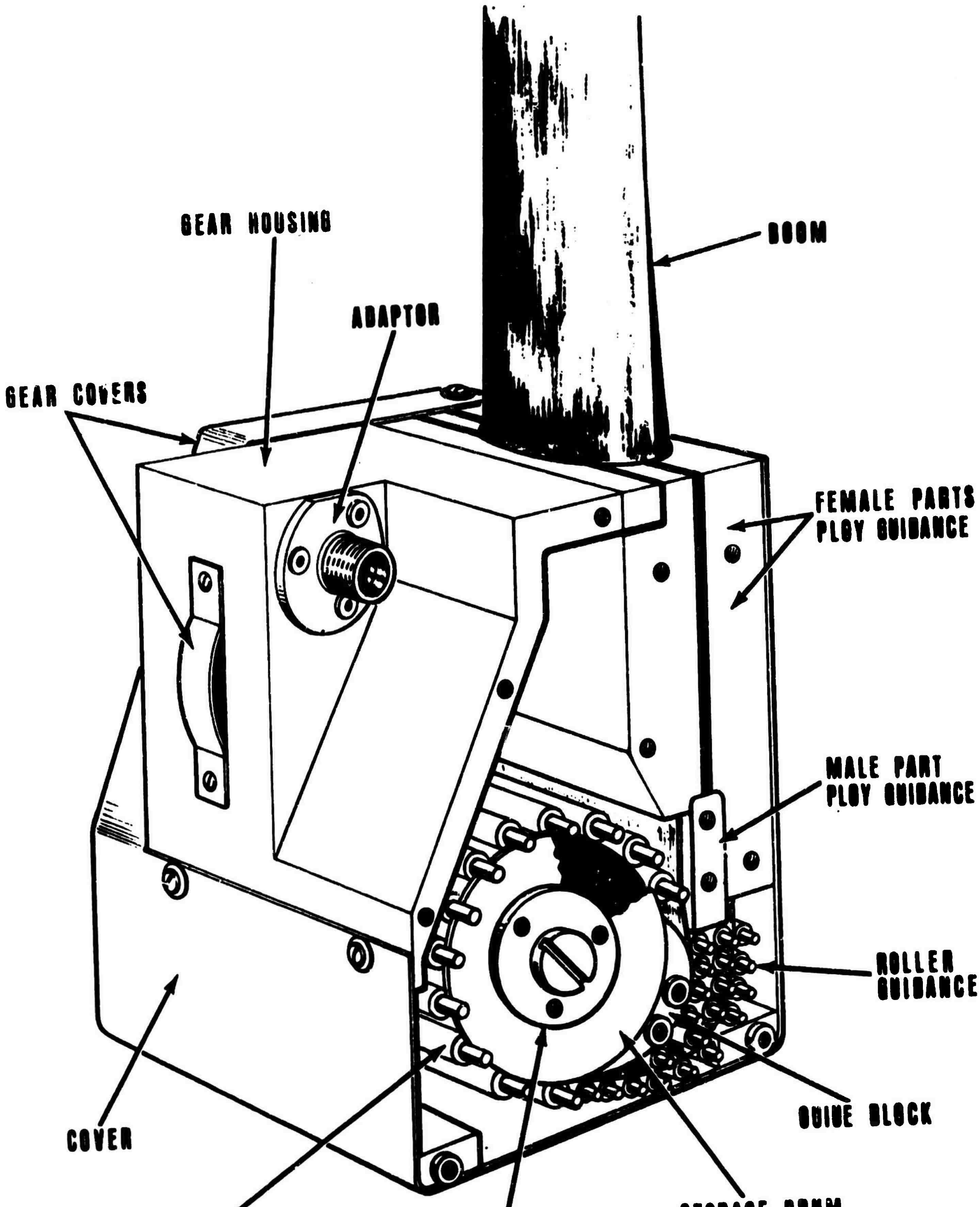


FIGURE 7 BOOM CONTAINER WITH SIDEPLATE REMOVED

boom containers and control box might someday be incorporated into a backpack maneuvering unit. As a consequence the back frame (see Figure 1) was designed solely for test and evaluation purposes and not as flight hardware. Since there was a safety factor of 6 on the de Havilland Multi Degree of Freedom Rig, this factor was also applied to the back frame. This satisfied all the safety requirements for de Havilland personnel to evaluate and test the system.

The back frame was manufactured from aluminum alloy tube with an aluminum alloy web plate and gussets. Welded construction was used throughout.

The attachment of the back frame to the operator was by means of quick release belts. These were arranged into a shoulder harness with a crossover on the chest and separate belts around the thighs. There was some doubt on the optimum layout for these belts and the decision as to the configuration was left to the development phase of the program.

V DEVELOPMENT

This section will describe the post-design development of the boom attachment system. The principles involved in the deployment mechanism had previously been reduced to practice by de Havilland and required adaptation and repackaging to meet the requirements of this system.

A. Boom Containers

Most of the development work on these items was concentrated on reducing the input torques to an acceptable level. This was essential to avoid extensive modification to the control box and degrading the entire system concept. Four problem areas emerged during the course of development. Three of these were directly connected with the problem of high input torques and were resolved satisfactorily whilst the fourth arose as a result of modifications necessary to solve the preceding problems and was not itself remedied. These problem areas are described below.

1. Guidance Development

The boom container guidance system has two basic parts; the roller guidance which leads the flattened STEM off the cassette to the ploy guidance; and the ploy guidance (see Figure 7), which supports the STEM during its transfer from flat to normal tubular shape. It must be realised that normally the guidance systems have only two functions; firstly to control the position of the STEMs, and secondly to support them when transferring compressive loads (or in the case of the ploy guidance any applied load). Thus the ideal guidance system would be one which would impose no friction loading on the forming booms during boom extension and retraction.

It would be very difficult to assess analytically the magnitude of guidance friction, and, since this was a newly designed unit, no empirical information was available. Consequently, in the design stage an assumed pessimistic guidance efficiency of 50% was used.

When a boom is retracted, strain energy is supplied to the STEMs in flattening them and storing them on the drum. For the boom containers this results in a calculated maximum drum torque of 9.3 in. lb.

Thus, assuming a 50 % guidance efficiency, the maximum torque at the drum would be 18.6 in. lb.

This figure of 18.6 in. lb. max. was that around which the boom container and control box was designed, controlling gear ratios, gear sizes, slip clutch torque capacities, etc.; and thus it was very important that the actual drum torque should not exceed this figure.

The basic guidance operation was checked immediately after the first boom container had been assembled without gearing. It was found that (a) the torque at the storage drum necessary to extend or retract the booms was nearly three times greater than the allowable design maximum, (b) the inner of the three co-axial booms buckled as it emerged from the ploy guidance, (c) even if the booms were forcefully joined at initial extension they tended to separate as extension continued, (d) the designed boom end plug was unsatisfactory, as it caused local buckling of the STEM. Apart from the tip plug fault, which was simply unsatisfactory design, the other phenomena were related to a single cause: the basic ploy guidance shape was wrong. This steadily became more apparent as modifications were made to the ploy guidance in an attempt to improve its operation. Partial success was achieved; buckling and the problems of booms not joining on initial extension were eliminated; but drum torques could not be lowered below twice the design maximum, and the booms still tended to separate during extension.

Two other problems became apparent. Firstly the ploy guidance material was unsatisfactory, and secondly a guide block forming part of the roller guidance was not up to its task. These two problems were minor in comparison with that posed by the basically wrong ploy guidance shape, but for completeness will be enlarged on. For strength reasons the ploy guidance casting material chosen was an aluminum powder filled epoxy resin. Sliding contact between STEM and guidance released aluminum powder from the guidance surface and this powder built up inside the unit, thus posing a potential hazard to bearings. The guide block, made of Teflon, was unsatisfactory for three reasons: accurate positioning was difficult to achieve; heli-coil inserts stripped out of the block; and the parts after machining were bowed, due to relief of the cast-in stresses.

In order to overcome these guidance problems, and in the hope of reducing the retraction torque at the drum to an acceptable figure the following action was taken. All ploy guidances were remade. The filler was changed from aluminum powder to

Molycote and chopped strand glass fibre . The guide blocks were redesigned and remanufactured.

On re-assembly and checking of basic guidance operation, all aspects of operation were satisfactory except that retraction torques at the the drum were still high, around 30 in. lb. Modifications such as fluting and Teflon spraying of the ploy guidance to reduce contact area and friction were tried with little effect. It was found that retraction torques of 25 in. lb. max. could be obtained with a 0.030 in. gap between the male and female parts of the ploy guidance and this was reluctantly accepted as the best that could be obtained. To get retraction torque down to 25 in.lb. required that the roller guidance roller bushings be lubricated. Without lubrication these rollers would not rotate.

That concludes the guidance development description but the question remains could this figure have been improved on and was the design figure of 9.3 in. lb. realistic. As already mentioned the theoretical retraction torque is calculated from the strain energy transferred to the booms during storage or retraction. The design calculation was based on the nominal STEM thickness of 0.004 in.; however, a sample of the STEM tube was found to be 0.00435 in. thick. Based on this figure the calculated design torque becomes 12 in. lb. Had the design been based on this figure, the actual figure of 25 in. lb max. would have been acceptable. However, as already mentioned, the assumed guidance efficiency of 50% was considered pessimistic and this justifies the use of the nominal boom thickness in the design calculation. It is interesting to note that retraction torques without the STEMs passing through the guidance were measured at about 12 in. lb. It is obvious from the development description that friction in the ploy guidance was largely responsible for the high retraction torques. The obvious solution is to reduce the friction coefficient between the booms and the ploy guidance. As stated this was attempted when the guidance parts were sprayed with Teflon. To get a durable spray coat this would have to be cured hard and the temperatures involved would be incompatible with the basic guidance material. To rectify this by changing the basic ploy guidance material would have been prohibitively expensive as such methods as lost wax casting or profile machining would have had to be investigated. Had this been known in the design stage this or perhaps some alternative ploy guidance system could have been included in the design development stage.

2. Gearing Development

As designed the gear train in the boom container consisted of three spur gears and a worm and wheel giving an overall reduction ratio of 36.4:1. This high reduction was necessitated by the relatively low torque capacity of the flexidrive shafts.

Following the guidance development the units were fully assembled. Testing of units soon showed that excessive input drive torques, incompatible with known drum torques, were present. This was traced to the worm gearing in the units. It was assumed this was the result of mis-alignment between worm and wheel and considerable effort was expended correcting this. From subsequent operation it was concluded that the structure was not rigid enough to maintain a set alignment. To cure the assumed inconsistent worm and wheel alignment, this part of the unit was re-designed to make the arrangement more rigid and easier to align. The advantages of the modified arrangement were:

(a) Worm wheel shaft length was considerably reduced and wheel position was such as to minimize shaft deflection.

(b) Alignment between worm and wheel was far easier as this became part of the worm housing sub-assembly.

(c) Positioning of worm housing between side plates was no longer super-critical.

Functional tests on the modified and re-assembled boom containers were satisfactory and the units were assembled to the back frame. Design Acceptance Test procedures commenced. During the course of this testing manual operation became steadily more difficult. On stripping down the units considerable worm wheel wear was noted. Concluding that this wear was due to mis-alignment, overloading or a combination of both, it was decided to redesign the unit gear train, eliminating the worm gearing and substituting spur gearing.

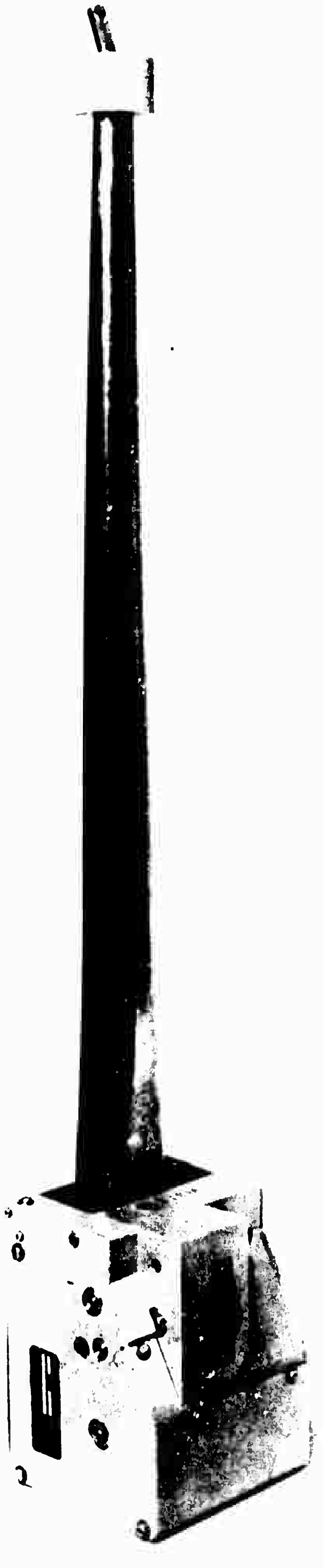


Figure 8 FINAL BOOM CONTAINER APPEARANCE

This has resulted in a slightly more complicated unit but the drive efficiency is greatly improved. This improvement has been offset by a reduction in the gear ratio to 29.5:1 necessary due to space limitation. An unfortunate feature of the re-design from the system point of view was that, the right-angle drive having been eliminated, the path of the flexidrive shaft was changed. This has not affected system operation but the shafts would probably have to be sheathed to avoid damage to an operator's pressure suit.

The gearing development problem can be summarized in one sentence. Due to the low guidance efficiency, gear efficiencies unobtainable with the design set were essential. The developed boom container appearance can be seen in Figure 8.

3. Boom Clamp

In the initial design a boom clamp was incorporated. Its intended action is described in Section IV (sub-section C). It was realized during the ploy guidance development that increasing the gap between the male and female parts of the ploy guidance was going to reduce the effectiveness of the boom clamp. This gap increased from 0.010 to 0.030 in. approximately. As the total movement of the clamp as designed was 0.015 in. (at a constant velocity ratio) and as it was necessary that during extension or retraction of the booms the clamp would not touch the booms, its effectiveness became insignificant. It was discarded in the final stage of the gear development. An attempt was made to improve its action by doubling the movement to 0.030 in., which was the maximum obtainable without a complete redesign. Even with this movement the clamping effect was small compared with the required effect.

Elimination of the boom clamp means that any compressive or tensile boom loads above that reacted by friction in the ploy guidance are reacted back through the drive mechanism; and, due to the cassetting of the boom, some movement of the boom into or out of the boom container will occur.

The amount of movement depends on the cassette size, the degree of fullness of the cassette (i.e. the boom extended length), and whether the load is applied following extension or following retraction. After retraction, for example, the boom is tightly wrapped around the drum. If the drum is then locked with the boom extended 3 ft. and a tension load applied the boom will extend a fraction of an inch. Under a compression load the boom would retract as much as a foot as the boom elements expand out to the cassette walls.

There is some doubt whether a pressure suited astronaut would be able to operate a boom clamp on the boom container between his legs when strapped into the system. Since the ploy guidance is now developed, on any future units a satisfactory boom clamp design could be arrived at.

4. The Boom

Two boom problems emerged during development. The first and most easily cured was one commonly occurring on STEM units - that of backwinding. Backwinding is a local folding back of the flattened STEM, generally at the point where the STEMs are attached to the drum. It is caused when the boom is in compression and occurred in this case at that part of the boom which is unsupported in the unit (i.e., the short length of STEM between the fixing point on the drum and the point when the STEMs begin to cassette). It is important to eliminate backwinding, as it results in a boom's inability to take compressi e loads. It was cured by fitting reinforcing leaves on either side of the boom at the drum.

The second problem was that of arriving at a satisfactory boom end plug design and fixing arrangement. In the design arrangement the end plug was attached to all three elements of each boom and was of such a shape that on assembly it locally buckled the boom. The plug shape was changed and the buckling eliminated but the fixing arrangement caused excessively high winding torques at the drum by forcing the STEMs to clamp around the male part of the ploy guidance during the last few feet of retraction. The clamping effect was eliminated by attaching the plug to only one of the STEMs forming the boom. This was necessarily the forward wound STEM, to achieve satisfactory joining of the STEMs during initial extension.

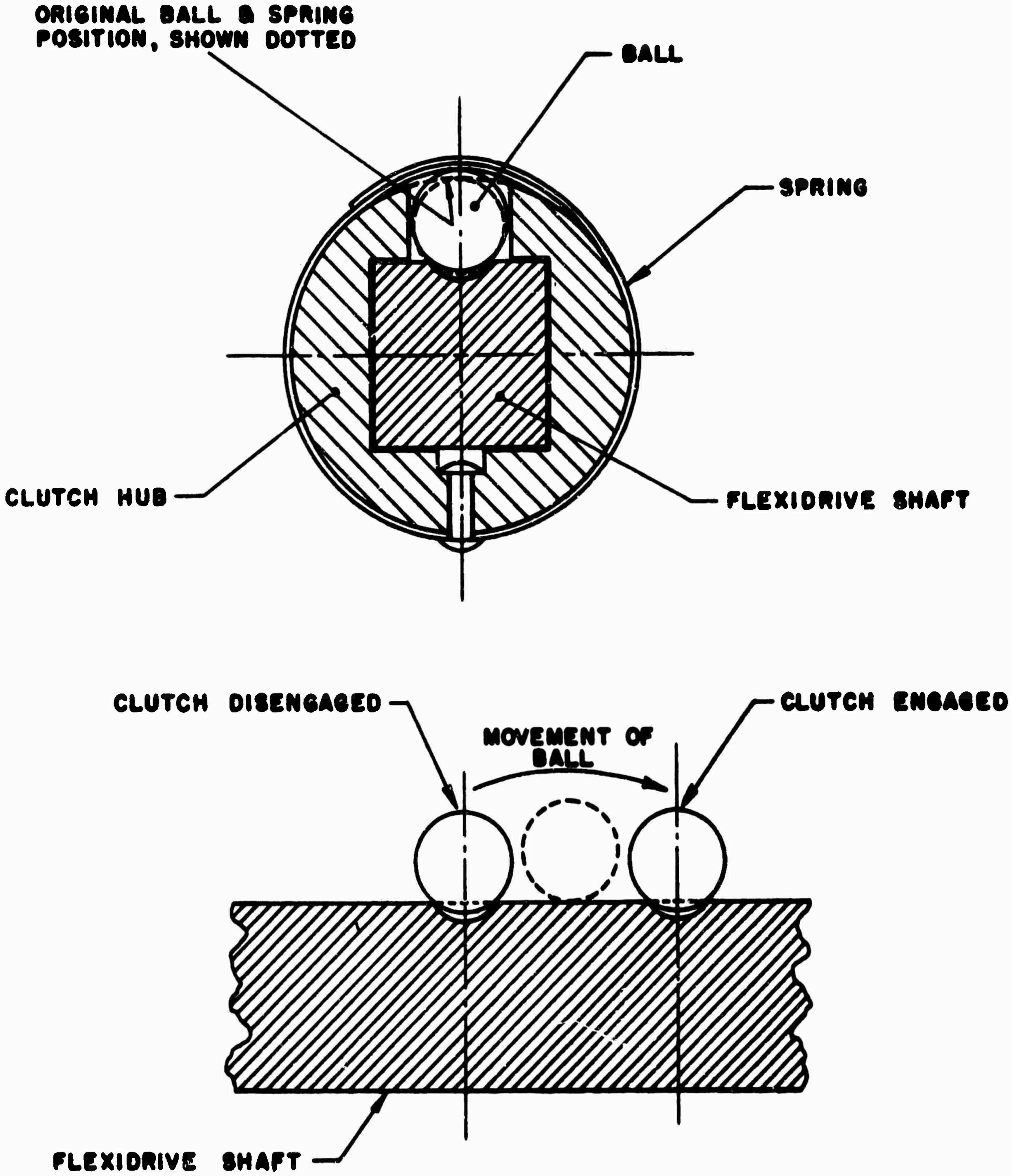


FIGURE 9 BALL AND DETENT ARRANGEMENT CLUTCH DISENGAGE MECHANISM

As far as unit function goes, the final boom end plug arrangement was completely acceptable; but from the system point of view the sharp exposed ends of the STEM tubes would be a hazard to the user's space suit. These ends are only exposed in the first few inches of extension and a simple shroud attached to the end plug and covering the exposed edges would cure the problem. Any future units would include this.

B. Control Box

Regarded as an individual unit rather than part of a system, there were remarkably few development problems; and those that did occur were either predicted in the design stage or were of a minor nature.

The one predicted development area was in the clutch operation for connection or disconnection of drive to individual boom containers. This operation is described in Section VI (sub-section B. 4a). As designed, clutch engagement was not positive and clutch hold-in force was too low. This was cured by installing a bigger ball and a weaker spring in the detent mechanism, (see Figure 9).

In the first case, when the ball was in the detent the spring ceased to act and the clutch throw out forces moved the ball out of the detent, thereby bringing the spring into action. The end result was either complete disengagement or the point being reached where the spring reaction transferred by the ball balanced the clutch throw-out force.

In the second case, bigger ball is in permanent contact with

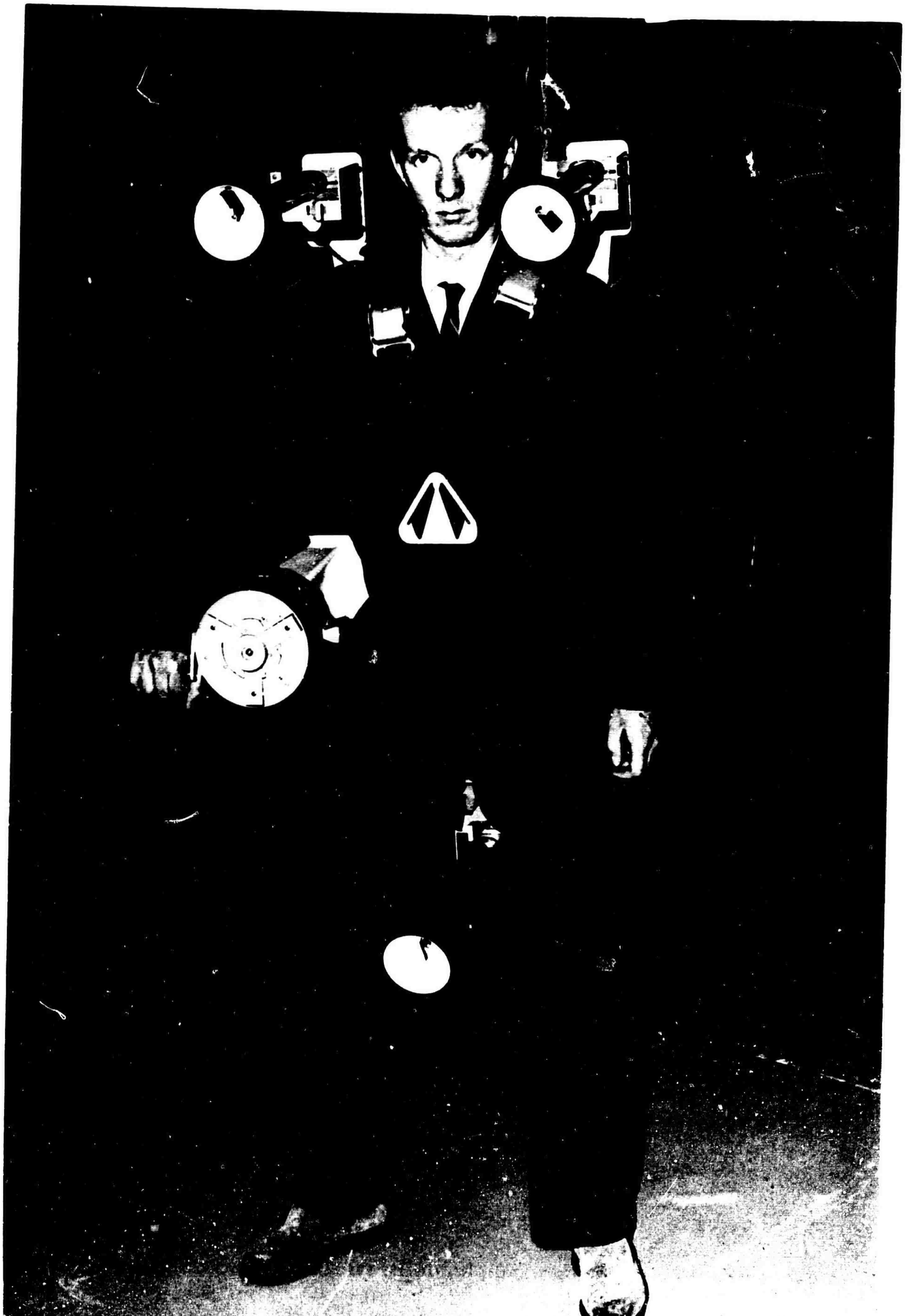


Figure 10 HARNESS ARRANGEMENT

the spring in or out of the detent; when in the detent, reaction to the spring force is large enough to more than balance the clutch throw out force. A weaker pre-loaded spring was installed. The pre-load supplied the reaction necessary for positive engagement, whilst the lower rate eased the transfer from detent to detent (i.e. from clutch engaged to clutch disengaged).

Regarded as part of a system, a development problem on the control box did occur as a result of the high drive torque on the boom containers.

A design feature of the control box was the slipping clutches. These were included to protect the control box gearing, boom container gearing, and the connecting flexidrives from damage in the event of excessive drive torques being applied either from the motor or hand crank. It was not possible to adjust the hand crank slip clutch to take the higher torques resulting from the increased boom container retraction torques. Thus it was possible to extend all three units together but not possible to retract them without the clutch slipping, boom container retraction torques being higher than extension torques. For this reason the hand crank slip clutch was locked by means of pins.

This has the effect that, when operating the system manually, the operator must observe full extension or retraction either visually or by feel through the drive to avoid subjecting the system to excessive torques. For example, full retraction is accomplished when the boom tip plug contacts the male part of the boom container ploy guidance. However, the handle can still be turned a few more times before failure, due to the orsional flexibility of the flexidrive shaft. Thus feel is not the best guide.

On any future similar units we would correct this situation by using a slipping clutch of higher capacity, such as that used on the motor drive.

C. Harness

The harness arrangement (see Figure 10) was designed during the development phase, the prime requirements being considered as comfort and adaptability to de Havilland personnel who had to hang in it on our multi-degree of freedom test rig during function and Design Acceptance Tests. The success of the existing harness arrangement can be judged by the fact that five people have, at different times, been accomodated in it, one hanging for a period of approximately 1-1/2 hours.

The final arrangement of system to operator attachment was by means of quick release belts (see Figure 2). There is a diagonal shoulder harness with an adjustable cross over position and separate belts round the thighs. The back frame surface facing the operators back was padded for operator comfort.

D. BACK FRAME

The control box support flange was found to be neither stiff enough in its support of the control box nor adequately attached to the remainder of the back frame. This was cured by attaching a channel section to the flange and through the tubular frame members (see Figure 1).

VI DETAILED HARDWARE DESCRIPTION

A. Boom Containers

The drive connection to the boom containers is made at the input shaft, which has a broached square hole to receive the flexidrive inner cable (see Figure 11). This and all other drive shafts in the unit are of stainless steel. Retention of the flexidrive is achieved with the adaptor, which also contains one of the input shaft bearings. A stainless steel 26 tooth gear on the input shaft drives a 72 tooth stainless steel gear mounted between bearings on the second drive shaft. One end of this shaft projects beyond the bearing and supports another 26 tooth stainless steel gear. All of the above mentioned parts are mounted on a machined aluminum alloy housing, which in turn is pinned in position between two aluminum alloy side plates. A third side plate, external to the other two, positioned and supported on spacers, completes the boom container framework. Between these three side plates all the detail parts are positioned and secured,except those mentioned above.

The gear housing is so positioned that the cantilevered 26 tooth gear engages accurately with a similarly cantilevered 72 toothed stainless steel gear. This gear is mounted on a shaft whose two bearings are supported by the two adjacent side plates

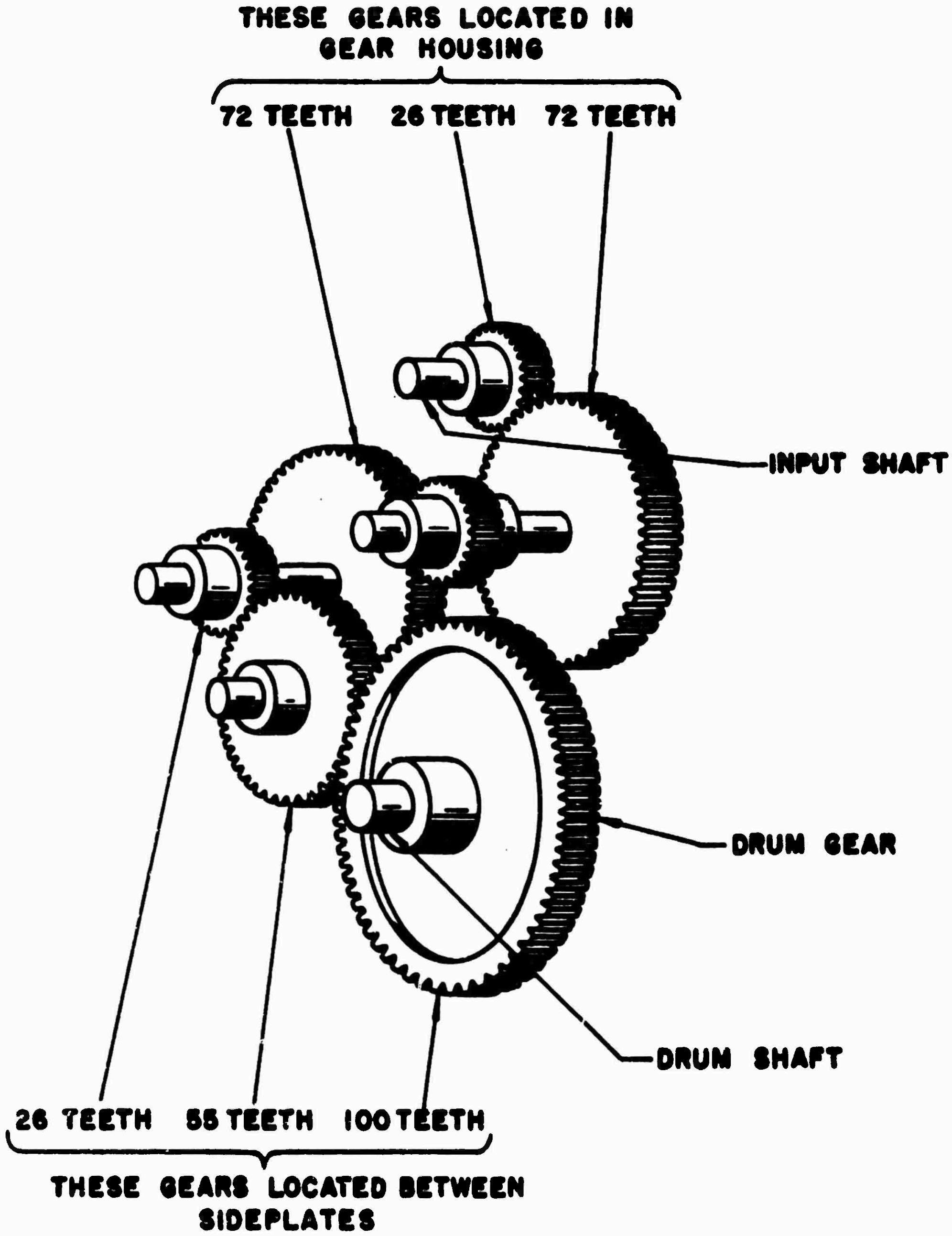


FIGURE 11 BOOM CONTAINER GEARING

enclosing the remainder of the drive gearing to the drum. Between these bearing and on the same shaft is mounted a 26 toothed stainless steel gear which drives the 100 toothed aluminum alloy drum gear via a 55 tooth stainless steel idler gear. This gearing gives an overall reduction ratio of 29.4:1.

The drum gear is keyed to the drum shaft and is mounted external to the two drum shaft bearings, one of which is located in the mid side plate. The other drum shaft bearing is located in the support shaft which forms part of the stop assembly. This support shaft is secured by screws to the adjacent side plate after final adjustment of the stop assembly. This aluminum alloy support shaft has four functions. Firstly, it supports one end of the storage drum via a bearing on the support shaft. Secondly, it locates two guide rods diametrically opposed on either side of the drum shaft, which stops the stop nut from rotating relative to the drum and yet allows it to move axially along the threaded drum shaft. Thirdly, it provides the limit stop to the nut, which on contacting the support shaft locks the drum, thus preventing further boom extension. Fourthly, the support shaft can be rotated after final assembly of the container via the screw driver slot provided on the side plate locating spigot, for final adjustment of limited extension length.

The aluminum alloy drum is connected to the drum shaft by screws which pass through one of the drum cheeks and into a flange on the shaft. The bronze stop nut guide rod ends opposite to those located in the support shaft are located in a housing which is supported on a bearing by the drum shaft.

The three STEMs forming the boom are fixed on the drum by three screws which locate in press nuts. On either side of the three STEMs are reinforcing leaves fixed by the same three screws.

Twelve aluminum alloy cassette rollers surround the storage drum and are supported in bearings in the side plates. These rollers are all located on the same pitch circle and are equally spaced, enclosing about 250° of arc. Over the remaining length of arc, continuous cassette support is given by the teflon guide block, which has the dual purpose of providing both cassette and 'roller' guidance. The guide block encloses two steel inserts which locate in the side plates, thereby positioning the block.

The two innermost STEMs on the drum are backwound; i.e., the outer surface of the STEM (defined in its natural tubular state) are innermost on the drum, whilst the outer STEM is forward wound (outer surface facing outward on the drum). The forward wound STEM exits from the cassette and passes through one roller guidance set, whilst the two backwound STEMs pass through the other set together. The roller guidance set for the forward wound element is comprised only of aluminum alloy rollers supported by bronze bushes in the side plates. The roller guidance for the backward wound elements comprises rollers in one half of the set and the Teflon guide block on the other. The roller guidance directs the flattened STEM through a 90° direction change, whereupon the STEMs enter the ploy guidance.

The ploy guidance comprises three separate parts, all of which are located and fixed in position between the side plates. Ploy guidance material is Molycote and chopped strand glass fibre filled epoxy resin.

An aluminum alloy insert cast into the inner of the three ploy guidance parts (called the male guidance part) is used to support that part and locate the end position of the other two. The male part is cantilever supported because the STEM tubes wrap themselves around each other just after the support point, thus completely enclosing the part. The two female parts locate on either side of the male. Aluminum alloy inserts in the female guidance serve to strengthen it and house heli-coil inserts for side plate and cover attachment. A fourth part of the guidance, cut from one of the female parts and originally intended as part of the boom clamp, was reincorportated as part of the fixed guidance by attaching it to the gear housing.

The aluminum alloy boom end plug is attached to the forward wound STEM by two screws and an end plug clamp. The clamp acts as a washer and fits between the screw heads and the STEM. The end plug houses a stainless steel lug in a spherical bearing for attachment of an attachment device (to be customer supplied). A spherical bearing is retained in the boom end plug by a retaining ring.

1. CENTRE SHAFT
2. PLANETARY GEAR
3. HANDLE TO MOTOR DRIVE CHANGE OVER KNOB
4. RETRACTION-EXTENSION CONTROL
5. HANDLE
6. MANUAL DRIVE SLIP CLUTCH
7. ELECTRIC MOTOR
8. BOOM DRIVE ENGAGEMENT-DISENGAGEMENT LEVER
9. PUSH BUTTON
10. LEVER SUPPORT PLATE
11. GUIDE SHAFTS

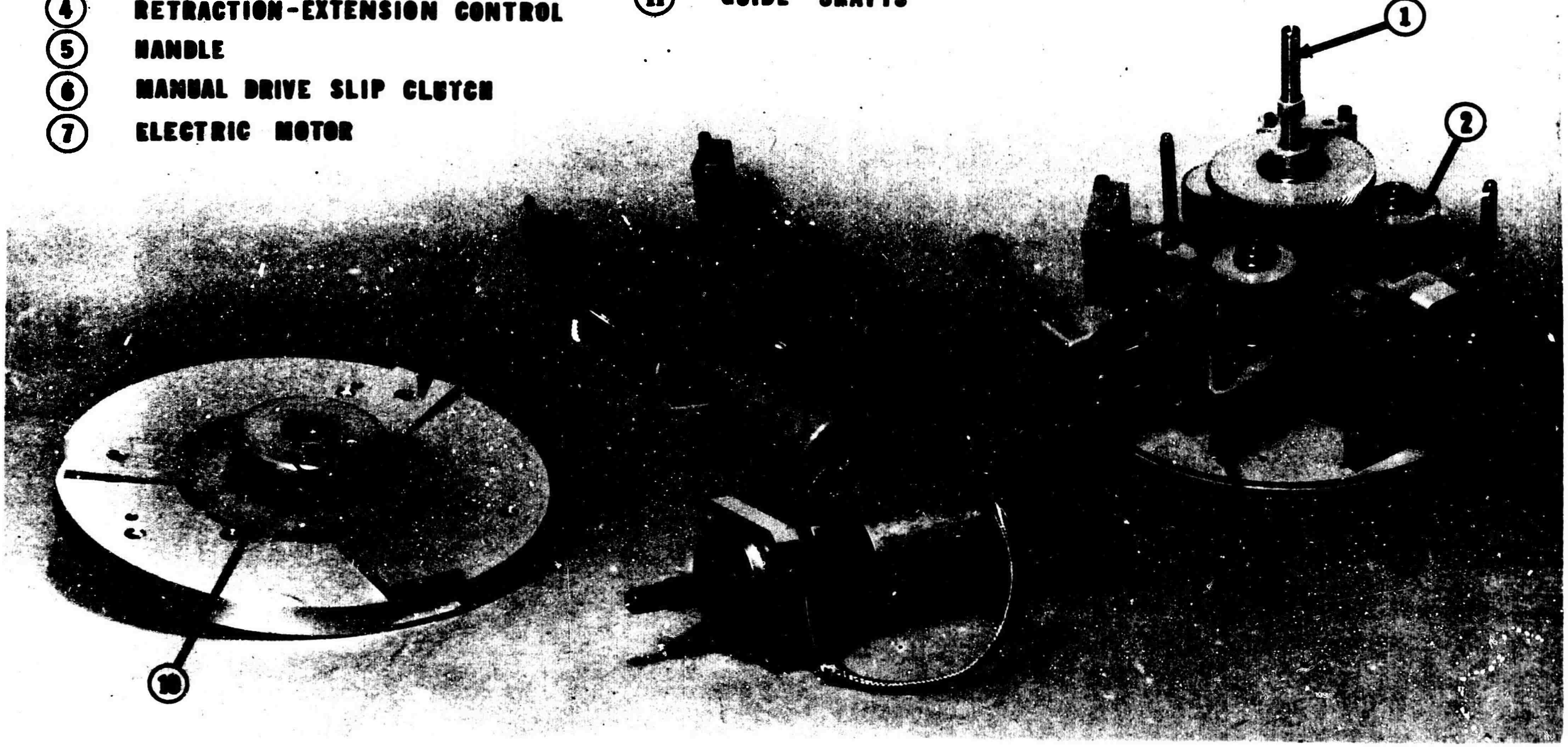


FIGURE 12 CONTROL BOX SUB-ASSEMBLIES

Two aluminum alloy pillars at the back face of the container provide the means of attaching the containers to the back frame. These pillars have four heli-coil inserts for the attachment screws.

Two covers enclose the container and are attached to it by screws at the ploy guidance, on the gear housing, at an anchor nut, and at a cover fixing block attached to one of the side plates.

B. CONTROL BOX

1. Framework and Major Gear Shafting

Three aluminum alloy circular plates (see Figure 12) separated axially by aluminum alloy supports to which they are attached and located by screws and dowel pins, form the basic structure of the control box. Centrally located and supported on bearings in the front and rear plates is the stainless steel centre shaft. Three shafts, called the flexidrive shafts, whose axes are equally spaced on a pitch circle concentric with the centre shaft lie between the end plate and the centre plate.

Bearings in the centre plate support one end of each of the three shafts whilst adaptors, fixed and located by screws and dowel pins in the end plate, hold the bearings which support the other ends. A broached square hole in that end of the flexidrive shafts passing through the adaptor mates with the flexidrive. The flexidrive is held in position by a cap nut which screws onto the adaptor.

2. Manual Drive Gear Path

The aluminum alloy handle is supported at either end by bearings on the handle shaft. The handle shaft is pinned to the aluminum alloy crank arm, which in turn is screwed and pinned to the stainless steel drive coupling. This coupling is keyed to the input handle shaft and secured by a central screw. This coupling is necessary as the handle assembly has to be removed to remove the control box cover. The input handle shaft is supported by bearings in two of the supports separating the front and central plates. The handle shaft axis is perpendicular to the centre shaft axis and towards the front of the control box. On the input handle shaft, between the two supports, is a 120 toothed stainless steel spur gear which drives the 36 toothed stainless steel gear of the hand crank slip-clutch. The slip-clutch is on a stainless steel shaft supported by bearings in two of the supports. On the same shaft is a 40 toothed stainless steel helical pinion which drives an 80 toothed aluminum alloy helical gear located but not fixed on the centre shaft.

The helical gear is fixed and co-axial with one half of a dog-tooth clutch set. They are free to rotate on bearings about the centre shaft but restricted from moving axially along it. The other half of the dog-tooth clutch is located on the centre clutch assembly, where it is fixed and co-axial with an 84 toothed stainless steel spur gear and one half of another dog-tooth clutch set. The centre clutch assembly is free to rotate on bearings about the centre shaft and also free to move axially along the shaft, although this movement is controlled by the motor manual drive change over mechanism. (Selection of manual drive moves the centre clutch assembly along the centre shaft until the first mentioned dog-tooth set engages). The 84 toothed gear is in permanent engagement, regardless of its axial movement, with three 60 toothed aluminum alloy planetary gears which are free to rotate about their respective flexidrive shafts but fixed axially. Each of these gears is attached to and co-axial with one half of a dog-tooth clutch set. The other half of the [illegible] is part of the clutch disengage mechanism and is fixed to rotate with the flexidrive shaft but free to move axially along it. The limits of this axial movement are set by the ball-detent arrangement between the clutch disengage assembly (see Figure 13) and the flexidrive shaft. Each limit corresponds to engagement or disengagement of the clutch set on the shaft and on engagement the shaft is locked with the 60 toothed gear and thus driven by the handle.

3. <u>Motor Drive Gear Path</u>

A 24 volt DC motor is attached to the rear of the back plate by four screws such that the splined motor shaft protrudes into the control box. On this shaft a 60 toothed stainless steel motor gear is located. It is positioned by spacers and secured by a nut and locking washer. The electrical connections are made from the motor to a terminal block also located on the rear face of the back plate. The motor gear mates with the 84 toothed stainless steel gear of the motor slip clutch, which is located and pinned on the centre shaft at the extreme rear end of the control box. Only on motor drive does the centre shaft transmit torque. Half of a dog-tooth clutch set is located and fixed on an integral and concentric flange on the centre shaft. This clutch locates with its other

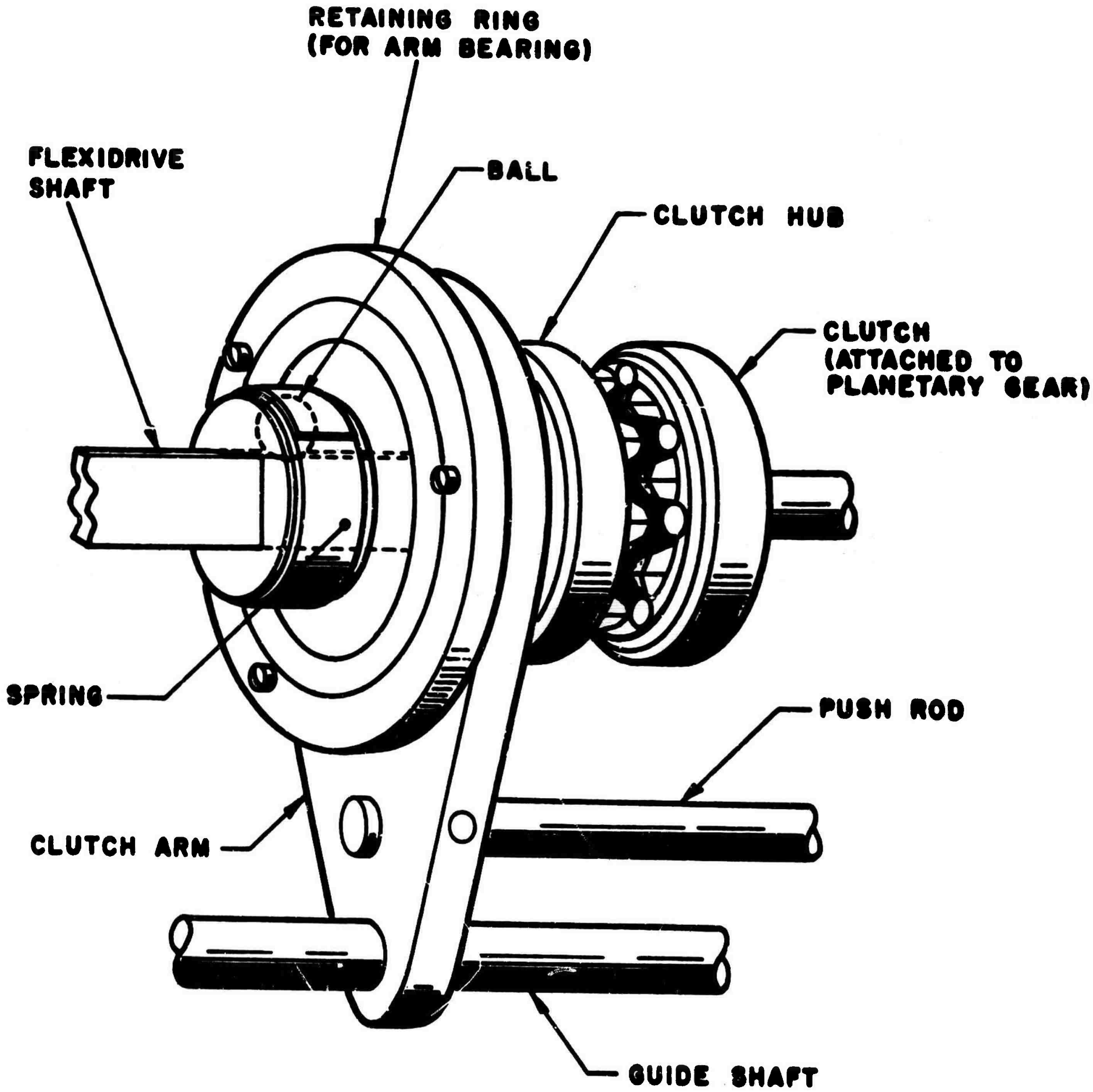


FIGURE 13 CLUTCH DISENGAGE MECHANISM

THIS MECHANISM CONNECTS OR DISCONNECTS THE DRIVE TO INDIVIDUAL BOOM CONTAINERS half on the center clutch assembly when the assembly is moved into engagement along the shaft by the motor manual drive change over mechanism. This action automatically disengages the center clutch assembly from the manual drive gearing through the helical gear. Drive from the 84 toothed gear of the center clutch assembly passes through the planetary gearing as described in the previous section.

4. Manual Controls

a. Boom Drive Engagement or Disengagement

The controls for engagement or disengagement of drive to the individual boom containers are located on the exterior face of the front plate. These consist of three aluminum alloy levers pivoted on an aluminum alloy lever support plate attached to the front plate. These levers are equally spaced and radially disposed, presenting a Y configuration.

When all three boom container drives are disconnected, the three levers lie in radial slots in the front plate, such that their outer surfaces are flush with the outer surface of the plate. A push button is located on the portion of the center shaft protruding from the front plate. A compression spring seated on the front plate and co-axial with the centre shaft keeps this push button away from contact with the inner ends of the three levers (i.e. at the junction of the Y configuration). The outer ends of each lever are connected to stainless steel push rods which pass through the control box parallel to the center shaft and locate in the clutch actuating arm of the clutch disengage mechanism. The actuating arm is attached to the rest of the mechanism by a bearing retained in the arm and on the clutch hub by retaining rings. Three stainless steel guide shafts located in and between the back and centre plates support the actuating arms thus preventing any binding of the aluminum alloy clutch hubs of the clutch disengage assemblies on the flexidrive shafts.

Manual depression of the push button on the front plate causes all three levers to rock about their pivots, their outer ends emerging from the slots in the front plate. Movement of the levers is transmitted through the push rods to the clutch actuating arms, which in turn brings the clutch sets on the flexidrive shafts into engagement. Manual depression of any lever reverses this operation and causes disengagement of the concerned boom container drive.

The limits of axial movement on the clutch disengage mechanism are set by the ball and detent arrangement between the clutch disengage assembly and the square portion of the flexidrive shaft. A stainless steel ball is retained in the clutch hub by a spring rivetted to the hub. This ball is free to move in a direction perpendicular to the axis of the flexidrive shaft in a hole in the hub. Two-part spherical detents in the square portion of the flexidrive shaft form seats for the ball in the clutch hub. The clutch disengage assembly has its position and total movement limited by the ball seating in the detents of the flexidrive shaft.

b. Handle to Motor Drive Change Over Mechanism

Observed from the outside of the control box, the control for changing from manual to motor drive appears as a round black plastic knob seated on a slot in a contoured block. A stainless steel rod called the control rod passes through the knob, and a retaining ring on the rod supports a washer which acts as the seat of a compression spring. The other end of the spring seats on a counterbored face in the knob. The entire spring is concealed by the knob and its action is to force the knob against the contoured block, called the control rod guide. The contour on the control rod guide is such that the spring action forces the knob to take one of three positions, corresponding to manual, motor or neutral drive.

The control rod passes through a slide fit hole in a stainless steel pivot mounted on an aluminum alloy pivot block. The pivot block and control rod guide are attached on opposite sides of one of the supports between the rear and centre plate. The end of the control rod screws into a stainless steel pivot located in the arm of the centre clutch assembly. Movement of the knob into the two extreme positions results in the control rod pivoting about the pivot block, thereby moving the centre clutch assembly into engagement with either the motor or handle drives via the clutches on the centre shaft or on the helical gear assembly. When the knob is in the motor drive position, the control rod depresses the actuator of a microswitch. This energizes the motor power input circuitry up to the retraction-extension switching.

In the neutral drive position, midway between the extreme positions, neither the handle nor motor drives are engaged and the microswitch is not actuated. Operation of any of the other controls on the control box will have no effect whatsoever when the change over mechanism is in this neutral position.

c. Retraction-Extension Control for Motor Drive

This control is located on the side of the control box and its mechanism is supported on another of the supports between the back and centre plates. The mechanism consists of two microswitches attached to an aluminum alloy switch support bracket, which in turn is attached to the plate separating support. A stainless steel shaft is supported in the bracket and fixed by retaining rings. An aluminum alloy switch arm rotates about this shaft and is in contact with a spring which forces the arm against the microswitch actuators. Two handle locating tubes are attached to the arm and the switch handle by two screws. The locating tubes fit in a [illegible] gate cut in the cover. The gate provides three positions for the handle and these correspond with motored boom extension, motored boom retraction, and neutral. The two extreme gate positions allow the arm to operate one or another of the microswitches under the action of the beryllium copper spring. In the middle or neutral position, the arm is restricted from operating either of the switches by the limited depth of the neutral gate in the cover.

5. Lubrication and Cover Fitting

All the gears in the control box with the exception of the helical gear set were lubricated with General Electric G.300 grease. This grease was used as it is supposedly spaceworthy. Its lack of good lubricating ability precluded its use on the helical gear set, where a molybdenum disulphide filled grease was used. The cover is an aluminum alloy tube which fits over the control box framework, seating on the rear plate and secured by screws to the front plate. To fit the cover it is necessary that the handle assembly, control rod, and control rod guide of the handle motor drive change over mechanism, and the handle switch and handle locating tubes be removed. When the cover is in place these parts are reassembled through slots in the cover.

6. Back Frame and Control Box Support

The basic structure of the back frame is made up of aluminum alloy tubes welded together to form an 'A' frame but with the legs of the 'A' joined and V-braced. A plate welded onto the apex of the 'A' frame forms the supporting and fixing surface for the saddle position boom container. A cover welded onto the plate effectively forms a saddle and prevents container-to-pressure-suit contact. Plates welded to brackets, which in turn are welded to the ends of the two 'A' frame legs, form the locating and fixing surfaces for the two shoulder located boom containers.

An aluminum alloy plate is welded to the frame and covers the area between the two cross members of the modified 'A' frame. This plate, foam padded, forms the interface with the operator's back. On the reverse side of the frame, a similar but smaller plate is welded to the frame. This plate extends beyond the 'A' frame to support the eccentrically positioned control box support, which in turn supports the control box.

The control box support is an aluminum alloy box section tube, seam welded. An external flange welded onto the tube at one end provides the means for attaching the control box, whereas a plate welded across the other end provides the means for attaching the support to the back frame. Both attachments are made by screws. Eight screws pass through clearance holes in the external flange and into threaded holes in the back plate of the control box. Four screws pass through clearance holes in the plate attached to the back frame and screw into press nuts located in the plate at the end of the control box support.

An aluminum alloy channel section joist runs the width of the eccentric plate welded to the back frame. It helps secure the plate to the tubular frame and strengthens the eccentric portion of the plate, thus preventing excessive flap of the control box.

Triangular shaped aluminum alloy sheet plates are welded into each corner of the back frame, formed by the joint of the sides of the 'A' frame and the cross member joining the two legs. Essentially, their purpose is to strengthen the frame, but they also serve as attachment points for the upper ends of the diagonal shoulder harness.

Similar eyebolts pass through either side of the tubular frame several inches from the apex of the 'A'. The hooks on the lower ends of the diagonal shoulder harness and the hooks on two thigh belts attach to these.

The remaining ends of the two thigh belts are located in slots in the saddle cover over the saddle position boom container.

7. Harness Arrangement

The diagonal shoulder harness is simply two domestic quality auto seat belts. The diagonal is not formed by the harness crossing, but by the belts folding back on themselves in an aluminum alloy buckle.

Each shoulder belt passes through two slots in the buckle, thus the buckle has a total of four slots. This arrangement permits easy adjustment of the buckle and retention of buckle position on harness release, and yet allows no change of buckle position relative to the harness during final harness tightening. Two quick release latches complete the diagonal shoulder harness.

The two thigh straps are simply auto seat belt webbing with one end of each strap located in double slots in the saddle cover of the back frame and the other ends attached to hooks.

VII TESTING

This section will cover the Design Approval Testing (D.A.T.) and some of the functional tests relating to the system as a whole, and not reported in the section on development.

Bending tests were carried out on the STEM booms fitted to a breadboard model of the boom container. These were unsatisfactory, the root strength being too low to meet the requirement of reacting a 45 ft. lb. bending moment at 4 feet extension. The booms were not completely representative of those to be fitted to the actual boom containers and theoretical considerations led to the belief that the booms would in practice meet the specified strength. Tests on the system's booms proved otherwise. An explanation for this is given in Appendix II. As stated in the appendix, in the interests of package size and weight the boom containers were not modified in order to increase their strength. The D.A.T. was modified to ensure that the booms were not subjected to failure loads by reducing boom length at which the design torque was to be applied to arm's length (2 ft. 9 in.) instead of 4 feet.

As mentioned in the development section the first attempt at the D.A.T. was dis-continued following breakdown of the worm gearing in the boom containers. This occurred before vertification of the system's ability to meet the modified design loads.

In its final form the system satisfactorily passed the modified D.A.T. although some interesting points arose. Firstly, manual operation of boom extension and retraction with the system unattached to a work surface, yet supported in our six degree of freedom simulator, caused some coinsiderable movement of the system about the longitudinal (head-feet) axis of the operator, i.e. in a plane at right angles to the plane of handle rotation. The degree of movement appeared to increase with increased handle torques, as for boom retraction. Secondly, although handle torques were quite low, the operator had a natural tendency to react the handle rotation reaction forces by grasping the control box with his free hand, thereby reacting forces through the shoulders and two arms, rather than through the harness arrangement. This appeared to reduce the operator's movement and fatigue. None of these effects occurred when the booms were motored in and out.

Thirdly, the operator had difficulty observing full retraction of the booms, particularly of the one between his legs. This was because the boom containers were mounted too far back for the operator to visually observe them without physical difficulty.

Fourthly, with the system attached to a flat work surface even a 2 ft. 9 in. boom extension was too great for the operator to be able to apply a 45 ft. lb. torque. With a 6 in. extension and a 15 in. arm on the torque wrench the operator was just able to apply such a torque, but only with great effort and only for a few seconds. It should be noted that the operator applied a pure torque. With a single armed torquing device, the tendency of the operator was to work in the normal single handed manner, thereby applying both a torque and a direct load to the work surface, which of course was reacted back through the attachment system. Due to the relative weakness of the system's booms in reacting direct loads,it was ensured that when the operator applied a torque using a single armed device he reacted the load applied by one hand on the handle of the device by exerting an opposite and presumably equal force on the device at the work axis, i.e. in the longitudinal axis of a bolt attached perpendicular to the work surface.

It is felt that with the added incumberance of a restrictive pressure suit the ability to apply pure torque will degenerate. This will result in direct loads being applied back through the attachment system. In addition methods will have to be devised for locking the torquing device to the nut or screw, etc., to prevent slip off with possible high system dynamic loading.

The above mentioned problem is a maximum when torques are applied in the plane of the work surface. The good tensile and compressive strength of the booms, proved to meet specification during the course of D.A.T., removes the problem when applying torques in a plane perpendicular to that of the work surface. However, results of tests in the Air Force's five degree of freedom simulator indicate that most work is done in the plane of the work surface. Thus, the designed and developed boom attachment system has an inherent weakness. This is a result of specification inadequancy with respect to direct loading in the plane of the work surface. Unfortunately, this spotlights another possible operational inadequancy of the developed system. This is the problem of dynamic loading during the process of attachment. These dynamic load effects are likely to be maximized by the built in nature of the boom containers.

All system phases of the D. A. T. were carried out in the six degree of freedom simulator mentioned earlier in this section. In order to assess some of the foregoing, it will be necessary to describe the de Havilland simulator and the degree of simulation present.

A requirement for the simulator was that it give good simulation of freedom in the plane of a work surface and of translation in a direction perpendicular to the plane of the work surface. This was obtained by a simple counter weight system. This consisted of a single wire rope supported on pulleys attached to a 40' high roof girder. The ropes hung down, one end attached to the backframe of the boom attachment system as shown in Figure 1, and the other to the counterweight box. Simulation restrictions in the important degrees were as follows:-

a) Torsion of the wire rope in the rotational plane of the work surface.

b) Pendular restoring force in the two translational degrees.

c) Pulley friction and counterweight inertia in the vertical direction.

Restriction (a) is negligible. Restriction(b) was unimportant as the test plan dictated no translation. Restriction (c) only effected initial wind out loads when attached to the work surface.

The method of attaching the back frame to the wire rope effected the simulation in the rotation plane perpendicular to the work surface.

The movement of the operator was least restricted in that plane in which most movement occurred during manual extension and retraction of the booms when unattached to the work surface.

CONCLUSION

The study of the best configuration for a three point attachment system resulted in the selection of one boom located at either shoulder and one between the legs. The boom containers were built in at the attachment frame so that the booms extended parallel to each other. It was considered that this would simplify the process of attachment.

As explained in section VII on testing, although the system was designed to withstand 45 ft. lb. torque at 4 ft. extension it was in fact only tested to a torque of 45 ft. lb. at 2 3/4 ft. extension. The boom containers were 1.6 lb. overweight.

De Havilland was not able to ascertain whether the system was capable of operation by a pressure suited astronant as no suits were available. Comments on this capability appear later in this section.

Within the limitations outlined above the system performed to specification. Testing indicated that it was possible to apply considerable torque to a work surface - almost to the limit of the operator's physical ability - without failure of the system and without appreciable deflection of the operator. The system will be very useful for evaluation purposes and for research into the forming of specifications for future attachment systems.

Having used the system, some general observations of the systems performance and design are in order. Opinions arising from these observations are those of the writer and result from a very limited test program:

Firstly, tests in the low gravity simulator indicate that motor drive for boom extension or retraction is preferrable to manual drive. Operation of the manual drive was fatiguing and caused appreciable unproductive movement of the system.

Secondly, the load-applying abilities of an operator appear to be greatly reduced when strapped into the system. This is not surprising, as using the developed system is analogous to working sitting in a chair, tied to the back rest with both feet off the floor. It is felt that a great improvement might be obtained if the operator's feet were the only part of his body attached to the system. In this way he would be able to react loads through his body in a relatively normal manner. This would be impossible for a ground evaluation model unless simulation was achieved under water, where the operator's body weight could be counter-balanced by buoyancy forces.

Thirdly, although it is desirable to keep control forces to a minimum, it is also necessary that these forces be sensed by the operator so that he knows when he is controlling. Due to the restrictions of a space suit, control movement should generally be kept low. The importance of these considerations is going to depend on pressure suit technology.

The controls of the developed Boom Attachment System require little movement, but some control forces are too low and others are too high. The requirements for a low mobility suit might be relatively high force gated controls, whereas, for high mobility suits could be relatively low force ungated position controls.

Fourthly, the specification did not require the booms to be capable of reacting direct loads in the plane of a work surface other than as implied by the 45 ft. lb. torque reaction ability. The minimum direct load applied at the tip of a boom extended 4 ft. sufficient to cause failure is approximately 5 lb. Such a figure indicates the systems inability to react appreciable direct loads in the plane of a work surface and the inability of individual booms to react dynamic loads such as might occur during the process of attachment. Unless an operator is using minimum reaction power tools he is likely to apply appreciable direct loads back through the system. This tendency is likely to increase with decreasing pressure suit mobility.

This section will conclude with a brief outline of the approach that could conceivably be taken by de Havilland in producing a second generation boom attachment system based on experience with the present system and advances in STEM technology that have taken place since de Havilland designed the boom containers of the present system.

One of the major advances in STEM technology in the last year has been the BI-STEM. (U. S. & Canadian Patents Pending.) It shows significant advances in deployment mechanism packaging density for a given boom strength. This is achieved by employing two front-to-front nested underlapped or open STEM elements, instead of a single overlapped element. The boom strength can thereby be doubled, whilst the weight increases by 30% and the deployment mechanism volume is reduced by as much as 50%. For a boom container size of approximately 5 in. x 5 in. x 10 in. the application of the BI-STEM principle could result in a fourfold increase in system reaction capability. The boom deployed from such a container would be 1.38 in. diameter and 0.007 in. thick and the total weight, complete with drive motor would be approximately 7.5 lb.

The boom container to frame or other interface connection would be achieved using a sprung lockable ball joint. The containers would be located as for the present system or conceivably with the 'between the legs' container relocated at the hip. Each boom container would have its own drive motor, power being supplied from a battery pack of suitable capacity located in the back frame or some related piece of equipment. Deployment control would be achieved by an ungated, low force, self cancelling control located on the astronaut's belly.

The frame would include leg supports with attachment only at the astronaut's feet. Possibly the leg frame would be articulated to avoid interference with unusual structures and to enable any stance to be adopted. If articulated the leg joints would be lockable.

Such a system would be capable of reacting direct loads of the order of 75 lb. and dynamic loads on individual booms of the order of 25 lb. all at 4 ft. extension. The torque reaction capability would be increased to at least 90 ft. lb.

APPENDIX I

MATERIALS REPORT

This Appendix has two main sections:

Section I Evaluation of the designed boom attachment system materials.

Section II Recommendations regarding future space-worthy models.

SECTION I

A. Introduction

A full scale survey of all materials used in the boom attachment system was carried out. Any errors in materials, specifications, and finishes found by the Materials and Processes Group during the design stages were of a minor nature. Drawings were corrected immediately and the parts were made to drawing or a materials substitution issued correcting the situation so that the finished parts were acceptable. These errors are not listed in this report.

B. Boom Containers

These units contain the boom, storage drum, gear system to drive the drum, and a guidance for the deploying boom.

(1) Boom Guidance

This consists of a roller system for the flattened STEM and a ploy guidance system.

Many problems were experienced in the development of this part of the system due to high frictional loads being generated, mostly in the ploy guidance.. These were partly design and partly materials problems and were covered elsewhere in the report.

It is significant to note that solid film lubrication arrangements must be made with a view to ensuring replenishment of the film from a reservoir - an applied film that is not renewed will not give reliable long term service. Thus, moulding the lubricant into the guidance is good practice.

Teflon spraying of the ploy guidance was attempted, but no satisfactory results were obtained. However, Teflon application for effective utilization of its properties is not simple; and a more sophisticated development approach to this method may be capable of producing better results.

To obtain really low loading once optimum guidance design has been determined, it may be necessary to look to a dual system of solid film lubrication aided by injection of fluid lubricants from wicks installed in areas of high loading—along the flutes for instance.

It was found necessary to lubricate the roller guidance bushes to ensure rotation. Both silicone and mineral oils were used for this purpose, the better lubricity being provided by standard SAE 10 mineral oil.

(2) Gears

Initial loading on the gear train was too high due to the large mechanical reduction (36.4: 1) required. Overloading of the worm gearing and unit structure occurred. This problem was handled from a mechanical rather than a materials viewpoint.

A more rigid structure was designed and spur gearing substituted for worm gearing.

Future work would include reduction of wear and friction, through materials studies.

(3) Drum Shaft

This 0.25 in. diameter shaft was manufactured from 303 series stainless steel. Stress analysis (see Appendix II) showed the shaft to be the critical failure member. Yield strength of such members can be greatly improved by use of heat treated 400 series stainless steel.

(4) Boom Clamp

This was not considered, as it was withdrawn from the finished unit.

C. <u>Stainless Steel Booms</u>

These are standard boom elements formed by the usual SPAR process. The application is routine and the strength values achieved in the formed boom compatible with normal practice. Thus, the boom buckling problems are correctly attributed to excessive loading, and no reason was seen for any metallurgical investigation other than the routine tensile testing.

No problems with edge rippling or relaxation of the formed tube were found on cycling the boom in the revised design, confirming the suitability of the material condition for the application.

No surface finish of edge treatment of the boom was carried out.

D. Control Box.

(1) Gear Shaft

The output torque from the control box is transmitted via a gear to a 303 series stainless steel shaft called the flexidrive shaft, which has a 0.126 in. square hole broached longitudinally in its end to receive the square end of the flexidrive.

(2) Manual Controls

The handle to motor drive changeover mechanism control knob is manufactured from nylon 101 and is adequate for ground use only.

(3) Slip Clutches

These bought out parts are satisfactory for ground operation.

E. Flexible Drive

These performed satisfactorily under ground conditions where no lubrication problems are to be expected.

F. Harness

This was not studied.

G. Back Frame

Provided suitable inspection of the welded joints is carried out, the frame is suitable for present and future applications.

SECTION II

A. Introduction

Some thought was devoted to the possible areas of development which would be required if the unit was to be judged spaceworthy. Solutions to all of the problems envisaged would not be found without more advanced work being done.

An outline of the problems follows, with recommendations regarding the course of action to be taken.

B. Boom Containers

(1) Boom Guidance

Further work could be done to reduce the frictional loading. Various solid lubricant fillers could be moulded into the guidance once an optimum design has been chosen. Also, wick lubrication would be investigated as outlined in Section 1.

Lubrication of the rollers would also have to be developed. Neither mineral oil (because of its volatility) nor silicone oil (because of its creep properties and poor lubricity) is desirable.

Development would center around the low vapour pressure diester lubricants, moulded bushings containing solid lubricants, or a reservoir of replenishing lubricant, either solid or liquid.

(2) Gears

More consideration would have to be given to the compatibility of the gears; in general aluminum alloy gears pose reliability problems in applications such as this. It is not possible to harden or protect the teeth and the result is often wear or deformation of the teeth due to slight misalignment, cocking, etc, causing design limits to be exceeded. Combinations such as Be Cu on stainless steel or Fiberfil on Be Cu should be considered.

(3) Clamp

If this is to be included in later models, the material to be used will probably present no problem.

(4) Drum Shaft

Future models should consider use of a hardenable stainless steel -410 or 416. This would enable the shaft to be heat treated to RC30-34, which gives it excellent wear and deformation resistance.

C. Stainless Steel Booms

Stainless steel appears to be quite highly reflective; in actual fact it is not - it absorbs over 50% of the solar radiation incident upon it. It is also a poor thermal conductor. Thus we have two problems - the booms will tend to bend due to thermal assymetry and they will get hot.

Bending over a 10 ft. length should not be serious but the boom temperature may rise to a point where contact with the boom could cause damage to a space suit, aluminised Mylar or Nylon cord, etc.

Therefore, the absorbtivity of the boom should be reduced for space applications, by the application of silver electroplate or vapour deposited aluminum.

No particular care is taken over the edge or the raw stainless steel stock ; although it is ordered as #5 edge., which is rounded, it is sometimes rough. A rough edge could be a hazard and so an extra finishing operation could be provided to stock for space applications to ensure a smooth edge.

D. Control Box

Failure of any component in this box could cause failure of a boom to deploy or retract. This automatically fails the whole unit, as there is not a safety device in the design giving alternative drive to a boom with a defective train. Thus great care would have to be taken to build into the system the highest reliability.

(1) Control Rods

In spite of increase in diameter of these rods, a stainless steel such as 416 should be used to give a wider margin of safety.

(2) Gear Shaft

The same argument applies here as in (1).

(3) Handles

Nylon is not a generally acceptable spaceworthy material; in addition, if it were exposed to solar radiation it is probable it would heat to beyond its maximum service temperature.

Thus, plated Delrin or Teflon should be used for space applications.

(4) Slip Clutches

Complete analysis of the springs, components, lubrication and modus operandi of these clutches under space environmental conditions would be necessary, including selection of materials, freedom from cold welding, galling, etc.

E. **Flexible Drive**

Ordinary steels are used in the construction of this drive system. They are in general not used for space applications.
The main problem lies, however, in assuring adequate lubrication of the drive in space.

Probably the best approach to the problem would be a sealed or semi-sealed system lubricated with a non-silicone, low vapour pressure grease, loaded with a solid lubricant such as MoS_2 of MS_2.

Development work would be required.

F. **Harness**

Again we require a space approved material, this time in a fabric form, and again its absorbtivity must be low.

Possibly a woven Mylar, coated with vapour deposited aluminum, would suffice, though a Nylon would be stronger.

G. **Back Frame**

This appears to be satisfactory provided it, too, is protected from the absorbtion of too much thermal energy.

APPENDIX II

STRESS ANALYSIS RESULTS

1 **Bending of Booms Due to Applied Torque**

The critical boom in bending is the boom between the operator's legs. For this boom,

$$M = R \times l$$

$$= 13.4 \times 48$$

$$= 644 \text{ lb. in.}$$

From Equation (8)

$$M_{crit} = \frac{0.96\, EI_{\parallel}\, nt}{d\,(h + 2R_1 \sin 0)}$$

$$= 666 \text{ lb. in.}$$

$\therefore$ Reserve Factor $=$ 1.035

When bending tests were performed on a 48 inch long specimen of boom, it failed at a lower bending moment than that predicted by theory. When the specimen was examined the instability failure was found to be in the innermost element of the three. Also, the buckled area was not only in the region of the cross section curved to radius R_2 (as predicated by theory), but in addition had spread into the region curved to R_1.

This unpredicted failure is obviously a function of the unit length; and, as the unit length of 5 inches was outside the current state-of-the-art, no previous experience of this type of failure existed.

When the reduced bending strength was discovered by test, it was decided to pursue the development of the short unit at the expense of the strength requirements. This was done to keep package size and weight to a minimum.

II. Booms Under End Load

A compression test was performed on a 48 inch long specimen of boom with no resulting damage. Thus, the boom is considered able to withstand end loads up to 100 pounds without any reduction in subsequent performance.

III. Stress Analysis of the Gears

To assess the integrity of the gears, the torques and associated tooth forces throughout the system's gear trains were required to be known. For an approximate (but pessimistic) value of torques and tooth loads through the gear train, efficiencies of 95% for each gear interface and 60% for each flexible drive shaft were assumed to act at the pitch circle radii.

The critical mode of the gear train was found to be tooth bending, and to ensure integrity a minimum reserve factor of 2.0 on the 0.2% proof stress was considered the minimum requirement. On this basis, all of the gears were found satisfactory, the minimum reserve factor being 2.83 on the 26 toothed stainless steel gear on the second pass from the drum gear of the boom container.

IV. Stress Analysis of the Shafts

The critical mode of the only shaft stressed, the drum shaft of the boom container, was found to be pin bearing at the drum gear attachment point. A reserve factor of 2.6 at this point renders this the critical failure mode of the entire system, excepting the booms. All other shafts were more lightly loaded and passed by comparison.

V. Stress Analysis of the Boom End Plug Assembly

The loads applied to the plug were:

(a) 100 pounds along the axis of the boom.

(b) 13.4 pounds transverse to the axis of the boom.

The most critical component was the adhesive pad dispenser attachment lug for loading case (b), the reserve factor being 1.16 on 0.2% proof stress.

VI. The Support Frame

The support frame was designed to commercial standards with a minimum allowable reserve factor of 6.0 on the most critical component.

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13. ABSTRACT The low gravity field environment experienced in space presents problems to astronauts attempting to perform work outside their spacecraft. This problem can be defined in part as the difficulty to maneuver and the difficulty to keep station relative to a surface on which work is being carried out. This is mainly due to the greatly reduced restraining forces available to a space worker, particularly the lack of friction force as a result of body weight reaction. To become effective in space, man must develop means of controlling reactions to work loads.

Means have been studied to overcome this problem by restraint devices such as handholds, belts, and harnesses. Another means, and that which this report describes, is attachment to a work surface by rigid booms. A rigid boom attachment system offers a worker the advantage of being able to neglect the effect of moderate work loads.

Design and development of an evaluation model of such a system using unfurlable booms - an application of de Havilland's patented STEM (Storable Tubular Extendible Member) principle was carried out by de Havilland along guide lines laid by the United States Air Force and within the restrictions of a limited budget. The developed system should prove of great use in narrowing the guide lines to an optimum system.

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Astronaut Attachment Astronaut Constraint Extra-Vehicular Activity Positioning in Space Space Maintenance Unfurlable Structures						

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