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CRANFIELD INSTITUTE OF TECHNOLOGY

DEPARTMENT OF AIRCRAFT DESIGN

M.Sc. THESIS

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DESIGN PROJECT 1974/75 A 74 AIRCRAFT

ELEVATOR DESIGN

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SUMMARY

Part 1 of this thesis contains the symmetric loading calculations and evolution of the design of elevator for the A 74 Reduced Take off and landing aircraft. Part 2 contains the analysis of final design of elevators. The aircraft is designed to the specifications of DES 7400 and satisfies the British Civil Airworthiness Requirement.

A 74 project aircraft has longitudinal control from the tail plane and elevator combination. Tail plane is used for trimming and elevators for pitch controls. All control surfaces are hydraulic power operated. The tailplane uses an inverted supercritical section.

The elevator is a four piece arrangement with duplicated hydraulic actuator system. It is of round nose type and the elevator hinge line is at 0.6 C. perpendicular to the aircraft centre line. As elevators are fully power operated mass balancing is not provided.

Section 5 in part 1 gives the schemes considered in the design of elevator leading to the final choice of each major part. Section 6 gives the description of final design. Elevator is made of conventional built up construction using Aluminium alloy L 72 sheets and L 65 forgings. Part 2 gives the detailed stress analysis of the final design in accordance with normal aircraft design practice.

ACKNOWLEDGEMENT

The author would like to thank all members of staff of the Department of Aircraft Design, particularly the project supervisors for their help and encouragement throughout the preparation of this design.

CONTENTS

PART I - DESIGN SYNTHESIS AND LOADING

<u>SECTIONS</u>	<u>Page No.</u>
1.0 Introduction	1
1.1 Notations	2
1.2 List of drawings	4
1.3 References	5
2.0 Design Philosophy	6
2.1 Introduction	6
2.2 Survey of elevators in some existing aircrafts	6
2.3 Discussion	8
3.0 Elevator Layout	
3.1 Introduction	9
3.2 Two-piece elevator scheme	10
3.3 Three hinge elevator scheme	10
3.4 Four-piece elevator	11
4.0 Review of Construction Technique	
4.1 Introduction	13
4.2 Construction methods	13
4.3 Selection of material	14
5.0 Design Schemes	
5.1 Introduction	15
5.2 Schemes for support - structure and actuator	16
5.3 Scheme 1	16

SECTIONSPage no.

5.4	Scheme 2	18
5.5	Scheme 3	22
5.6	Schemes for elevator structure	24
5.7	Elevator spar	24
5.8	Elevator trailing edge	26
5.9	Elevator tip fairing	27
6.0	Description of Final Design	
6.1	Elevator layout geometry	29
6.2	Actuators	29
6.3	Actuator hinge bracket	29
6.4	Nose cell skins	30
6.5	Aft cell skins	30
6.6	Non-hinge ribs	31
6.7	Hinge ribs	31
6.8	Spar	31
6.9	Trailing edge position	32
6.10	False spar on trailing edge	32
6.11	Out board elevator tip fairing	32

LIST OF APPENDICES

Appendix 1	Symmetric loading calculations	33
Appendix 2	Elevator design cases, Bending Moment and Torque distribution	61
Appendix 3	Shear flow distribution	76
Appendix 4	Preliminary Design of four-piece elevator	79
Appendix 5	Optimisation of Hinge positions	83
Appendix 6	Elevator torsional stiffness requirement	96

PART II - ANALYSIS

SECTIONS

Page no.

1	Shear force, bending moment and Torque distribution on elevator	104
2	Shear Flow distribution in elevator	112
3	Aft cell skins and stiffeners	118
4	Nose cell skin	125
5	Non-hinge rib	130
6	Hinge rib	145
7	Spar	157
8	Hinge brackets	162
9	Trailing edge brackets	171
10	Weight Estimate	176

PART I

DESIGN SYNTHESIS AND LOADING

SECTION 1

INTRODUCTION

SECTION 1INTRODUCTION.A-74 PROJECT AIRCRAFT.

A-74 project aircraft is short haul civil transport of 150 passenger capacity and capable of operating from 1220 m long runways. The design considerations and detailed specifications are given in reference 1 - DES 7400. Salient features of design and major considerations are recounted here for completeness of this report and to follow up the evolution of the design of elevators.

In order to get lowest possible ground noise foot print, the engines are located on raised pylons between the wing and tail plane, giving maximum noise shielding by wing and tail plane surfaces. RTOL capability dictated the need for high lift devices consisting of triple slotted trailing edge flaps and leading edge slats on over 70% of wing span. Maximum cruise speed of over 0.8 M and low fuel consumption at reduced speed are obtained through a high aspect ratio unswept wing with super critical sections. The fuselage is of circular cross section and it is possible to stretch the fuselage to accomodate upto 180 passengers by providing additional sections forward and aft of centre section. Low wing design helps in keeping the floor level of fuselage low for easy loading and unloading and in reducing the under carriage length. The aircraft has four high bypass ratio Rolls Royce RB 410 type engines which reduce the fatigue problems of tail planes and controls which are in the wake of engine exhaust.

The tail plane also has supercritical section which is inverted as normal flight balance loads are downward. The large high lift devices and rear mounted engines necessitate a rather large tail plane and elevator surface for balancing. Tailplane is used for normal trimming and elevators for longitudinal control.

1.1 NOTATION

The following symbols are used in this report.

L_T	Tail load	N
$L_{T\alpha}$	Tail load due to incidence	N
$L_{T\eta}$	Tail load due to elevator deflection	N
L_{T_1}	Incremental tail load in lg to ng manoeuvre	N
L_{T_2}	Incremental tail load in ng to lg in manoeuvre	N
$L_{T_1\eta}$	Elevator component of L_{T_1}	N
$L_{T_1\alpha}$	Incidence component of L_{T_1}	N
$L_{T_2\eta}$	Elevator component of L_{T_2}	N
$L_{T_2\alpha}$	Incidence component of L_{T_2}	N
L_{T_1}	Balancing tail load in steady flight	N
L_e	Elevator load	N
$L_{e\alpha}$	Incidence component of elevator load	N
$L_{e\eta}$	Elevator load due to deflection	N
T	Net engine thrust	N
Z_T	Height of thrust line above CG	m

l'_T	Distance of L_T from CG	m
$\bar{c}$	Standard mean chord of wing	m
M_0	Pitching moment at zero lift	Nm
C_{m_0}	Pitching moment coefficient at zero lift	
AUM	Maximum All up mass 58,900	KGm
V	Velocity	m/sec
$a_1 a_2 a_1 a_2$	Lift curve slope	per rad.
R	Damping coefficient for short period motion	
J	Natural damped frequency of short period motion	
S	Wing area	m^2
S_T	Tail plane area	m^2
μ	Aircraft relative density	
$\bar{V}$	Tail volume coefficient	
K_B	Radius of gyration of aircraft in pitch	
ϵ	Down wash angle	
η	Elevator deflection	
θ	Pitch angle	
C_{EL}	Elevator chord aft of hinge line	m

1.2 LIST OF DRAWINGS

The following is the list of drawings included in this report.

A 74 - E1	Elevator Geometry
A 74 - E2	General Arrangement of Elevators
A 74 - E3	Elevator Details
A 74 - E4	Elevator Spar
A 74 - E5	Non-hinge rib at station 1880
A 74 - E6	Hinge rib at station 1420
A 74 - E7	Actuator and hinge bracket arrangement
A 74 - E8	Details of bracket

1.3 REFERENCES

- 1 A 74 - Project Aircraft Design specifications
DES 7400 by Prof. Howe and Mr. Ward.
- 2 Cranfield Institute of Technology -
Department of Aircraft Design -
Hand book of detail design.
- 3 Cranfield Institute of Technology -
Department of Aircraft Design -
Stressing Data Sheets.
- 4 Bruhn E.F. Analysis and design of airplane
structures.
- 5 Roark. Formulas for stress and strain.
- 6 Hand book of Aeronautics. Volume 1.
Structural principles and data.
- 7 Aircraft Design department supplementary
notes. DES 8, DES 13 and DES 545.
- 8 Aircraft Design department supplementary
notes. DES 730
- 9 Botman M. NLL report S 446.

SECTION 2

DESIGN PHILOSOPHY

DESIGN PHILOSOPHY.

2.1 INTRODUCTION.

In the design of modern aircraft it is very necessary to keep the initial manufacture and developing cost low and the operating cost due to maintenance absolutely minimum to compete with a shrinking world airline market brought about by inflatory economic conditions.

Though the elevator constitutes only a small percentage of the total structural weight every component is designed to keep the weight minimum.

All the detail components are designed to use the same item wherever possible to reduce the initial tooling and development cost. Also the use of new structural materials like fibre reinforced plastics are limited with a view of keeping the development costs low.

In a control surface the structural integrity is of paramount importance and it is considered necessary to have 'Fail Safe Principles' of design incorporated in the elevator support structure and actuator system.

2.2 SURVEY OF ELEVATOR CONTROLS IN SOME EXISTING AIRCRAFTS.

Design features and general arrangement of elevators of few recent long and medium haul civil transport aircrafts studied is presented below to give some idea about the recent trend in elevator design.

(1) A 300 B Airbus.

A 300 B employs fully powered flying controls operated by hydraulic power with no reversion to manual control. Hydraulic power supplies are triplexed to 3 jacks at each control surface. Jack sizes are

chosen so that full control over flight envelope is available with 2 jacks.

Each elevator is a single spar skin rib assembly without stringers, the material being copper based Aluminium Alloy. Trailing edge is a honey comb sand-wich construction. Each elevator is supported by seven self lubricating Teflon bearings, multiple support ensuring fail safe arrangement.

Pitch control is by elevators and trim obtained by moving the tail plane and effectiveness of trim a low speed increased by increasing tail-plane camber through a deflection of elevator neutral position. A duplex centrifugal feel system and an auto-pilot units are provided in control circuit.

(2) Lockheed Tristar.

Flight controls are irreversible serve-actuator powered. Four independent hydraulic systems assure that no change in flying qualities will result from a system failure. Safe return and landing is possible with only one system operation.

Pitch control is provided by all moving horizontal stabiliser with a mechanically linked elevator powered by four independent hydraulic systems. Because it is more effective aerodynamic surface than conventional fixed stabiliser/elevator combination the flying stabiliser ensures positive control upto and beyond diving speed.

(3) BAC - 111

BAC - 111 has variable incidence tail plane and two piece elevator. Elevator is a three hinge elevator and trim tabs are operated by electric actuator. Hydraulic system has duplicated actuators and manual override.

(4) VC - 10

The major design criteria in the design of the elevator was the high velocity exhaust jet from the engines contributing to acoustic fatigue. So the elevator structure is made of a simple rectangular box of honey-comb panels

and a solid web of honey comb at trailing edge.

The elevators are four piece elevators with fully powered hydraulic and has no manual over ride. As the elevators are fully powered, the elevators are not mass balanced.

(5) F 28 Fellowship.

Elevator has irreversible double hydraulic actuator hinged to a double horn fitting on the forward auxillary spar. The torsion box is formed by upper and lower panels in honey-comb sandwich structure and trailing edge solid honey comb structure. Each elevator half section is mounted on three hinges perpendicular to spar.

2.3 DISCUSSION.

From the above survey it is clear that there is a preference for fully powered control systems with no manual over ride. Also there is no necessity for mass balancing with fully powered control systems. Most aircrafts use triplex or quadraplexed hydraulic actuator systems.

Honey-comb panel construction is favoured for rear mounted high velocity jet engines. Otherwise spar skin rib assembly with Alluminium Alloy is used. Though the A 74 project is a rear engined aircraft the engines are high by-pass ratio low velocity jet engines.

ELEVATOR

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SECTION 3

ELEVATOR LAYOUT

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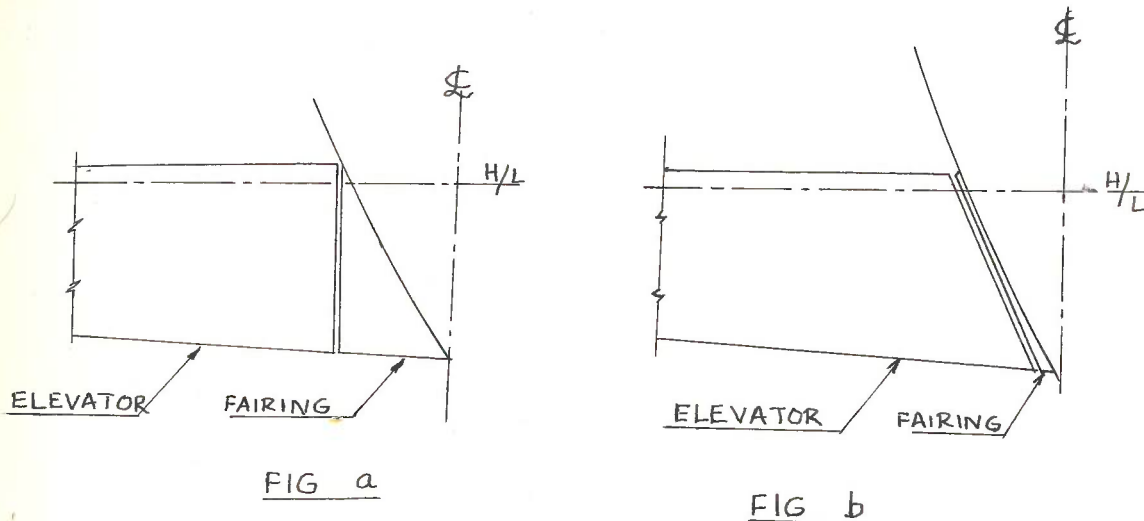
SECTION 3.

ELEVATOR LAYOUT.

3.1 Introduction.

The tail plane of A 74 aircraft has an Aspect ratio 6 and a full length elevator with hinge line at 0.6 chord. There is no sweep back and the elevator hinge line is perpendicular to the centre line of aircraft.

To match the elevator inboard edge to the aft fuselage it is necessary to have a fairing as shown in sketches below.



The layout shown in figure a has a large fairing attached to the fuselage tail cone. As A 74 aircraft has a variable incidence tailplane and as the tailplane is used for trimming this large fairing will produce excessive drag. Hence

layout shown in figure b is preferred for elevator inboard edge.

3.2 Two-piece elevator scheme.

The first possibility is to have one piece elevator with two hinges on either side. But the selection of a large single piece elevator will need large cross sectional area in order to satisfy allowable deflection criterion and hence not a suitable layout.

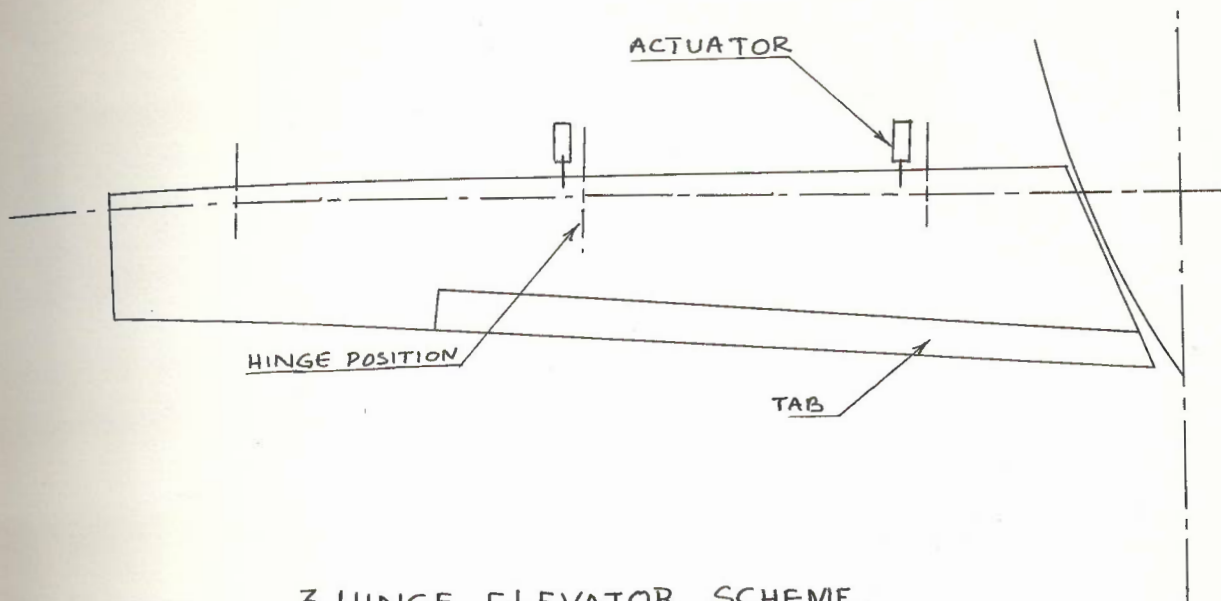
3.3 Three hinge elevator scheme.

The next alternative is to have a multi-hinge elevator. The use of multiple hinges with at least three positions for actuators is effective in providing fail-safe consideration.

The control surfaces have considerable stiffness in the chordwise direction compared to the normal direction. Hence when the elevator is fully deflected this elevator stiffness will introduce additional load on the tail plane in case of multi-hinged design. The use of multi-hinge elevator for a large span elevator is to be considered in the context of these additional loads on the tail plane.

Also the analysis of elevator becomes more difficult if the flexibility of hinge bracket and effect of tail plane bending are to be taken into account. In some cases the hinge bracket flexibility aggravates the overall loading. It will be necessary to confirm the analysis with extensive testing programme.

Three hinge elevator with duplicated actuator on both sides is a suitable alternative to having four piece elevator scheme. This will again need to take to account of the effect of failure of one actuator, because of jamming and hydraulic system should be designed to provide for over riding.

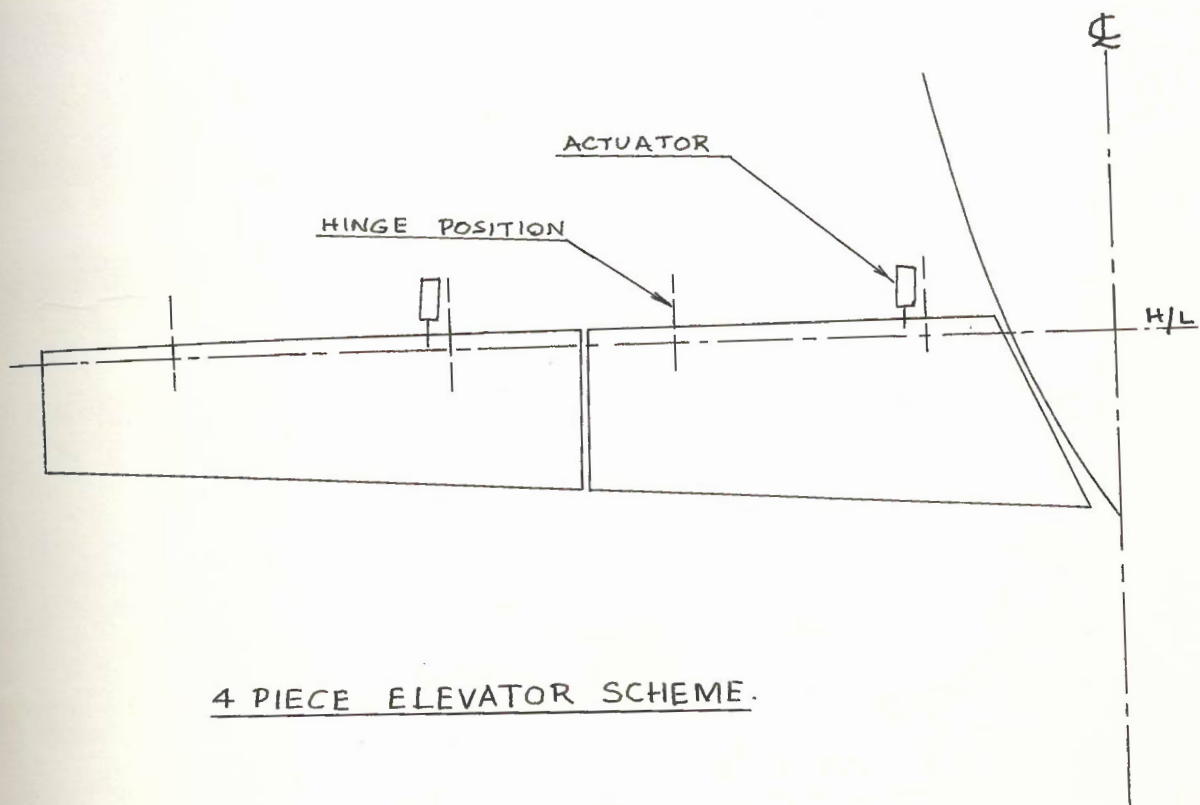


3 HINGE ELEVATOR SCHEME.

3.4 Four piece elevator scheme.

The choice of two elevators per side has the advantage of increasing the fail-safe characteristics. Two possible layout investigated are: Division of elevator into two pieces (a) of equal area (b) of equal span. Division into two equal areas is preferred as this gives a better distribution of loads between two pieces.

In Appendix C the optimisation of hinge positions are carried out rigourously for equalisation of moments at the hinge supports and the mid-point between the supports. Then the hinge positions are optimised for equal deflections of the tips and midspan. The positions as given by optimising the deflections are chosen for design as this is a better criteria for preventing the control surfaces getting jammed under loading.



4 PIECE ELEVATOR SCHEME.

SECTION 4

REVIEW OF CONSTRUCTION TECHNIQUE

SECTION 4REVIEW OF CONSTRUCTION TECHNIQUES.4.1 INTRODUCTION.

The choice of construction technique must not be taken in isolation for the elevator only, but it should be part of an integrated scheme adopted for the whole aircraft structure. In addition to giving an efficient structure the effect of fatigue and deflection of control surfaces is also an important consideration.

Using a structure made of honey comb panels is justified when fatigue caused by engine noise or the jet efflux from engines warrant extra cost and complexity involved. In A-74 project aircraft, no serious acoustic fatigue problems were present on account of the high bypass ratio engines used. Therefore conventional built up construction is adopted.

4.2 Some of the construction methods investigated are given below.

(a) Bonding: Redux bonding of the structure has been adopted for many aircrafts especially for the skin-stringer panels. The advantages are elimination of stress concentrations due to rivets and consequent improvement of fatigue characteristics of panel. However this method is not adopted for elevator mainly because other parts of the aircraft do not use bonding and the additional manufacturing costs for only elevators were not justified.

(b) Rivetting: Tail plane and elevators are of supercritical section and it is imperative to provide a clean aerodynamic surface. For this reason it is necessary to provide counter sunk rivets for attachment of skin. On primary structure counter sinking is done only if gauges are 1.2 mm or more.

Instead it is decided to use spin dimpling of skin and light rib flanges where they are to be rivetted. The spar flange is counter sunk to suit the rivet and nose skin panels are again spin dimpled to give a smoother top surface. The same method is used for attaching skin to hinge ribs which have 1.6 mm flange thickness.

It is proposed to use Avdel rivets in the trailing edge portion where there is no access due to the thin trailing edge section. Avdel rivets are also used for attaching the nose cell skin ribs. All joints are wet rivetted, using red sealant PL 1005. This provides certain amount of bonding effect and prevents corrosion damage.

4.3 SELECTION OF MATERIAL.

Aluminium alloy specification L 72 is used for elevator. The stress levels in the skin panels determined the use of aluminium alloy skins. As the elevator is in jet efflux of engines and because of the low fatigue life of L 73 material, L 72 material is preferred for skins, ribs and spar,

SECTION 5

DESIGN SCHEMES

SECTION 5

DESIGN SCHEMES.

5.1 INTRODUCTION.

This section describes the schemes considered for designing of each major component of elevators and the evolution of final design. This is done considering the special characteristics of A-74 project aircraft like the use of super critical section for tail plane and since the trim load is normally down section is inverted. Also the mandatory requirement of PCAR is complied with in the selection of final design.

The design evolved in two major parts. The first part includes schemes for positioning of actuator, hinge, bracket and tail plane trailing edge structure; The second is for the detail parts like spar, torsion box and trailing edge of elevator structure itself. For both the schemes will be considered as a whole and then detailed merits of each part discussed.

5.2 SCHEMES FOR SUPPORT STRUCTURE AND ACTUATOR.

For design purposes the elevator support structure and actuator for the inboard elevator at station 1330 MM from aircraft centre line is considered. The need to accommodate the actuator and the hinge bracket on tail plane spar along with the actuator bracket on elevator makes the support at station 1330 MM, deciding factor of selection. The inverted supercritical section of tail plane dictates that any fairing to accommodate the actuator should be on the top surface.

Earlier, in the design, it was decided to have two jacks per elevator to provide duplex hydraulic actuators. This also reduces the size of jack and hence the fairing if the jack is to be mounted on top of the tail plane.

5.3 SCHEME 1.

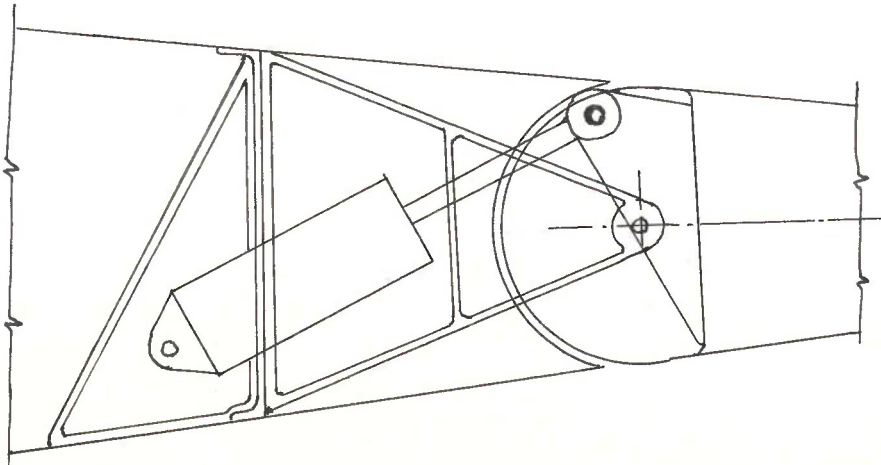
In this scheme the actuators are placed aft of rear spar in the trailing edge of tail plane. Actuator bracket on the elevator is inside the trailing edge position. The hinge line is at middle of spar. For the trailing edge bracket on the tail plane the actuators are on either side of it. See Fig. 5.1 of scheme for further details.

(a) Actuator position.

Positioning of actuator in the region between rear spar and nose cell of elevator is very convenient for maintenance purposes. It is easy to provide removable inspection panels on the trailing edge structure for frequent inspection of elevator hinges. Since the whole actuator was inside the trailing edge there is no need to provide any fairing and consequent drag problems.

One of the major problems encountered in this scheme is the small lever arm available for the actuator and the relative increase in size of jack. Also as the actuator mounting brackets on the rear spar is far from the rib it will be necessary to provide extensive back up structure.

Another alternative investigated was to position the actuator inside the tail plane box with a cut out in spar to accomodate the actuator. But owing to the complexity of scheme and difficulty in inspection and servicing of items this scheme was discarded.



(b) Hinge brackets.

Hinge brackets at actuator positions are more difficult to design because of the need to accomodate the actuators and brackets at the same place. Major factors considered in the design of hinge brackets are listed below.

- (1) Easy accessibility to hinges for inspection and maintenance. It should be possible to replace the elevator without much difficulty.
- (2) There should be adequate clearance between actuator and structure. A minimum of 15 mm is considered essential.
- (3) Elevator controls support structure should be designed to carry a sideways inertia

load of 12 times the weight. This load is small and the actuator brackets are designed to carry the side load.

(4) Self-aligning bearings are provided to take care of any slight non-alignment of supports.

(5) Hinge bolts should have positive locker device and some means of preventing over tightening.

In scheme 1 separate brackets are provided for actuators and hinge support on elevator. This complicated arrangement was not favoured and discarded in favour of a single bracket.

The trailing edge ~~bracket~~ on rear spar housed a single bearing which carried all the reaction. This needed a rather big bearing and extensive back up structure on elevator.

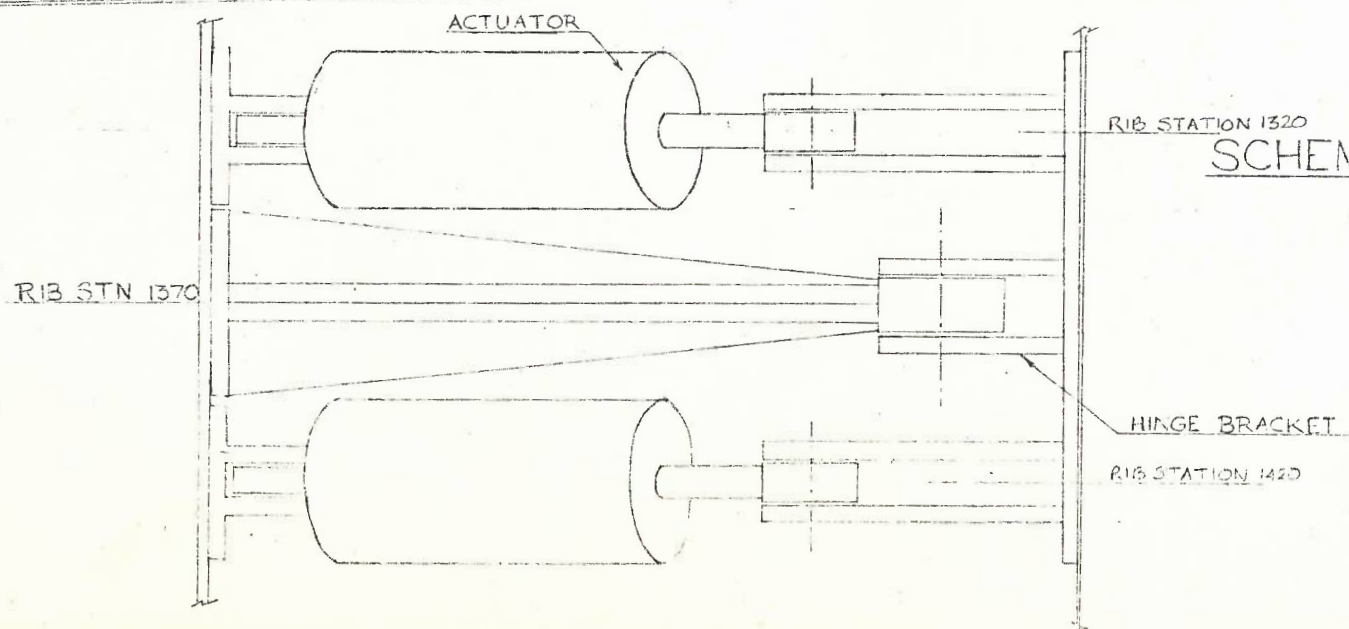
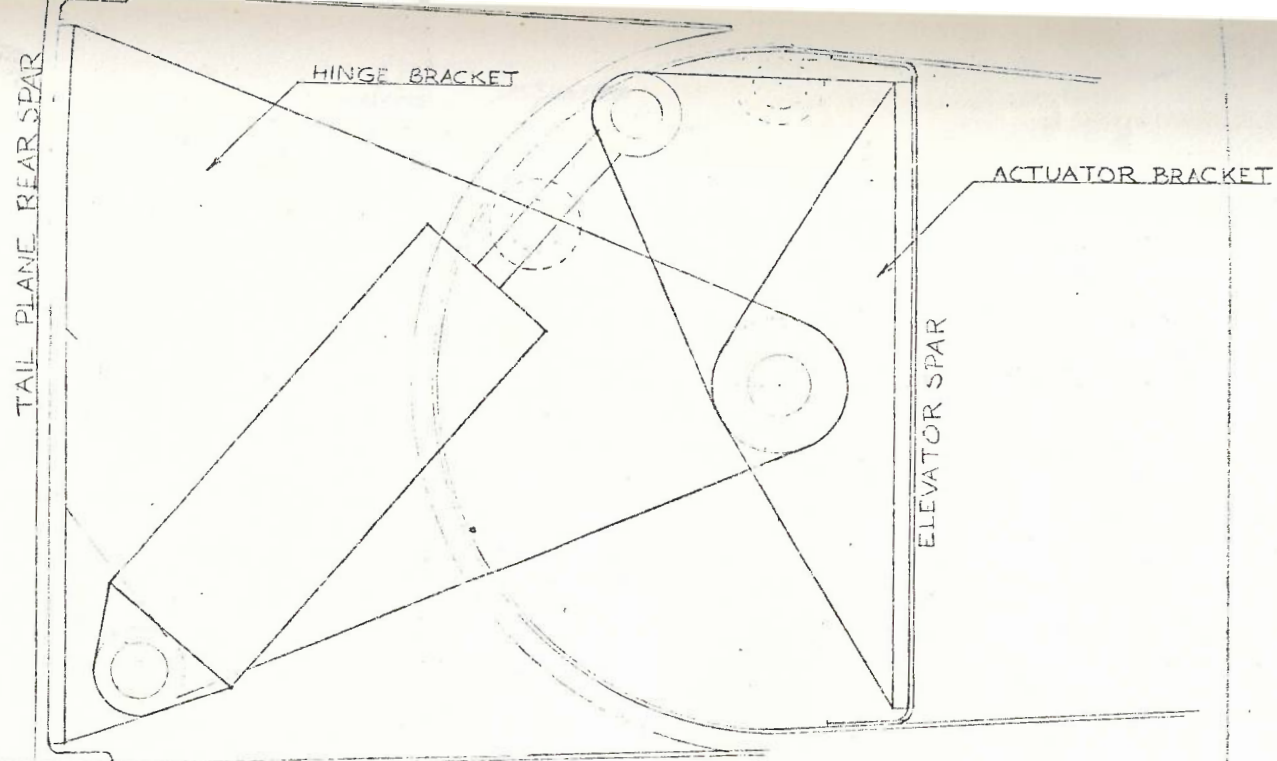
5.4 SCHEME 2.

Positioning the actuator on the top surface of tail plane with a fairing over it is the major modification from scheme 1. Hinge is at the centre of the elevator and the support bracket details are as in scheme 1. Fig. 2 gives details of this scheme.

(a) Actuator position.

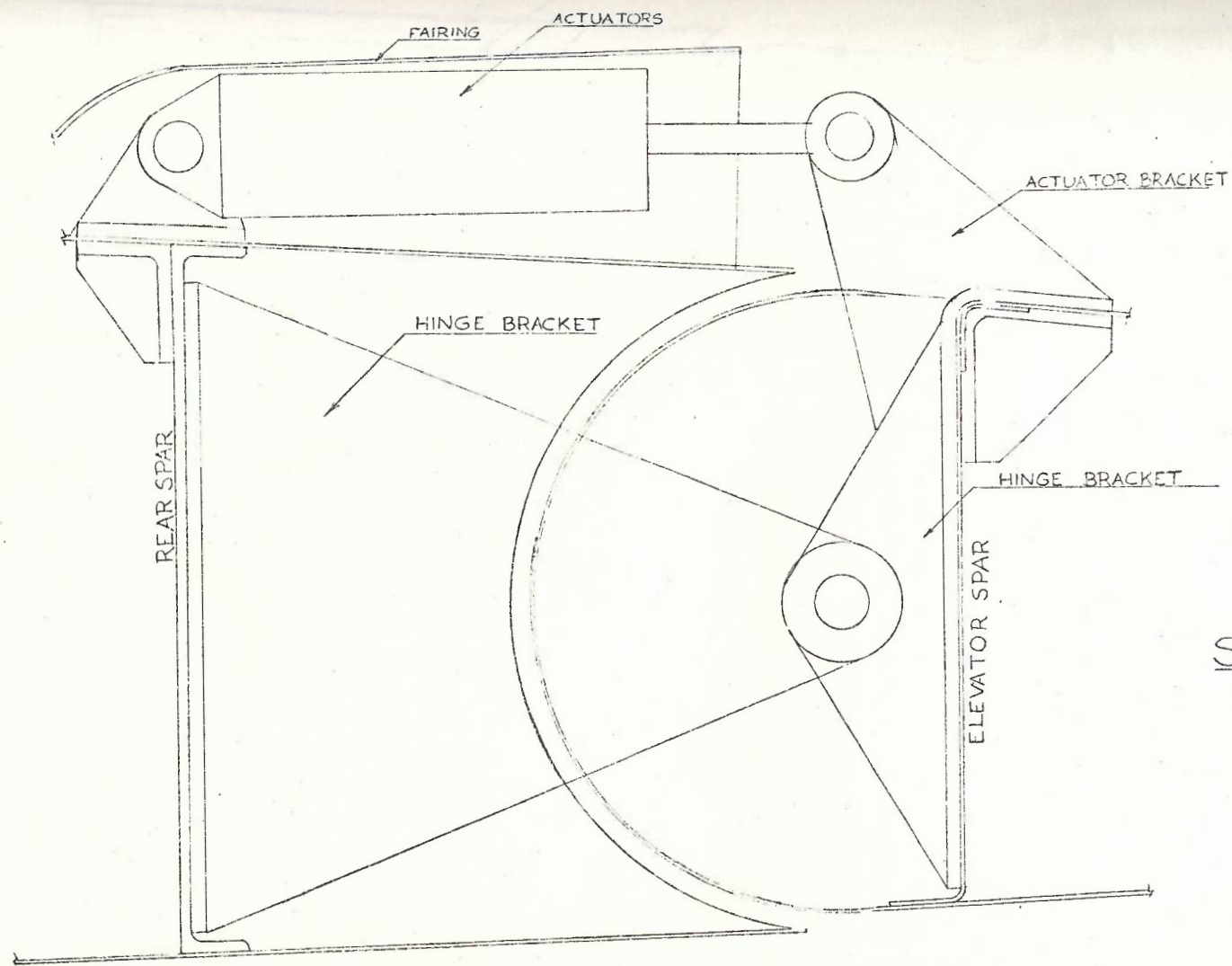
The actuator is positioned on the top surface because of inverted superscritical section of tail plane. This position is also good from accessibility and inspection point of view. The lever arm is more than what was available in scheme 1.

The major disadvantage comes from the additional drag because of the large fairing required. Increased drag is to be avoided, unless otherwise very essential, and so this scheme is not favoured.

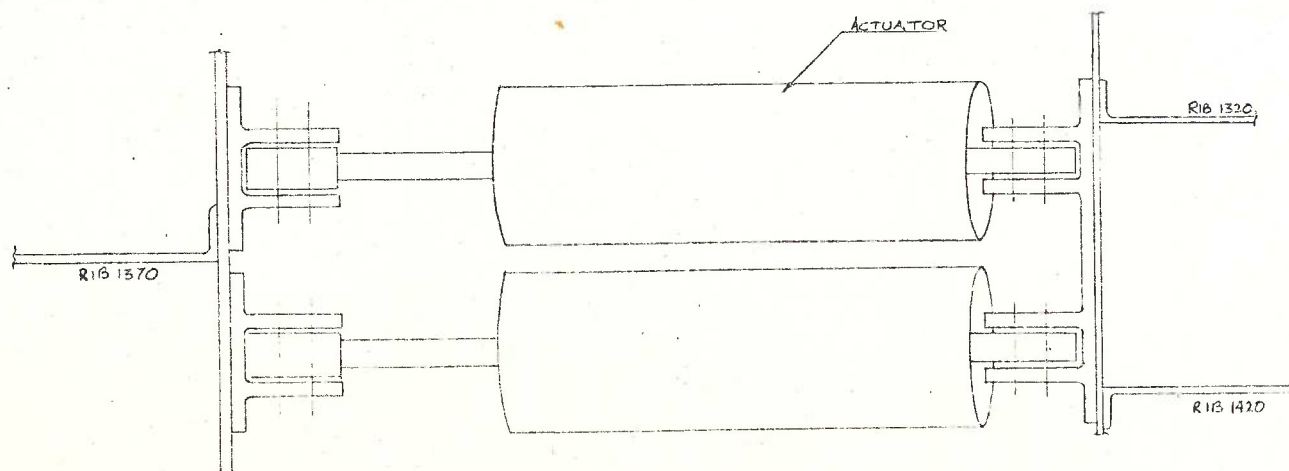
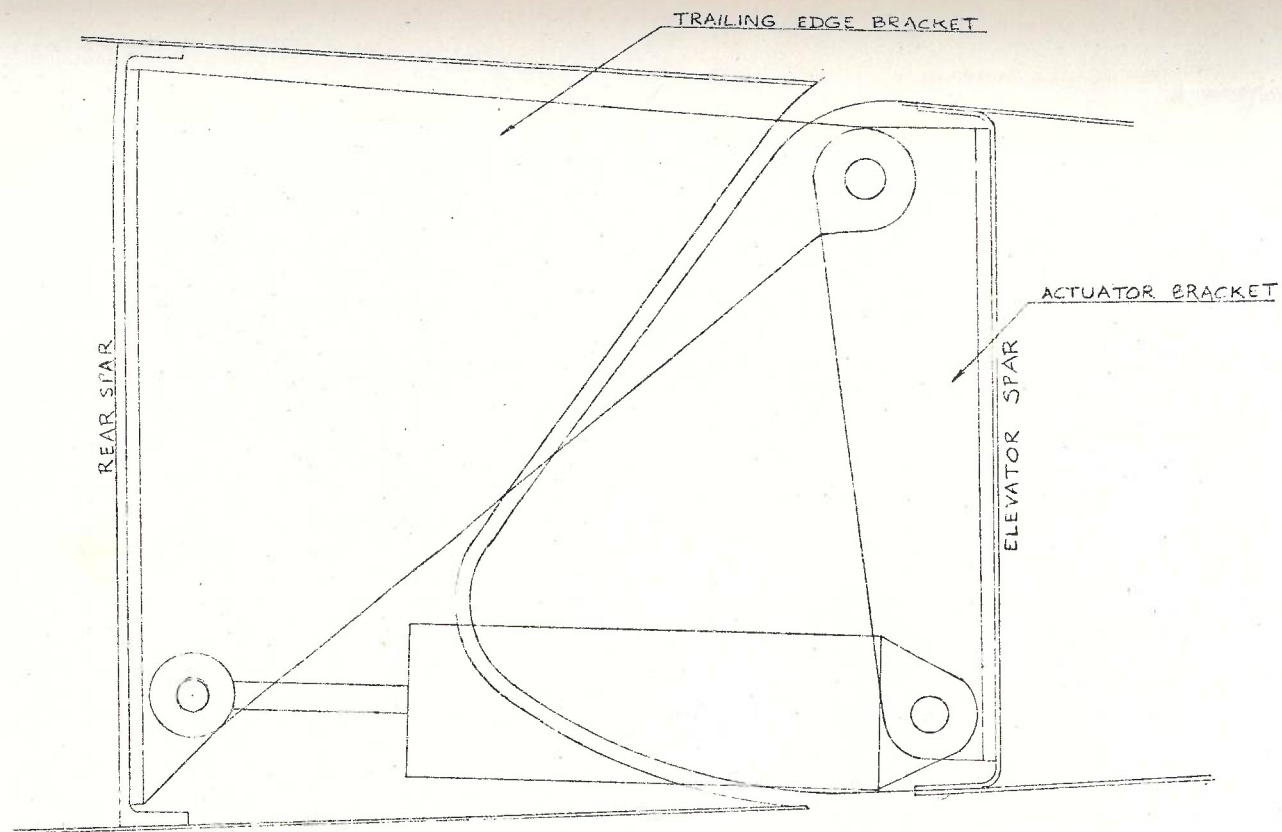


SCHEME 1

SCALE 1:1



SCHEME 2



SCHEME 3

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(b) Hinge bracket.

The hinge bracket is machined forging picking up bolts both on the elevator spar and top surface. A part of spar flange is cut away to accomodate the bracket and a boom angle introduced in the aft box. This scheme was not favoured in view of the actuator and hinge support bracket of scheme 3. Another possible alternative is to have two seperate brackets, one for top actuator and one for hinge. This way the spar flanges need not be cut.

5.5 SCHEME 3.

The details of this scheme are shown in Fig. 3. The actuator is placed between the rear spar and nose cell of elevator. The most important change is that the elevator hinge is shifted to the top of the elevator spar. The actuator is picked up at the bottom flange of the rear spar.

(a) Actuator position.

The position of actuator in trailing edge position of tail place provides easy access for inspection and maintenance. This position of actuator and hinge gives maximum possible lever arm for actuator and hence the jack size is minimised. Also this avoids additional fairing and consequent increase in drag. So this position of actuator is adopted in preference to schemes 1 and 2.

(b) Hinge brackets.

This scheme has a single bracket for actuator and hinge support. The bracket is attached to the forward face of elevator spar and it is not necessary to cut away the spar flange. Also the bracket gives maximum lever arm for actuator.

Trailing edge support brackets are duplicated. Each bracket is designed to take 80% of ultimate load in case of one bracket failing. This gives a chance to provide 'Fail safe' consideration for design of hinge support bracket.

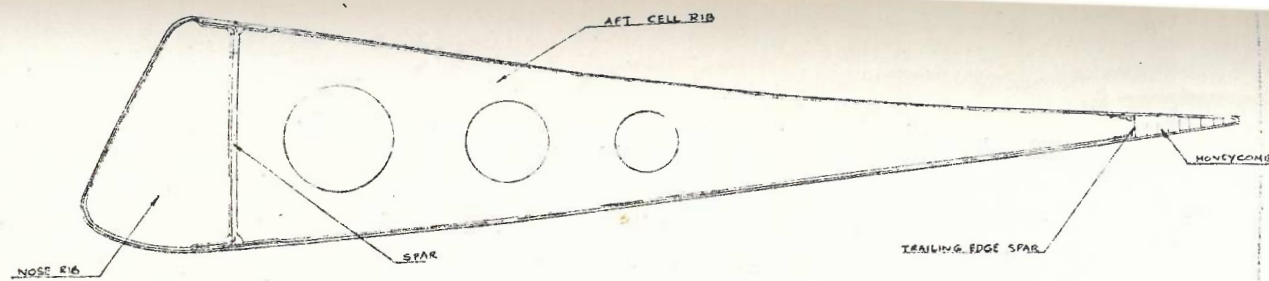


FIG. 6.5

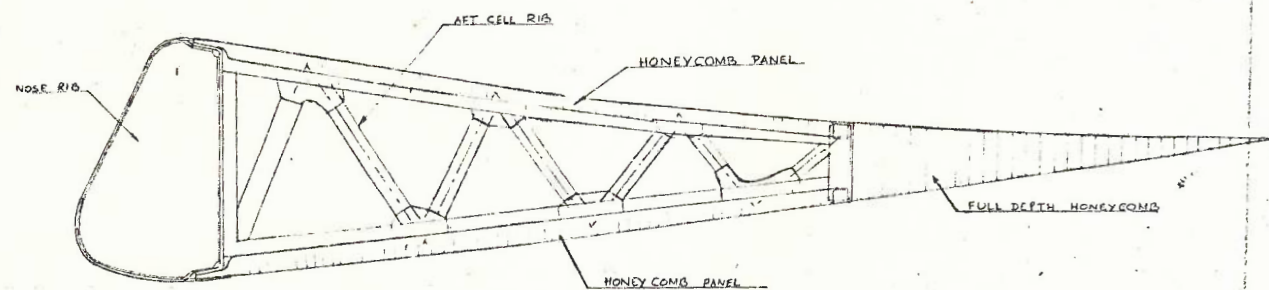


FIG. 6.4

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In conclusion scheme 3 was adopted as being the best compromise for all the aspects of design considered. This provided easy access, least aerodynamic drag, maximum lever arm for actuation and a possibility for 'fail safe design' of brackets.

5.6 SCHEMES FOR ELEVATOR STRUCTURE.

The A-74 aircraft has a novel feature in using the tail plane and wing for noise shielding from the rear mounted engines. However this will give the possibility of accoustic fatigue in their skin panels of components which are in the region. Also the jet efflux from engines flows partly over the tail plane and elevators. But as the A-74 aircraft has high bypass ratio, engine. Calculations showed there is no possibility of accoustic fatigue. Though a possible scheme using honey comb panels was studied briefly, this was discarded in favour of conventional single spar construction. Figs. 6.4 and 6.5 show the two schemes.

5.7 ELEVATOR SPAR.

The elevator spar is designed to carry bending moment and vertical shear acting on the elevator. But the spar also provides a joint between the nose cell and aft cell skins. Some of the other considerations for design of spar are the need to give a smooth joint between nose and aft cell skins. To avoid a step on the elevator contour, it is decided to have a joining strap on the spar and to provide joggles on ribs. Also the joining straps help in providing a leak proof joint to prevent rain water entering the elevator box, causing corrosion problems. The schemes considered are shown in the sketches below.

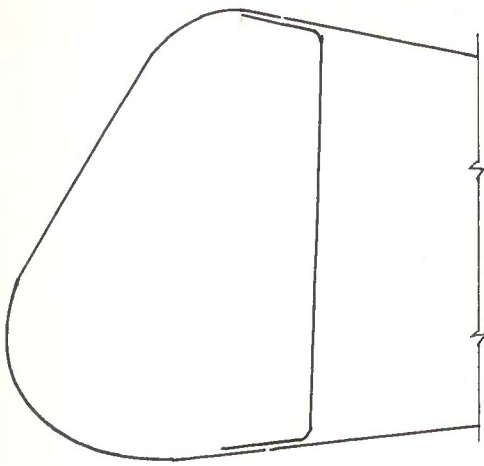
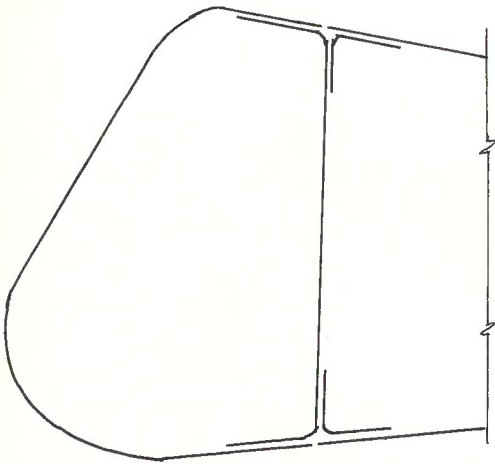
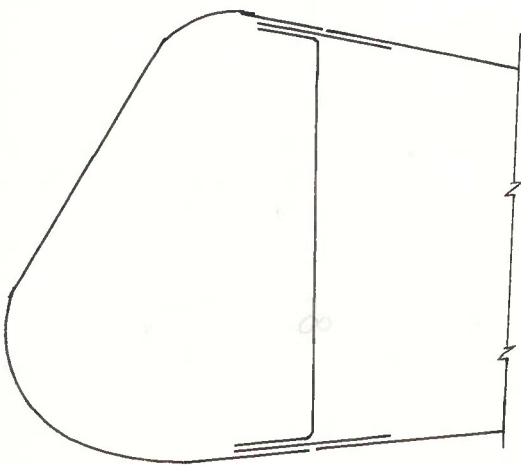
SCHEMES FOR SPARFIG 1FIG 2FIG 3

Figure 1.

This scheme provides for the ^{nose} cell and aft skins joined over the flange of spar. But this will cause serious problems of spar flange instability and so not adopted.

Figure 2.

This scheme provides for forward skin to be rivetted on spar flange and aft cell skin on to an angle section which forms part of spar boom. However, this scheme is not considered good, because of possibility of rain water getting inside and causing corrosion.

Figure 3.

This scheme provides for a joining strap on top of spar flange. This scheme gives the best compromise for all the above considerations. But the rib has to be joggled to accomodate the spar and joining strap thickness so that outside contour is maintained. This scheme is adopted for design.

5.8 ELEVATOR TRAILING EDGE.

It was decided to provide a false spar at trailing edge with a separte trailing edge position. Because of super critical section the trailing edge is quite sharp and presents manufacturing problems. Two possible alternatives investigated are shown below.

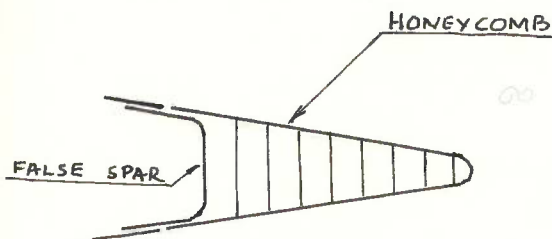
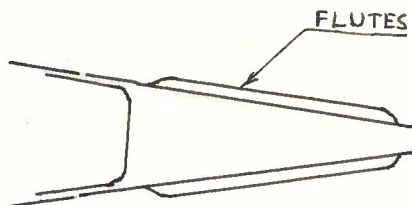
FIG. 1FIG. 2

Figure 1: This shows a full depth honey comb panel with a thin skin.

Figure 2: This shows a fluted skin bent at the trailing edge and attached to false spar.

The scheme of Fig.1 showing the honey comb core is selected because of the increased stiffness it provides for the section in the jet efflux of engines. Moreover other components like leading edge slats and spoilers make use of the honey comb construction and its use here does not pose any new problems.

6.9 ELEVATOR TIP FAIRING.

In order to give a smooth aerodynamic shape to the elevator tip this has to match with tail plane tip fairing. There are two alternatives to make a fairing of this type.

(1) Made of L 59 Aluminium Alloy sheets in two pieces and joined at the middle. L 59 is soft material and it is easy to form by hydraulic press or drop hammer process.

(2) Using glass fibre reinforce plastic to mould the fairing. Glass cloth of B S 3396 P6 with polyster resin to give a panel thickness of 1 mm is adequate to meet load requirements.

6.10 CALCULATION OF ACTUATOR DIMENSIONS

For inboard elevator

Max. reaction on inboard elevator = 20100 N.

Max. reaction torque on inboard

elevator = 13742 Nm ultimate.

Assuming one jack is to be designed to take 80% of proof load at hinge,

Max. torque to be resisted

$$\begin{aligned} \text{by jack} &= \frac{2}{3} \times 0.8 \times 13742 \\ &= \underline{7260 \text{ Nm.}} \end{aligned}$$

As jack has a lever arm of 180 mm under maximum deflection,

$$\text{Jack load} = \underline{40400 \text{ N}}$$

For a double acting jack from 4 way valve characteristics

$$F_{\text{max.}} = \frac{2}{3} p_s A$$

$$F_{\text{max}} = \text{Jack load}$$

$$A = \text{Area of jack}$$

$$p_s = \text{Supply pressure}$$

Assuming a supply pressure of 3000 psi,

$$\begin{aligned} A &= 40400 \times \frac{3}{2} \times \frac{1}{3000 \times 6895} \text{ m}^2 \\ &= \underline{2.9 \times 10^{-3} \text{ m}^2} \end{aligned}$$

$$\begin{aligned} \text{Diameter of piston} &= \sqrt{\frac{4 A}{\pi}} \\ &= \underline{60.6 \text{ mm}} \end{aligned}$$

Assuming a wall thickness of 5 mm (appx.)

$$\text{The outside dia. of jack} = \underline{70 \text{ mm}}$$

From an elevator deflection of $\pm 16^\circ$

$$\begin{aligned} \text{Stroke} &= R \theta = 2 \times .18 \times \frac{16}{180} \times \\ &= 0.11 \text{ m} = \underline{110 \text{ mm.}} \end{aligned}$$

SECTION 6

DESCRIPTION OF FINAL DESIGN

SECTION 6

DESCRIPTION OF FINAL DESIGN

6.1 ELEVATOR LAYOUT AND GEOMETRY

The elevator layout selected for A-74 project aircraft is given in Drawing A-74 - E 1. It is a four piece elevator scheme with inboard and outboard elevators having equal area. The inboard end rib of inboard elevator is at an angle to the hinge line requiring only a small fairing to be attached to the tail cone.

6.2 ACTUATORS

Actuators are positioned at the inboard hinge of each elevator. Drawing A 74 - E 7 gives the actuator positioning of the inboard elevator. Actuator position and brackets of outboard elevator are similar.

Elevator control is fully hydraulic with no manual reversion. Each elevator is driven by 2 actuators double acting type. Actuators are driven by separate hydraulic systems. As the elevators are fully hydraulic artificial feel system is provided in the control circuit.

6.3 ACTUATOR HINGE BRACKET

See Drawing A 74 - E 8

Actuator hinge bracket has two lugs, one on either side with hinge support bracket and actuator lug accommodated in the middle. This avoids out of plane bending of lugs. On the top there are S 80 steel bush of 15 mm I/D and at bottom steel bush of 12 mm I/D. Bracket is machined out of L 65 forging and surface finished to 30 microns. As elevator hinge brackets are class I parts magnetic or ultra-sonic crack detection is essential.

Bracket is attached to the front face of spar and back up structure by 4-S 96 - 6.3 mm ($\frac{1}{4}$ ") UNF bolts and castellated nuts with wire locking. The back up structure consists of a L65 block which is bolted on to the rib.

6.3 TAIL PLANE TRAILING EDGE BRACKET

See Drawing A-74 - E 8

Seperate trailing edge brackets are provided for each actuator. The trailing edge bracket is designed to carry 80% of ultimate load in the event of failure of one of the brackets. The trailing edge brackets are bolted to a channel section at top to give lateral stability against side loads. The brackets are attached to the tail plane hinge rib through rear spar and back up structure.

The bracket is machined out of L 65 forging. Bracket houses a AMPEP 15 GSW self aligning bearing. The bracket is attached by a group of 6 S 96 5/16" bolts and castellated nuts with wire locking. The tail plane trailing edge structure has removable panels with Zeus fasteners to facilitate inspection and maintenance of hydraulic actuators and hinge supports.

Hinge bolts are made from high tensile steel S 99 and has shoulder to prevent over tightening. Bolts are locked to class A standard using castellated nuts and wire locking.

6.4 NOSE CELL SKINS

See Drawing A-74 - E 3.

Nose cell skin is made of 0.8 mm thick Aluminium alloy sheets rivetted on to nose ribs by 3.14 mm ($\frac{1}{8}$ ") Avdel counter sunk rivets. Skin and ribs are spin dimpled to accomodate the 100 degrees C'SK. Drawing E 3 gives a section through the nose cell. It is necessary to ensure that the nose cell does not distort out shape and foul with trailing edge structure. Nose cell skin is attached on to finger plate and spar by C'SK Avdel rivets. 1.2 mm thick joint strap on spar flange ensures there is no spanwise step at fore and aft skin joint.

6.5 AFT CELL SKINS

See Drawing A 74 - E 3

Aft cell skins are made in two pieces and rivetted on to ribs and joint strap using L 69 CSK

rivets of 3.14 mm diameter. The skin and ribs are spin dimpled. Rivets in the trailing edge region where reaction cannot be provided are 100° C'SK Avdel rivets.

Stiffeners are attached to the skin with L 69 CSK rivets and the stiffeners are attached to the joint strap. Stiffeners have joggles at the end.

6.6 NON-HINGE RIBS

See Drawing A 74 - E 5

Ribs are placed at about 250 mm pitch with 2 end ribs and 4 hinge ribs at hinge stations. Rib spacing in between the hinge supports is decided by permissible shear buckling stress of panel. All intermediate bays have a stiffener to allow higher permissible shear stress in panel.

Non hinge ribs are made of Aluminium alloy L 72 pressings of 0.8 mm thick sheets. The ribs have joggles to accomodate the joint strap and false spar at the trailing edge so that a smooth aerodynamic contour is maintained on the outside.

6.7 HINGE RIBS

See Drawing A-74 - E 6

Hinge ribs carry a much higher load and are machined out of DTD 5020 plates. There are no lightening holes on hinge ribs and steps are provided at edges to accomodate joint strap and false spar.

6.8 SPAR

See Drawing A-74 - E 4

Spar is made from L 72 material 1.6 mm thick with bent flanges 18 mm wide. As the spar runs on the straight contour it is possible to provide a straight bent for spar flange. Lightening holes are provided in the spar to reduce the weight and to increase shear stiffness of web.

6.9 TRAILING EDGE POSITION

See Drawing A 74 - E 3

Trailing edge position is made of full depth honey comb case with 0.314 mm (30 SWG) skins which are bent at the trailing edge and is attached on to the flange of trailing edge false spar.

6.10 FALSE SPAR ON TRAILING EDGE

It is made of L 72 material 1.0 mm thick having 30 mm bent flange. The aft cell skins and honey comb facings are rivetted on to the false spar.

6.11 OUTBOARD ELEVATOR TIP FAIRING

This is made out of glass fibre reinforced panel 1.0 mm thick moulded to give the smooth fairing shape. Glass fibre cloth BS 3396/P6 5 layers in polyester resin is used for making these fairings. Fairing is attached to the mid rib using 3/16" CSK bolts and stiff nuts.

APPENDIX I

SYMMETRIC LOADING CALCULATIONS

APPENDIX 1SYMMETRIC LOADING CALCULATIONS1.1 INTRODUCTION

Loading calculations of A 74 project aeroplane was divided into separate groups doing different calculations. Symmetric loading calculations and manoeuvring loads were done by Mr. A.N. Campbell and the author. Specimens calculations give the details of calculations given in the table at the end of this section.

1.2 LOADING CASES INVESTIGATED

Table 1.1 gives the balance tail load for 3 aircraft weights of (1) AUM 58,900 KG (2) APS weight 40,000 KG (3) Landing weight 50280 KG at sea level, 6.5 KM and 9.2 KM altitude for forward and aft C.G positions.

From this six cases giving maximum tail loads in up and down direction is considered for calculation of manoeuvring loads from 1g to 2.5g and 2.5g to 1g conditions. See table 1.2 to 1.4.

Finally table 1.5 gives the manoeuvring cases for approach cases with full flaps. It is seen that case 21 of approach case with forward C.G position gives the critical cases. All other manoeuvring cases give only about half the value of this load.

SPECIMEN CALCULATIONS.

In steady flight conditions, the balance tail load is given by equation 14 on DES 8/6.

$$L_T (lg) = \frac{M_O + nW(h - H_O)\bar{C} + T Z_T - \frac{W}{g} K_B^2 \ddot{\theta}}{l_T}$$

Where,

$$M_O = \frac{1}{2} \rho V^2 S \bar{C} C_{m_O}$$

CASE 1.

All ^{up}weight = 58,000 Kg. mass at sea level.

C.G. forward, Power on
 $V_D = 206 \text{ m/sec}^2$

$$\begin{aligned} M_O &= \frac{1}{2} \times V^2 \cdot S \cdot \bar{C} C_{m_O} \\ &= -\frac{1}{2} \times 1.225 \times 206^2 \times 136 \times 3.84 \times 0.08 \end{aligned}$$

From DES 7400/12,

$$\rho = 1.225 \text{ Kg./m}^3$$

$$S = 136 \text{ m}^2$$

$$C_{m_O} = -0.07 - .01 = -0.08$$

$$\bar{C} = 3.84 \text{ ms.}$$

$$M_O = \underline{-1,087,000 \text{ Nm.}}$$

$$nW(h - H_O)\bar{C} = 1.0 \times (58,900 \times 9.81) (-.20 - .215) 3.84 = -33,240 \text{ Nm.}$$

$$h = 0.20 \text{ for forward C.G.}$$

$$H_O = 0.215$$

For power on condition, Thrust is equal to the drag.

$$C_D = 0.0174 + 0.0437 C_L^2$$

At sea level for $V_D = 206$, $W = 58,900$

$$C_L = \frac{nW}{\frac{1}{2} \rho V^2 S} = \frac{58,900 \times 9.81}{\frac{1}{2} \times 1.225 \times 206^2 \times 136}$$

$$= 0.1635$$

$$C_D = 0.0174 + 0.0437 \times 0.1635^2$$

$$= 0.01857$$

$$T = D = \frac{1}{2} \rho V^2 \cdot S \cdot C_D$$

$$= \frac{1}{2} \times 1.225 \times 206^2 \times 136 \times 0.01857$$

$$= \underline{65,600 \text{ N.}}$$

$$Z_T = -3.45 \text{ m.}$$

$$T \cdot Z_T = -65,600 \times 3.45$$

$$= \underline{-226,200 \text{ Nm.}}$$

$$l_T = 12.4 \text{ m.}$$

Balance Tail load for steady level flight

$$L_T = - \frac{1,346,440}{12.4} = \underline{-108.6 \text{ KN.}}$$

(b) FLIGHT MANOEUVRING LOADS.

For the solution of pitching moment equation

R - Damping coefficient for short period motion, and

J - Natural damped frequency of short period motion.

From BCAR Chapter D 3-2 Appendix 1,

$$R = \left[1 + \frac{d\epsilon}{d} \right] \frac{1}{4} \frac{S_T}{S} \frac{l_T^2}{K_B^2} a_{1T} + \frac{a_1}{4} - \frac{l_T^2}{2K_B^2} (m_q) \text{ less tail}$$

$$m_q = 0$$

$$\text{Where } S_T = 33.58 \text{ m}^2$$

$$S = 136 \text{ m}^2$$

$$l_T = 12.4 \text{ m for forward C.G}$$

$$K_B = \frac{I}{M} = \frac{29.8 \times 10^5}{58,900} = 7.1 \text{ m.}$$

$$\text{For } V_D = 206 \text{ m/sec}^2 \text{ at sea level. from DES 7400}$$

$$a_{1T} = 4.8$$

$$a_1 = 6.1$$

$$\frac{d\epsilon}{d\alpha} = 0.47$$

$$R = 2.855$$

$$J = \left[\frac{W \bar{C} a_1 H_m}{2g \rho S K_B^2} - R^2 \right]^{\frac{1}{2}}$$

H_m - Manoeuvre margin

$$= \frac{-1}{a_1} \left[\frac{dC_m}{d\alpha} \right]_{\text{less tail}} + \frac{a_{1T}}{a_1} \left[1 - \frac{d\epsilon}{d\alpha} \right] \bar{V}$$

$$- \frac{l_T m_{\dot{q}}}{\mu \bar{C}} + \frac{\bar{V} a_{1T}}{2\mu}$$

$$\text{Where } m_{\dot{q}} = 0$$

$$\bar{V} - \text{tail volume coefft.} = \frac{S_T l_T}{S \bar{C}} = \underline{0.797}$$

$$\mu - \text{Aircraft relative density} = \frac{W}{g \times S l_T}$$

$$= \underline{28.5}$$

$$H_m = -(h_Q - H) + \frac{a_{1T}}{a_1} \left[1 - \frac{d\epsilon}{d\alpha} \right] \bar{V} + \frac{\bar{V} a_{1T}}{2\mu}$$

$$= -(0.2 - 0.215) + \frac{4.8}{6.1} (1 - 0.47) 0.797 + \frac{0.797 \times 4.8}{2 \times 28.5}$$

$$H_m = \underline{0.414}$$

Hence,

$$J = \frac{58,900 \times 6.1 \times 0.414}{2 \times 1.225 \times 136 \times 7.1^2} - 2.855^2 \quad \frac{1}{2}$$

$$= 5.06$$

ELEVATOR ANGLE REQUIRED FOR STEADY ROTATION. (η_s)

$$\eta_s = \frac{-W H_m (n-1)}{\frac{1}{2} \rho V^2 S a_2 \bar{V}}$$

$$a_2 = 0.65 a_{1T}$$

$$\eta_s = \frac{58,900 \times 9.81 \times 0.414 (2.5 - 1.0)}{\frac{1}{2} \times 1.225 \times 206^2 \times 136 \times 0.797 \times .65 \times 4.8}$$

$$\eta_s = \underline{0.0408 \text{ radians.}}$$

For an instantaneous application of elevator

$$\eta_{\max} = \frac{-W H_m (n-1) \Sigma}{\frac{1}{2} \rho V^2 S a_2 \bar{V}}$$

$$\text{Where } \Sigma = \left[1 + e^{\frac{-R\pi}{J}} \right]^{-1}$$

Referring to Fig. 1 of BCAR D 3-2 Appendix 1

$$\text{For } \frac{J}{R} = \frac{5.06}{2.855} = 1.77 \quad \Sigma = 0.85$$

$$\eta_{\max} = \underline{0.0347 \text{ radians.}}$$

For instantaneous application of elevator, the incremental tail load

$$\begin{aligned}
 L_{T_0} &= - \frac{W H_m \bar{C} (n-1)}{l_T} \approx \\
 &= - \frac{58,900 \times 9.81 \times 0.414 \times 3.84 \times (2.5 - 1) \times .85}{12.4} \\
 &= \underline{-94.5 \text{ KN}}
 \end{aligned}$$

Form DES 13/6 equation 12

L_T - Increment in tail load due to incidence effect is given by

$$\begin{aligned}
 L_{T_\alpha} &= W \frac{S_T}{S} (n-1) \frac{a_{1T}}{a_1} \left[1 - \frac{d\epsilon}{d\alpha} + \frac{a_1}{2\mu} \right] \\
 &= 58,900 \times 9.81 \times \frac{33.58}{136} (2.5 - 1) \frac{4.8}{6.1} \times \\
 &\quad \left[1 - 0.47 + \frac{6.1}{2 \times 28.5} \right] \\
 &= \underline{+107.6 \text{ KN}}
 \end{aligned}$$

Total tail load at first half cycle of motion from instantaneous application of elevator

$$L_{T_0} + L_{T_\alpha} = -94.5 + 107.6 = \underline{13.1 \text{ KN}}$$

This is rather too severe and so BCAR D 3-2 suggests

$$\begin{aligned}
 \eta_1 &= \eta_s \frac{\Sigma}{\left[1 + \frac{3.3 R}{K} \right]^{\frac{1}{2}}} \quad \text{for } l_g \text{ to } 2.5 \\
 &= \underline{0.029 \text{ radians.}} \\
 2 &= \eta_s \left[1 - \frac{\Sigma}{1 + \frac{3.3 R}{K}} \right] \quad \text{for } n_g \text{ to } l_g. \\
 &= \underline{0.0118 \text{ radians.}}
 \end{aligned}$$

$$\text{Where } K = \frac{2 \hat{t}}{\Sigma \eta_s} \frac{d\eta}{dt}$$

$$\hat{t} = \frac{W}{g S V} = \frac{58,900}{1.225 \times 136 \times 206} = 1.72$$

$$K = 4J + R = 23.55$$

$$L_{T1} = \frac{L_{To}}{\left[1 + \frac{10}{K} R\right]^{\frac{1}{2}}} = \frac{94.5}{1 + \frac{10 \times 2.855}{23.55}} = 63.2 \text{ KN.}$$

$$L_{T2} = - \frac{L_{To}}{\left[1 + \frac{10}{K} R\right]^{\frac{1}{2}}} = -63.2 \text{ KN}$$

Contribution from elevator deflection are given by

$$L_{T1} = \frac{L_{To}}{\left[1 + \frac{3.3}{K} R\right]^{\frac{1}{2}}} = \frac{(-94.5)}{1.182} = -80.0 \text{ KN.}$$

$$\begin{aligned} L_{T2} &= L_{To} \sum \left\{ - \frac{1}{\left[1 + 3.3 \frac{R}{K}\right]^{\frac{1}{2}}} \right\} \\ &= 94.5 \left[\frac{1}{0.85} - \frac{1}{1.182} \right] = 0 \end{aligned}$$

CHECKED MANOEUVRE.

BCAR Chapter D 3-2 Appendix 1 gives empirical relations to solve equations of motion in this case.

$$\eta_{\max} = \eta_s K_3$$

$$\text{Where } K_3 = 1 + 0.68 \left[\frac{\phi}{R} \right]^{1.7} \left[\frac{1}{J/R + 1} \right]^{0.15 + 1.5 \frac{J}{R}}$$

$$\begin{aligned} \phi &\text{ greater of } (J^2 + R^2)^{\frac{1}{2}} \\ &= (5.06 + 2.855)^{\frac{1}{2}} \end{aligned}$$

Or,

$$\frac{V_e}{2V_A} \left[\frac{W}{g \rho S V} \right] = 4.98$$

$$V_e = V = 206 \text{ m/sc}$$

$$V_A = 111.4 \text{ m/sc}$$

$$\phi = 4.98$$

$$\text{Choose value of } \phi = 5.81$$

$$\text{Substituting } \eta_{\max} = 0.0408 \times 1.13 = \underline{0.0462 \text{ radians.}}$$

Maximum increment of elevator deflection during checked manoeuvre

$$\eta_1 = \eta_{\max} K_4$$

$$K_4 = K_3 \sin \theta_1$$

$$= K_3 \sin \left[\frac{\pi}{2 + 2.8 \frac{1}{W/R}} 1.09 \right]$$

$$= 1.125$$

$$\eta_1 = 1.125 \times 0.0408$$

$$= \underline{0.046 \text{ radians.}}$$

Total increment in tail loads are

$$L_{T1} = L_{T0} K_1 \text{ for lg to ng manoeuvre}$$

$$K_1 = \left[\frac{\phi}{R} \right]^{1.2} \left[\frac{0.64}{0.48 \left[\frac{J}{R} \right]^{1.3} + 1} \right]$$

$$K_1 = 0.75$$

Therefore,

$$L_{T1} = 94.5 \times 0.75 = \underline{-70.8 \text{ KN.}}$$

$$\begin{aligned}
 L_{T_2} &= -L_{T_0} K_2 \quad \text{for ng to lg manoeuvre.} \\
 K_2 &= \left[\frac{\phi}{R} \right]^{1.38} \left[\frac{0.92}{0.48 \left(\frac{J}{R} \right)^{1.3} + 1} \right] \\
 &= \underline{1.22}
 \end{aligned}$$

Therefore

$$L_{T_2} = -94.5 \times 1.22 = \underline{116.0 \text{ KN.}}$$

The elevator components of tail loads are given by,

$$\begin{aligned}
 L_{T_1 \eta} &= \frac{L_{T_0}}{\sum} \frac{\eta_1}{\eta_s} = L_{T_0} \frac{K_4}{\sum} \quad \text{for lg to ng.} \\
 &= \underline{-102.6 \text{ KN}}
 \end{aligned}$$

$$L_{T_2} = \frac{L_{T_0} \eta_2}{\sum \eta_s} = L_{T_0} \frac{K_5}{\sum}$$

$$\text{Where } K_5 = K_3 \sin(\tan^{-1} y + \pi/2)$$

$$\text{And } y = \frac{B}{C \phi - a_1 \frac{H \eta}{V} \frac{2}{\phi/R}}$$

$$B = \left[1 - \frac{d\epsilon}{d\alpha} + \frac{a_1}{2\mu} \right] = 3.06$$

$$C = \left[1 + \frac{d\epsilon}{d\alpha} \right] \frac{a_1}{\mu} = 0.248$$

Substituting other values, $\hat{y} = -1.83$

$$K_5 = 1.13 \sin(-.479)$$

$$\text{Therefore } L_{T_2} = \underline{+60.2 \text{ KN.}}$$

$$\text{Tail load in steady 1 g flight } (L_{T_{1g}}) = \underline{-108.6 \text{ KN}}$$

$$\text{Increment in tail load due to lg to ng manoeuvre} = \underline{-70.8 \text{ KN.}}$$

$$\begin{aligned} \text{Tail load for lg to ng manoeuvre} &= -108.6 - 70.8 \\ &= \underline{-179.4 \text{ KN}} \end{aligned}$$

$$\ddot{\theta} = - \frac{L_{T_1} l_T}{W K_B^2} g = \underline{0.295 \text{ rad/sec}^2}$$

For ng to lg manoeuvre,

$$L_T = L_{T_{2.5g}} + L_{T_2} = -119.0 + 116.0 = \underline{3.0 \text{ KN}}$$

$$\ddot{\theta} = \frac{L_{T_2} l_T}{W K_B^2} g = \underline{0.482 \text{ rad/sec}^2}$$

INCIDENCE AND ELEVATOR COMPONENT OF TAIL LOAD.

In order to calculate the torque on tail plane and loads on elevator the total aerodynamic load on the tail plane is to be separated into incidence and elevator component.

$$L_T = L_{T\alpha} + L_{T\eta}$$

Where,

$L_{T\alpha}$ - incidence component.

$L_{T\eta}$ - elevator component.

(a) Manoeuvring from steady 1.0g flight to 2.5g

$$\begin{aligned} L_{wb} &= n W - L_{T_{1g}} = 9.81 \times 58,900 - (-108,600) \\ &= \underline{686.500 \text{ KN}} \end{aligned}$$

$$\alpha_b = \frac{L_{wb}}{a_1 q s} \quad (\text{w.r.t. zero lift line})$$

Where,

$$a_1 = 6.15 \text{ from A 74/12}$$

$$q = \frac{1}{2} \rho V^2 = \frac{1}{2} \times 1.225 \times 206^2 = 26,000$$

$$\alpha_b = \frac{686,000 \times 57.3}{6.15 \times 26,000 \times 136}$$

$$= 1.81 \text{ degrees.}$$

Body setting angle = 3°

$$\alpha_b \text{ w.r.t. body centre line} = -1.19^\circ$$

From A 74/12

$$\text{the downwash angle } \epsilon = +0.35^\circ$$

In steady 1.0 g condition the aircraft is trimmed by moving the all moving tail plane and so the elevator component $L_{T\eta_{1.0g}} = 0$

$$L_{T_{1.0g}} = -108.6 \text{ KN}$$

In manoeuvre from 1.0g to 2.5g

$$L_{T_{2.5g}} = -119 \text{ KN}$$

$$L_{T_1} = -70.8 \text{ KN}$$

$$L_{T_1\eta} = -102.6 \text{ KN.}$$

$$\begin{aligned} \text{Therefore, } L_{T_{1\alpha}} &= L_{T_1} - L_{T_1\eta} \\ &= -70.8 - (-102.6) \\ &= 31.8 \text{ KN} \end{aligned}$$

Total incidence component of tail load,

$$\begin{aligned} L_{T\alpha} &= L_{T\alpha_{1.0g}} + L_{T_{1\alpha}} \\ &= -108.6 + 31.8 \\ &= -76.8 \text{ KN.} \end{aligned}$$

Total elevator component of tail load,

$$\begin{aligned}
 L_T \eta &= L_{T_{1.0g}} + L_{T_1} \eta \\
 &= 0 - 102.6 \text{ KN}
 \end{aligned}$$

Check $L_T + L_T = L_T$

$$-76.8 - 102.6 = -179.4$$

Also, $L_{T(1g)} + L_{T_1} = L_T$

$$-108.6 - 70.8 = -179.4$$

Both are full filled

Manoeuvring from 2.5g to 1.0g.

Balance load at initial 2.5g steady flight

$$L_{wb} = nW - L_{T_{2.5g}}$$

$$= 2.5 \times 9.81 \times 58,900 - (-119,000)$$

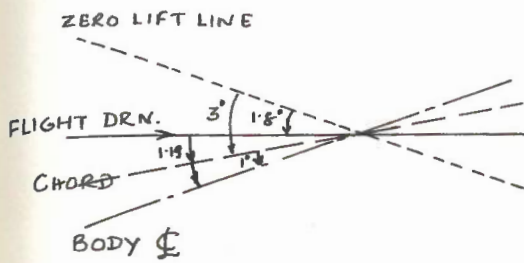
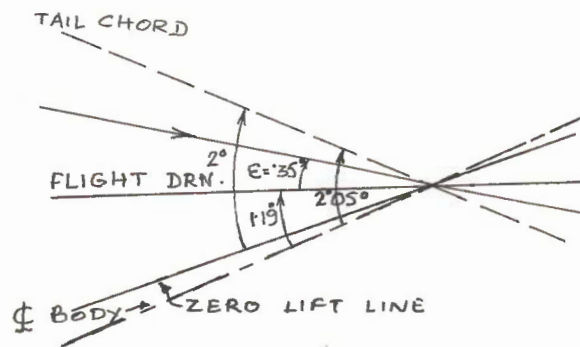
$$= 1,562,000 \text{ N.}$$

$$\begin{aligned}
 \alpha_b &= \frac{L_W}{q S_T a_{1T}} = \frac{1,562,000 \times 57.3}{26,000 \times 33.58 \times 4.8} \\
 &= +4.12^\circ \quad (\text{w.r.t. zero left line.})
 \end{aligned}$$

$$\alpha_b \text{ w.r.t body centre line} = +1.12^\circ$$

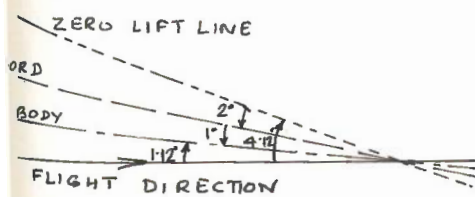
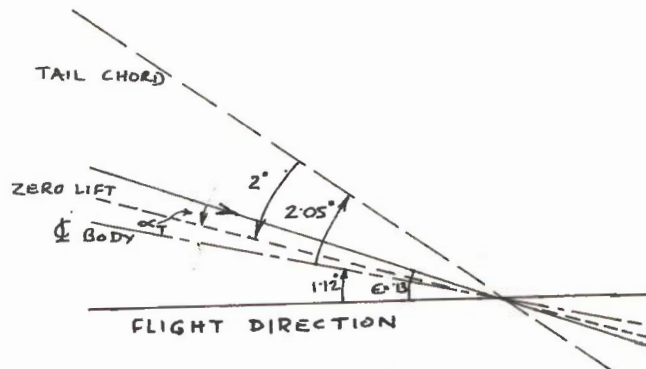
From A 74/12 the downwash angle ϵ at tail for $\alpha_b = 1.12^\circ$ 1.3.

To find out tail angle of attack at 2.5g manoeuvre, it is assumed that the steady 1.0g condition the elevator angle is zero.

WINGTAIL PLANE

Steady 1.0g condition.

Tail chord makes an angle 2.05° w.r.t. Body C.

WINGTAIL PLANE

2.5g condition

Incidence component of tail load $L_{T\alpha}$

$$\begin{aligned} L_T &= q S_T a_{1T} \alpha_T \\ &= -26,000 \times 33.58 \times 4.80 \times \frac{0.13}{57.3} \\ &= \underline{-9.5 \text{ KN.}} \end{aligned}$$

Total steady state load on tail plane for 2.5g,

$$L_{T_{2.5g}} = \underline{-119.0 \text{ KN.}}$$

$$L_{T\eta} = -119 - (-9.5) = \underline{-109.5 \text{ KN.}}$$

Manoeuvring from 2.5g to 1.0g

Total tail load, $L_{T_2} = \underline{116.2 \text{ KN}}$

$$L_{T_{2\eta}} = \underline{+60.2 \text{ KN}}$$

$$L_{T\alpha} = \underline{+55.8 \text{ KN}}$$

Total incidence component of tail load = $L_{T\alpha_{2.5g}} + L_{T_{2\alpha}}$

$$L_{T\alpha} = +55.8 - 9.5 = \underline{46.3 \text{ KN.}}$$

Total elevator component of tail load = $L_{T\eta_{2.5}} + L_{T_2\eta}$

$$L_{T\eta} = -109.5 + 60.2 = \underline{-49.3 \text{ KN.}}$$

Check,

$$L_T + L_T = 46.3 - 49.3 = \underline{-3.0}$$

$$L_{T_{2.5}} + L_{T_2} = -119 + 116.0 = \underline{-3.0}$$

Both conditions are satisfied.

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L _T KN	L _{T₁} KN	L _{T₂} KN	Total loads KN
1	58,900	0	FWD	V _D	2.5	-119.0		+116.0	- 3.0
2	58,900	0	FWD	V _D	1.0	-109.0	-70.60		-179.0
3	58,900	0	AFT	V _D	2.5	- 41.0		+ 57.6	+16.6
4	58,900	0	AFT	V _D	1.0	- 81.0	-37.4		-118.4
5	58,900	6.5	FWD	V _D	2.5	-114.5		+112.0	- 2.5
6	58,900	6.5	FWD	V _D	1.0	-104.0	-67.0		-171.0
7	58,900	6.5	AFT	V _D	2.5	- 36.4		+ 54.0	+17.6
8	58,900	6.5	AFT	V _D	1.0	- 76.4	-34.0		-110.4
9	58,900	9.2	FWD	V _D	2.5	- 85.6		+111.0	+25.4
10	58,900	9.2	FWD	V _D	1.0	- 72.9	-64.5		-137.4
11	58,900	9.2	AFT	V _D	2.5	- 5.9		+ 57.6	+51.7
12	58,900	9.2	AFT	V _D	1.0	- 76.7	-37.4		-114.1

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt KG	Alt Kms	C.G	Vel		L _T KN	L _{T1} KN	L _{T2} KN	Total load KN
13	58,900	0	FWD	V _C	2.5	-100.3		+114.0	+13.7
14	58,900	0	FWD	V _C	1.0	- 89.2	-69.6		-158.8
15	58,900	0	AFT	V _C	2.5	- 21.4		+ 57.3	+35.9
16	58,900	0	AFT	V _C	1.0	- 60.0	-36.5		- 96.5
17	58,900	6.5	FWD	V _C	2.5	-100.3		+114.0	+13.7
18	58,900	6.5	FWD	V _C	1.0	- 89.2	-67.1		-156.3
19	58,900	6.5	AFT	V _C	2.5	- 21.4		+ 53.7	+32.3
20	58,900	6.5	AFT	V _C	1.0	- 60.2	-33.7		- 93.9
21	58,900	9.2	FWD	V _C	2.5	- 77.8		+112.0	+34.2
22	58,900	9.2	FWD	V _C	1.0	- 63.0	-64.5		-127.5
23	58,900	9.2	AFT	V _C	2.5	+ 2.39		+ 57.3	+61.69
24	58,900	9.2	AFT	V _C	1.0	- 66.8	-36.5		-103.3

TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L_T KN	L_{T1} KN	L_{T2} KN	Total load KN
25	58,900	0	FWD	V_A	2.5	-62.35		+113.0	+50.65
26	58,900	0	FWD	V_A	1.0	-37.40	-69.2		-106.6
27	58,900	0	AFT	V_A	2.5	+19.8		+ 55.2	+75.0
28	58,900	0	AFT	V_A	1.0	- 5.8	-35.6		- 41.40
29	58,900	6.5	FWD	V_A	2.5	-62.30		+109.0	+46.7
30	58,900	6.5	FWD	V_A	1.0	-38.0	-63.3		-101.3
31	58,900	6.5	AFT	V_A	2.5	+19.8		+ 70.17	+90.5
32	58,900	6.5	AFT	V_A	1.0	- 5.8	-41.7		- 47.5
33	58,900	9.2	FWD	V_A	2.5	-62.0		+111.0	+49.0
34	58,900	9.2	FWD	V_A	1.0	-37.7	-62.1		- 99.8
35	58,900	9.2	AFT	V_A	2.5	+19.0		+ 55.2	+74.0
36	58,900	9.2	AFT	V_A	1.0	-39.8	-35.6		- 75.4

TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	VEL		L _T KN	L _{T1} KN	L _{T2} KN	Total load
1	40,000	0	FWD	V _D	2.5	-111.0		+74.8	-36.2
2	40,000	0	FWD	V _D	1.0	-106.0	-47.5		-153.5
3	40,000	0	AFT	V _D	2.5	- 60.0		+46.2	-13.8
4	40,000	0	AFT	V _D	1.0	- 89.0	-30.2		-119.2
5	40,000	6.5	FWD	V _D	2.5	-107.0		+71.7	-35.3
6	40,000	6.5	FWD	V _D	1.0	-101.0	-45.2		-146.2
7	40,000	6.5	AFT	V _D	2.5	- 55.5		+44.8	-10.2
8	40,000	6.5	AFT	V _D	1.0	- 84.0	-28.6		-112.6
9	40,000	9.2	FWD	V _D	2.5	- 76.4		+69.0	-74.0
10	40,000	9.2	AFT	V _D	1.0	- 70.0	-42.0		-112.0
11	40,000	9.2	AFT	V _D	2.5	- 23.2		+46.5	+23.3
12	40,000	9.2	AFT	V _D	1.0	- 51.0	-28.7		- 79.7

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L_T KN	L_{T_1} KN	L_{T_2} KN	Total load
13	40,000	0	FWD	V_C	2.5	-92.4		+74.5	-17.9
14	40,000	0	FWD	V_C	1.0	-86.0	-47.4		-133.4
15	40,000	0	AFT	V_C	2.5	-39.8		+46.4	+ 6.6
16	40,000	0	AFT	V_C	1.0	-67.0	-29.1		- 96.1
17	40,000	6.5	FWD	V_C	2.5	-92.4		+71.5	-20.9
18	40,000	6.5	FWD	V_C	1.0	-86.0	-49.0		-135.0
19	40,000	6.5	AFT	V_C	2.5	-39.6		+50.0	+10.4
20	40,000	6.5	AFT	V_C	1.0	-67.0	-31.2		- 98.2
21	40,000	9.2	FWD	V_C	2.5	-67.7		+69.0	+ 1.3
22	40,000	9.2	FWD	V_C	1.0	-60.3	-41.6		-101.9
23	40,000	9.2	AFT	V_C	2.5	-14.0		+50.0	+36.0
24	40,000	9.2	AFT	V_C	1.0	-40.6	-30.2		- 70.8

(1) TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L_T KN	L_{T1} KN	L_{T2} KN	Total load
25	40,000	0	FWD	V_A	2.5	-41.0		+77.3	+36.0
26	40,000	0	FWD	V_A	1.0	-25.1	-49.0		-74.1
27	40,000	0	AFT	V_A	2.5	+14.3		+48.0	+62.6
28	40,000	0	AFT	V_A	1.0	-34.1	-31.0		-65.0
29	40,000	6.5	FWD	V_A	2.5	-41.0		+78.0	+37.0
30	40,000	6.5	FWD	V_A	1.0	-25.0	-46.5		-71.6
31	40,000	6.5	AFT	V_A	2.5	+14.3		+65.3	+79.6
32	40,000	6.5	AFT	V_A	1.0	-34.1	-38.5		-72.6
33	40,000	9.2	FWD	V_A	2.5	-41.0		+78.0	+37.0
34	40,000	9.2	FWD	V_A	1.0	-25.0	-45.0		-70.0
35	40,000	9.2	AFT	V_A	2.5	+14.3		+61.2	+75.5
36	40,000	9.2	AFT	V_A	1.0	-34.1	-35.2		-69.3

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt KG	Alt Kms	C.G	Vel		L_T KN	L_{T1} KN	L_{T2} KN	Total load
1	50,280	0	FWD	V_D	2.5	-115.5		+98.5	-17.0
2	50,280	0	FWD	V_D	1.0	-101.5	-64.1		-171.6
3	50,280	0	AFT	V_D	2.5	- 49.8		+50.1	+0 .3
4	50,280	0	AFT	V_D	1.0	- 85.0	-32.9		-117.9
5	50,280	6.5	FWD	V_D	2.5				
6	50,280	6.5	AFT	V_D	1.0	-104.1	-57.8		-161.9
7	50,280	6.5	AFT	V_D	2.5				
8	50,280	6.5	AFT	V_D	1.0	- 76.48	-29.4		- 76.48
9	50,280	9.2	FWD	V_D	2.5				
10	50,280	9.2	FWD	V_D	1.0	- 71.96	-55.3		-127.6
11	50,280	9.2	AFT	V_D	2.5				
12	50,280	9.2	AFT	V_D	1.0	- 44.34	-24.7		- 69.04

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L _T KN	L _{T₁} KN	L _{T₂} KN	Total load
13	50,280	0	FWD	V _C	2.5	-95.0		+98.5	+ 3.5
14	50,280	0	FWD	V _C	1.0	-85.8	-61.0		-146.8
15	50,280	0	AFT	V _C	2.5	-28.0		+50.3	+22.3
16	50,280	0	AFT	V _C	1.0	-72.6	-32.9		-105.5
17	50,280	6.5	FWD	V _C	2.5				
18	50,280	6.5	FWD	V _C	1.0	-88.03	-56.6		-144.63
19	50,280	6.5	AFT	V _C	2.5				
20	50,280	6.5	AFT	V _C	1.0	-60.41	-30.0		- 90.41
21	50,280	9.2	FWD	V _C	2.5				
22	50,280	9.2	FWD	V _C	1.0	-61.96	-55.6		-117.56
23	50,280	9.2	AFT	V _C	2.5				
24	50,280	9.2	AFT	V _C	1.0	-34.34	-25.1		- 59.44

1.1 TABLE OF MANOEUVRE LOADING CASES IN SYMMETRIC MANOEUVRES

Case	Wt. KG	Alt Kms	C.G	Vel		L _T KN	L _{T1} KN	L _{T2} KN	Total load
25	50,280	0	FWD	V _A	2.5	-52.5		97.9	45.4
26	50,280	0	FWD	V _A	1.0	-31.5	-60.4		-91.9
27	50,280	0	AFT	V _A	2.5	+16.9		49.7	+66.6
28	50,280	0	AFT	V _A	1.0	- 4.36	-32.65		-36.01
29	50,280	6.5	FWD	V _A	2.5				
30	50,280	6.5	FWD	V _A	1.0	-31.86	-55.1		-86.96
31	50,280	6.5	AFT	V _A	2.5				
32	50,280	6.5	AFT	V _A	1.0	- 4.24	-27.5		-31.74
33	50,280	9.2	FWD	V _A	2.5				
34	50,280	9.2	FWD	V _A	1.0	-31.68	-54.0		-85.68
35	50,280	9.2	AFT	V _A	2.5				
36	50,280	9.2	AFT	V _A	1.0	- 4.25	-23.8		-28.05

TABLE 1.2

MANOEUVRING LOADS

Cases	Alt Km	C.G	Vel	Weight Kg	η	J	R	J/R	K_4	η_S rad.	η_1 rad.	L_{T_0} KN	Σ
A	0	FWD	V_D	58,900	1	5.06	2.85	1.79	.922	-.040	.046	-94.5	.85
B	6.5	AFT	V_A	58,900	1	4.66	2.71	1.72	.925	-.060	.055	-41.2	.86
C	6.5	AFT	V_A	40,000	1	3.33	2.30	1.45	.95	-.114	-.108	-50.9	.89
D	0	FWD	V_D	40,000	1	3.27	2.52	1.30	.97	-.027	-.026	-68.4	.92
E	0	FWD	V_D	50,280	1	4.48	2.63	1.59	.936	-.041	-.031	-84.1	.87
F	0	FWD	V_D	50,280	1	2.86	2.86	1.00	1.0	-.021	-.021	-55.9	.96

TABLE 1.2 (CONTINUED)

Cases	$L_{T1\eta}$ KN	L_1 KN	L_{T1} KN	K_3	H_m	K_5	η_2	$L_{T2\eta}$ KN	L_{T2}	$L_{T2\infty}$
A	-102.6	-70.8	31.8	1.13	.414	-.54	.022	60.2	116.2	51.8
B	-44.2	-41.7	2.5	1.14	.168	-.29	-.017	13.7	70.7	57.0
C	-54.3	-38.5	15.8	1.22	.296	-.71	-.081	40.6	65.3	24.7
D	-72.1	-47.5	24.6	1.27	.408	-.74	.020	54.8	74.8	20.0
E	-89.0	-64.1	24.9	1.17	.42	-.62	.022	59.2	98.5	39.3
F	-58.2	-37.4	20.8	1.38	.204	-.86	.018	50.0	57.6	7.6

TABLE 1.3

TAIL PLANE LOADS IN 1g TO ng MANOEUVRE.

Cases	1.0 steady balance			1g to 2.5g			Total (KN)		
	L_T	$L_{T\infty}$	$L_{T\eta}$	L_{T_1}	$L_{T_1\eta}$	L_{T_1}	L_T	$L_{T\infty}$	$L_{T\eta}$
A	-108.6	-108.6	0	-70.8	-102.6	31.8	-179.4	-76.8	-102.6
B	- 5.8	- 5.8	0	-41.7	- 44.2	2.5	- 47.5	- 3.3	- 44.2
C	- 34.1	- 34.1	0	-38.5	-54.3	15.8	- 72.6	-18.3	- 54.3
D	-106	-106	0	-47.5	- 72.1	24.6	-153.5	-81.4	- 72.1
E	-107.5	-107.5	0	-64.1	- 89.0	24.9	-171.6	-82.6	- 89.0
F	- 81.0	- 81.0	0	-37.4	- 58.2	20.8	-117.9	-60.2	- 58.2

TABLE 1.4

TAIL PLANE LOADS IN 2.5g TO 1g MANOEUVRE

cases	2.5g to 1g			2.5g steady			Total (KN)		
	L_{T_2}	$L_{T_2\eta}$	$L_{T_2\alpha}$	L_T	$L_{T\alpha}$	$L_{T\eta}$	L_T	$L_{T\alpha}$	$L_{T\eta}$
A	116.2	60.2	51.8	-119	- 9.5	-109.5	- 3	42.3	- 49.3
B	70.7	13.7	57.0	19.8	78	- 59.0	90.5	134.5	- 45.7
C	65.3	40.6	24.7	14.3	- 75.6	89.9	79.6	- 50.9	130.5
D	74.8	54.8	20.0	-111.0	-124.5	13.5	- 36.2	-104.5	68.3
E	98.5	59.2	39.3	-115.5	- 71.0	- 44.5	- 17.0	- 31.7	14.7
F	57.6	50.0	7.6	- 49.8	- 27.8	- 22	0.3	-20 .2	28.0

TABLE 1.5

MANOEUVRING LOADS IN LANDING

Cases	η	Vel.	CG	Weight Kg	$L_{T_{1g,2g}}$	L_{T_o}	$L_{T_{1,2}}$	$L_{T_{\eta 1,2}}$	$L_{T\alpha}$ (1g,2g)	$L_{T_{\eta}}$ (1g,2g)	$L_{T_{\alpha}}$ (1g,2g)	$L_{T\alpha}$ KN	$L_{T_{\eta}}$ KN	$L_{T\eta}$ KN
21	1	92.6	FWD	52,000	-253	-55	-41.7	-60	18.2	- 7.9	-245	-226	69.9	-294
22	1	92.6	AFT	52,000	-233	-27.7	-28.5	-43	15.3	0	-233.8	-218	-43.8	-262
23	2	92.6	FWD	52,000	-256		68.48	46	22.1	-58.4	-198	-175	-12.1	- 187
24	2	92.6	AFT	52,000	-210		46.5	24	21.6	- 9.4	-201	-179	15.5	-163

APPENDIX 2

ELEVATOR DESIGN CASES AND

BENDING MOMENT AND TORQUE DISTRIBUTION

APPENDIX 2.ELEVATOR LOADING AND DESIGN CASES.AND BENDING MOMENT AND TORQUE DISTRIBUTION.

For calculation of elevator loads the chordwise load distribution due to incidence given in A 74 - 19 and chordwise load distributions due to control deflections given in A 74 - 23 are used^{and} as the tail plane is of a super critical section the polynomial distribution of DES 545 is not used.

ELEVATOR DESIGN CASE.

From Appendix 1, Table 1 the critical case for elevator is case 21, the aircraft manoeuvring from 1.0g to 2.0g.

From BCAR D3 - 2.3 the loads are to be found for the case of symmetric manoeuvring from steady level flight conditions at airspeed equal to V_F in the flaps at take off or landing (whichever is critical) to a manoeuvring load factor of 2.0.

From Appendix 1, Table 1, the critical case for elevator is case 21, low speed manoeuvring with flaps at landing and AVW of 52,000 Kg. The loads for this case are noted below taken from Table 1.5.

$$\begin{aligned} \text{Total load steady 1.0 flight } L_{T_{1g}} &= \underline{-253,000 \text{ N}} \\ \text{Tail load due to elevator } L_{T_{1.0}} &= \underline{-7,900 \text{ N}} \\ \text{Tail load due to incidence } L_{1.0} &= \underline{-245,000 \text{ N}} \end{aligned}$$

In manoeuvre:

$$\begin{aligned} \text{Total tail load in manoeuvre} &= \underline{-41,700 \text{ N}} \\ \text{Tail load due to elevator deflection} &= \underline{-60,000 \text{ N.}} \\ \text{Tail load due to incidence} &= \underline{+18,200 \text{ N.}} \end{aligned}$$

$$\text{Total tail load due to incidence} = \underline{-226,800 \text{ N.}}$$

$$\text{Total tail load due to elevator} = \underline{-67,900 \text{ N.}}$$

$$\text{Total tail load} = \underline{-294,700 \text{ N.}}$$

From low speed chordwise load distribution due to incidence given in A 74 - 19

$$\begin{aligned} L_{T_e \alpha} &= 0.35 L_{T \alpha} = 0.35 \times -226.8 \\ &= \underline{-79.4 \text{ KN.}} \end{aligned}$$

Similarly from chordwise load distribution due to elevator deflection given in A 74 - 23

$$\begin{aligned} L_{T_e \eta} &= 0.2275 L_{T \eta} = 0.2275 \times -67.9 \\ &= \underline{-15.4 \text{ KN.}} \end{aligned}$$

$$\begin{aligned} \text{Total load on elevator} &= -79.4 - 15.4 \\ &= \underline{-94.8 \text{ KN.}} \end{aligned}$$

Choosing ultimate design factor 1.5,

$$\begin{aligned} \text{Factored load per side on elevator} &= \frac{-94.8 \times 1.5}{2} \\ &= \underline{-71.1 \text{ KN.}} \end{aligned}$$

CASE 2

Elevator load is calculated for case no. 1
High speed manoeuvre at AVW 58,9000 Kg.

$$\text{Total tail load} = L_T = \underline{-179.6 \text{ KN.}}$$

$$\text{Total incidence load} = L_{T \alpha} = \underline{-77.6 \text{ KN.}}$$

$$\text{Total elevator deflection} = L_{T \eta} = \underline{-102.0 \text{ KN.}}$$

From high speed load distribution due to incidence given in A 74 - 19, the elevator load due to incidence is $0.26 L_T$.

Therefore total load on elevator is given by,

$$\begin{aligned} L_{T_e} &= 0.26 \times (-77.6) + 0.2275 \times (-102.0) \\ &= \underline{43.4 \text{ KN.}} \end{aligned}$$

Therefore the case 21 is critical for elevator loads.

FOUR PIECE ELEVATOR SCHEME.

The layout of four piece elevator scheme is fixed after optimisation and is as shown in Fig. of Appendix 2. The calculation of Bending moments and torques are given in Table 2.1-2.4 and bending moment and torque diagrams are given in Fig. 2.4 - 2.9

CALCULATION OF BENDING MOMENT AND TORQUE.

From the symmetric loading cases investigated the critical loading case is that of low speed manoeuvring case in the flaps and forward C.G position.

$$\text{Total tail load} = \underline{-294.7 \text{ KN.}}$$

$$\text{Tail load due to incidence } L_T = \underline{-226.8 \text{ KN.}}$$

$$\text{Tail load due to elevator deflection } L_T = \underline{-67.9 \text{ KN.}}$$

From the chordwise load distribution for slow speed given in A 74 - 19 and A 74 - 23 replotted in Fig. 2.2 and Fig. 2.3 load on elevator is

$$0.35 L_{T\alpha} + 0.2275 L_{T\eta}$$

$$\begin{aligned} \text{Total load on elevator } L_{T_e} &= 0.35 \times 226.8 + 0.2275 \times 67.9 \\ &= 79.4 + 15.4 \\ &= \underline{94.8 \text{ KN.}} \end{aligned}$$

$$\text{Factored load per side} = \frac{94.8 \times 1.5}{2} = \underline{71.1 \text{ KN.}}$$

From the load distribution along the span Fig. 2.1 the percentage of area on a strip 0.355 m. at 0.8875 from = 7.06%.

$$\begin{aligned} \text{The load on this strip} &= \frac{71.000 \times 7.06}{100} \\ &= \underline{5128.3 \text{ N.}} \end{aligned}$$

The elevator position is from 0.71m
given to 7.100m from Φ

From the chordwise load distribution due to incidence at any section,

20.2 cm.²(60.5%) has C.G. 0.281 C_{EL} aft of hinge line.

10.8 cm.²(26.2%) has C.G. 0.77 C_{EL} aft of hinge line

5.0 cm.²(13.3%) has G.G 0.0575 C_{EL} fwd of hinge line

Therefore,

$$\begin{aligned}\text{Torque at any section} &= L_{T_e} \times C_{EL} (0.281 \times .605 + 0.262 \\ &\quad \times .77 - 0.133 \times 0.0575) \\ &= \frac{0.364 C_{EL} L_{T_e}}{\end{aligned}$$

From the chordwise load distribution at any section due to control deflection,

18.0 cm.²(81.8%) has C.G. 0.3325 aft of hinge line.

4.0 cm.²(18.2%) has C.G. 0.0 575 fwd. of hinge line.

$$\begin{aligned}\text{Torque at any section} &= L_{T_e} \times C_{EL} (.818 \times .3325 \\ &\quad - .182 \times .0575) \\ &= \frac{0.262 C_{EL} L_{T_e}}{\end{aligned}$$

From the loading on elevator for this case,

$$L_{T_e} = L_{T_{e\alpha}} + L_{T_{e\eta}}$$

$$L_{T_{e\alpha}} = 0.8335 L_{T_e}$$

$$L_{T_{e\eta}} = 0.1625 L_{T_e}$$

$$L_{T_e} = 0.8335 L_{T_e} + 0.1625 L_{T_e}$$

Load on strip at 0.8875 m. from $\mathcal{C} = \underline{5128.3 \text{ N.}}$

Chord aft of hinge line $C_{EL} = 1.130 \text{ m.}$

Load on strip due to incidence $L_{Te\alpha} = 0.8335 \times 5128.3$
 $= \underline{4294.7 \text{ N.}}$

Load on strip due to elevator
 deflection $L_{Te\gamma} = 0.1625 \times 5128.3$
 $= \underline{833.6 \text{ N}}$

Incremental Torque due to incidence $= 0.364 \times C_{EL} \times L_{Te}$
 $\Delta T_{\alpha} = \underline{1766.9 \text{ Nm.}}$

Incremental torque due to deflection $= \underline{246.8}$

Total incremental torque $= 1766.9 + 246.8$

$\Delta T = \underline{2013.7 \text{ Nm.}}$

TABLE 2.1SHEAR FORCE AND BENDING MOMENT ON INBOARD ELEVATOR

	STN Dist for m	Load on strap N	Hinge Reaction N	Shear N	Bending Moment Nm
	0.8875	5128.3		5128.3	0
	1.2425	5078.5		10206	1820.0
A ▲	1.3700		-20376	-10170	3121.0
	1.5975	4987.3		- 5183	807.0
	1.9525	4891.9		- 291	-1033.0
	2.3075	4767.6		4476	-1136.0
	2.6625	4676.4		9152	453.0
B ▲	2.900		-18150	- 8998	2626.0
	3.0175	4560.3		- 4436	1568.0
	3.3725	4435.9		0	0

A & B - hinge positions.

TABLE 2.2SHEAR FORCE AND BENDING MOMENT ON OUTBOARD ELEVATOR

	STN m	LOAD N	Hinge Reaction N	Shear N	Bending Moment Nm
	3.7275	4290.8		4290	0
	4.0825	4166.5		8457	1523.0
C ▲	4.3300	-	-20059	-11602	3616.0
	4.4375	3988.2		- 7614	2613.0
	4.7925	3847.2		- 3767	- 89.3
	5.1475	3656.5		- 111	-1426.0
	5.5025	3461.7		3350	-1466.0
	5.8575	3233.6		6583	- 276.0
	6.2125	2922.7		9505	-2060.0
D ▲	6.2300		-14027	-4522	2266.0
	6.5675	2549.6		1970	700
	6.9225	1969.2		0	0

C & D - Hinge positions.

2.3 TORQUE ON INBOARD ELEVATOR

STATION m	LOAD N	C_{EL} m	$L_{Te\alpha}$ N	$L_{Te\eta}$ N	$\cdot 364 L_{Te\alpha} C_{EL}$	$\cdot 262 L_{Te\eta} C_{EL}$	ΔT Nm	T Nm
0.8875	5128.3	1.130	4294.7	833.6	1766.9	246.8	2013.7	2013.7
1.2425	5078.5	1.106	4252.8	825.6	1711.7	238.8	1950.5	3964.2
A $\blacktriangle$ 1.3700								-10054.8
1.5925	4987.3	1.081	4176.6	810.7	1643.2	229.4	1882.0	- 8172.8
1.9525	4891.9	1.056	4096.2	795.6	1574.4	219.9	1793.9	- 6378.9
2.3075	4767.6	1.032	3992.3	775.2	1499.5	209.5	1709.0	- 4669.9
2.6625	4676.4	1.007	3916.1	760.2	1436.1	200.6	1636.7	- 3033.2
B $\blacktriangle$ 2.900								
3.0175	4560.3	0.983	3819.0	741.3	1366.3	190.6	1556.9	- 1476.0
3.3725	4435.9	0.958	3714.0	721.6	1295.5	180.9	1476.4	0
							14019.1	

A and B hinge positions

2.4 TORQUE ON OUTBOARD ELEVATOR

STATION m	LOAD N	C_{EL} m	$L_{Te\alpha}$ N	$L_{Te\eta}$ N	$364 L_{Te\alpha} C_{EL}$	$262 L_{Te\eta} C_{EL}$	ΔT Nm	T Nm
3.7275	4290.8	0.934	3593.5	696.5	1221.1	170.1	1391.2	1391.2
4.0825	4166.5	0.909	3489.4	677.0	1154.4	161.2	1315.6	2706.8
C ▲ 4.330								-7191.0
4.4375	3988.2	0.884	3339.9	648.0	1074.4	150.0	1224.4	-5966.6
4.7925	3847.2	0.860	3221.8	626.4	1008.1	140.8	1148.9	-4817.7
5.1475	3656.5	0.835	3062.3	594.2	930.6	129.9	1060.5	-3757.2
5.5025	3461.7	0.811	2898.6	562.1	854.4	119.2	973.6	-2783.6
5.8575	3233.6	0.786	2707.6	525.9	744.5	108.1	882.6	-1901.0
6.2125	2922.7	0.761	2447.1	475.6	667.8	94.7	762.5	-1138.5
D ▲ 6.2300								
6.5675	2549.6	0.737	2134.7	414.2	571.7	79.8	651.5	- 487.0
6.9225	1969.2	0.712	1649.0	320.2	427.3	59.7	487.0	0
							9897.8	

FIG. 21 TAIL PLANE SPANWISE LOAD DISTRIBUTION
DUE TO INCIDENCE AND ELEVATOR

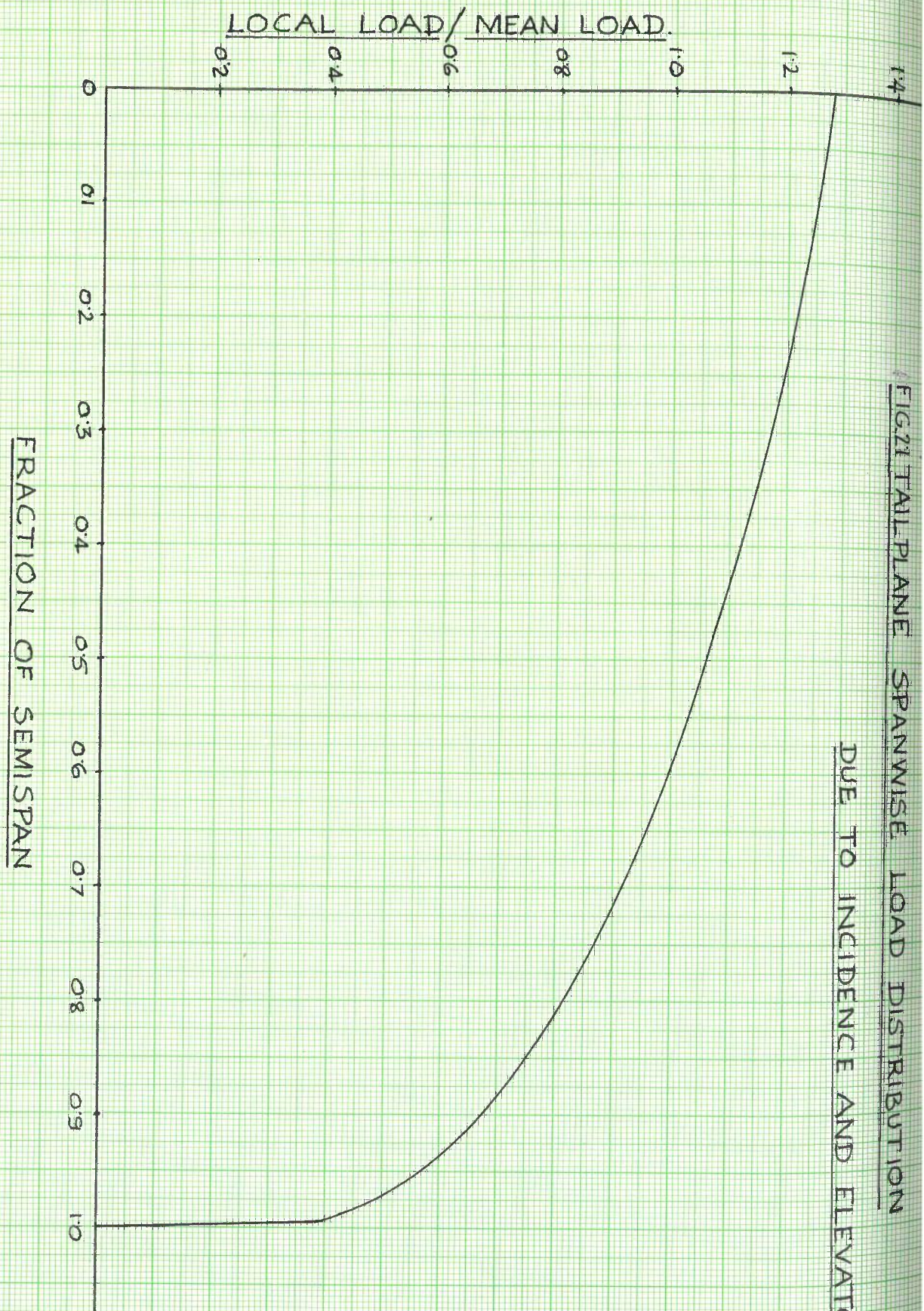


FIG 22 TAIL PLANE CHORDWISE LOAD DISTRIBUTION

DUE TO INCIDENCE

LOCAL LOAD / MEAN LOAD

2.0
1.5
1.0
0.5
0
-0.5
-1.0
-1.5
-2.0

0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0

FRACTION OF CHORD

$M=0.8$

$C_L=1.0$

$M=0.5$

$C_L=0.4$

FIG 2.3

CHORDWISE LOAD DISTRIBUTION DUE TO

CONTROL DEFLECTIONS

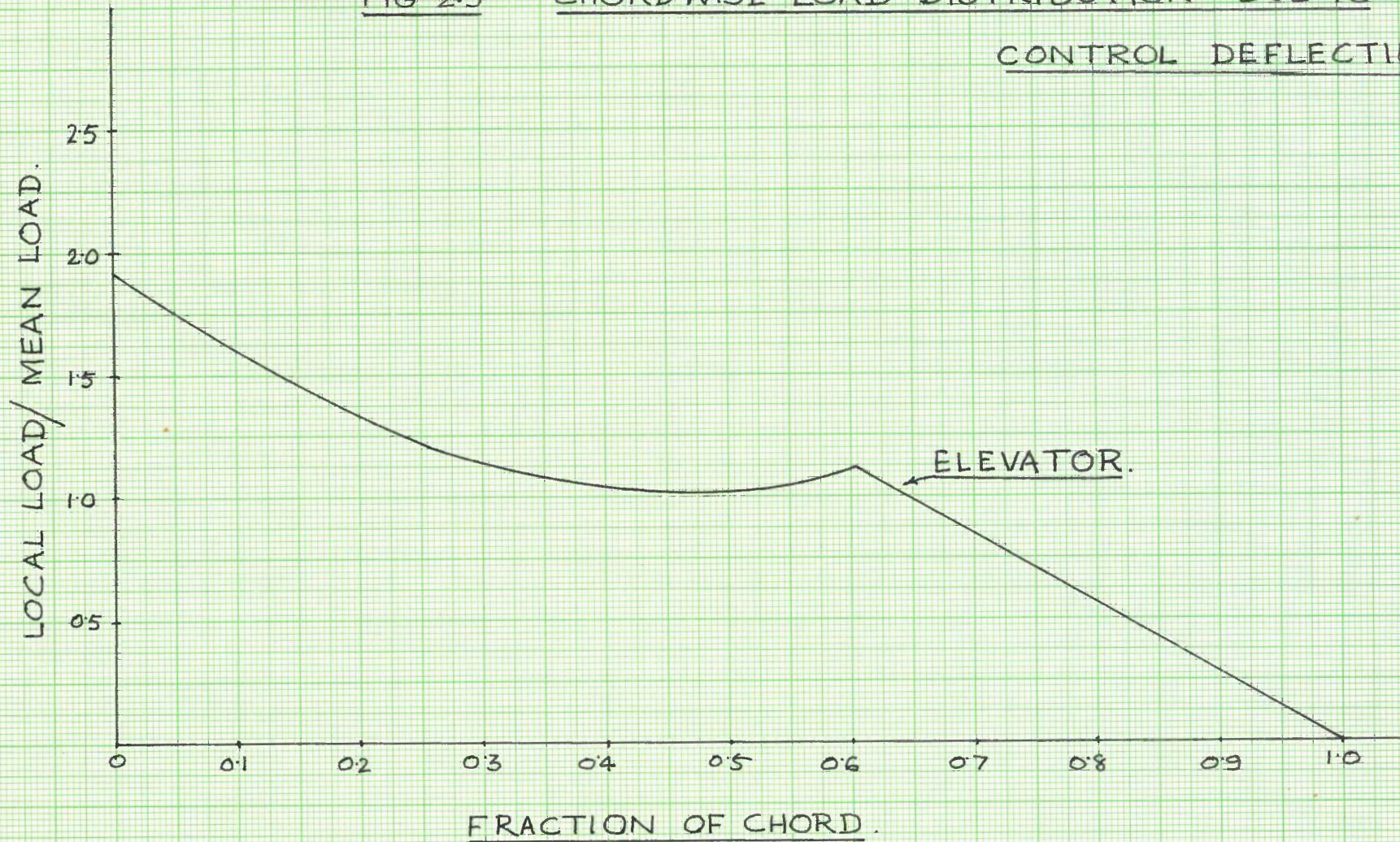


FIG 2.4 SHEAR ON INBOARD ELEVATOR

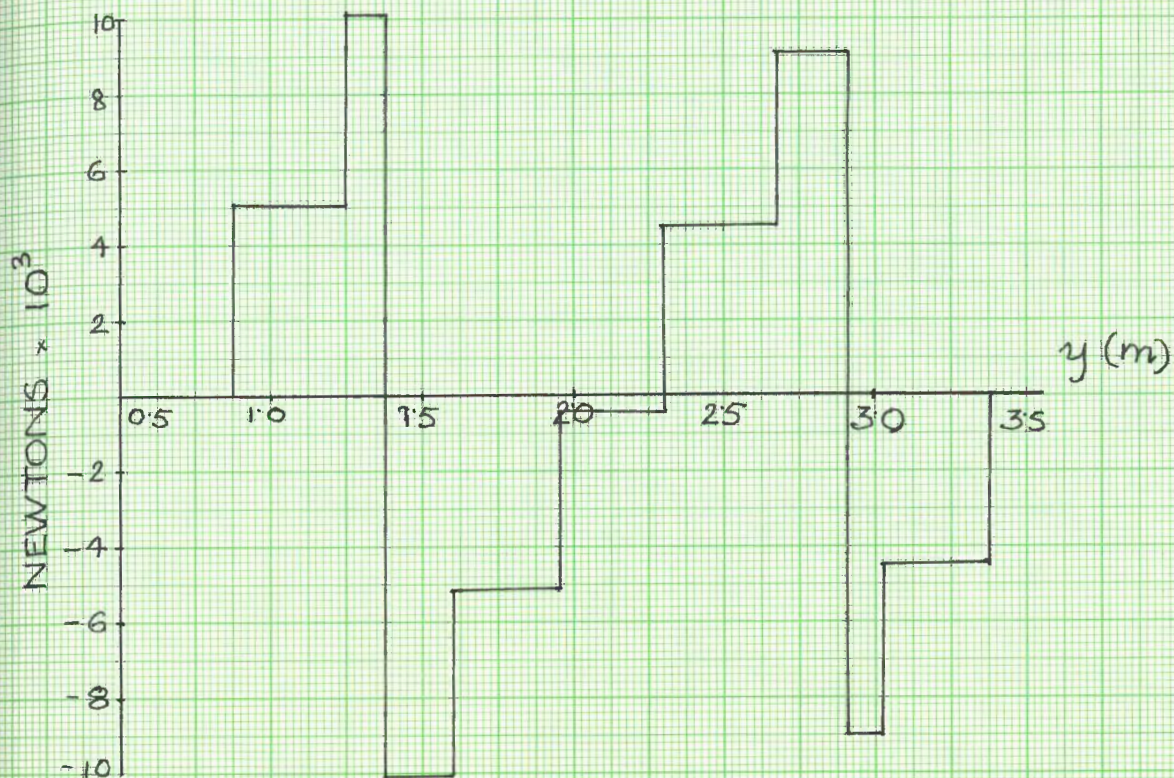


FIG 2.5 BENDING MOMENT ON INBOARD
ELEVATOR

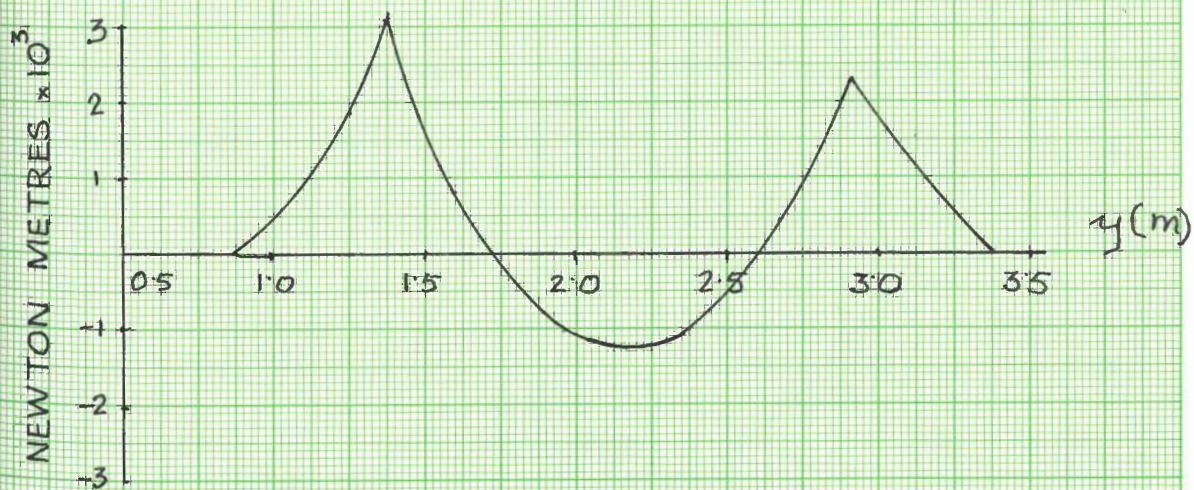


FIG. 2.6 SHEAR ON OUTBOARD ELEVATOR.

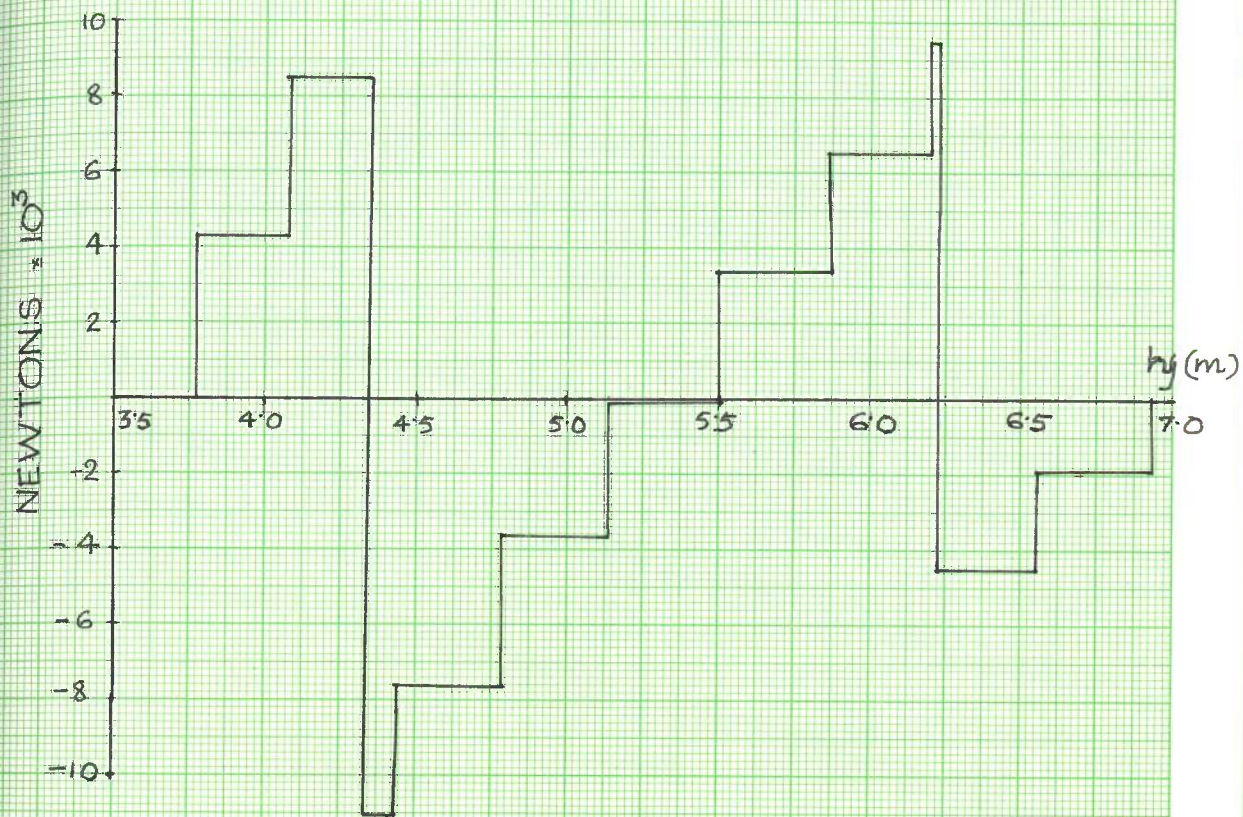


FIG. 2.7 BENDING MOMENT ON OUTBOARD ELEVATOR

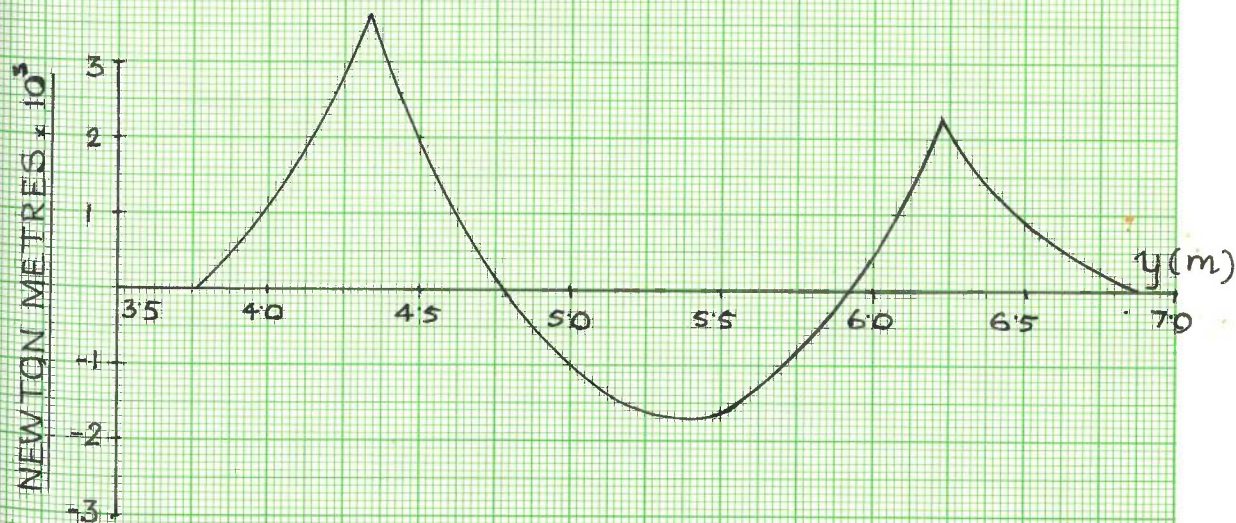


FIG 2.8 TORQUE ON INBOARD ELEVATOR

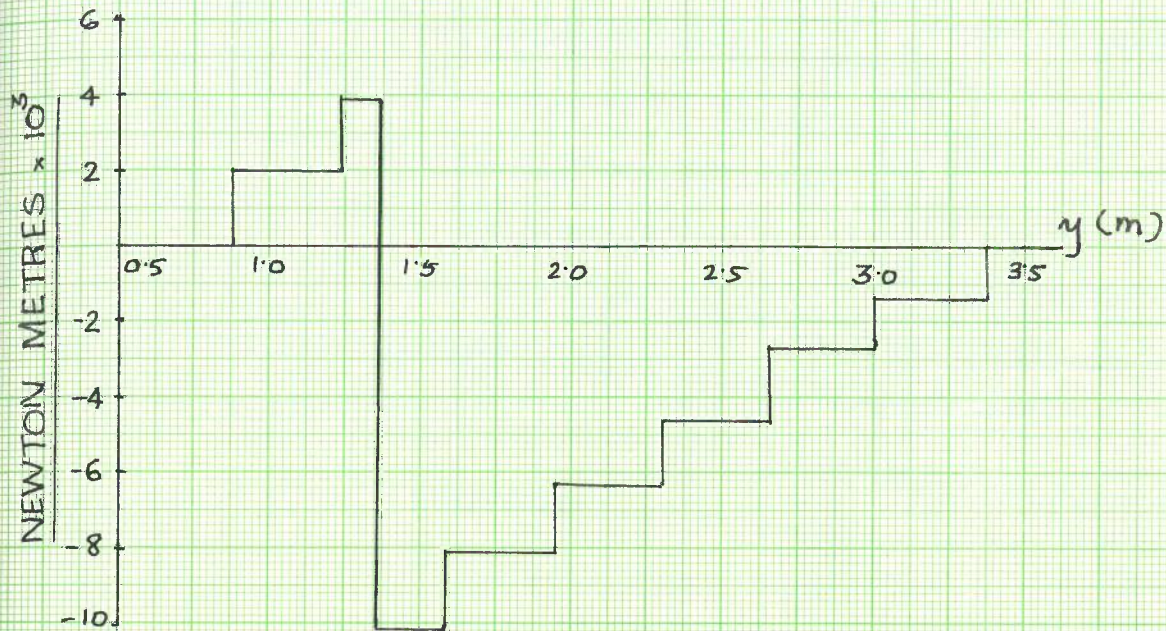
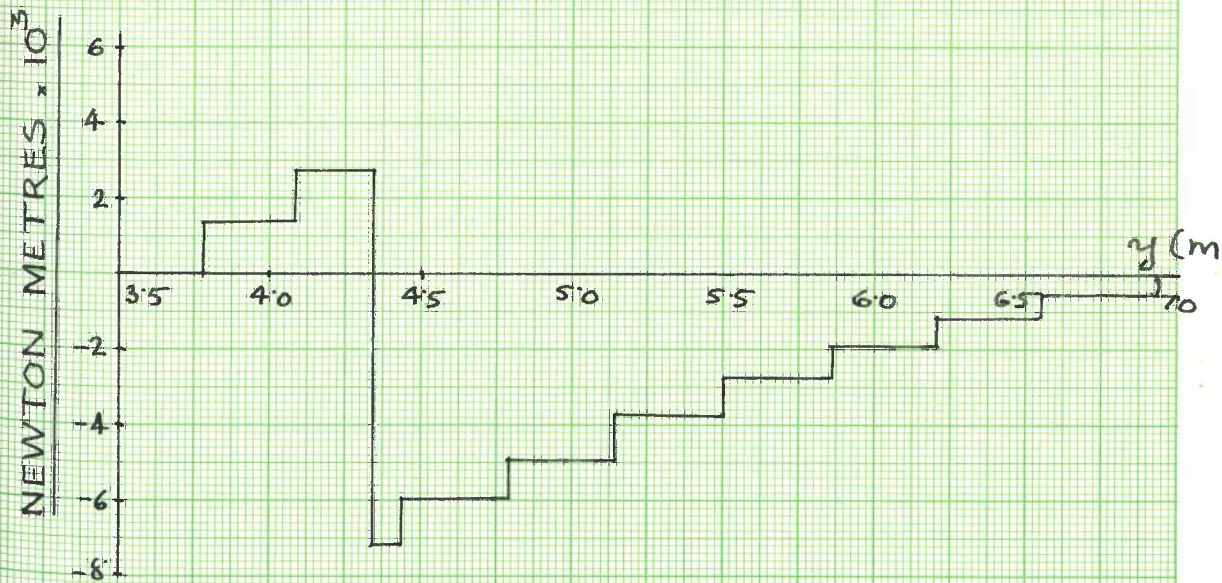


FIG 2.9 TORQUE ON OUTBOARD ELEVATOR



APPENDIX 3

SHEAR FLOW DISTRIBUTION

APPENDIX 3

SHEAR FLOW DISTRIBUTION

3.1 INTRODUCTION

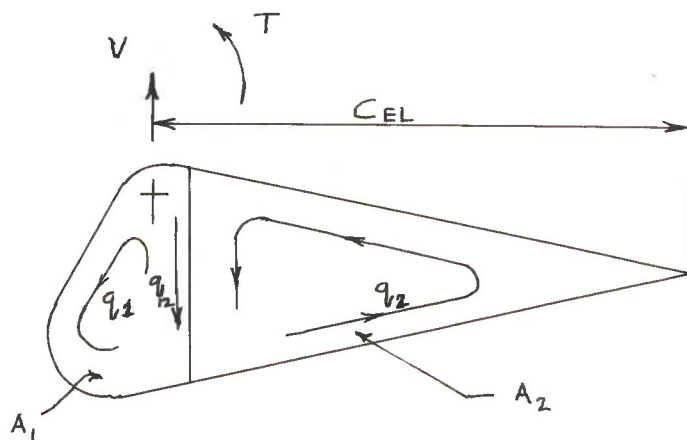
The elevator is basically a two cell box with a main spar dividing the cross section into a forward nose cell with a small trailing edge false spar. The hinge is located at the top of forward cell and spar is at constant distance $0.0625 C_{EL}$ aft of hinge line where C_{EL} is the elevator chord aft of hinge line.

Aerodynamic loading on the elevator is restored into a vertical shear force component and torque about the hinge line. This shear flow distribution in the elevator nose and aft cell depending on the thickness of sheets in the cell and section geometry. Considering the Engineering theory of torsion, the shear flows are obtained by,

- (1) Vertical equilibrium of shear.
- (2) Overall equilibrium of torque.
- (3) Equalisation of rate of twist of each cell which is compatibility condition.

It is assumed that vertical shear is carried by the spar web only. The equations are solved to obtain the shear flows q_1 , q_2 and q_{12} in the nose cell, aft cell and spar web.

3.2 TYPICAL SECTION OF ELEVATOR.



$$\text{Area of nose cell } A_1 = 0.03375 C_{EL}^2$$

$$\text{Area of aft cell } A_2 = 0.0875 C_{EL}^2$$

$$\text{Length of skin in nose cell } P_1 = 0.675 C_{EL}$$

$$\text{Length of skin in aft cell } P_2 = 1.85 C_{EL}$$

$$\text{Height of spar web } P_{12} = 0.20625 C_{EL}$$

3.3 VERTICAL EQUILIBRIUM.

Considering vertical shear as reacted by spar web,

$$q_{12} = \frac{V}{0.20625 C_{EL}} = 4.848 \frac{V}{C_{EL}}$$

3.4 OVERALL EQUILIBRIUM ABOUT SPAR.

(a) Considering only shear V,

$$2 A_1 q_1 + 2 A_2 q_2 = V \times b$$

$$2 \times 0.03375 C_{EL} q_1 + 2 \times [0.0875 C_{EL} q_2] = V \times 0.0625 C_{EL}$$

$$1.08 q_1 + 2.8 q_2 = \frac{V}{C_{EL}} \quad (1)$$

Equation the rate of twist of each cell,

$$\begin{aligned} \frac{1}{2A_1 G_1} \left[q_1 \frac{P_1}{t_1} + (q_1 - q_2 - q_{12}) \frac{P_{12}}{t_{12}} \right] \\ = \frac{1}{2A_2 G_2} \left[q_2 \frac{P_2}{t_2} + (q_{12} + q_2 - q_1) \frac{P_{12}}{t_{12}} \right] \end{aligned}$$

Substituting,

$$2.648q_1 - 2.004q_2 = 2.244 \frac{V}{C_{EL}} \quad (2)$$

Solving equations (1) & (2),

$$q_1 = 0.865 \frac{V}{C_{EL}}$$

$$q_2 = 0.0235 \frac{V}{C_{EL}}$$

(b) Torque only.

For overall equilibrium at spar,

$$2 A_1 q_1 + 2 A_2 q_2 = -T$$

$$2 \times 0.03375 C_{EL}^2 q_1 + 2 \times 0.0875 C_{EL}^2 q_2 = -T$$

$$0.0675q_1 + 0.175q_2 = -\frac{T}{C_{EL}^2} \quad (3)$$

From equation (2) for equating the twist about spar,

$$2.648q_1 - 2.004q_2 = 0 \quad (4)$$

Solving equations (3) & (4),

$$q_1 = -3.347 \frac{T}{C_{EL}^2}$$

$$q_2 = 4.422 \frac{T}{C_{EL}^2}$$

3.5 Overall shear flow in elevator cells are given below,

$$q_1 = 0.865 \frac{V}{C_{EL}} - 3.347 \frac{T}{C_{EL}^2}$$

$$q_2 = 0.0235 \frac{V}{C_{EL}} - 4.422 \frac{T}{C_{EL}^2}$$

$$q_{12} = 4.848 \frac{V}{C_{EL}}$$

APPENDIX 4

PRELIMINARY DESIGN OF ELEVATOR

APPENDIX 4PRELIMINARY DESIGN OF FOUR PIECE ELEVATOR4.1 INTRODUCTION

The plan form and geometry of elevators are known by now and is as shown in A 74 - E 1. The position of hinges are fixed by optimisation and is given in Appendix 5. The loads coming on the elevator and the bending moment and shear force distribution are done in Appendix 2. The shear force bending moment and torque distributions are calculated considering bays of arbitrary width and not on rib positions.

4.2 SKIN DEFLECTION OF AFT CELL PANEL UNDER AIR PRESSURE

It is considered reasonable to permit a deflection of 0.2% of local depth of section while designing control surfaces. In this calculation it is assumed that all pressure loading is acting on one side of elevator. The curves of deflection parameter $\frac{P}{E} (b/t)^4$ versus $\frac{\delta}{t}$ as given in

R.Ac.S. 02 - 09 - 07 is used for calculation. From Table 2.1 of Appendix 2 the maximum pressure loading is 8500 N/m².

Considering a panel of 250 mm x 1090 mm of 1.2 mm thickness L 72 Aluminium alloy material

$$\frac{P}{E} \left[\frac{b}{t} \right]^4 = \frac{8500}{72.4 \times 10^9} \left(\frac{250}{1.2} \right)^4 = 450$$

From R.Ac.S. 02 - 09 - 07

$$\frac{\delta}{t} = 2.20$$

$$\text{Skin deflection } \delta = 2.2 \times 1.2 = \underline{2.64 \text{ mm}}$$

Depth of spar at this section = 225 mm

Permissible deflection = 0.02 x 225

$$= \underline{4.5 \text{ mm}}$$

So the skin deflection for a panel of this size is within the permissible limit.

4.3 PERMISSIBLE SHEAR STRESS IN PANEL

From Table 2.1 of Appendix 2 the maximum shear and torque occurs on panel just inboard of inboard hinge of inboard elevator.

$$\text{Shear} = \underline{10206 \text{ N}}$$

$$\text{Torque} = \underline{3964.2 \text{ Nm}}$$

Elevator chord aft of hinge line $C_{EL} = 1.090 \text{ m}$

From Appendix 3 the shear flows in cell

$$\begin{aligned} q_1 &= 0.865 \frac{V}{C_{EL}} - 3.347 \frac{T}{C_{EL}^2} \\ &= \underline{-3068 \text{ N/m}} \end{aligned}$$

$$\begin{aligned} q_2 &= 0.0235 \frac{V}{C_{EL}} - 4.422 \frac{T}{C_{EL}^2} \\ &= \underline{14533 \text{ N/m}} \end{aligned}$$

For a nose skin of 0.8 mm thickness and aft skin 1.2 mm thickness

$$\begin{aligned} \text{Shear stress in nose cell} &= 3.835 \text{ MN/m}^2 \\ \text{Shear stress in aft cell} &= 12.1 \text{ MN/m}^2 \end{aligned}$$

4.4 SHEAR BUCKLING OF AFT CELL SKIN PANEL

Considering ribs are positioned at 250 mm pitch shear buckling stress for a panel of 250 mm x 1090 mm and 1.2 m thickness is calculated.

From ESDU 71005 for $\frac{b}{a} = 0.25$ $K = 5.0$

$$\frac{\tau_{crit}}{E_T} = K \left[\frac{t}{b} \right]^2$$

$$\frac{\tau_{crit}}{E_T} = 5.0 \times \left(\frac{1.2}{250} \right)^2 = 0.00014$$

$$\text{From DES } \tau_{crit} = \frac{306}{8.34} \text{ MN/m}^2$$

It is seen that the ribs have to be closer to prevent shear buckling or additional stiffeners are to be provided to reduce panel size. It is decided to provide a stiffener in the middle which has sufficient stiffness to provide simply supported condition for panel.

For $\frac{b}{a} = 0.125$ $K = 4.9$

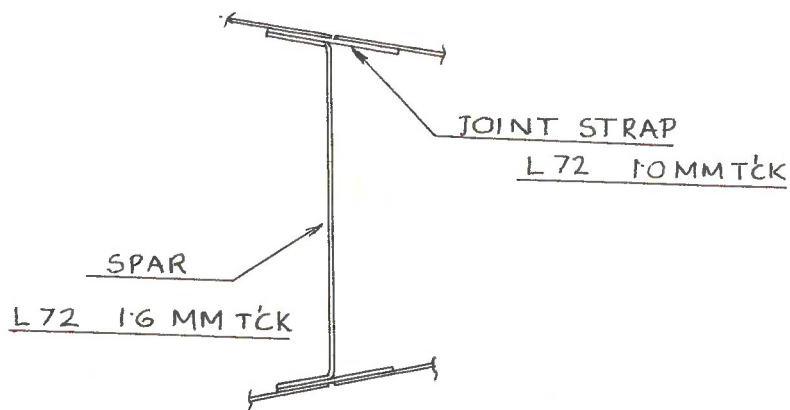
$$\frac{\tau_{crit}}{E_T} = 4.9 \times \frac{1.20^2}{125} = 0.00045$$

$$\tau_{crit} = 32.69 \text{ MN/m}^2$$

So for preliminary design the ribs at 250 mm pitch and a stiffener is provided in the middle.

4.5 SPAR

Spar is made of 1.6 mm thick L 72 sheet with 18 mm flange length. From Table 2.1 maximum bending moment at spar is 3121 Nm. Depth of spar at this section is 220 mm. Considering 30 x t width of skin is also effective in bending.



$$I_{NA} = 3.14 \times 10^{-6} \text{ m}^4$$

$$\begin{aligned} \text{Bending stress} &= \frac{3121 \times 0.11}{3.14 \times 10^{-6}} \\ &= 111.2 \text{ MN/m}^2 \end{aligned}$$

Considering one end free and one end simply supported for the flange $K = 0.58$

$$\begin{aligned} \frac{\sigma_{crit}}{E_T} &= K \left[\frac{t}{b} \right]^2 \\ &= 0.58 \times \frac{1.6}{18}^2 = 0.0045 \end{aligned}$$

From DES 7306

$$\sigma_{crit} = 180 \text{ MN/m}^2$$

Hence spar section assumed is reasonable.

APPENDIX 5

OPTIMISATION OF HINGE POSITIONS

50 OPTIMISATION OF HINGE POSITIONS.

(a) By equalising the Bending moments at the supports and in the centre of elevator.

51 INBOARD ELEVATOR.

For this optimisation a 100 KN load on the elevator is considered. This load is distributed along the span as given in A 74 - and for the sake of analysis a linear variation of load intensity is assumed between the inboard and outboard end.

$$p_t \text{ load at tip of inboard elevator} = \underline{7350 \text{ N/m.}}$$

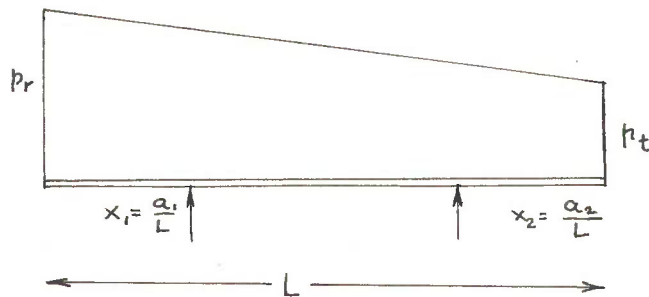
$$p_r \text{ load at root of inboard elevator} = \underline{8740 \text{ N/m.}}$$

$$p_o = p_r - p_t = \underline{1390 \text{ N/m.}}$$

$$h_r \text{ height of spar at root} = 0.255 \text{ m.}$$

$$h_t \text{ height of spar at tip} = 0.218 \text{ m.}$$

$$\Delta h = h_r - h_t = 0.037 \text{ m.}$$



Equalising the moments at hinges,

$$\frac{p_r x_1^2}{p_t x_2^2 + \frac{p_o}{3} x_2^2} = \left[\frac{h_r - \Delta h x_1}{h_t + \Delta h x_2} \right]^2$$

For $X_2 = 0.3$ and substituting the values for other terms $X_1 = 0.296$.

Calculating the hinge reactions P_1 and P_2 and Moments M_1 and M_2 .

$$\frac{P_2 (1 - X_1 - X_2)}{L} = \frac{1}{2} p_t (1 - X_1) + \frac{1}{6} p_o (1 - X_1)^3 - \frac{1}{2} p_r X_1^2$$

For $X_1 = 0.3$

$X_2 = 0.296$

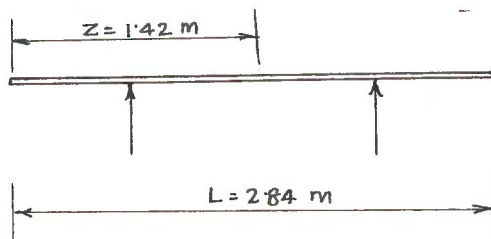
$P_2 = 10820 \text{ N}$

$P_1 = \text{Total load} - P_2$
 $= 23652.3 - 10820$
 $= 12832.3 \text{ N.}$

M_1 moment at Support A $= \frac{1}{2} p_r a_1^2 = 3308.65 \text{ Nm.}$

M_2 moment at Support B $= \frac{1}{2} p_t a_2^2 + \frac{1}{6} p_o \frac{a_2^3}{1}$
 $= 2912.9 \text{ Nm.}$

Calculation of M_3 - max. moment on between supports.



$$a_2 = 0.3 \times 2.84$$

$$\begin{aligned} M_3 &= \frac{1}{2} p_t Z + \frac{1}{6} \times p_o Z^3 - P_2 (Z - a_2) \\ &= \frac{1}{2} 7350 \times 1.42^2 + \frac{1}{6} \times 1390 \times 1.42^3 - 12832.2 \\ &\quad (1.42 - 0.852) \\ &= \underline{646.38 \text{ Nm.}} \end{aligned}$$

The moments at hinges and at the middle of span for $X_2 = 0.25$ and $X_2 = 0.2$ are tabulate in table and graph of $\frac{M_2}{M_3}$ and $\frac{I_2}{I_3}$ are plotted.

TABLE 1.

INBOARD ELEVATOR.

CASE	X_2	X_1	M_1 Nm	M_2 Nm	M_3 Nm
1	0.30	0.296	3308.6	2912.9	646.4
2	0.25	0.2495	2360.7	2016.6	7.32
3	0.20	0.2025	1510.9	1317.7	-1767.9

OUTBOARD ELEVATOR

$$\begin{aligned} h_r \text{ Height of spar at root} &= 0.218 \text{ m.} \\ h_t \text{ Height of spar at tip} &= 0.160 \\ \Delta h = h_r - h_t &= 0.058 \text{ m.} \end{aligned}$$

Loading again is assumed to be varying linearly from root to tip.

$$p_r \text{ loading at the root} = \underline{7350 \text{ N/m.}}$$

$$p_t \text{ loading at the tip} = \underline{3850 \text{ N/m.}}$$

$$p_o = p_r - p_t = \underline{3500 \text{ N/m.}}$$

TABLE 2
OUTBOARD ELEVATOR.

CASE	X_2	X_1	M_1 Nm	M_2 Nm	M_3 Nm
1	0.30	0.2756	3520.0	2381.86	2157.6
2	0.25	0.214	2112.0	1631.10	522.7
3	0.20	0.180	1505.0	1029.19	-1453.6

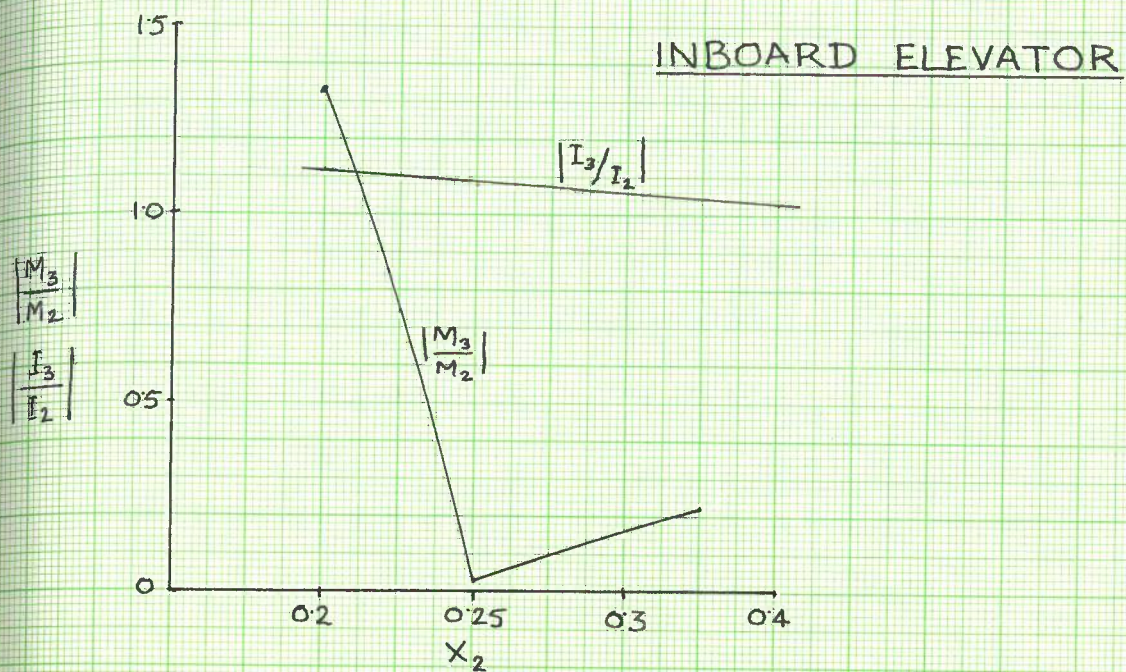
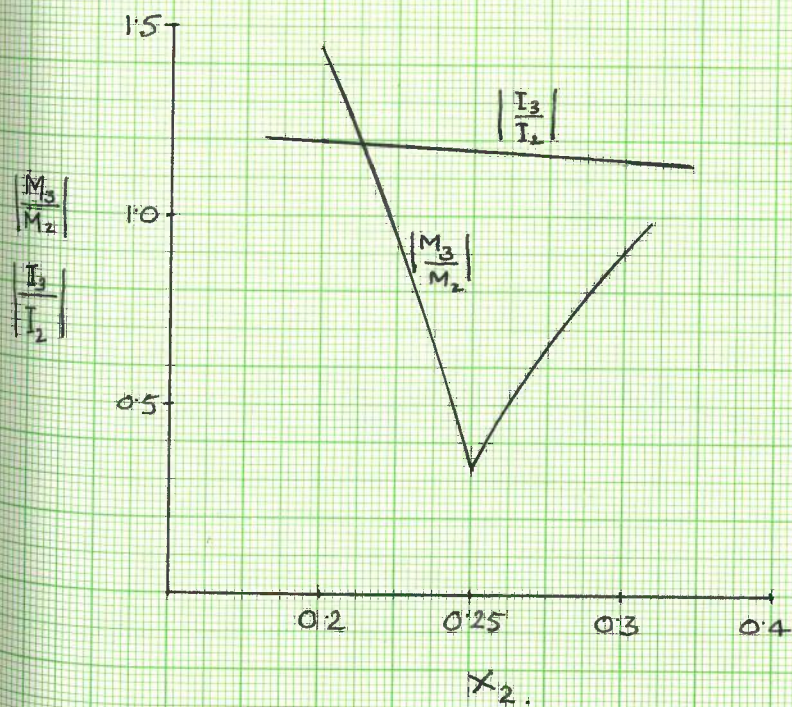
For equating the $\frac{M}{I}$ at the hinge and the centre of span, the moment of inertia is assumed to be proportional to height of spar² the values are tabulated in table 3 for inboard and outboard elevator.

TABLE 3INBOARD ELEVATOR

CASE	X_2	M_2	M_3	$\frac{M_3}{M_2}$	Ht. at hinge B (m)	Ht. at middle spar	$\frac{I_3}{I_2}$
1	.30	2912.9	646.38	0.2219	0.232	0.241	1.079
2	.25	2016.6	7.32	0.0036	0.230	0.241	1.0979
3	.20	1317.4	-1767.9	1.341	0.227	0.241	1.1272

OUTBOARD ELEVATOR

CASE	X_2	M_2	M_3	$\frac{M_3}{M_2}$	Ht. at hinge B (m)	Ht. at middle spar C	$\frac{I_3}{I_2}$
1	.30	2381.86	2157.56	0.9058	0.177	0.188	1.1281
2	.25	1631.09	522.7	0.3204	0.174	0.188	1.1674
3	.20	1029.19	-1453.6	1.4120	0.171	0.188	1.2087

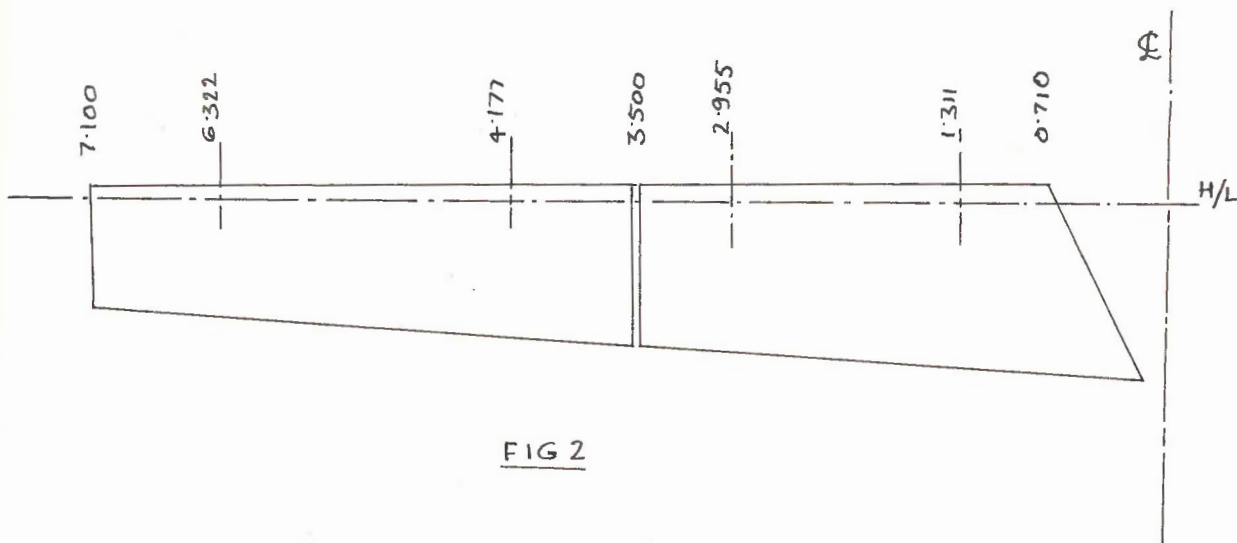
FIG 1 $\left| \frac{M_3}{M_2} \right|$ AND $\left| \frac{I_3}{I_2} \right|$ VS X_2 .OUTBOARD ELEVATOR

These values of $\frac{M_3}{M_2}$ and $\frac{I_3}{I_2}$ are plotted against the position of hinge B in Fig 1. From graph $\frac{M_3}{M_2}$ and $\frac{I_3}{I_2}$ meet at $x_2 = 0.2095$ for inboard elevator and $x_2 = 0.2105$ for outboard elevator.

Therefore for $x_2 = 0.2095$

$$\frac{p_r X_1^2}{p_t X_2^2 + \frac{p_o}{3} X_2^3} = \left[\frac{h_r - \Delta h X_1}{h_t + \Delta h X_2} \right]^2$$

Therefore on substitution, $X_1 = 0.2118$



The position of hinges for optimised $\underline{M}$ values at hinge and midspan are given in Fig 2.1

However these values were not used as a more appropriate criteria for optimisation is taken as equalisation of deflections at tip and the midspan of elevators.

52 OPTIMISATION OF HINGE POSITIONS FOR DEFLECTION.

For the calculation of deflection the load acting on the elevator is divided into strips and moment is found out along the span. Also unit load is applied at each tip and the middle and unit moment is found out at each section. $\frac{Mm}{Kd^2}$

is found at all three stations.

A number of iterations were carried out to get the deflections equal at midspan and at tips. Only the last two iterations are given in table.

INBOARD ELEVATOR ITERATION 1

STN m from	Load N	Reaction N	Shear N	Bending moment M	m_1	m_2	m_3	$\frac{d^2}{100}$	$\frac{Mm_1}{Kd^2}$	$\frac{Mm_2}{Kd^2}$	$\frac{Mm_3}{Kd^2}$
0.8875	5128		5128	0	.1775			6.81	0		
1.2425	5078		10206	1820	.5325			6.55	148.0		
▲ 1.3700		-20495	-10289	3121							
1.5975	4987		- 5302	780	.5624	.0945	.1137	6.25	70.00	11.79	14.18
1.9525	4891		- 411	-1102	.4102	.2425	.2912	5.90	- 76.82	- 45.0	-54.39
2.3075	4767		4356	-1247	.2582	.3895	.3012	5.66	- 56.88	- 85.81	-66.35
2.6625	4676		9032	300	.1060	.5370	.1237	5.38	5.91	29.00	6.89
▲ 2.9100		-18030	- 8998	2535							
3.0175	4560		- 4435	1567		.5325		5.10		163.6	
3.3725	4435		0	0		.1775		4.84		0	
									$\Sigma = \frac{91}{K}$	$\Sigma = \frac{72}{K}$	$\Sigma = \frac{-99.28}{K}$

▲ Hinge positions

INBOARD ELEVATOR LAST ITERATION

STN	Load N	Reaction N	Shear N	Bending moment M	m_1	m_2	m_3	$\frac{d^2}{100}$	$\frac{Mm_1}{Kd^2}$	$\frac{Mm_2}{Kd^2}$	$\frac{Mm_3}{Kd^2}$
0.8875	5128.3		5128.3	0	.1775			6.81	0		
1.2425	5078.5		10206	1820.0	.5325			6.55	147.96		
▲ 1.3700		-20376	-10170	3121.0							
1.5975	4987.3		- 5183	807.0	.5617	.0966	.1037	6.25	72.52	12.47	13.38
1.9525	4891.9		- 291	-1033.0	.4086	.2472	.2912	5.90	- 71.53	-43.28	-50.98
2.3075	4767.6		- 4476	-1136.0	.2539	.3982	.2962	5.66	- 50.95	-79.92	- 59.44
2.6625	4676.4		-1152	453.0	.1024	.5490	.1087	5.38	8.62	46.22	9.09
▲ 2.9000		-18150	-8998	2626.0							
3.0175	4560.3		- 4436	-1568.0		.5325		5.10		163.7	
3.3725	4435.9		0	0		.1775		4.84		0	
									$\Sigma = \frac{106.69}{K}$	$\Sigma = \frac{99.19}{K}$	$\Sigma = \frac{-87.95}{K}$

▲ Hinge positions

OUTBOARD ELEVATOR ITERATION 5

STN	Load N	Reaction N	Shear N	Bending moment M	m_1	m_2	m_3	$\frac{d^2}{100}$	$\frac{Mm_1}{Kd^2}$	$\frac{Mm_2}{Kd^2}$	$\frac{Mm_3}{Kd^2}$
3.7275	4290.8		4291	0	.1775			4.58			
4.0825	4166.5		8457	1567	.5325			4.37	190.9		
▲ 4.3300		-19984	-11527	3660							
4.4375	3988.2		- 7538	2460	.7279	.050	.0537	4.12	434.6	30.0	32.06
4.7925	3847.2		- 3691	- 200	.5813	.2153	.2312	3.92	- 29.6	- 11.0	- 11.8
5.1475	3656.5		- 35.0	-1485	.4429	.3806	.4087	3.68	-178.7	-153.58	-164.9
5.5025	3461.7		3426	-1380	.2942	.5459	.3562	3.46	-117.3	-217.72	-142.0
5.8725	3233.6		6660	- 200	.1497	.7112	.1812	3.24	- 9.2	- 43.9	- 12.0
6.2125	2922.7		9582	2184	.001	.8769	.0037	3.03	0.7	658.3	26.0
▲ 6.2200		-14100	- 4598	2275							
6.5675	2549.6		- 1969	700		.5325		2.85		130.8	
6.9225	1969.2		0	0		.1775		2.62		0	
									$\Sigma = \frac{291.4}{K}$	$\Sigma = \frac{392}{K}$	$\Sigma = \frac{-298}{K}$

▲ Hinge positions

OUTBOARD ELEVATOR LAST ITERATION

STN	Load N	Reaction N	Shear N	Bending moment M	m_1	m_2	m_3	$\frac{d^2}{100}$	$\frac{Mm_1}{Kd^2}$	$\frac{Mm_2}{Kd^2}$	$\frac{Mm_3}{Kd^2}$
3.7275	4290.8		4290	0	.1775			4.58	0		
4.0825	4166.5		8457	1540	.5325			4.37	187.5		
▲ 4.3300		-20059	-11602	3776					463.0		
4.4375	3988.2		- 7614	2613	.7280	.0035	.0537	4.12	463.0	2.1	35.0
4.7925	3847.2		- 3767	- 189.3	.5820	.212	.2312	3.92	- 13.1	- 15.0	- 5.0
5.1425	3656.5		- 111	-1426	.4450	.375	.4087	3.68	-172.5	-145.0	-158
5.5025	3461.7		3350	-1466	.2960	.538	.3612	3.46	-129.0	-228.0	-152.0
5.8575	3233.6		6583	- 276	.1530	.700	.1862	3.24	- 11.5	- 62.0	- 16.0
6.2125	2922.7		9505	2060	.0072	.863	.0087	3.03	4.9	586.0	6.0
▲ 6.2300		-14027	-4502	2266							
6.5675	2549.6		- 1970	700		.5325		2.85		130.0	
6.9225	1969.2		0	0		.1775		2.62		0	

$$\Sigma = \frac{229.3}{K} \quad \Sigma = \frac{265.0}{K} \quad \Sigma = \frac{-290.0}{K}$$

▲ Hinge positions

Optimised positions of hinge are shown in figure.

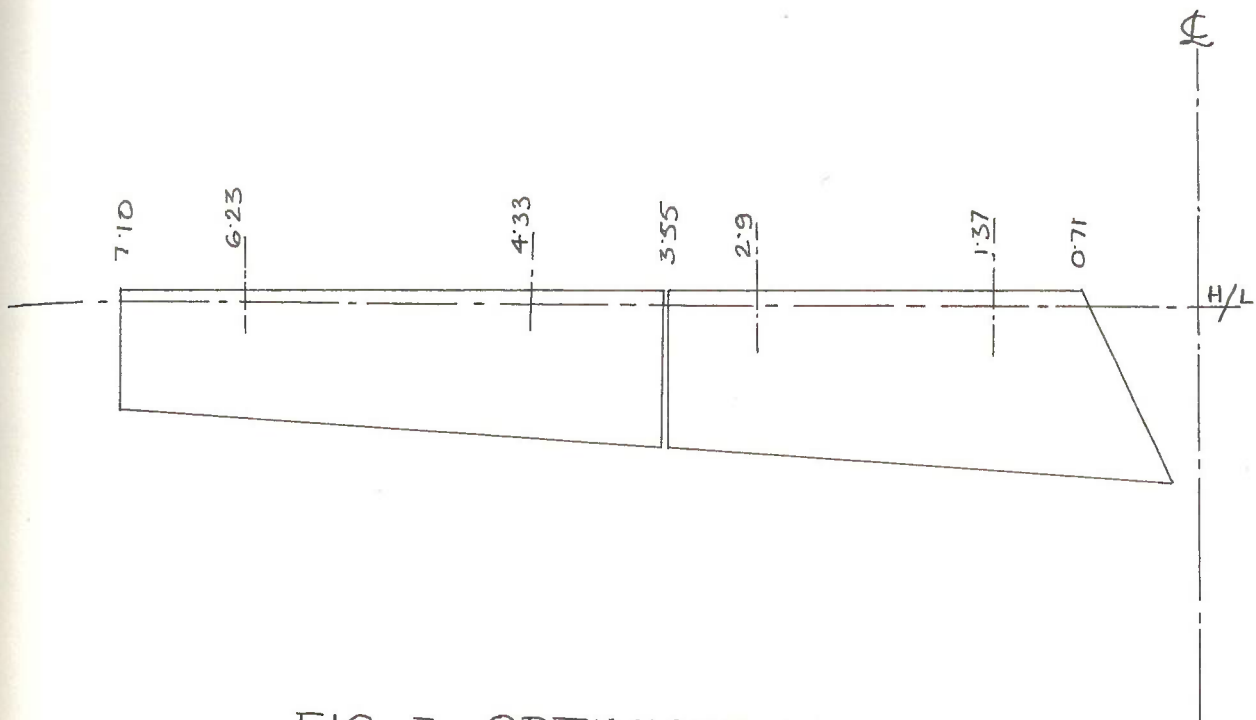


FIG. 3 OPTIMISED HINGE POSITIONS

APPENDIX 6

ELEVATOR TORSIONAL STIFFNESS REQUIREMENT

APPENDIX 6Elevator Torsional Stiffness Requirement.

Elevator is designed to meet BCAR requirements for torsional stiffness and bending stiffness of elevator over hang as given in Chapter 3 -9 Appendix 7.2 and 7.4.

6.1 Torsional stiffness criterion.

Torsional stiffness criterion is given by,

$$K = \frac{1}{V_D} \left[\frac{T_e}{b_e C_e^2} \right]^{\frac{1}{2}} (1 - M^2)^{\frac{1}{4}}$$

Where,

V_D - Aircraft design diving speed (ft./s)

T_e - Elevator torsional stiffness, measured between sections 0.05 b_e from port and starboard tips of the elevator.

b_e - Elevator span (ft.)

C_e - Geometric mean chord of elevator aft of hinge line.

From the table 5 of BCAR D 3-9 Appendix 7.2 for a plain elevator K should not be less than 0.012.

Minimum value of torsional stiffness is given by,

$$T_e = b_e C_e^2 \left[\frac{K \times V_D}{1 - M^2} \right]^{\frac{1}{4}}$$

Only the aft cell of torsion box is considered for the calculation of effective stiffness. All the dimensions are converted to feet and substituted in the equation.

$$\text{Design diving speed} = 206 \text{ m/sec} = \underline{676 \text{ ft./s}}$$

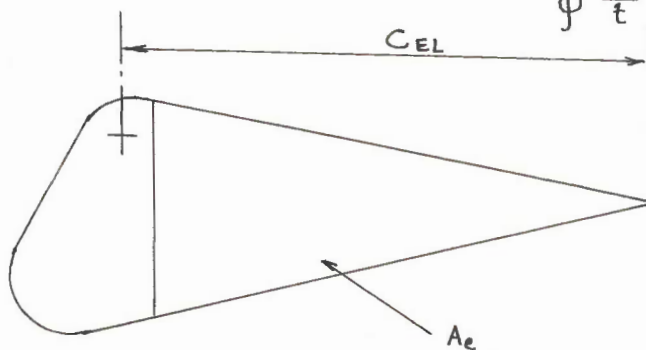
$$\text{Elevator span} = 14.2 \text{ m.} = \underline{46.5 \text{ ft.}}$$

$$\text{Geometric mean chord} = 0.945 \text{ m.} = \underline{3.1 \text{ ft.}}$$

$$\begin{aligned} T_e &= 3.1 \times 46.5 \times \frac{0.012 \times 676^2}{(1 - 0.605)^4} \quad (1) \\ &= \underline{15,200 \text{ lb. ft. / rad.}} \end{aligned}$$

Using Bredt-Batho theory of torsion,

$$\text{Polar moment of inertia } J = \frac{4 A_e^2}{\oint \frac{ds}{t}}$$



$$A_e = 0.0875 C_{EL}^2$$

$$P_e = 1.85 C_{EL}$$

C_{EL} at station y from centre line of a/c is given by,

$$C_{EL} \text{ at } y = \left[2.98 - \frac{1.23}{7.1} y \right] 0.4$$

$$\frac{t}{J} = \frac{\oint \frac{ds}{t}}{4 A_e^2}$$

$$= \frac{1.85 C_{EL}}{4 \times (0.0875 C_{EL}^2)^2}$$

$$= \frac{60}{14.2(2.98 - .173y)}$$

y	$\frac{t}{J}$
0	1.0
6.5	1.4
13.0	2.29
18.5	3.39
23.0	4.52

t/J is plotted against y in Fig. 1.

Area of under the curve from 3.5 ft from centre line to 19.5 ft. from centre line is 40.9

$$\text{Torsional stiffness } T_e = \frac{I}{\theta} = \frac{GJ}{L}$$

When

$$y = L$$

$$T_e = \frac{144 \times 4 \times 10^6 \times t}{2 \times 40.9}$$

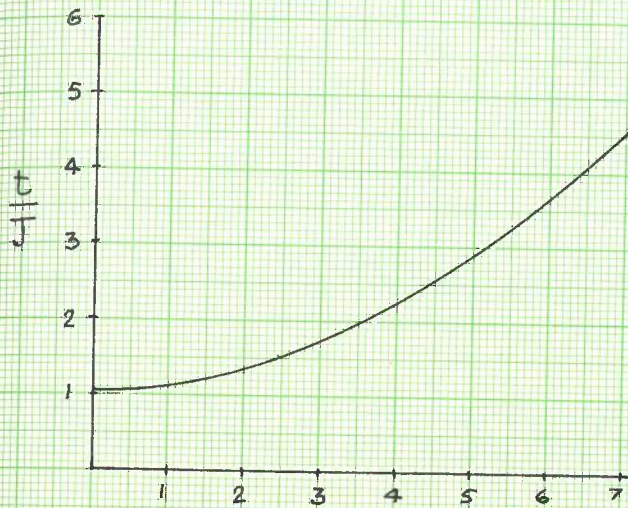
Equating this value to minimum torsional stiffness obtained in equation (1)

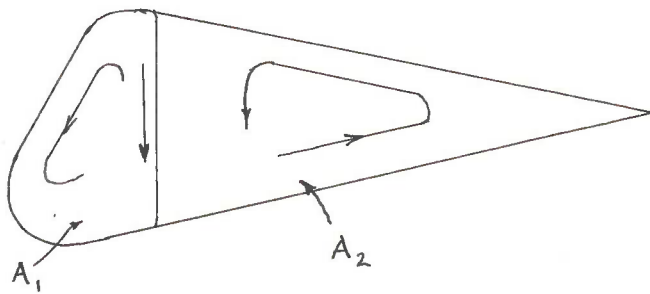
$$\begin{aligned} t_{\text{minimum}} &= \frac{15,200 \times 2 \times 40.9}{144 \times 4 \times 10} \\ &= 0.00216 \text{ ft.} \\ &= \underline{0.4 \text{ mm.}} \end{aligned}$$

The skin thickness of Aft skin panel is 1.2 mm. and this gives adequate margin against the torsional stiffness criterion.

6.2 Four-Piece Elevator.

In this case only the torsional stiffness requirement of outboard elevator is calculated.

FIG 1 $\frac{t}{J}$ Vs SEMISPANSEMISPAN - METRES



$$A_1 = 0.0216 C_{EL}^2$$

$$P_1 = 0.675 C_{EL}$$

$$A_2 = 0.0875 C_{EL}^2$$

$$P_2 = 1.85 C_{EL}$$

$$P_{12} = 0.206 C_{EL}$$

Rate of twist of cell under only torsion loading is calculated in the shear flow analysis of elevator

$$\left[\frac{d\theta}{dz} \right]_{\text{cell 2}} = \frac{1}{2A_2 G} \left[q_2 P_2 + q_2 P_{12} - q_1 P_{12} \right]$$

From shear flow analysis of elevator under torsion load,

$$q_1 = 3.576 \frac{T}{C_{EL}}$$

$$q_2 = 4.355 \frac{T}{C_{EL}}$$

$$C_{EL} = 0.83 \text{ m.} \quad \text{length } L = 3.55 \text{ m.}$$

The wall thickness of forward box is 0.8 mm. Aft box 1.2 mm. and span 1.6 mm.

On substitution the rate of twist is given as,

$$\left[\frac{d\theta}{dz} \right] = \left[\frac{1}{2 \times 0.0875 C_{EL}^2 \times 29 \times 10^9} \left(\frac{4.355 \times 1.85}{1.2 \times 10^{-3}} + \frac{4.355 \times .2065}{1.6 \times 10^{-3}} - \frac{3.576 \times .2065}{1.6 \times 10^{-3}} \right) \right]$$

Twist θ is,

$$\theta = \frac{8.16 \quad T L}{2 \times 0.0875 \times 29 \times 10^6}$$

$$\begin{aligned} \frac{T}{\theta} &= \underline{30,000 \text{ Nm/rad.}} \\ &= \underline{22,100 \text{ lb ft/rad.}} \end{aligned}$$

As against a minimum torsional stiffness required 152,000 lb ft/rad this gives adequate margin in torsional stiffness criteria.

6.3 Flexural stiffness criteria.

BCAR Chapter D 3-9 gives the flexural bending criteria,

$$K = \frac{1}{V_D} \left[\frac{F_e}{S_e' d'} \right] (1-M^2)^{\frac{1}{4}}$$

Where, F_e - Flexural stiffness of elevator overhang at outboard hinge (lb ft/rad.)

S_e' - Elevator area of overhang including area forward of hinge line (ft²)

d' - Distance from outer hinge to tip along hinge line (ft.)

For plain elevators K should be greater than or equal to 0.09

S_e' - Area of overhang = 7.45 ft.²

d' - Distance of hinge line = 2.85 ft.

V_D = 676 ft/sec

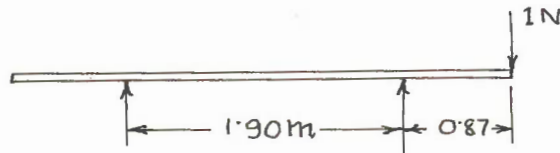
M = 0.605

Minimum value of flexural stiffness F_e

$$\begin{aligned} F_e &= \left[\frac{0.09 \times 676}{(1 - 0.605)^{\frac{1}{4}}} \right]^2 \times 7.45 \times 2.85 \\ &= \underline{123,000 \text{ lb.ft/rad.}} \end{aligned}$$

Deflection consists of 2 parts,

- (1) Deflection of overhang under unit load when considered as cantilever.
- (2) Deflection due to rotation of spar at hinge position.



$$\text{Deflection } \delta_1 = \frac{W L_1^3}{3 E I}$$

$$I = 2.1 \times 10^{-6} \text{ m}^4$$

$$\begin{aligned} \delta_1 &= \frac{1 \times 0.87^3}{3 \times 72.4 \times 10^9 \times 2.1 \times 10^{-6}} \\ &= 0.146 \times 10^{-6} \text{ m.} \end{aligned}$$

Rotation θ
at hinge

$$= \frac{M L_2}{3 E I}$$

$$= \frac{.87 \times 1.90}{3 \times 72.4 \times 10^9 \times 2.1 \times 10^{-6}}$$

$$\theta = 0.36 \times 10^{-6} \text{ rad.}$$

$$\delta_2 = L_1 \theta = 0.314 \times 10^{-6} \text{ m.}$$

$$\text{Total deflection } \delta = 0.56 \times 10^{-6} \text{ m.}$$

$$\text{Flexural stiffness} = \frac{M}{\theta}$$

$$\text{Where, } \theta = \frac{\delta}{L}$$

$$F_e = \frac{0.87 \times .87}{.56 \times 10^{-6}}$$

$$= \underline{1.35 \times 10^6 \text{ Nm/rad.}}$$

$$= \underline{1,000,000 \text{ lb ft/rad.}}$$

So there is adequate margin against the minimum flexural stiffness required.

PART II

ANALYSIS

SECTION 1

SHEAR FORCE, BENDING MOMENT AND
TORQUE DISTRIBUTION ON ELEVATOR

SECTION 1SHEAR FORCE, BENDING MOMENT AND TORQUEDISTRIBUTION ON ELEVATOR.

The procedure for calculation of shear force bending moment and torque distribution is same as it was done in Appendix 2 and specimen calculations are not repeated here. As the position of ribs are decided by this stage the load distribution on each bay is calculated. It should be noted that hinge reactions and torque are now equally divided between two ribs at each hinge. Refer drawing A 74 - E2 for details of rib and spar positions.

The calculations are given in tables 1 to 4 and the shear force, bending moment and torque distributions on inboard and outboard are plotted in Fig. A1 to A 6.

RIB STATION MM FROM ϕ	LOAD ON STRIP N	$L_{T\eta}$ N	L_{Te} N	C_{EL} M.M	$0.262 L_{T\eta} C_{EL}$ NM	$0.364 L_{Te} C_{EL}$ NM	TORQUE ΔT NM
710	2950	480	2470	1138	143	1020	1163
920	2920	475	2445	1124	140	1000	1140
1120	2908	473	2435	1105	137	980	1117
1320	1390	226	964	1100	65	386	451
1420	2862	466	2394	1090	133	950	1083
1630	3410	555	2855	1070	156	1110	1266
1880	3380	549	2831	1055	152	1085	1237
2130	3320	540	2780	1040	147	1050	1197
2380	3270	532	2738	1020	142	1015	1157
2630	3210	522	2688	1000	137	980	1117
2880	540	88	452	990	23	163	186
2920	2645	430	2215	980	113	790	903
3130	2610	424	2186	968	107	770	877
3340	2560	416	2144	950	100	742	842.
3550						Σ	13736

RIB STATION MM FROM ϕ	ΔV N	ΔT Nm	ΔV_{RIB} N	ΔT_{RIB} Nm	REACTION N	SHEAR N	BENDING MT. Nm	TORQUE Nm.
710	2950	1163	1475	581		1475	0	581
920	2920	1140	2935	1152		4410	309.7	1733
1120	2908	1117	2914	1131		7324	1191.7	2864
1320	1390	451	2149	784	-10051	-578	2656.5	3223
1420	2862	1083	2126	767	-10051	-8303	2882.1	-9327
1630	3410	1266	3136	1174		-5267	1096.4	-8153
1880	3380	1237	3395	1256		-1879	-245.3	-6897
2130	3320	1197	3350	1217		1578	-738.3	-5680
2380	3270	1157	3295	1177		3025	-393.8	-4503
2630	3210	1117	3245	1137		4773	774.5	-3366
2880	540	186	1875	652	-8966	8018	2753.9	-2714
2920	2645	903	1592	545	-8966	827	2787.1	-2169
3130	2610	887	2627	890		-6547	1412.2	-1279
3340	2560	842	2635	858		-3920	588.9	-421
3550			1285	421		-1285	0	0

WELL IS IN OUTBOARD ELEVATOR SHEAR, B.M & TORQUE

RIB STATION MM FROM $\frac{1}{2}$	LOAD ON STRIP N	$L_{T\eta}$ N	L_{Te} N	C_{EL} M.M	$0.262 L_{T\eta} C_{EL}$ NM	$0.364 L_{Te} C_{EL}$ NM	TORQUE ΔT Nm
3560							
3800	2810	457	2353	940	112	805	917
4040	2755	448	2277	920	107	760	867
4280	2705	440	2265	900	104	741	845
4380	1200	195	1005	890	45	326	371
4685	3235	525	2710	875	120	863	983
4990	3185	517	2868	865	117	840	957
5295	2955	480	2475	850	107	765	872
5600	2850	463	2387	825	100	716	816
5905	2680	436	2244	805	92	658	750
6210	2520	410	2110	790	85	606	691
6250	344	56	308	765	11	86	97
6500	1868	304	1564	750	60	426	486
6750	1715	279	1436	730	53	382	435
7000	1392	226	1166	710	42	300	342

RIB STATION MM FROM $\frac{1}{2}$	ΔV N	ΔT Nm	ΔV_{RIB} N	ΔT_{RIB} Nm	REACTION N	SHEAR Nm	B.M Nm	TORQUE Nm
3560			1405	458		1405	0	458
3800	2810	917	2782	892		4187	337.2	1350
4040	2755	867	2725	852		6912	1342.1	2206
4280	2705	845	1902	608	-9564	-750	3000.9	-1874
4380	1200	371	2217	677	-9564	-8097	2925.9	-5885
4685	3235	983	3210	970		-4887	456.3	-4915
4990	3185	957	2960	864		-1927	-1034.3	-4501
5295	2955	872	2902	844		975	-1621.9	-3207
5600	2850	816	2675	783		3650	-1324.6	-2424
5905	2650	750	2600	720		6250	-211.4	-1704
6210	2520	691	1432	394	-6414	1268	1694.9	-1310
6250	344	97	1106	292	-6413	-4040	1745.6	-1018
6500	1868	486	1791	460		-2249	736.4	-588
6750	1715	435	1552	388		-696	174.4	-170
7000	1392	342.	696	171		0	0	0

FIG 1.1 SHEAR ON INBOARD
ELEVATOR

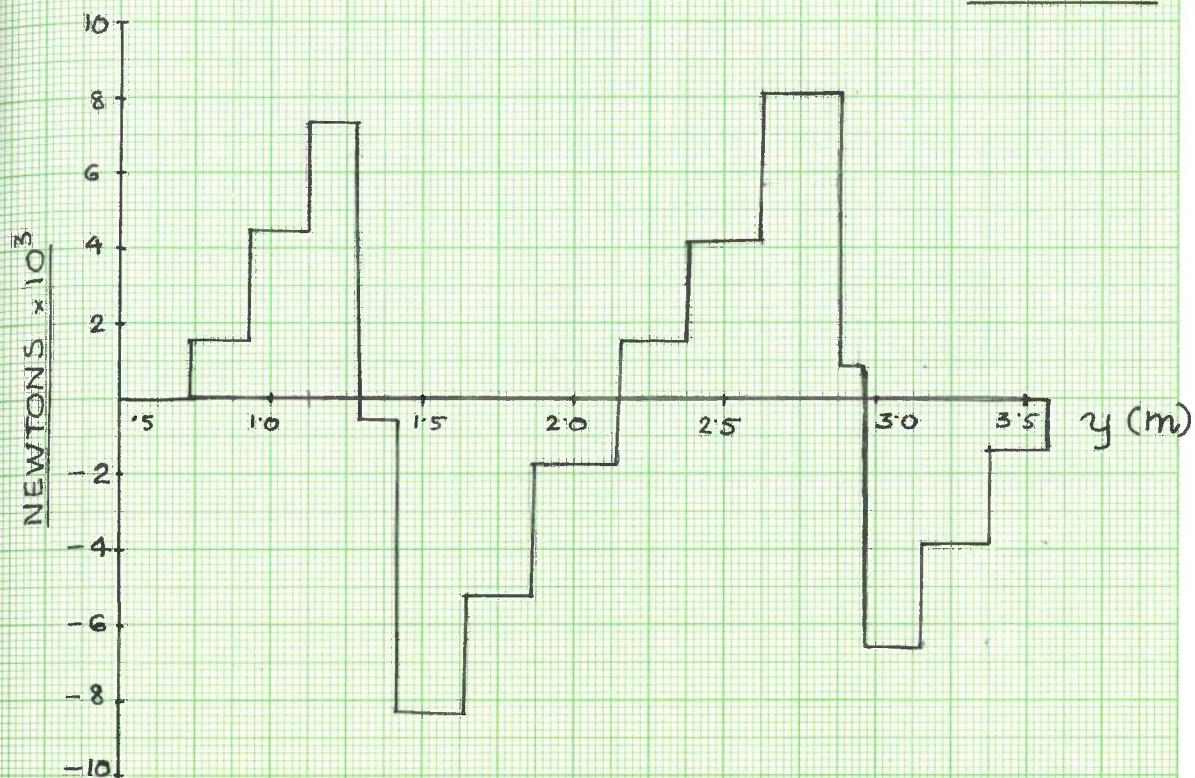


FIG 1.2 BENDING MOMENT ON
INBOARD ELEVATOR

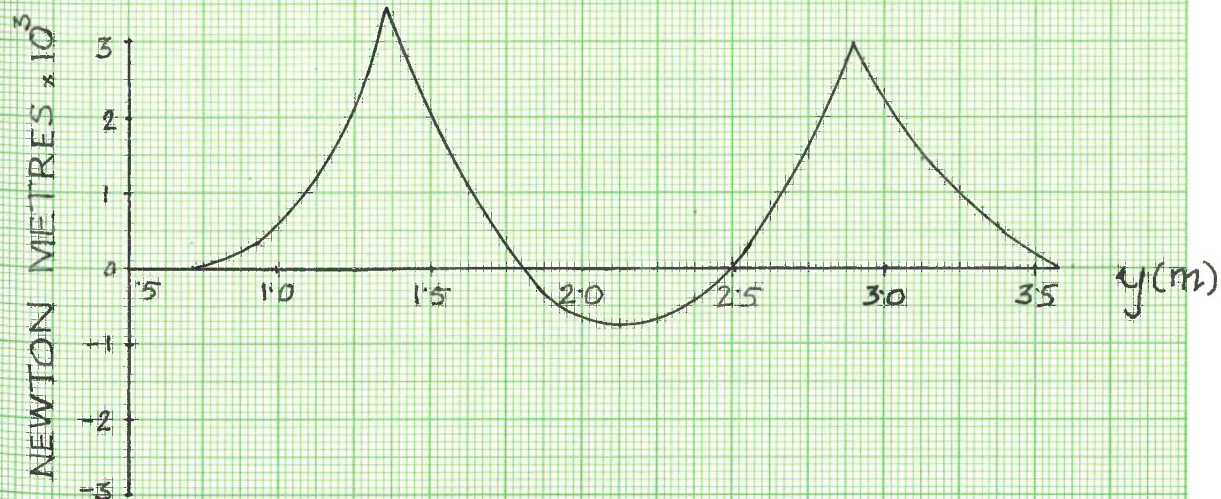


FIG 1.4 SHEAR ON OUTBOARD
ELEVATOR

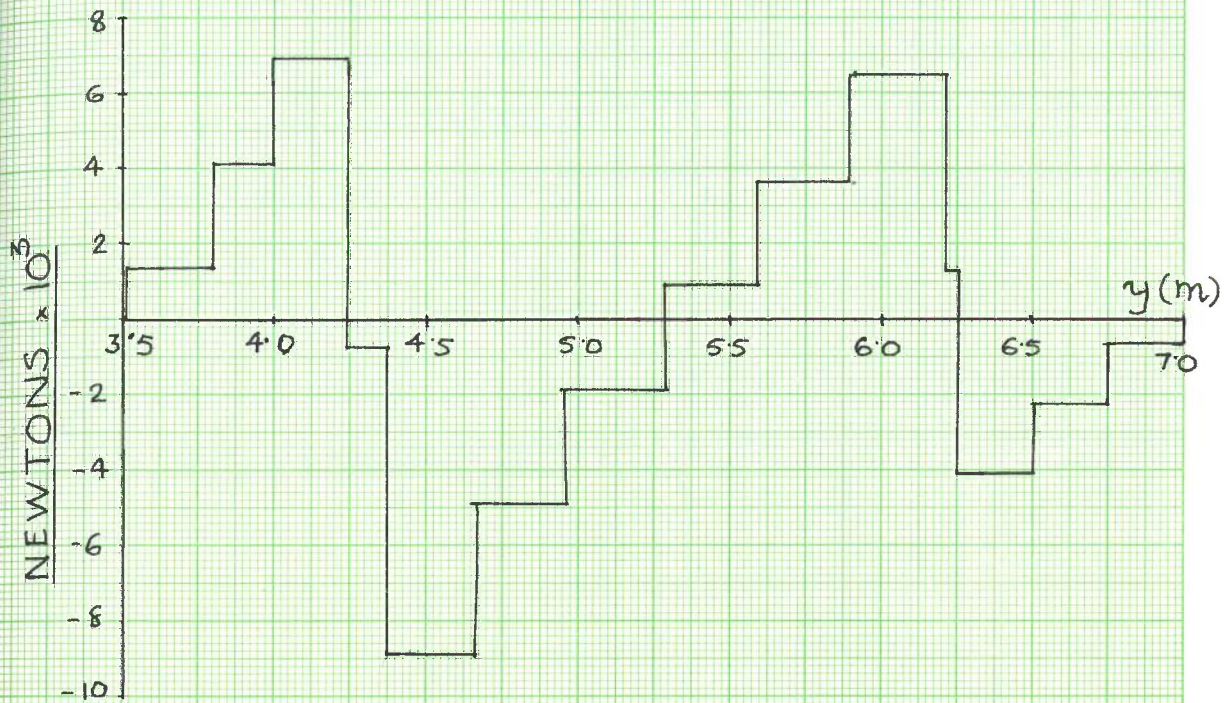


FIG 1.4 BENDING MOMENT ON
OUTBOARD ELEVATOR

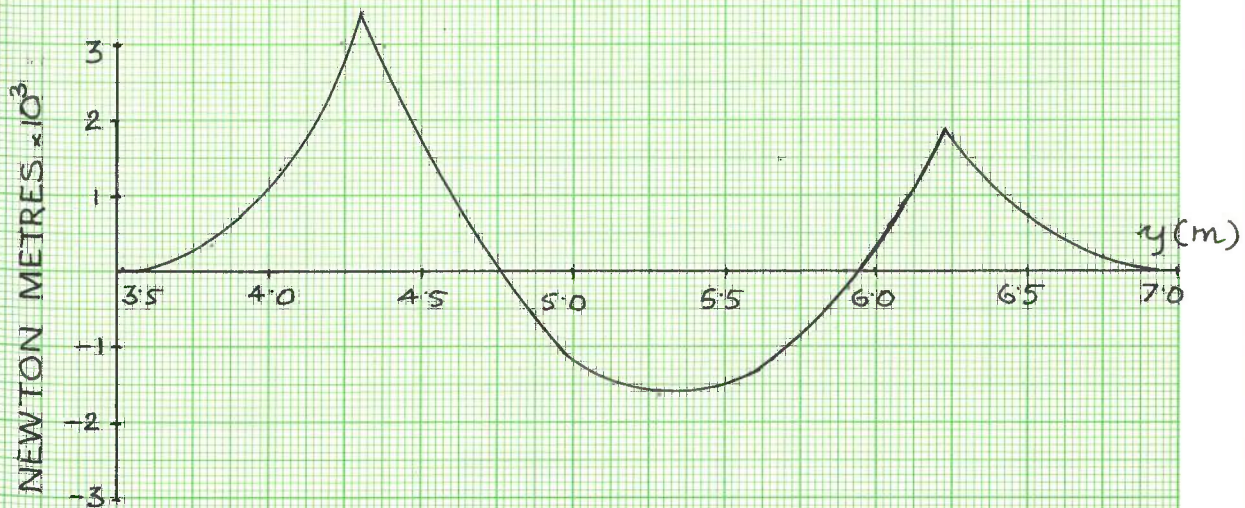


FIG 1.5 TORQUE ON INBOARD

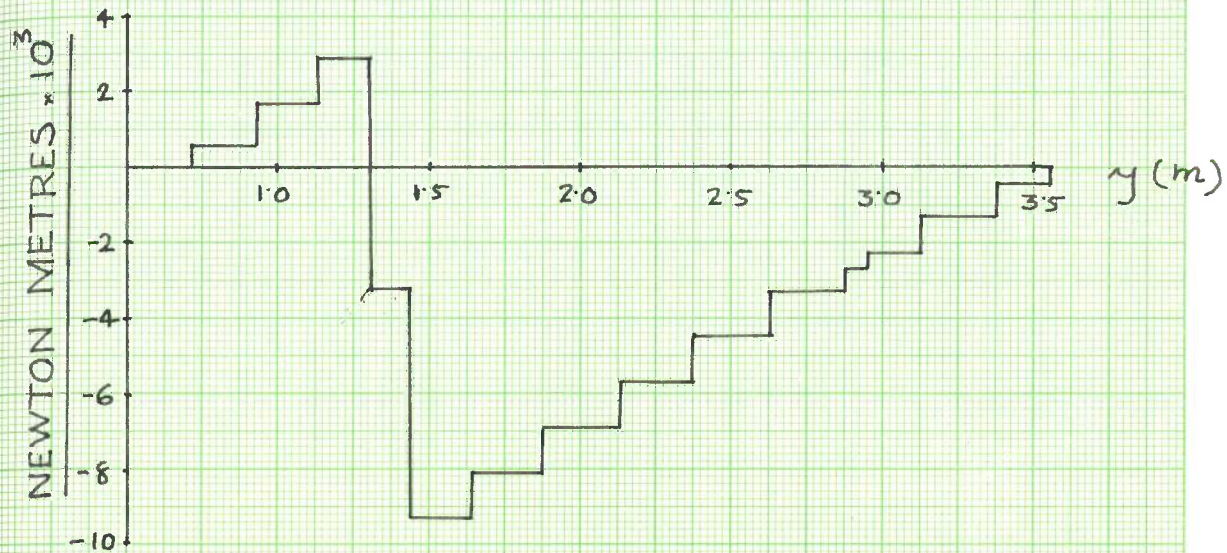
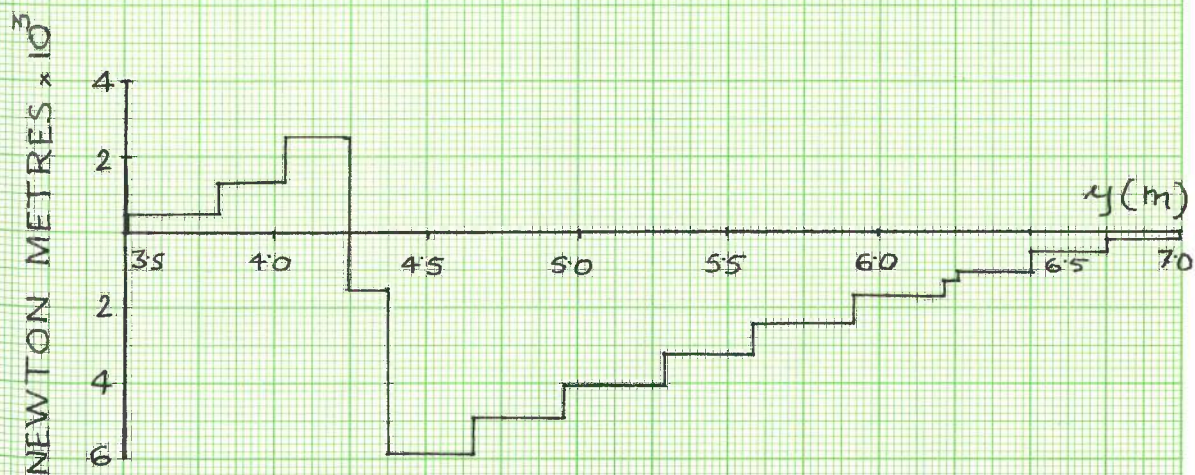
ELEVATOR

FIG 1.6 TORQUE ON OUTBOARD

ELEVATOR

SECTION 2

SHEAR FLOW DISTRIBUTION IN ELEVATOR

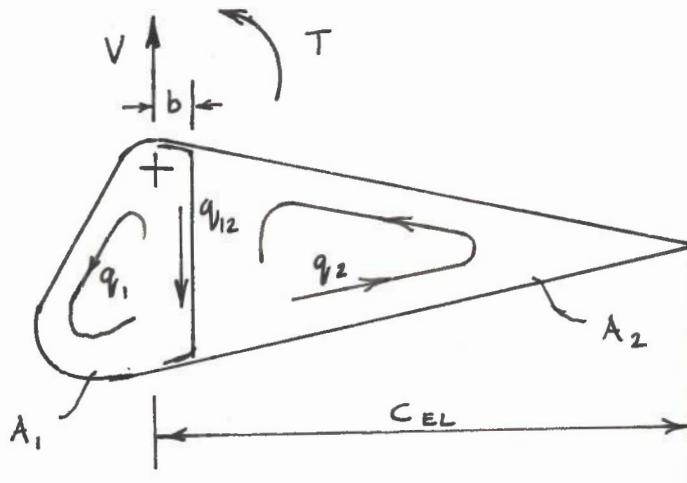
SECTION 2

SHEAR FLOW DISTRIBUTION IN ELEVATOR.

2.1 Introduction.

Shear flow analysis given in this section is done as per DES 730. As the shear flow analysis is already carried out for one set of values of skin thickness the details are not repeated.

2.2 Section geometry and sign convention.



$$\text{Area of nose cell } A_1 = 0.03375 C_{EL}^2$$

$$\text{Area of aft cell } A_2 = 0.0875 C_{EL}^2$$

$$\text{Wall length of nose cell } P_1 = 0.675 C_{EL}$$

$$\text{Wall length of aft cell } P_2 = 1.85 C_{EL}$$

$$\text{Wall length of spar } P_{12} = 0.20675 C_{EL}$$

It is assumed that vertical shear is carried by web.

2.3 CASE 1

$$t_1 = 0.8 \text{ mm.} \quad t_2 = 1.2 \text{ m.} \quad t_{12} = 1.6 \text{ m.}$$

Details of shear flow distribution for this thickness of skins is given in Appendix 3 under preliminary design of elevators. Only the values of shear flows are quoted here.

$$q_1 = 0.865 \frac{V}{C_{EL}} - 3.347 \frac{T}{C_{EL}^2}$$

$$q_2 = 0.0235 \frac{V}{C_{EL}} - 4.422 \frac{T}{C_{EL}^2}$$

$$q_{12} = 4.848 \frac{V}{C_{EL}}$$

2.4. CASE 2

$$t_1 = 0.8 \text{ mm.} \quad t_2 = 1.0 \text{ mm.} \quad t_{12} = 1.6 \text{ mm.}$$

(a) For Vertical equilibrium.

Vertical shear is carried by the spar.

$$q_{12} = \frac{V}{0.2065 C_{EL}} = 4.848 \frac{V}{C_{EL}}$$

For overall equilibrium,

$$2 A_1 q_1 + 2 A_2 q_2 = V b$$

Substituting,

$$1.08 q_1 + 2.8 q_2 = \frac{V}{C_{EL}} \quad (1)$$

Equating the rate of twist of each cell,

$$\frac{1}{2A_1G_1} \left[q_1 \frac{P_1}{t_1} + (q_1 - q_2 - q_{12}) \frac{P_{12}}{t_{12}} \right]$$

$$= \left[\frac{1}{2A_2G_2} \right] \left[q_2 \frac{P_2}{t_2} + (q_{12} - q_2 - q_1) \frac{P_{12}}{t_{12}} \right]$$

Substituting,

$$2.648 q_1 - 2.184 q_2 = 2.244 \frac{V}{C_{EL}} \quad (2)$$

Solving equations (1) and (2)

$$q_1 = 0.842 \frac{V}{C_{EL}}$$

$$q_2 = 0.0325 \frac{V}{C_{EL}}$$

(b) For Torque only,
For overall equilibrium

$$0.0675 q_1 + 0.175 q_2 = -\frac{T}{C_{EL}} \quad (3)$$

Equating the rate of twist from equation(2)

$$2.648 q_1 - 2.184 q_2 = 0 \quad (4)$$

Solving,

$$q_1 = -3.576 \frac{T}{C_{EL}}$$

$$q_2 = -4.335 \frac{T}{C_{EL}}$$

Therefore shear flow distribution is,

$$q_1 = 0.842 \frac{V}{C_{EL}} - 3.576 \frac{T}{C_{EL}}$$

$$q_2 = 0.0325 \frac{V}{C_{EL}} - 4.335 \frac{T}{C_{EL}}$$

$$q_{12} = 4.848 \frac{V}{C_{EL}}$$

Shear flow distribution in each bay is obtained on substitution of appropriate values of shear and torque.

TABLE 2.1

SHEAR FLOW IN INBOARD ELEVATOR

Panel	Shear N	Torque Nm	C_{EL} m	q_1 N/m	q_2 N/m	q_{12} N/m
1	-1475	581	1.138	- 400	- 1980	6280
2	4410	1733	1.124	-1190	- 5988	19100
3	7324	2864	1.105	-2000	-10117	32200
4	- 578	-3648	1.100			
5	-8303	-9327	1.090	19590	34421	37000
6	-5667	-8153	1.070	20020	32376	25700
7	-1879	-6893	1.050	19230	27358	8700
8	1578	-5680	1.040	18900	23235	7280
9	3025	-4505	1.020	17090	19270	14500
10	4773	-3366	1.000	15070	14085	21600
11	8018	-2714	0.990	13650	12418	24300
12	827	-2163	0.980	8308	10019	4080
13	-6547	-1279	0.968	1330	5890	33000
14	-3920	- 421	0.950	625	1963	20000

TABLE 2.2

SHEAR FLOW IN OUTBOARD ELEVATOR

Panel	Shear N	Torque Nm	C_{EL} m	q_1 N/m	q_2 N/m	q_{12} N/m
1	1405	458	0.940	-1318	-2889	2350
2	4187	1350	0.920	-4175	-7175	7170
3	6912	2206	0.900	-7000	-11973	11750
4	- 750	-1874	0.890			
5	-8097	-5885	0.875	11900	33880	-43600
6	-4887	-4915	0.865	16933	28768	-27200
7	-1927	-4501	0.850	18942	27547	-10900
8	975	-3207	0.825	16790	20827	5690
9	3650	-2424	0.805	16470	16550	21860
10	6250	-1704	0.790	16900	12236	38200
11	1268	-1310	0.765	8910	9909	32000
12	-4040	-1098	0.750	2070	-8503	-28000
13	-2249	- 588	0.731	1040	-4818	-14800
14	- 696	- 170	0.710	267	-1457	- 4720

SECTION 3

AFT CELL SKINS AND STIFFENERS

SECTION 3AFT CELL SKINS AND STIFFNERS3.1 INTRODUCTION.

Aft cell skins are analysed for shear buckling and deflection of skin under air loading. Preliminary design of elevator in Appendix has shown that shear buckling is more predominant criterion. Shear flows in skin panels are taken from Tables 2.1 and 2.2. Stiffeners are designed to provide simply supported condition for skins.

3.2 SKIN SHEAR BUCKLING.

Skin shear stresses and reserve factors are given in Tables 3.1 and 3.2. The stiffeners are assumed to provide simply supported edge conditions. Permissible shear stresses are calculated from ESDU 71005.

$$\tau_{crit} = K E_T (t/b)^2$$

$$\text{For } a/b = \frac{250}{1090} = 0.23 \quad K = 5.0$$

$$\frac{\tau_{crit}}{E_T} = 5.0 \times \left(\frac{1.2}{250} \right)^2 = 0.00014$$

$$\tau_{crit} = 8.34 \text{ MN/m}^2$$

For panels with stiffener in the middle the panel size is 125 mm x 1090 mm.

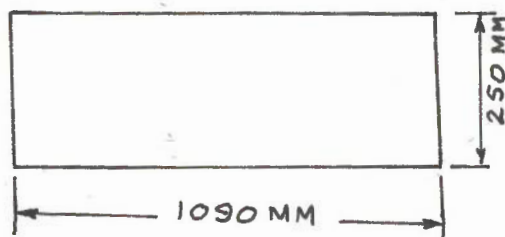
$$\frac{\tau_{crit}}{E_T} = 4.9 \times \left(\frac{1.2}{125} \right)^2 = 0.00045$$

$$\tau_{crit} = 32.69 \text{ MN/m}^2$$

3.3 SKIN DEFLECTION OF PANEL.

The skin deflection of panel under proof loading should not exceed 1/50 of depth of elevator at the station.

Considering a panel without the effect of stiffner,



L 72 1'2 MM T'ck

Maximum air pressure at the section is 8500 N/m².

Using R.Ae.Soc. 02.09.07

$$\frac{P}{E} (b/t)^4 = \frac{8500 \times 250^2}{72.4 \times 10^9 \times 1.2} = 460.1$$

$$\text{For this value of } \frac{P}{E} (b/t)^4 \quad \frac{\delta}{t} = 2.25$$

$$\text{Deflection of panel} \quad \delta = 2.7 \text{ mm.}$$

$$\begin{aligned} \text{Permissible deflection} \quad \delta_c &= \frac{1}{50} \times 220 \\ &= 4.4 \text{ mm.} \end{aligned}$$

$$\text{Reserve factor} = 1.6$$

R.F 1'6

The presence of stiffner decreases the deflection of panel. As the panel already has a margin against deflection detailed calculations are not done.

3.4 STIFFENER.

The stiffener is able to provide simply supported end conditions against shear buckling.

Considering a panel of 125 mm x 1090 mm
for $a/b = 0.11$ $K = 4.9$ from ESDU 71005

From R.Ae.Soc. 02.03.02 for $K = 4.9$
value of $\mu = 1.63$.

$$\mu = \left[\frac{I_c b}{a^2 t^3} \right]^{\frac{1}{2}} = 1.63$$

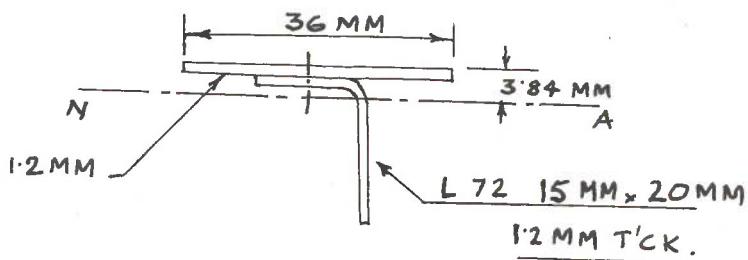
a - Panel width.

b - stiffener spacing.

$$I_c = \frac{2 \times a^2 t^3}{b} = 1.63^2 \times 250^2 \times 1.2^3$$

$$= 2180 \text{ mm}^4$$

30 times the width of skin effective in bending .



Item	A	y	Ay	y'	Ay'	I.
1	43.2	0.6	25.9	3.2	443	-
2	18	1.8	32.4	2.0	72	-
3	24	11.2	269	7.5	1350	800
	85.2		327.3		1865	
					800	
					$\Sigma = 2665$	

$$I_{NA} = 2665 \text{ mm}^4$$

Therefore stiffener moment of inertia is adequate to provide simply supported condition.

3.5 DIRECT STRESSES IN SKIN DUE TO AIR PRESSURE

When the skin deflects under air pressure direct stresses are produced in the skin.² From R. Ae. Soc. 02.09.07 the value of $\frac{p}{E} (t/b)^2$ is estimated from the value of $\frac{\delta}{t} \cdot \frac{E}{t}$

For panel 250 mm x 1090 mm the maximum deflection is found to be 2.7 mm from section 3.3

$$\frac{\delta}{t} = 2.25 \quad \text{for} \quad \frac{p}{E} (b/t)^4 = 460.1$$

$$\frac{\sigma_2}{E} (b/t)^2 = 90$$

$$\sigma_2 = 145 \text{ MN/m}^2$$

0.1% proof stress of material L 72 is 231 MN/m²

RF 1.6

Reserve factor = 1.6

3.6 ATTACHMENT OF AFT CELL SKIN TO THE FRONT STRAP

Maximum shear flow in panel is 34420 N/m from table 3.2.

$$\begin{aligned} \text{Shear load} &= 34420 \times 0.250 \\ &= 8600 \text{ N.} \end{aligned}$$

Selecting nine rivets 5/32" diameter L 69 counter sunk head, load for rivet = 950 N.

$$\text{For } \frac{D}{t} = 0.3 \quad \text{from CIT S-12-13}$$

$$K_u = 0.55 \quad K_p = 0.43$$

$$S_o = 2.56 \text{ KN}$$

$$S_u = 0.53 \times 2560 = 1360 \text{ N.}$$

$$S_p = 0.43 \times 2560 \times 1.5 \times .7 = 1150 \text{ N.}$$

Reserve factor = 1.21 1.21

3.7 INTER RIVET BUCKLING.

Using R.Ae.Soc. 02.01.08,

$$\frac{\sigma_{crit}}{E_T} = K (t/l)^2 \quad K = 1.23 \text{ for CSK}$$

$$= 1.23 \times \frac{1.2}{30}^2 = 0.00195$$

From DES 7306/2

$$\sigma_{crit} = 140 \text{ MN/m}^2$$

$$\text{Applied shear stress} = \frac{34420}{1.2 \times 10^{-3}}$$

$$= 28.5 \text{ MN/m}^2$$

Reserve factor > 2

RF > 2

TABLE 3.1INBOARD ELEVATOR AFT CELL SHEAR STRESSES

Panel	q_2 N/m	t mm	Applied shear stress MN/m ²	No. Of Stiffner	Allowable shear Stresses MN/m ²	RF
1	- 1980	1.2	1.65	0	8.34	> 2
2	- 5988	1.2	4.99	0	8.34	1.67
3	-10117	1.2	8.48	1	32.69	> 2
4		1.2		0		
5	34421	1.2	28.7	1	32.69	1.14
6	32376	1.2	26.9	1	32.69	1.21
7	27350	1.2	22.5	1	32.69	1.45
8	23235	1.2	19.3	1	32.69	1.70
9	19270	1.2	16.0	1	32.69	> 2
10	14805	1.2	12.3	1	32.69	> 2
11	12418	1.2		0		
12	10019	1.2	8.49	1	32.69	> 2
13	5890	1.2	4.91	0	8.34	1.7
14	1963	1.2	1.63	0	8.34	> 2

TABLE 3.2OUTBOARD ELEVATOR AFT CELL SKIN SHEAR STRESSES

Panel	q_1 N/m	t mm	Applied shear stress MN/m ²	No. of Stiffner	Allowable shear stress MN/m ²	RF
1	- 2289	1.2	1.91	0	8.34	> 2
2	- 7175	1.2	5.98	0	8.34	1.4
3	-11970	1.2	9.98	1	32.69	> 2
4		1.2		0		
5	33880	1.2	28.2	1	32.69	1.16
6	28760	1.2	24.0	1	32.69	1.36
7	27547	1.2	22.9	1	32.69	1.49
8	20817	1.2	17.3	1	32.69	1.89
9	16550	1.2	13.8	1	32.69	> 2
10	10250	1.2	10.2	1	32.69	> 2
11	9870	1.2		0		
12	8630	1.2	7.1	1	32.69	> 2
13	4840	1.2	4.0	0	8.34	> 2
14	1470	1.2	1.21	0	8.34	> 2

SECTION 4

NOSE CELL SKIN

DRAWING NUMBER A 74-E 2

R.F.

SECTION 4.NOSE CELL SKIN.

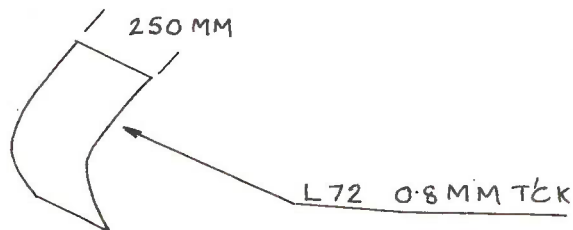
Refer Drawing A 74 - E 2

4.1 INTRODUCTION.

Analysis of nose cell is carried out considering the nose section as a curved panel. The method given in Chapter 13.6 of structural principles and data is followed. As the nose cell is not a semi-circular section a further check on shear buckling stress is carried out, considering it as a flat panel for the portion of nose cell where it is flat.

4.2 SHEAR BUCKLING.

Considering a nose cell panel on the inboard elevator,



Equivalent radius = 110 mm.

$$\frac{b^2}{Rt} = 706$$

From Fig. 13.42 B, K = 90

DRAWING NUMBER A 74 - E 2

R.F

$$\frac{\tau_{crit}}{E_T} = K (t/b)^2$$

$$\begin{aligned}\tau_{crit.} &= 90 \times 72.4 \times 10^9 \frac{0.8}{250}^2 \\ &= \underline{67 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Maximum applied stress from table 4.1} \\ &= \underline{25.0 \text{ MN/m}^2}\end{aligned}$$

$$\text{Reserve factor} \quad > 2 \quad > 2$$

Considering only the straight portion of nose cell of 100 mm. wide,

$$\begin{aligned}\text{For } a/b \text{ of } 0.4 \quad K &= 5.8 \\ \frac{\tau_{crit}}{E_T} &= 5.8 \times \frac{0.8}{100}^2\end{aligned}$$

$$\begin{aligned}\tau_{crit.} &= \underline{27 \text{ MN/m}^2} \\ \text{Max. applied stress} &= \underline{25 \text{ MN/m}^2} \quad 1.07 \\ \text{Reserve factor} &= 1.07\end{aligned}$$

However this is a very conservative assumption as the effect of curvature on panel is completely neglected. Also no allowance is made to take into account of the reduction in radius as depth of section decreased with increasing span. This again is a conservative assumption.

4.3 ATTACHMENT OF NOSE SKIN.

$$\begin{aligned}\text{Max. shear flow in panel} &= 20020 \text{ N/m} \\ \text{Total load} &= 20020 \times 0.25 \\ &= 5050 \text{ N.}\end{aligned}$$

Providing 9 numbers CSK 5/32" L 69 rivet on top and 5/32" Avdel rivet on bottom surface.

DRAWING NUMBER A 74 E 2

R.F

$$\text{Load per rivet} = \frac{5050}{9} = \underline{560 \text{ N.}}$$

From CIT S - 12 - 19 for Avdel rivets,

$$K_u = 0.42 \quad K_p = .33$$

$$S_o = 2.53 \text{ KN}$$

$$S_u = 0.42 \times 2530 = 1000 \text{ N.}$$

$$S_p = 1.5 \times 0.7 \times .33 \times 2530 = \underline{870 \text{ N.}}$$

$$\text{Reserve factor} = 1.57 \quad 1.57$$

4.4 INTER RIVET BUCKLING.

Using R.Ac.Soc. 02.01.08

$$K = 1.23 \text{ for CSK rivets.}$$

$$\frac{\sigma_{crit}}{E_T} = 1.23 \frac{0.8}{30}^2$$

$$\sigma_{crit} = \underline{63 \text{ MN/m}^2}$$

$$\text{Applied shear stress} = \underline{25 \text{ MN/m}^2} \quad \text{RF} > 2$$

$$\text{Reserve factor} > 2$$

TABLE 4.1INBOARD ELEVATOR NOSE CELL SHEAR STRESS.

Panel	q_1 N/m	t mm	τ_1 MN/m ²	τ_{crit} MN/m ²	R.F
1	- 400	0.8	0.5	67.0	> 2
2	-1190	0.8	1.49	67.0	> 2
3	-2000	0.8	2.50	67.0	> 2
4					
5	19590	0.8	24.5	67.0	> 2
6	20020	0.8	25.00	67.0	> 2
7	19230	0.8	24.1	67.0	> 2
8	18900	0.8	23.6	67.0	> 2
9	17900	0.8	21.4	67.0	> 2
10	15070	0.8	18.9	67.0	> 2
11					
12	13650	0.8	17.0	67.0	> 2
13	8306	0.8	10.0	67.0	> 2
14	1330	0.8	1.66	67.0	> 2

TABLE 4.2OUT BOARD ELEVATOR NOSE CELL SHEAR STRESSES

Panel	q_1 N/m	t mm	τ_{crit} MN/m ²	τ_1 MN/m ²	R.F
1	- 1318	0.8	67.0	1.61	> 2
2	- 4175	0.8	67.0	5.21	> 2
3	- 7000	0.8	67.0	8.75	> 2
4		0.8			
5	117990	0.8	67.0	22.5	> 2
6	16930	0.8	67.0	21.2	> 2
7	18940	0.8	67.0	23.5	> 2
8	16740	0.8	67.0	20.8	> 2
9	16490	0.8	67.0	20.5	> 2
10	16960	0.8	67.0	21.1	> 2
11		0.8			
12	2070	0.8	67.0	2.56	> 2
13	1040	0.8	67.0	1.31	> 2
14	267	0.8	67.0	0.31	> 2

SECTION 5

ANALYSIS OF NON-HINGE RIB

SECTION 5ANALYSIS OF NON-HINGE RIB.

Rib at station 1880 Inboard elevator.
Refer to Drawing A 74-E5 for details.

5.1 INTRODUCTION.

Rib at station 1880 mm. from centre line of aircraft is taken as a typical non-hinge rib for analysis. All non-hinge ribs are made of Aluminium Alloy L 72 pressings with lightening holes in the web to decrease weight and increase stiffness of the panel. The nose portion of rib is shaped as shown in the drawing. The nose cell and aft cell have separate ribs and are rivetted on to the spar.

Ribs are analysed to react incremental shear force and torque given by the steps in Table 2 of section 1. The air loading on half the bay on either side of rib is contributing to the shear force and torque increments on the rib.

5.2 LOADING AND SHEAR FLOW.

Incremental loads on the rib at station 1880,

$$\text{Shear } V = 3395 \text{ N.}$$

$$\text{Torque } T = 1174 \text{ Nm.}$$

From Section 2 the analysis of shear flow in the box is given as,

$$q_1 = 0.865 \frac{V}{C_{EL}} - 3.347 \frac{T}{C_{EL}^2}$$

$$q_2 = 0.0235 \frac{V}{C_{EL}} - 4.422 \frac{T}{C_{EL}^2}$$

$$q_{12} = 4.848 \frac{V}{C_{EL}}$$

Substituting in the equations above the value of V and T ,

$$q_1 = \underline{-750 \text{ N/m}}$$

$$q_2 = \underline{-4594 \text{ N/m}}$$

$$q_{12} = \underline{15600 \text{ N/m.}}$$

Shear flow in the web $q_w = q_{12} + q_2 - q_1$

$$q_w = \underline{11756 \text{ N/m.}}$$

5.3 MOMENT BALANCE AT THE SPAR.

$$\text{Area of nose cell} = 0.03375 C_{EL}^2$$

$$\text{Area of aft cell} = 0.0875 C_{EL}^2$$

Bending moment due to shear flow,

$$\text{Nose cell} = 2 A_1 q_1 = \underline{57.2 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

$$\text{Aft cell} = 2 A_2 q_2 = \underline{-912 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

$$\text{Net moment} = \underline{969.2 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

The load distribution on the elevator due to control deflection are plotted in Fig. 5.1. From this the bending moment due to air loading is obtained.

Bending moment due to air load at spar

$$\text{Nose cell} = \underline{-103.5 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

$$\text{Aft cell} = \underline{1071.9 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

$$\text{Net moment} = \underline{968.4 \text{ Nm.}} \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

Bending moment on rib is balanced at the spar.

5.4 SHEAR FORCE AND BENDING MOMENT DIAGRAM.

Detailed calculation of shear force and bending moment at different stations are presented in Table 1. The bending moment and shear force diagrams are presented in Fig. 5.2 and Fig. 5.3

TABLE 5.1

SHEAR FORCE DISTRIBUTION ON RIB.

STATION % CHORD	h m	AIRLOAD % ON STRIP	AIRLOAD N	SHEAR N	$-q_u h$	$-q_w h_s$	$h_s - h$ m	$q_w (h_s - h)$	Net shear N
0.575	0.22	13.8	468	468	165				633
▲ 0.625	0.215	14.3	485	953	161	-2526	0	0	$\frac{11114}{-1412}$
0.675	0.184	14.15	480	1433	161	-2526	0.031	-142	-1074
0.725	0.148	14.00	475	1908	161	-2526	0.067	-308	-765
0.775	0.113	13.70	465	2373	161	-2526	0.102	-468	-400
0.825	0.085	13.00	442	2815	161	-2526	0.130	-596	-146
0.875	0.059	9.40	319	3134	161	-2526	0.156	-717	52
0.925	0.33	5.65	192	3326	161	-2526	0.182	-836	125
0.975	0.009	1.95	69	3395	161	-2526	0.206	-940	0

▲ Spar position.

TABLE 5.2BENDING MOMENT DISTRIBUTION ON RIB.

Station % of	NetShear N	B.M due to shear Nm	BM due to air loading	Net BM Nm
0.575	633	35.0	9.5	44.5
▲ 0.625	$\frac{1114}{-1412}$	$\frac{57.2}{-912.0}$	$\frac{103.5}{1071.5}$	$\frac{160.7}{159.5}$
0.675	-1074	-660	731.9	71.9
0.725	- 765	-468	460.6	- 7.4
0.775	- 400	-219	262.7	-43.7
0.825	-146	-182	123.7	-58.3
0.875	52	- 84.5	44.9	-39.6
0.925	125	- 19.2	9.4	- 9.8
0.975	0	0	0	0

▲ SPAR POSITION

LIGHT RIB AT STN 1880

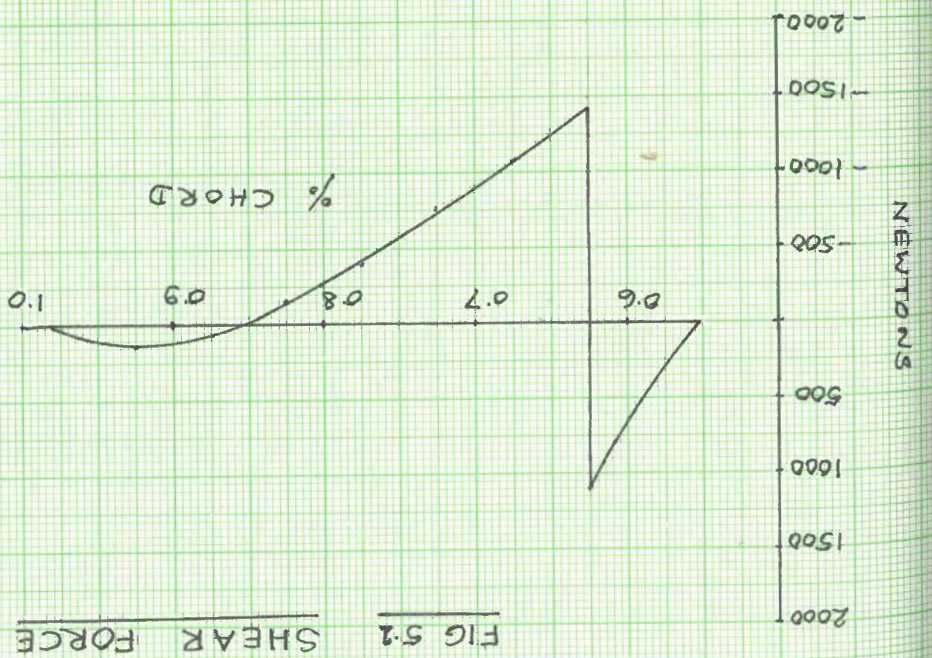
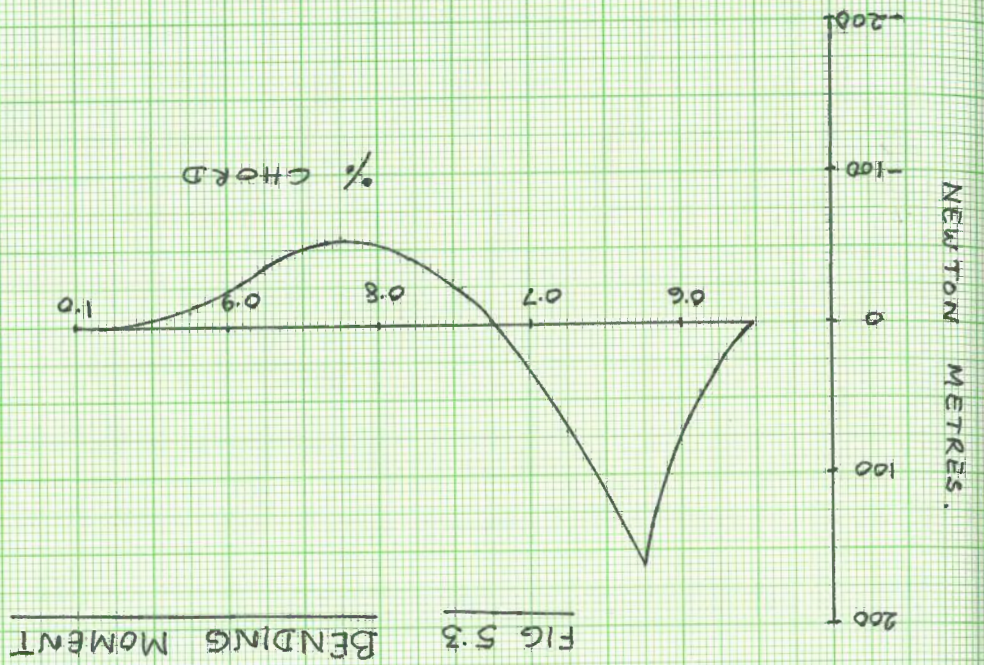
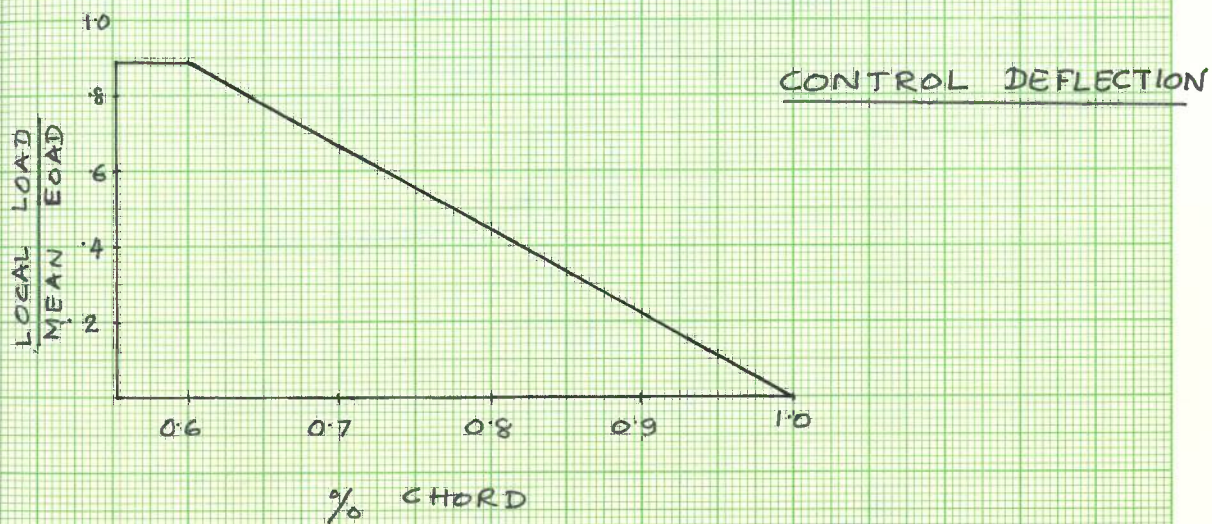


FIG 5.1



DRAWING NUMBER A 74 - E 5

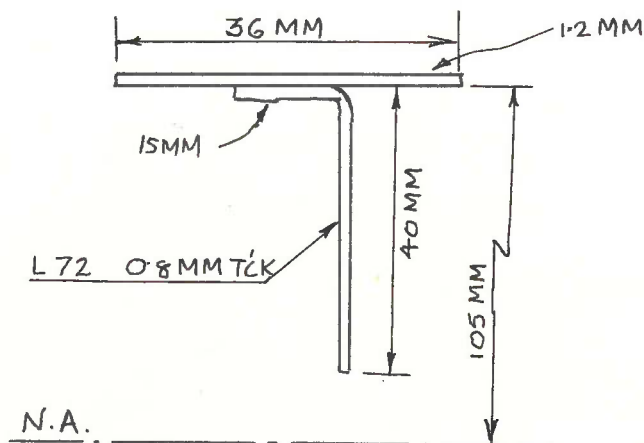
R.F.

Analysis of rib in bending.Moment of inertia of aft rib in bending.

For analysis it is assumed that 30 times the thickness of skin is effective in bending.

$$30 \times t = 30 \times 1.2 = 36 \text{ mm.}$$

$$\text{Depth of web at spar} = 210 \text{ mm.}$$



Item	A mm ²	y mm	A y ² mm ⁴	I ₀ mm ⁴
1	36	105	3.47 x 10 ⁵	-
2	12	104	1.30 x 10 ⁵	-
3	32	80	2.05 x 10 ⁵	5.3 x 10 ³
			7.32 x 10 ⁵	
			.053 x 10 ⁵	
			<u>7.373 x 10⁵</u>	
			$I_{NA} = 2 \times \Sigma$	

DRAWING NUMBER A 74 - E 5

R.F.

$$\begin{aligned}
 I_{NA} &= 2 \times 7.37 \times 10^5 \text{ mm}^4 \\
 &= \underline{1.474 \times 10^{-6} \text{ m}^4}
 \end{aligned}$$

Maximum bending moment in the aft rib occurs at spar section. From Table 5-2 Bending moment is 160.7 Nm.

$$\begin{aligned}
 \text{Bending stress in flange} &= \frac{160.7 \times 0.105}{1.474 \times 10^{-6}} \\
 &= \underline{10.9 \text{ MN/m}^2}
 \end{aligned}$$

5.4 Check against local instability of rib flanges.

$$\text{Flange thickness} = 0.8 \text{ mm.}$$

$$\text{Flange width} = 15 \text{ mm.}$$

One end is assumed free and other end is simply supported.

$$\sigma_{\text{crit.}} = K E_T (t/b)^2$$

$$K = 0.58 \text{ for above condition.}$$

$$\begin{aligned}
 \frac{\sigma_{\text{crit.}}}{E_T} &= 0.58 \times \frac{0.8}{15}^2 \\
 &= 0.00165
 \end{aligned}$$

From DES 7306/2

$$\sigma_{\text{crit.}} = \underline{120 \text{ MN/m}^2}$$

$$\begin{aligned}
 \text{Applied stress} &= \underline{10.9 \text{ MN/m}^2} \\
 \text{Reserve factor on flange instability} &
 \end{aligned}$$

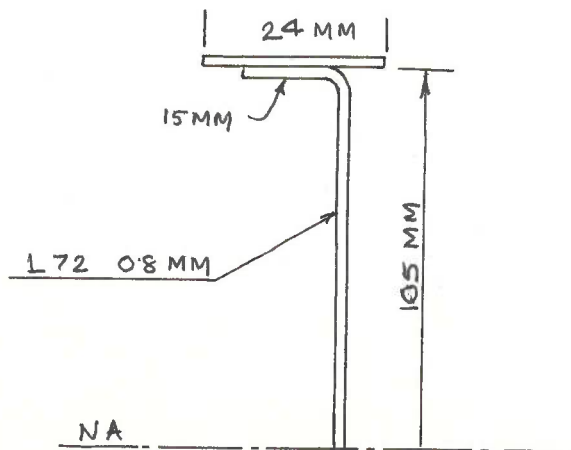
> 2

DRAWING NUMBER A 74 - E 5

R.F.

Nose cell rib at station 1880.

This rib also has same thickness 0.8mm. and 15 mm. flange. There are no lightening holes on nose rib. Moment of inertia is calculated as before.



Item	A mm ²	y mm	Ay ² mm ⁴	I _o mm ⁴
1	24	105	2.65x10 ⁵	-
2	12	104	1.30x10 ⁵	-
3	32	50	0.80x10 ⁵	0.36x10 ⁵
$\begin{array}{r} 4.75 \\ .36 \\ \hline 5.11 \times 10^5 \end{array}$				

$$\begin{aligned}
 I_{NA} &= 2 \times 5.11 \times 10^5 \text{ mm}^4 \\
 &= \underline{1.022 \times 10^{-6} \text{ m}^4}
 \end{aligned}$$

$$\begin{aligned}
 \text{Bending stress in flange} &= \frac{160.7 \times 0.105}{1.022 \times 10^{-6}} \\
 &= \underline{16.4 \text{ MN/m}^2}
 \end{aligned}$$

DRAWING NUMBER A 74 - E 5

R.F.

From section 5.4 allowable stress for local instability is 120 MN/m.

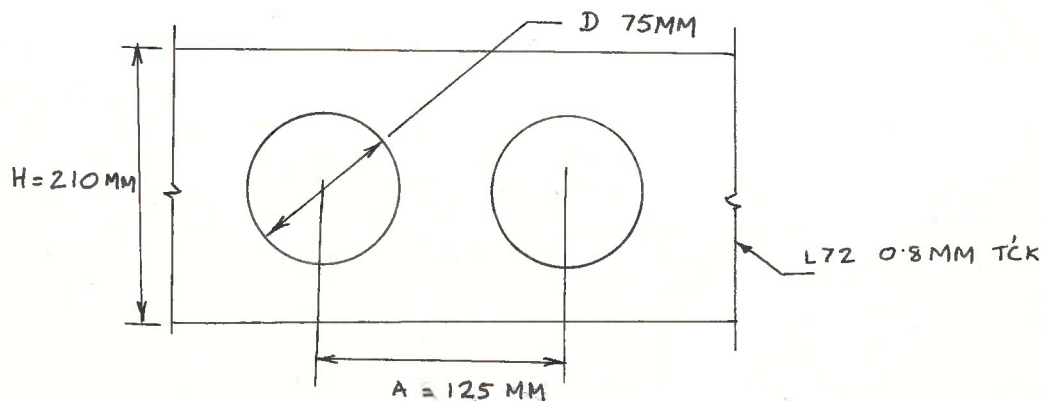
Reserve factor on flange instability

> 2

5.6 Shear buckling of aft rib.

Shear web analysis is carried out as per NLL Report S 446 by M. Botman.

(National Luchvaart Laboratorium)



$$\frac{H}{t} = \frac{210}{0.8} = 260 \quad \frac{A}{H} = \frac{125}{210} = 0.6$$

$$\frac{D}{A} = \frac{75}{125} = 0.6$$

Buckling stress for a plain panel without lightening holes = $2.8 \times 10^3 \text{ lb/in}^2$

From Fig. 2 of the report

$$\frac{\tau'_b}{\tau_b} = 1.1$$

DRAWING NUMBER A 74 - E 5

R.F.

$$\begin{aligned}\tau_{crit} &= 1.1 \times 2.8 \times 10^3 \text{ lb/m}^2 \\ &= 3060 \text{ lb/m}^2 = \underline{21.1 \text{ MN/m}^2}\end{aligned}$$

$$\text{Shear flow in rib} = 4594 \text{ N/m}$$

$$\begin{aligned}\text{Applied shear stress} &= \frac{4594}{0.8 \times 10^{-3}} \\ &= \underline{5.75 \text{ MN/m}^2}\end{aligned}$$

$$\text{Reserve factor} = \frac{21.1}{5.75}$$

> 2

5.7 Shear buckling of nose rib.

$$\text{Shear flow in rib} = 750 \text{ N/m.}$$

$$\begin{aligned}\text{Applied shear stress} &= \frac{750}{0.8 \times 10^{-3}} \\ &= \underline{0.9 \text{ MN/m}^2}\end{aligned}$$

From ESDU 71005 for all sides simply supported $b/a = 0.5$, $K = 5.8$

$$\frac{\tau_{crit.}}{E_T} = 5.8 \times \frac{0.8}{100}^2 = 0.000371$$

From DES 7306/2

$$\tau_{crit} = \underline{25.6 \text{ MN/m}^2}$$

Reserve factor on shear buckling

> 2

5.8 Inter rivet buckling.

The rib has smaller thickness than the skin and hence the inter rivet buckling stress of rib flange is calculated using R.Ac. 02.01.08

DRAWING NUMBER A 74 - E 5

R.F.

$$\sigma_{crit.} = K E_T (t/l)^2$$

K = 1.23 for counter sunk rivets.

t = Thickness, l - rivet pitch.

$$\frac{\sigma_{crit.}}{E_T} = 1.23 \times \left[\frac{0.8}{1} \right]^2$$

l mm	$\sigma_{crit.}$ MN/m
20	115
25	77
30	44.7
35	35.0
40	27.0
50	22.8

$$\text{Shear stress in aft cell flanges} = \underline{5.75 \text{ MN/m}^2}$$

However because of the super critical section of elevator a rivet pitch of 30 mm is used.

Reserve factor

> 2

Rivet pitching in nose cell rib is taken as a minimum of 2 rivets between notches.

DRAWING NUMBER A 74 - E 5

R.F.

5.9 Rib-Flange skin attachment.

Flange thickness 0.8 mm. material
 L 72 Rivets $\frac{1}{8}$ " diameter L 69, 100° csk
 for aft cell.

$$\text{Shear flow} = 1412 \text{ N/m.}$$

$$\text{Length of flange} = 1.05 \text{ m.}$$

$$\text{Shear load on flange} = \underline{1470 \text{ N.}}$$

From CIT S-12-01 single shear strength
 of rivet $S_o = 1.64 \text{ KN.}$

Using a reduction factor of 0.7 the
 design load is the smaller of

$$S_u = K_u S_o \quad S_p = K_p S_o$$

From CIT S-12-13,

$$\text{For } t/d = 0.25 \quad K_u = 0.48$$

$$K_p = 0.38$$

$$S_u = 0.48 \times 1.4 = \underline{0.69 \text{ KN}}$$

$$S_p = 0.38 \times .7 \times 1.5 \times 1.4 = \underline{0.60 \text{ KN.}}$$

Design load is 600N. in single shear
 when rivets are provided at 30 mm. pitch
 33 rivets have an applied load of 45N.
 per rivet.

RF > 2

Where it is not possible to provide
 reaction for solid rivets $\frac{1}{8}$ " Avdel rivets
 are used.

$$\text{Single shear strength } S_o = 1.82 \text{ KN}$$

$$\text{From CIT S-12-19 for } t/d = 0.25$$

$$K_u = 0.35 \quad K_p = 0.27$$

DRAWING NUMBER A 74 - E 5

R.F.

$$S_u = 0.35 \times 1.82 = 634 \text{ N.}$$

$$S_p = 0.27 \times 1.5 \times .7 \times 1.82 = \underline{515 \text{ N.}}$$

As against an applied load of 45N

RF > 2

(b) Nose cell.

$$\text{Shear flow} = 750 \text{ N/m.}$$

$$\text{Flange length} = 0.34 \text{ m.}$$

$$\text{Total shear load} = \underline{255 \text{ N.}}$$

A total of 15 rivets are provided
Therefore applied load of only 17 N per rivet.

RF > 2

5.10 Bearing strength of rivets on rib.

$$\text{Applied load on rivet} = 45 \text{ N.}$$

$$\text{Bearing stress} = \frac{45}{0.8 \times 3.18 \times 10^{-6}}$$

$$= \underline{18 \text{ MN/m}^2}$$

Allowable bearing stress for L 72 401MN/m² RF > 2

SECTION 6

ANALYSIS OF HINGE RIB

SECTION 6

Analysis of hinge rib at station 1420 on Inboard Elevator.

Refer drawing A 74 - E 6.

6.1 INTRODUCTION.

Rib at station 1420 is taken to show the analysis of a typical hinge rib. In this case the moment due to airloading only balances at the spar. There is an input of shear and moment at spar coming from the hinge reaction and torque.

6.2 LOADING.

Air loading which contributes to incremental shear and torque are taken from Table 1.1 as contributed by half bay width on either side of rib.

Incremental shear due to
airloading = 2126 N.

Incremental torque due to
air loading = 767 Nm.

Vertical load and torque increment due to reactions at hinge are taken from Table 1.2 of Section 1.

Vertical reaction on each
hinge rib = -10051 N.

Reaction torque on each
hinge rib = - 6104 N.

Net vertical reaction = - 7925 N.

Net torque = - 6104 N.

6.3 SHEAR FORCE AND BENDING MOMENT DISTRIBUTION.

The shear and bending moment calculations are similar to the analysis of section 5.

$$q_1 = 0.865 \frac{V}{C_{EL}} - 3.347 \frac{T}{C_{EL}^2} = \underline{10900 \text{ N/m.}}$$

$$q_2 = 0.0235 \frac{V}{C_{EL}} - 4.422 \frac{T}{C_{EL}^2} = \underline{22180 \text{ N/m}}$$

$$q_{12} = 4.848 \frac{V}{C_{EL}} = \underline{-35100 \text{ N/m.}}$$

$$q_w = q_{12} + q_2 - q_1 = \underline{-23820 \text{ N/m.}}$$

Calculations are given in Table 6.1 and 6.2 and shear force and B Mt diagrams on Fig. 6.1 and Fig. 6.2

TABLE 6.1

SHEAR FORCE ON RIB 1420.

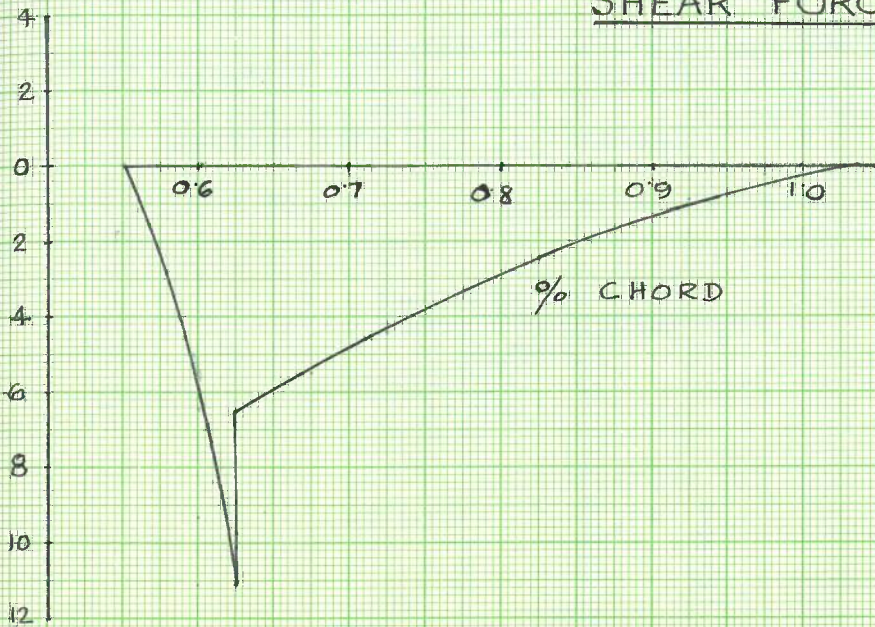
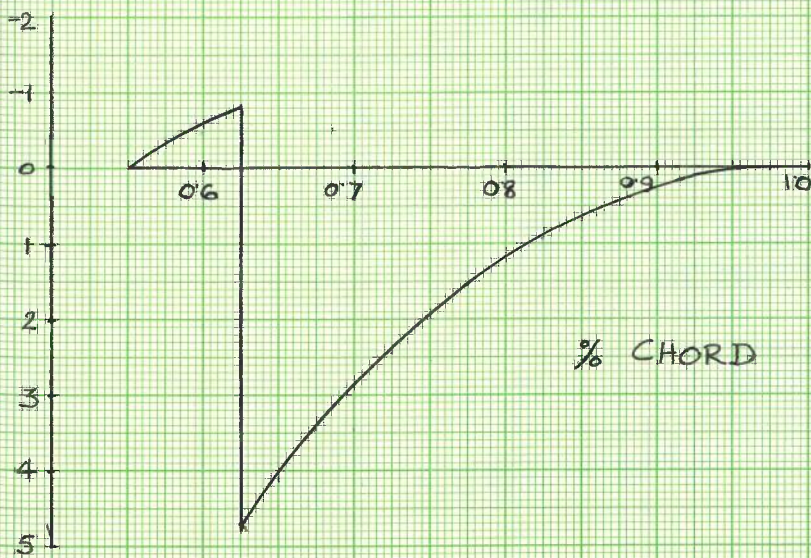
STATION % C	h m	$-q_1 h$	$h_s - h$ m	$q_2 (h_s - h)$	AIR LOAD %	AIR LOAD N	SHEAR N	REACTION N	NET SHEAR N
0.575	.228	-2486			13.8	293	293		- 2193
▲ 0.625	.223	-2430	0	0	14.3	304	597	-10051	<u>-11884</u> - 6574
0.675	.191	-2430	.032	720	14.15	300	897	-10051	- 5554
0.725	.154	-2430	.069	1530	14.00	297	1194	-10051	- 4447
0.775	.117	-2430	.106	2350	13.7	291	1485	-10051	- 3336
0.825	.088	-2430	.135	2995	13.0	276	1761	-10051	- 2415
0.875	.061	-2430	.162	3590	9.4	198	1959	-10051	- 1622
0.925	.034	-2430	.189	4190	5.65	120	2079	-10051	- 902
0.975	.009	-2430	.214	4750	1.95	45	2125	-10051	- 296

▲ Spar position

TABLE 6.2SHEAR AND BENDING MOMENT ON RIB.

Station % of chord	Shear N	Moment due to shear Nm	Moment due to airload Nm	Net Moment Nm
0.575	- 293	-486	8	-480
0.625	<u>-11884</u> - 6574	<u>-832</u> 4460	<u>17.1</u> 279.6	<u>-815</u> 4739.6
0.675	- 5554	3250	188.1	3438.1
0.725	- 4447	2290	119.2	2409.2
0.775	- 3336	1465	67.1	1532.1
0.825	- 2415	892	31.1	923.1
0.875	- 1622	414	11.1	425.1
0.925	- 902	93.7	2.20	95.1
0.975	- 296	0		

FIG 6.1

SHEAR FORCEFIG 6.2 BENDING MOMENT

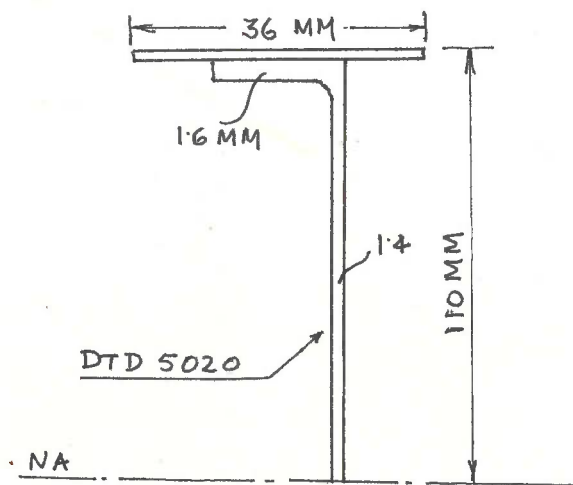
DRAWING NUMBER A 74 - E 6

R.F

6.3 ANALYSIS OF RIB 1420 IN BENDING.

The rib is machined from DTD 5020 material and has web of 1.4 mm. thick and flange 1.6 mm. thick, 30 x t skin width is assumed effective in bending.

$$30 \times t = 30 \times 1.2 = 36 \text{ mm.}$$



Item	A mm ²	y mm	Ay ² mm ⁴	I _o mm ⁴
1	36	110	3.96x10 ⁵	-
2	24	109	2.85x10 ⁵	-
3	3	55	4.74x10 ⁵	1.55x10 ⁵
			11.55	
			1.55	
			<u>13.10x10⁵</u>	

$$I_{NA} = 2 \times 1.31 \times 10^{-6} \text{ m}^4 = \underline{2.62 \times 10^{-6} \text{ m}^4}$$

DRAWING NUMBER A 74-E 6

R.F.

Maximum bending moment in the rib occurs at aft rib at the spar.

$$\text{Maximum bending stress} = 4739.2 \text{ Nm.}$$

$$\begin{aligned} \text{Bending stress} &= 4739.2 \times \frac{0.11}{2.62 \times 10^{-6}} \\ &= \underline{200 \text{ MN/m}^2} \end{aligned}$$

6.4 LOCAL INSTABILITY OF FLANGES.

$$\text{Flange thickness} = 1.6 \text{ mm.}$$

$$\text{Flange width} = 15 \text{ mm.}$$

One end is assumed simply supported and one end free.

$$\frac{\sigma_{\text{crit.}}}{E_T} = K(t/b)^2$$

$$\frac{\sigma_{\text{crit.}}}{E_T} = 0.58 \times \frac{1.6}{15}^2 = 0.0066$$

From DES 7306

$$\sigma_{\text{crit.}} = \underline{250 \text{ MN/m}^2}$$

$$\text{Applied stress} = \underline{200 \text{ MN/m}^2}$$

$$\text{Reserve factor} = \frac{250}{200}$$

1.25

$$\begin{aligned} \text{Nose rib bending stress} &= \frac{815 \times .11}{2.62 \times 10^{-6}} \\ &= \underline{34.5 \text{ MN/m}^2} \end{aligned}$$

Reserve factor

> 2

DRAWING NUMBER A 74 - E 6

R.F.

6.6 SHEAR BUCKLING OF RIB.

No lightening holes are provided on the rib. The shear buckling of panels supported on all sides supported as given in ESDU 71005 is used.

$$\text{For } b/a = 0.62 \quad K = 6.2$$

$$\frac{\tau_{\text{crit.}}}{E_T} = 6.2 \times \frac{1.4}{140}^2$$

$$= 0.00065$$

From DES 7306/2

$$\tau_{\text{crit}} = 44 \text{ MN/m}^2$$

$$\text{Shear flow in the rib} = 22180 \text{ N/m.}$$

$$\text{Applied shear stress} = \frac{22180}{1.4 \times 10^{-3}}$$

$$= 15.7 \text{ MN/m}^2$$

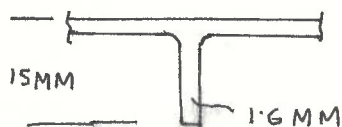
Reserve factor

> 2

6.7 STIFFENER INSTABILITY.

From ESDU 71005 Fig. 2 (R.Ae.S. 02.03.08.) for the panel having $K = 6.2$

$$\mu_{\text{crit.}} = 1.34$$



$$a = 220 \text{ mm.} \quad b = 140 \text{ mm.} \quad t = 1.4 \text{ mm.}$$

DRAWING NUMBER A 74 - E 6

R.F.

$$I_{xx} = 1800 \text{ mm}^4$$

Therefore,

$$\mu = \frac{1800 \times 140}{220^2 \times 1.4^3} = \underline{1.40}$$

$$\mu_{\text{crit}} = \underline{1.34}$$

$$\text{R.F.} = \underline{1.05}$$

1.05

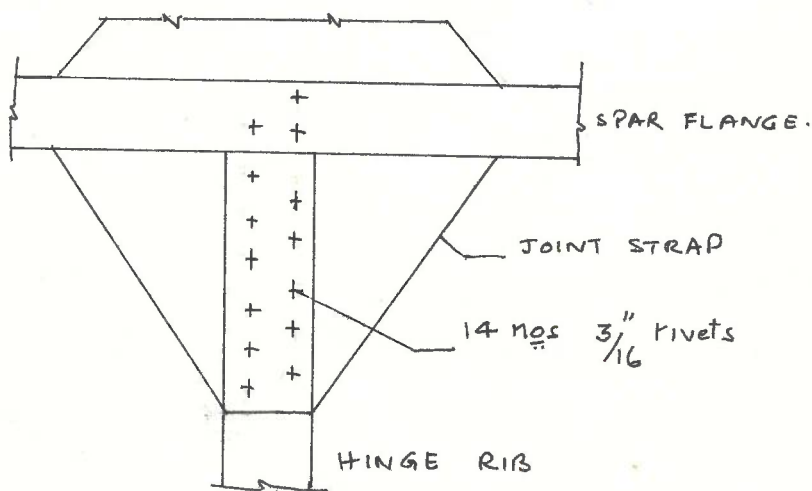
6.8 JOINT STRAP AT SPAR POSITION

Bending moment at spar is transmitted through 1.2 mm thick finger plate to nose rib.

$$\text{Bending moment at spar} = 4739.2 \text{ Nm.}$$

$$\text{Spar depth} = 223 \text{ mm.}$$

$$\text{End load on joining strap} = 21500 \text{ N}$$



$$\text{Load on each rivet} = \underline{15.4 \text{ KN.}}$$

DRAWING NUMBER A 74 - E 6

R.F.

$$\text{For } t/D = 0.3 \quad K_u = 0.53$$

$$K_p = 0.43$$

From DES 7306 strength of 3/16" rivet is 3.69 KN.

$$S_u = 0.53 \times 3.69 = 1.95 \text{ KN}$$

$$S_p = 0.43 \times 1.5 \times .7 \times 3.69 = 1.67 \text{ KN}$$

$$\text{Design load on rivet} = \underline{1.67 \text{ KN.}}$$

$$\text{Applied load on rivet} = \underline{1.54 \text{ KN}}$$

1.08

Applied finger stress.

$$\text{Tensile stress} = \frac{2 \times 1540}{30 \times 1.2 \times 10^{-6}}$$

$$= \underline{83.2 \text{ MN/m}^2}$$

$$\text{Shear stress} = \frac{2 \times 1540}{2 \times 30 \times 1.2 \times 10^{-6}}$$

$$= \underline{42.6 \text{ MN/m}^2}$$

$$\text{Bearing stress} = \frac{1540}{4.74 \times 1.2 \times 10^{-6}}$$

$$= \underline{271 \text{ MN/m}^2}$$

From CIT S - 14 - 02

Failing stress.

$$\text{Tensile stress} = 361 \text{ MN/m}^2$$

$$\text{Shear stress} = 193 \text{ MN/m}^2$$

Proof stress x 1.5 x 0.7

$$\text{Tensile stress} = \underline{314 \text{ MN/m}^2}$$

RF > 2

$$\text{Shear stress} = \underline{142 \text{ MN/m}^2}$$

RF > 2

$$\text{Bearing stress} = \underline{481 \text{ MN/m}^2}$$

RF 1.71

DRAWING NUMBER A 74 - E 6

R.F.

6.9 AFT SKIN RIB ATTACHMENT.

$$\text{Shear flow} = 22180 \text{ N/m.}$$

$$\text{Length of flange} = 1.09 \text{ m.}$$

$$\begin{aligned} \text{Total load on} \\ \text{flange} &= 24200 \text{ N.} \end{aligned}$$

Assuming 35 5/32" L 69 rivets at 30mm.
pitch, load on each rivet = 700 N.

Considering that Audel rivets will be
required in the area where reaction cannot
be provided for solid rivets.

$$\text{From CIT S - 12 - 19 for } t/d = 0.3$$

$$K_u = 0.42$$

$$K_p = 0.33$$

$$S_o \text{ for } 5/32" \text{ rivet} = 2.56 \text{ KN}$$

$$S_u = 0.42 \times 2.56 = \underline{1.15 \text{ KN}}$$

$$S_p = 0.33 \times 1.5 \times 0.7 \times 2.56 = \underline{0.89 \text{ KN}}$$

$$\text{Reserve factor} = 1.27$$

1.27

6.10 NOSE CELL RIB SKIN ATTACHMENT.

$$\text{Shear flow} = \underline{10900 \text{ N/m}}$$

$$\text{Length of flange} = 0.34 \text{ m.}$$

$$\text{Total load} = \underline{3710 \text{ N.}}$$

Providing 15 rivets $\frac{1}{8}$ " diameter L 69,

$$\text{Load on rivet} = \underline{247 \text{ N.}}$$

$$\text{Reserve factor} > 2$$

72

DRAWING NUMBER A 74 - E 6

R.F.

6.11 SPAR-RIB ATTACHMENT.

$$\text{Shear flow} = 23820 \text{ N/m.}$$

$$\begin{aligned} \text{Applied shear} \\ \text{load} &= 23820 \times 0.22 \\ &= \underline{5250 \text{ N.}} \end{aligned}$$

Using 10 numbers 5/32" rivet,

$$\text{Load for rivet} = \underline{525 \text{ N.}}$$

$$S_p = \underline{890 \text{ N.}} \text{ from section 6.9}$$

Reserve factor

1.70

SECTION 7

ANALYSIS OF SPAR

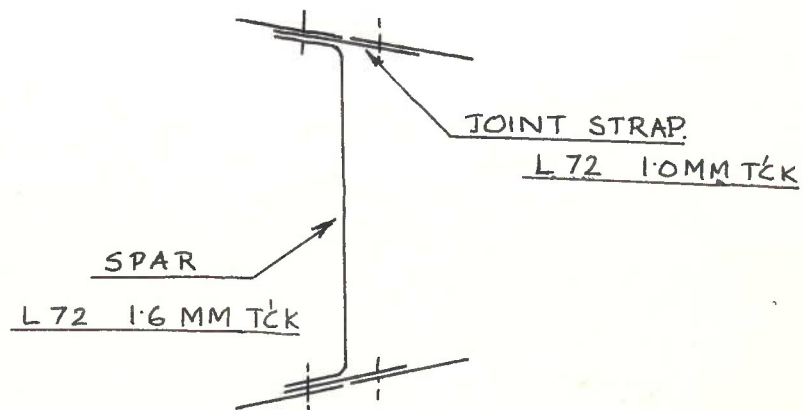
DRAWING A 74 - E 7

R.F

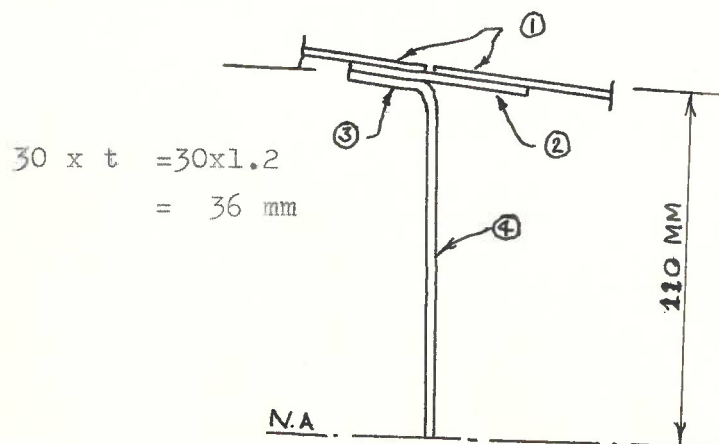
SECTION 7STRESS ANALYSIS OF INBOARD ELEVATOR SPAR.7.1 LOADING ON THE SPAR AND CROSS SECTION.

Bending moment and shear on spar is tabulated in Table I. The maximum bending moment in spar occurs at the inboard hinge section between ribs 6 and 7.

The spar section is shown below.



For calculation of section moment of inertia $30 \times t$ of the material is assumed effective in bending.

7.2 MOMENT OF INERTIA OF SECTION.

DRAWING NUMBER A 74 - E 7

R.F

As the section is symmetrical Neutral axis is in the middle. There is ^{no} lightening hole between rib 6 and 7.

Item	Area A mm ²	y mm	Ay ² mm ⁴	I _o mm ⁴
1	36	110	4.36x10 ⁵	-
2	30	109	3.57x10 ⁵	-
3	32	108	3.73x10 ⁵	-
4	86.5	27.0	0.63x10 ⁵	0.21x10 ⁵
			12.29x10 ⁵	
			.21x10 ⁵	
			12.50x10 ⁵ mm. ⁴	

$$\begin{aligned}
 I_{xx} &= 2 \times 12.50 \times 10^5 \text{ mm}^4 \\
 &= \underline{25.0 \times 10^{-7} \text{ m}^4}
 \end{aligned}$$

$$\begin{aligned}
 \text{Applied bending stress} &= \frac{3413.0 \times 0.11}{25 \times 10^{-7}} \\
 &= \underline{150 \text{ MN/m}^2}
 \end{aligned}$$

7.3 CHECK LOCAL INSTABILITY OF SPAR FLANGE

Flange thickness = 1.6 mm.

Spar width = 20 mm.

Assuming one end simply supported and one end free.

$$\begin{aligned}
 \sigma_{\text{crit.}} &= K (t/b)^2 \\
 &= 0.58 \times \frac{1.6}{20}^2 = 0.00371
 \end{aligned}$$

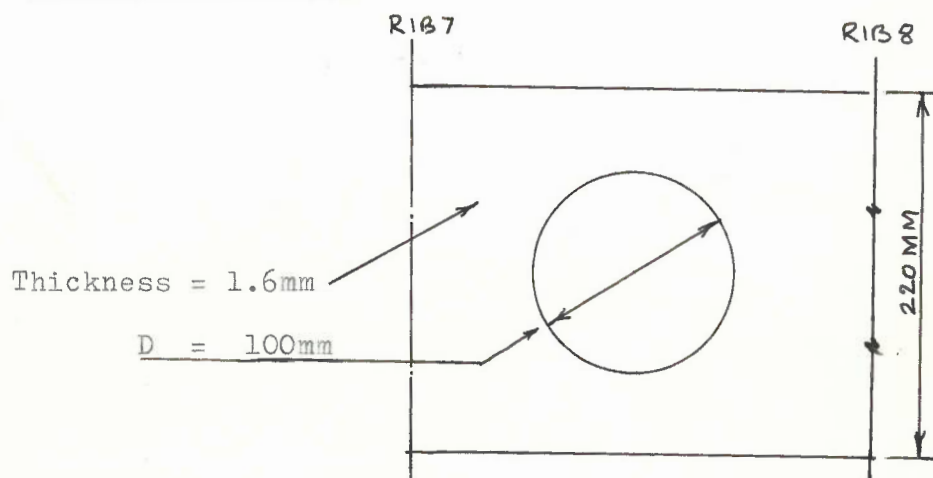
DRAWING NUMBER A 74 - E 6

R.F

From DES 7306/2

$$\text{Allowable critical stress} = \frac{175 \text{ MN/m}^2}{1.15}$$

$$\text{Reserve factor} = \frac{175}{150}$$

7.4 WEB SHEAR BUCKLING.

Shear buckling stress for a panel without lightening holes. From ESDU 71005 for $\frac{A}{H} = 0.88$ $K = 6.8$

$$\frac{\tau_{crit}}{E_T} = 6.8 \times \frac{1.6}{220}^2 = 0.000360$$

$$\tau_{crit} = \frac{26.0 \text{ MN/m}^2}{1.15}$$

$$\frac{H}{t} = \frac{220}{1.6} = 137$$

$$\frac{D}{A} = \frac{100}{250} = 0.4$$

DRAWING NUMBER A 74 - E 7

R.F.

From Fig. $\frac{\tau'_b}{\tau_b}$ Vs $\frac{H}{t}$ on extrapolation.

$$\begin{aligned}\frac{\tau'_b}{\tau_b} &= 0.86 \\ \frac{\tau_b}{\tau'_b} &= 0.86 \times 26.0 \text{ MN/m}^2 \\ &= \underline{22.4 \text{ MN/m}^2}\end{aligned}$$

Shear flow in panel.

From Table 11, shear in the web,

$$\begin{aligned}q_w &= -q_{12} + q_2 - q_1 \\ q_w &= -37000 + 34421 - 19590\end{aligned}$$

$$\begin{aligned}\text{Shear stress in panel} &= \frac{22100}{1.6 \times 10^{-3}} \\ &= \underline{13.8 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Reserve factor} &= \frac{22.4}{13.8} \\ &1.61\end{aligned}$$

7.5 Inter rivet buckling.

From R.Ac.Soc. Data Sheet 02.01.08

$$\sigma_{\text{crit.}} = K E_T (t/l)^2$$

For counter sunk rivets $K = 1.23$

For a rivet pitch of 18 mm. the outer rivet buckling stress of 0.8 mm. thick front box skin.

$$\begin{aligned}\frac{\sigma_{\text{crit}}}{E_T} &= 1.23 \frac{0.8}{18}^2 = \underline{0.00243}\end{aligned}$$

DRAWING NUMBER A 74 - E 7

R.F

From DES 7306/2

$$\sigma_{\text{crit.}} = \frac{175.6 \text{ MN/m}^2}{150}$$

$$\text{R.F.} = \frac{175.6}{150}$$

1.15

7.6 SPAR DEFLECTION AT MID SPAN.

For calculation of spar deflection a simplified uniformly distributed load and inertia distribution is assumed.

Deflection at mid span under these loading

$$\delta = \frac{w l^2}{384 E I} (5l^2 - 24 l_1^2)$$

In this case $l = 1.53 \text{ m.}$

$l_1 = 0.65 \text{ m.}$

Average value of u.d.l. on inboard elevator is 14200 N/m.

$$I = 2.5 \times 10^{-6} \text{ m}^4$$

$$E = 72.4 \times 10^9 \text{ MN/m}^2$$

$$\begin{aligned} \delta &= \frac{14700 \times 1.53^2 (5 \times 1.53^2 - 24 \times 0.65^2)}{384 \times 72.4 \times 10^9 \times 2.5 \times 10^{-6}} \\ &= 0.767 \times 10^{-3} \\ &= \underline{0.767 \text{ mm.}} \end{aligned}$$

Permissible deflection of control surface is 2% of depth of structure at section.

$$\begin{aligned} \text{Permissible deflection} &= 0.02 \times 220 \\ &= \underline{4.4 \text{ mm.}} \end{aligned}$$

Reserve factor

> 2

RF > 2

SECTION 8

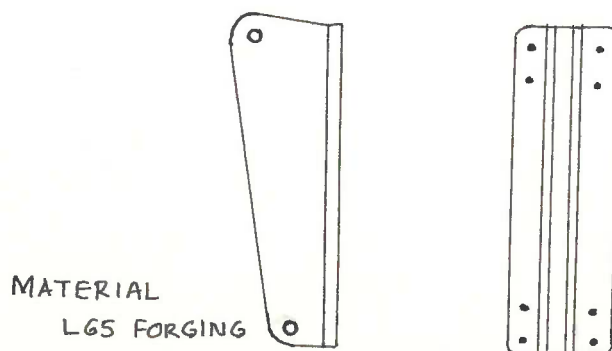
ANALYSIS OF HINGE BRACKETS

SECTION 8HINGE BRACKETS

DRAWING NUMBER A 74 - E 7 & E 8

R.F

8.1 This section gives stress analysis of hinge bracket at station 1370 on inboard elevator Drawing A 74 - E8 gives the details.

8.2 ACTUATOR HINGE BRACKET

The shortest effective moment arm for the actuator occurs at maximum elevator deflection.

$$\text{Maximum torque at hinge line} = \underline{13742 \text{ Nm}}$$

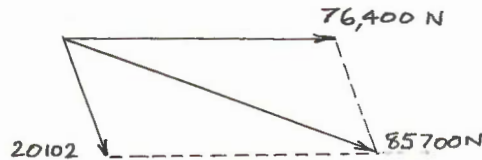
$$\text{Lever arm} = 0.180 \text{ m}$$

$$\begin{aligned} \text{Maximum load on actuator} &= \frac{13742}{0.18} \\ &= \underline{76,400 \text{ N}} \end{aligned}$$

By combining the maximum load due to actuator and hinge reaction using the parallelogram of forces for the elevator positioned at maximum deflection the maximum reaction is obtained.

DRAWING NUMBER A 74 E7 & E8

R.F



Assuming for 'fail safe considerations' in the event of one of the brackets failing the other should be able to resist 80% of ultimate load

8.3 SELECTION OF BEARING

Maximum reaction at the hinge
 bracket = 8740 KG

Load on each bearing = 0.80×8740
 = 6980 KG

From Ampep catalogue of self aligning standard spherical bearings 15 GSW bearing has a maximum static load of 8890 KG radially.

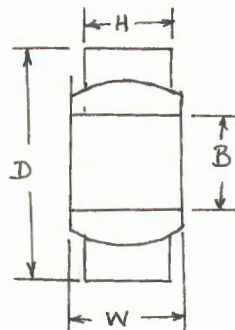
Design load = $1.5 \times 0.7 \times 8890$
 = 9240 KG

Reserve factor = $\frac{9240}{6980} = \underline{1.31}$

1.31

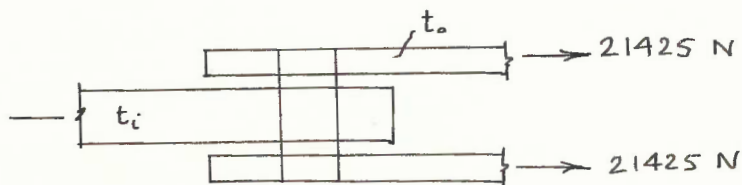
DRAWING NUMBER A 74 - E 8

R.F



dimensions in mm.

Part No.	B	W	D	H	Wt.
15 GSW	15	17	29	13.5	0.064

8.4 HINGE BOLT BENDING

Effective off set of

$$\text{bolts} = \frac{t_o}{2} + \text{gap} + \frac{t_i}{4}$$

$$= 2.5 + 3.0 + \frac{17}{4}$$

$$= \underline{9.75 \text{ mm.}}$$

Applied bending
moment

$$= 21425 \times 9.75 \times 10^{-3}$$

$$= \underline{200 \text{ Nm.}}$$

DRAWING NUMBER A 74 - E8

R.F

From R.Ac.S. 01.06.01

Form factor = 1.54

$$\text{Section modulus} = \frac{\pi d^3}{32} = \underline{3.32 \times 10^{-7} \text{ m}^3}$$

$$\text{For S 99 material } t_2 = \underline{1150 \text{ MN/m}^2}$$

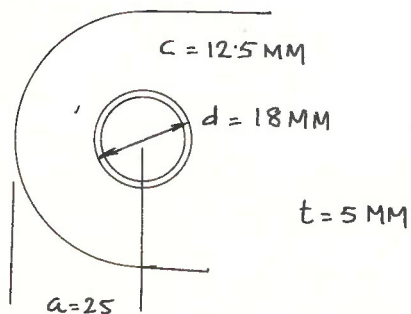
$$\begin{aligned} \text{Allowable bending moment} \\ \text{on bolt} &= 1.54 \times 1150 \\ &\times 10^6 \times 3.32 \times 10^{-7} \\ &= \underline{589 \text{ Nm}} \end{aligned}$$

Reserve factor > 2

RF > 2

8.5 LUG STRESSING.

Lug stressing is due as per the method given in DES 7312



$$\frac{a}{d} = \frac{25}{18} = 1.37$$

$$\frac{d}{t} = 3.6$$

Lug is analysed for a maximum load of 80% of ultimate.

$$\begin{aligned} \text{Applied load on each lug} &= 0.8 \times \frac{85700}{2} \\ &= \underline{34300 \text{ N}} \end{aligned}$$

Applied stress.

$$\begin{aligned} \text{Tensile stress} &= \frac{34300}{2 \times 5 \times 12.5 \times 10^{-6}} \\ &= \underline{272 \text{ MN/m}^2} \end{aligned}$$

DRAWING NUMBER A 74 - E8

R.F

$$\begin{aligned}\text{Shear stress} &= \frac{34300}{2 \times 25 \times 5 \times 10^{-6}} \\ &= \underline{136 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Bearing stress} &= \frac{34300}{18 \times 5} \\ &= \underline{380 \text{ MN/m}^2}\end{aligned}$$

From S - 14 - 02

Failing stress

$$\text{Tensile stress} = 64500 \text{ psi} = \underline{444 \text{ MN/m}^2}$$

$$\text{Shear stress} = 33000 \text{ psi} = \underline{211 \text{ MN/m}^2}$$

$$\text{As } \frac{a}{d} = 1.1$$

$$\begin{aligned}\text{Factor for shear} &= 1 - 0.4 (1.5 - 1.11) \\ &= 0.85\end{aligned}$$

$$\text{Bearing stress} = 132600 \text{ psi} = \underline{911 \text{ MN/m}^2}$$

Proof stresses

$$\begin{aligned}\text{Tensile stress} &= 56500 \times 1.5 \times 0.7 \\ &= 59000 \text{ psi} = \underline{406 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Shear stress} &= 27000 \times 1.5 \times .7 \times 0.85 \\ &= 26300 \text{ psi} = \underline{181 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Bearing stress} &= 90900 \times 1.5 \times 0.7 \\ &= 95000 = \underline{658 \text{ MN/m}^2}\end{aligned}$$

Allowable stress

Tension	= $\underline{406 \text{ MN/m}^2}$	1.5
Shear	= $\underline{181 \text{ MN/m}^2}$	1.39
Bearing	= $\underline{658 \text{ MN/m}^2}$	1.7

Actual loading of lug is only 21425 N
will give a Reserve factor > 2 in all
the three cases.

DRAWING NUMBER A 74 - E8

R.F

8.6 BEARING OF BOLT IN BUSH

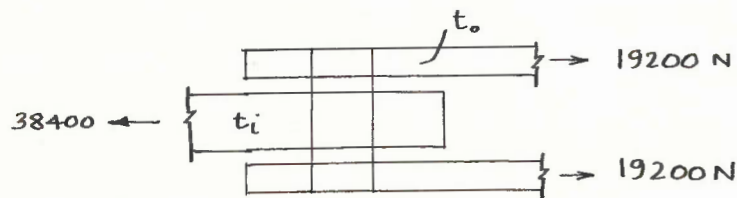
15 mm I/D S80 bush 1.5 mm thick
is provided in the actuator bracket.

$$\begin{aligned}
 \text{Bolt diameter} & 15 \text{ mm} \\
 \text{Bolt material} & \text{S99} \\
 \text{Applied bearing stress} & = \frac{68000}{10 \times 15 \times 10^{-6}} \\
 & = 450 \text{ MN/m}^2
 \end{aligned}$$

Allowable bearing stress for S 80
material from CIT S - 14 - 01

$$\begin{aligned}
 129900 \times 1.5 \times 0.7 & = 135500 \text{ psi} \\
 & = 930 \text{ MN/m}^2
 \end{aligned}$$

Reserve factor > 2

RF > 2 8.7 ACTUATOR BOLT BENDING

12 mm diameter bolt is used with 1.5 mm
thickness of bush material S 80.

$$\begin{aligned}
 \text{Effective off set of bolts} & = \frac{t_o}{2} + \text{gap} + \frac{t_i}{4} \\
 & = \frac{5}{2} + 3.0 + \frac{1.7}{4} \\
 & = 4.75 \text{ mm}
 \end{aligned}$$

DRAWING NUMBER A 74 - E 8

R.F

$$\begin{aligned}\text{Applied bending moment} &= 19100 \times 9.5 \times 10^{-3} \\ &= \underline{181 \text{ Nm.}}\end{aligned}$$

$$\text{Form factor for bolt} = 1.54$$

R.Ae.S. 01 - 06 - 01

$$\begin{aligned}\text{Section modulus} &= \frac{\pi d^3}{32} \\ &= \underline{1.69 \times 10^{-7} \text{ m}^3}\end{aligned}$$

0.2% proof stress for S 99 bolt
material is 1150 MN/m².

$$\begin{aligned}\text{Allowable bending moment on bolt} \\ &= 1.54 \times 1150 \times 1.69 \times 10^{-7} \\ &= \underline{300 \text{ MN/m}^2}\end{aligned}$$

$$\text{Reserve factor} \quad 1.66 \quad 1.66$$

8.8 BEARING OF BOLT IN BUSH

For 12 mm I/D and 1.5 mm thick S 80
bush bearing on S 99 bolt

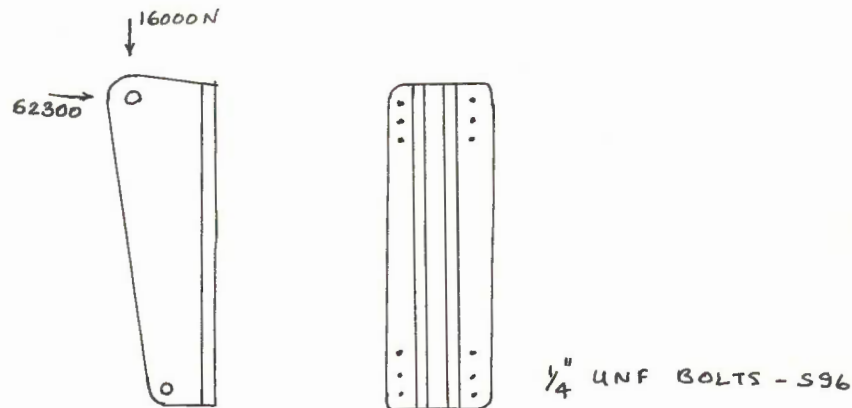
$$\begin{aligned}\text{Applied bearing stress} &= \frac{38200}{10 \times 12 \times 10^{-6}} \\ &= \underline{318 \text{ MN/m}^2}\end{aligned}$$

Allowable bearing stress from section
8.6 is 930 MN/m².

$$\text{Reserve factor} > 2. \quad > 2$$

DRAWING NUMBER A 74 - E 8

R.F

8. ACTUATOR BRACKET ATTACHMENT BOLTS

The fitting is designed to carry 80% of the loads at the station.

Attachment bolts are subjected to vertical shear due to reaction and tension load due to the moment produced by the actuator. The effect of offset on vertical reaction is to reduce the moment due to the actuator.

$$\begin{aligned}
 \text{Vertical reaction at hinge} &= 20100 \text{ N.} \\
 \text{Vertical load on bracket} &= 0.8 \times 20100 \\
 &= 16000 \text{ N} \\
 \text{Moment due to actuator} &= 0.8 \times 13740 \\
 &= 10993 \text{ Nm.} \\
 \text{Moment to be resisted} \\
 \text{by bolt group} &= 10993 - 800 \\
 &= 10193 \text{ Nm.}
 \end{aligned}$$

Assuming a lever arm of 160 mm between bolt groups

$$\begin{aligned}
 \text{Max. tension load in bolts} &= \frac{10193}{0.16 \times 6} \\
 &= 10.61 \text{ KN.} \\
 \text{Max shear load on bolts} &= \frac{16000}{12} \\
 &= 1.330 \text{ KN}
 \end{aligned}$$

DRAWING NUMBER A 74 - E 8

R.F

From DES 7306 for $\frac{1}{4}$ " UNF S 96 boltsSingle shear strength = 17.7 KNTension = 18.9 KNs = $\frac{1.33}{17.7}$ = 0.094t = $\frac{10.61}{18.9}$ = 0.55

From CIT S - 11 - 04

t' = 1.00

Reserve factor = $\frac{1.0}{.55}$ = 1.8 1.8

SECTION 9

ANALYSIS OF TRAILING EDGE BRACKETS

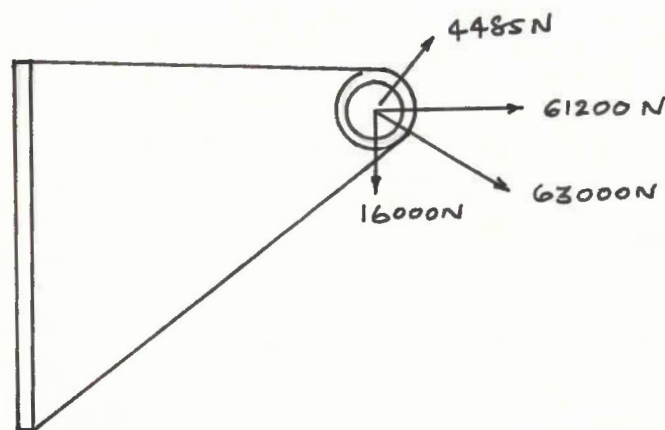
DRAWING NUMBER A 74 - E 8

R.F

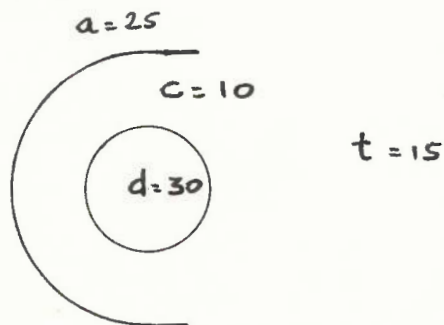
SECTION 9TRAILING EDGE BRACKETS

Tail plane edge brackets are designed to take up the hinge and actuator reactions and the side load produced by 12 times the weight of elevators. Each bracket is designed to carry 80% of the ultimate load at the hinge. Selection of bearing for bracket is given in Section 8.3.

The brackets are to be designed for the loads as shown below.



$$\text{Side load} = 12 \times 38.1 \times 9.81 = \underline{4485 \text{ N.}}$$

9.1 BEARING HOUSING

$$\begin{aligned} t &= 15 \text{ m} \\ \frac{d}{t} &= 2.0 \end{aligned}$$

DRAWING NUMBER A 74 - E 8

R.F

$$\begin{aligned}\text{Shear stress} &= \frac{P}{2at} = \frac{63000}{2 \times 10 \times 15 \times 10^{-6}} \\ &= 84 \text{ MN/m}^2\end{aligned}$$

$$\begin{aligned}\text{Applied tensile stress} &= \frac{63000}{2 \times 10 \times 15 \times 10^{-6}} \\ &= 210 \text{ MN/m}^2\end{aligned}$$

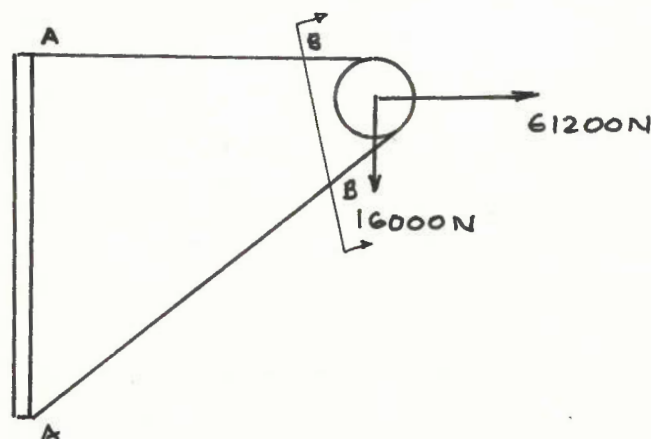
$$\begin{aligned}\text{Bearing stress} &= \frac{63000}{30 \times 15 \times 10^{-6}} \\ &= 140 \text{ MN/m}^2\end{aligned}$$

From section 8.5 permissible lug stresses are

$$\text{Tension} = 406 \text{ MN/m}^2 \quad \text{RF} > 2$$

$$\text{Shear} = 181 \text{ MN/m}^2 \quad \text{RF} > 2$$

$$\text{Bearing} = 658 \text{ MN/m}^2 \quad \text{RF} > 2$$

9.2 HINGE BRACKET IN BENDING

DRAWING NUMBER A 74 - E 8

R.F

$$\begin{aligned}\text{Moment at section 'AA'} &= 16000 \times 0.27 \\ &= \underline{4320 \text{ Nm.}}\end{aligned}$$

$$\begin{aligned}\text{Moment of inertia of section} &= \frac{1}{12} \times .010 \times .250^3 \\ &= \underline{13 \times 10^{-6} \text{ m}^4}\end{aligned}$$

$$\begin{aligned}\text{Applied bending stress} &= \frac{4320 \times 0.125}{1.3 \times 10^{-6}} \\ &= \underline{41.5 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Tension stress in bracket} &= \frac{61200}{250 \times 10 \times 10^{-6}} \\ &= \underline{24.5 \text{ MN/m}^2}\end{aligned}$$

$$\begin{aligned}\text{Permissible bending stress} \\ \text{in L 65} &= \frac{ft + t_1}{2} \\ &= \frac{463 + 401}{2} = \underline{432 \text{ MN/m}^2}\end{aligned}$$

$$R_b = \frac{41.5}{432} = \underline{0.096}$$

$$\begin{aligned}\text{Permissible stress in tension} &= ft \\ &= \underline{463 \text{ MN/m}^2}\end{aligned}$$

$$R_t = \frac{24.5}{463} = \underline{0.053}$$

$$\text{Reserve factor for combined} = \frac{1}{R_b + R_t} - 1$$

SECTION BB

$$\text{Depth of section} = 50 \text{ mm}$$

$$\begin{aligned}\text{Tensile stress} &= \frac{61200}{50 \times 10 \times 10^{-6}} \\ &= \underline{122 \text{ MN/m}^2}\end{aligned}$$

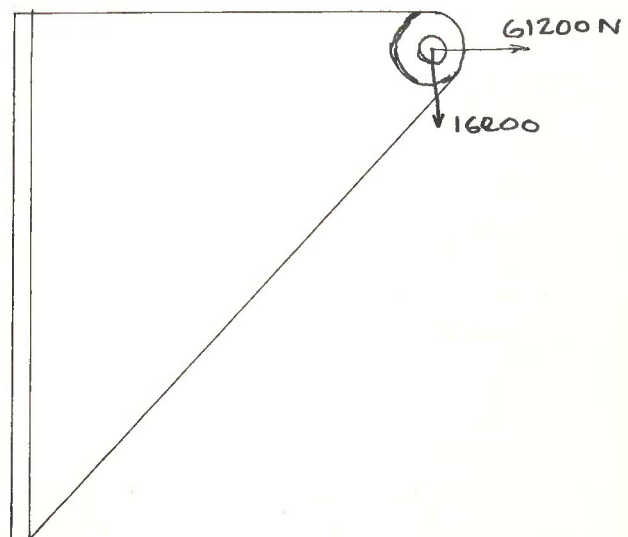
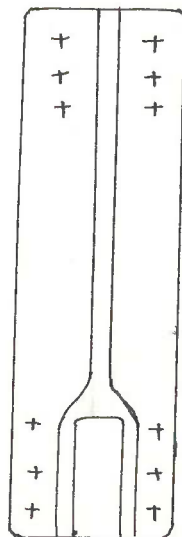
$$\text{Permissible tensile stress} = \underline{463 \text{ MN/m}^2}$$

$$\text{Reserve factor} > 2$$

$$RF > 2$$

DRAWING NUMBER A 74 - E 8

R.F

9.3 ATTACHMENT BOLTS

Tensile load is assumed to be taken up by 6 bolts at top.

$$\text{Tension per rivet} = \frac{61200}{6} = \underline{10,200 \text{ N.}}$$

Moment at spar 4320 Nm.

$$\begin{aligned} \text{Tension due to this moment} &= \frac{4320}{200} \\ &= \underline{21,600 \text{ N.}} \end{aligned}$$

$$\text{Tension per rivet} = \frac{21,600}{6} = \underline{3600 \text{ N.}}$$

$$\text{Tension load on rivet} = \underline{13800 \text{ N}}$$

$$\begin{aligned} \text{Shear load on rivet} &= \frac{16,000}{6} \\ &= \underline{5350 \text{ N.}} \end{aligned}$$

From DES 7306/3,

Allowable shear strength of $\frac{1}{4}$ " UNF S 96 bolt 17.7

$$\text{Allowable tensile stress} = \underline{18.9 \text{ KN}}$$

DRAWING NUMBER A 74 - E 8

R.F

$$S = \frac{5350}{18900} = 0.283$$

$$t = \frac{13800}{17700} = 0.78$$

From S - 11 - 04 of CIT,

$$t' = 1.0$$

$$\text{Reserve factor} = \frac{1.0}{0.78} = \underline{1.28}$$

RF 1.28

SECTION 10

WEIGHT ESTIMATE

SECTION 10WEIGHT ESTIMATE.10.1 INBOARD ELEVATOR.

$$\text{Area of aft cell skin} = 5.72 \text{ m}^2$$

$$\text{Area of nose cell skin} = 1.90 \text{ m}^2$$

Aft skins are of 1.2 mm thickness and nose skins 0.8 mm.

$$\begin{aligned} \text{Weight of aft cell skin} &= 5.72 \times 1.2 \times 10^{-3} \times 2768 \\ &= \underline{18.9 \text{ KG.}} \end{aligned}$$

$$\begin{aligned} \text{Wt. of nose cell skin} &= 1.90 \times 0.8 \times 10^{-3} \times 2768 \\ &= \underline{4.2 \text{ KG.}} \end{aligned}$$

10.2 RIBS.

The light ribs are made of 0.8 mm thick L 72 material with 3 lightening holes and has 15 mm flange width.

$$\text{Area of aft cell rib} = 0.0875 C_{EL}^2$$

$$\text{Area of nose cell rib} = 0.0375 C_{EL}^2$$

For rib with $C_{EL} = 1.04 \text{ m}$ taken as average,

$$\text{Area of aft cell rib} = 0.094 \text{ m}^2$$

$$\text{Area of nose cell rib} = 0.040 \text{ m}^2$$

$$\text{Area of flanges} = 0.036 \text{ m}^2$$

$$\begin{array}{l} \text{Area of lightening} \\ \text{holes} \end{array} = 0.016 \text{ m}^2$$

$$\text{Total area} = \underline{0.154 \text{ m}^2}$$

$$\begin{aligned}\text{Weight of rib} &= 0.154 \times 0.8 \times 10^{-3} \times 2768 \\ &= 0.34 \text{ KG.}\end{aligned}$$

$$\begin{aligned}\text{Total wt. of light ribs} &= 12 \times 0.34 \\ &= \underline{4.08 \text{ KG.}}\end{aligned}$$

$$\text{Area of nose hinge rib} = 0.041$$

$$\text{Area of aft cell rib} = 0.096$$

$$\text{Area of flanges} = 0.036$$

There are no lightening holes on rib.
Web is 1.4 mm thick and flanges 1.6 mm thick.

$$\begin{aligned}\text{Wt. of rib} &= (0.137 \times 1.4 + 0.036 \times 1.6) 2.768 \\ &= .1918 + .0576 = 0.69 \text{ KG}\end{aligned}$$

$$\text{Wt. of 4 hinge ribs} = \underline{2.76 \text{ KG}}$$

10.3 STIFFENERS.

$$\text{Area of stiffener} = 0.036 \text{ m}^2$$

$$\begin{aligned}\text{Total weight of stiffener} &= 0.036 \times 1.2 \times 2.768 \times 8 \\ &= \underline{0.95 \text{ KG}}\end{aligned}$$

10.4 SPAR.

Net area of spar after deducting the area
of lightening holes = 1.02 m²

$$\begin{aligned}\text{Weight of spar} &= 1.02 \times 1.6 \times 10^{-3} \times 2768 \\ &= \underline{4.51 \text{ KG}}\end{aligned}$$

10.5 HINGE BRACKETS.

$$\text{Volume of hinge bracket} = 0.000132 \text{ m}^3$$

$$\text{Weight} = 0.36 \text{ KG.}$$

$$\begin{aligned}\text{Total weight of hinge} \\ \text{brackets} &= \underline{0.72 \text{ KG}}\end{aligned}$$

- 10.6 Weight of trailing edge
honey comb = 1.0 KG
- Weight of other small
items not included = 1.0 KG
- Total weight of inboard elevator = 38.12 KG

10.7 OUTBOARD ELEVATOR.

SKINS

$$\text{Area of aft cell skin} = 2.92 \text{ m}^2$$

$$\text{Area of nose cell skin} = 0.97 \text{ m}^2$$

Aft skins are 1.2 mm thick and forward
skins 0.8 mm thick.

$$\begin{aligned} \text{Weight of aft cell skin} &= 2.92 \times 1.2 \times 10^{-3} \times 2768 \\ &= \underline{9.69 \text{ KG}} \end{aligned}$$

$$\begin{aligned} \text{Weight of nose cell skins} &= 0.97 \times 0.8 \times 10^{-3} \times 2768 \\ &= \underline{2.15 \text{ KG}} \end{aligned}$$

10.8 RIBS.

Light rib are pressed out of L 72 material
0.8 mm. thick and has three lightening holes and
has flange width 15 mm.

For an average rib of C_{EL} 0.825 m.,

$$\text{Area of aft cell rib} = 0.059 \text{ m}^2$$

$$\text{Area of nose cell rib} = 0.025 \text{ m}^2$$

$$\text{Area of flanges} = 0.024 \text{ m}^2$$

$$\begin{aligned} \text{Area of lightening} \\ \text{hole} \end{aligned} = 0.010 \text{ m}^2$$

$$\text{Net area of rib} = \underline{0.102 \text{ m}^2}$$

$$\begin{aligned}\text{Weight of light rib} &= 0.102 \times 0.8 \times 10^{-3} \times 2768 \\ &= \underline{0.282 \text{ KG}}\end{aligned}$$

$$\begin{aligned}\text{Total weight of} \\ \text{light ribs} &= \underline{2.53 \text{ KG}}\end{aligned}$$

$$\text{Area of nose hinge rib} = 0.028 \text{ m}^2$$

$$\text{Area of aft hinge rib} = 0.064 \text{ m}^2$$

$$\text{Area of flanges} = 0.026 \text{ m}^2$$

Considering these were again made of 1.4 mm web and 1.6 mm flange machined ribs with no lightening holes,

$$\begin{aligned}\text{Weight of rib} &= (1.6 \times 0.026 + 1.4 \times 0.0092) \times 1.4 \times 2.768 \\ &= 0.65 \text{ KG}\end{aligned}$$

$$\text{Total weight of 4 hinge rib} = \underline{2.6 \text{ KG}}$$

10.9 STIFFENERS.

$$\text{Total weight of stiffeners} = \underline{0.90 \text{ KG}}$$

10.10 SPAR.

$$\text{Area of spar} = 1.04 \text{ m}^2$$

$$\text{Weight of spar} = 1.04 \times 1.6 \times 2.768$$

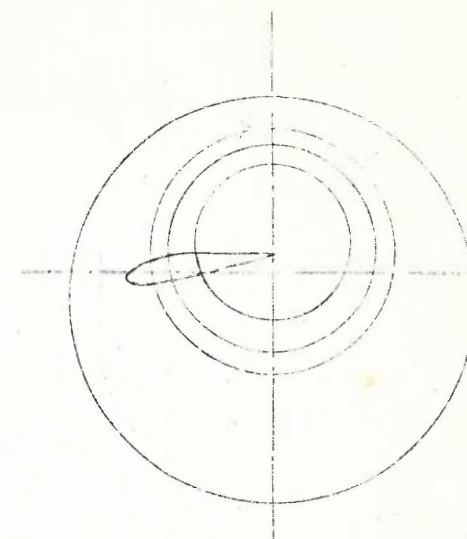
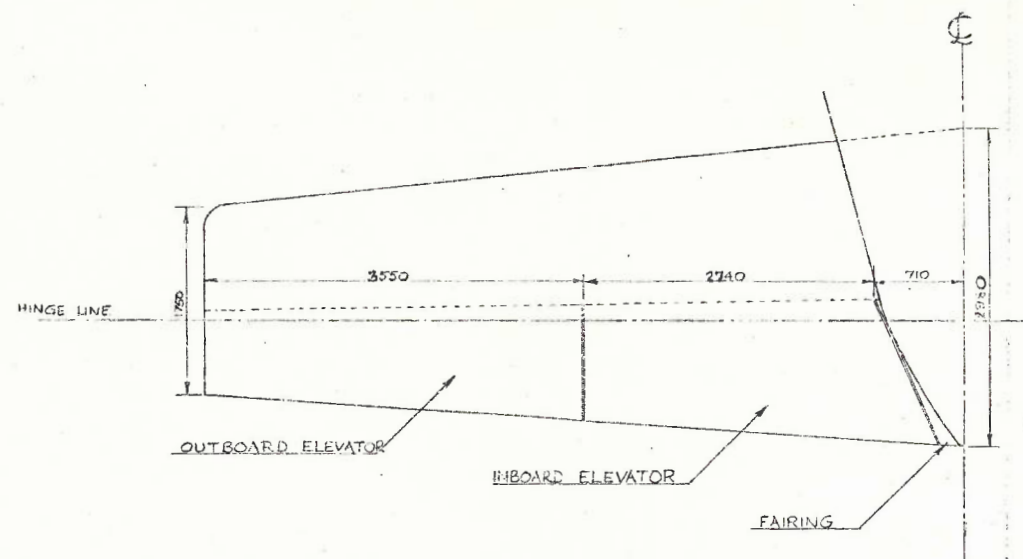
$$= \underline{4.60 \text{ KG}}$$

10.11 HINGE BRACKET.

$$\text{Volume of hinge bracket} = 0.00011 \text{ m}^3$$

$$\text{Weight of hinge bracket} = 0.3 \text{ KG}$$

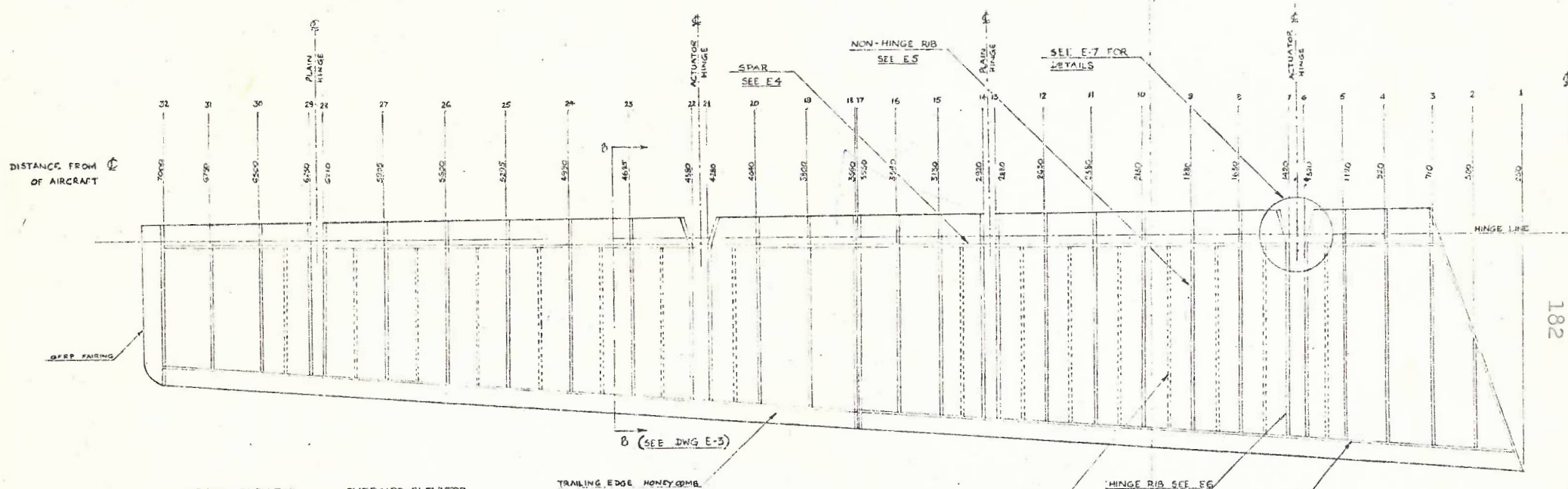
$$\begin{aligned}\text{Total weight of hinge} \\ \text{brackets} &= \underline{0.6 \text{ KG}}\end{aligned}$$



ELEVATOR GEOMETRY

ALL DIMENSIONS IN MM
 ELEVATOR CHORD/TAIL PLANE CHORD AT $H/L = 0.4$
 MAX. ELEVATOR DEFLECTION $\pm 16^\circ$

DRAWN EXTD 15/10/11	CHKD	APPV	SCALE 1:1.25	TITLE:- ELEVATOR GEOMET
COLLEGE OF AERONAUTICS CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD				DRAWING No. A74-E1 SHEET 1 OF 1



INBOARD ELEVATOR

OUTBOARD ELEVATOR

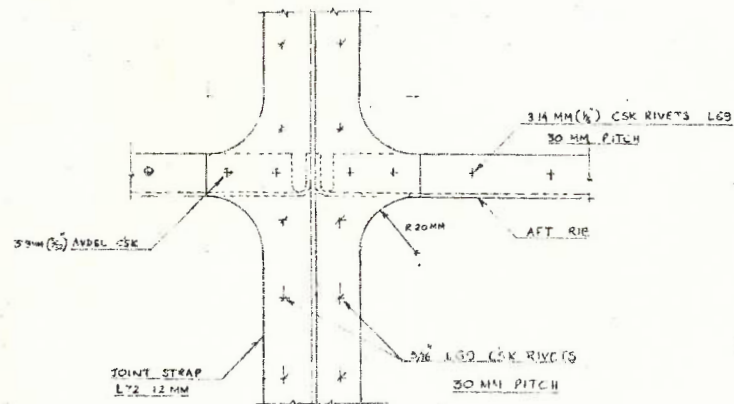
PANEL	AFT CELL THICKNESS MM	FORW CELL THICKNESS MM	PANEL	AFT CELL THICKNESS MM	FORW CELL THICKNESS MM
1-2	12	0.8	18-19	12	0.8
2-3	12	0.8	19-20	12	0.8
3-4	12	0.8	20-21	12	0.8
4-5	12	0.8	21-22	12	0.8
5-6	12	0.8	22-23	12	0.8
6-7	12	0.8	23-24	12	0.8
7-8	12	0.8	24-25	12	0.8
8-9	12	0.8	25-26	12	0.8
9-10	12	0.8	26-27	12	0.8
10-11	12	0.8	27-28	12	0.8
11-12	12	0.8	28-29	12	0.8
12-13	12	0.8	29-30	12	0.8
13-14	12	0.8	30-31	12	0.8
14-15	12	0.8	31-32	12	0.8
15-16	12	0.8			
16-17	12	0.8			

NOTES

1. ALL DIMENSIONS IN MM
2. SEE DRAWING E.3 FOR DETAILS

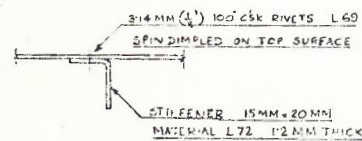
PORT AS DRAWN
STBD. OPPOSITE HAND

DRAWN	CHECKED	APPROVED	SCALE	TITLE
PM 20/10/00			1:10	G.A. OF ELEVATOR
COLLEGE OF AERONAUTICS CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD.				DRAWING No. A74-E2 SHEET 1 OF 1 SHEETS

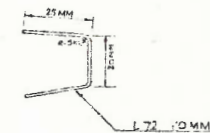


TYPICAL RIB ATTACHMENT

SCALE 1:1

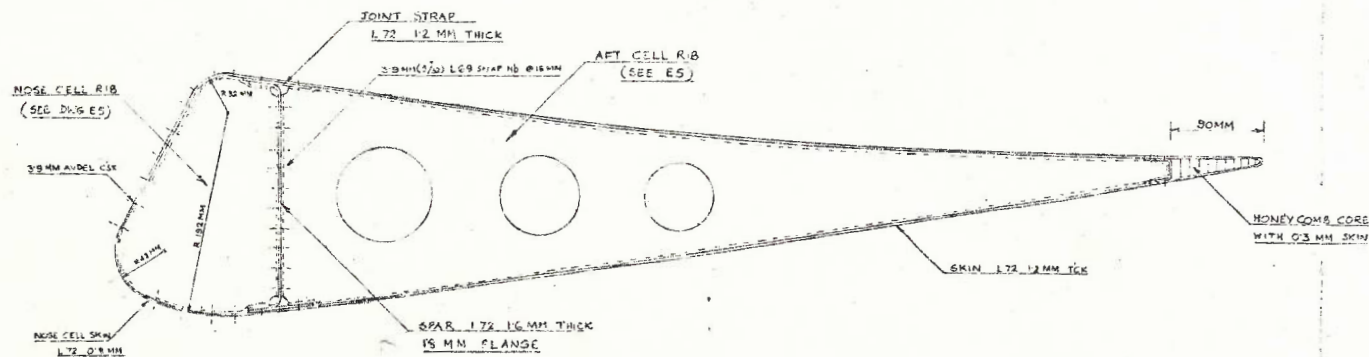


TYPICAL SECTION OF STIFFENER



TRAILING EDGE SPAR

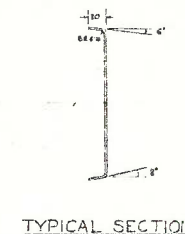
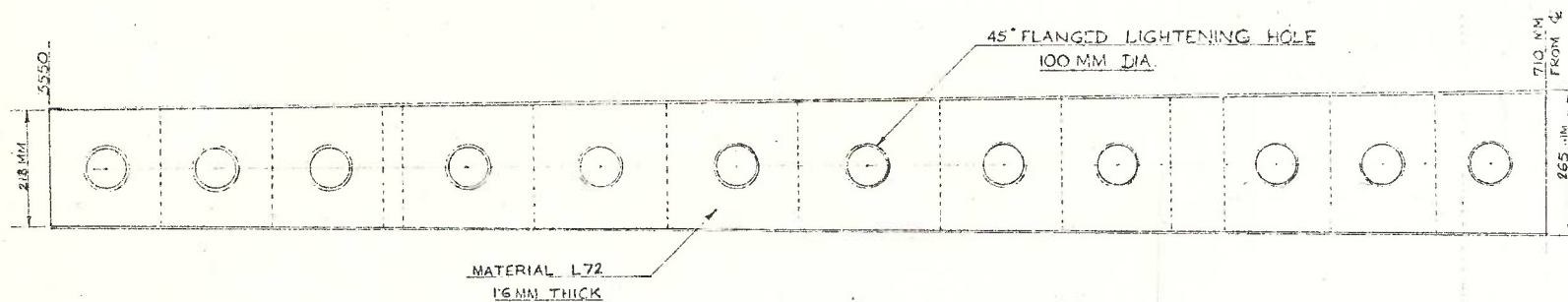
SCALE 1:1



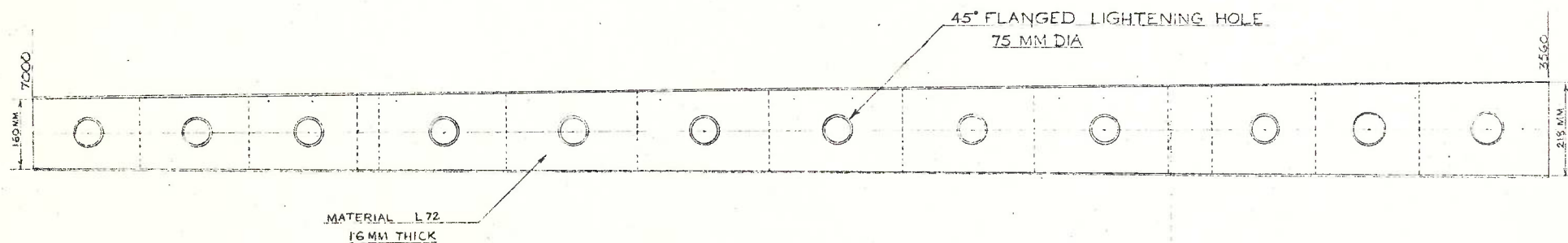
SECTION-BB (SEE E2)

SCALE 1:2

DRAWN	CHECK	APPROV	SCALE	TITLE:-
THOMAS			1:2	ELEVATOR DETAIL
COLLEGE OF AERONAUTICS				DR. WEST
CRANFIELD INSTITUTE OF TECHNOLOGY				A74-E3
CRANFIELD				SHEET

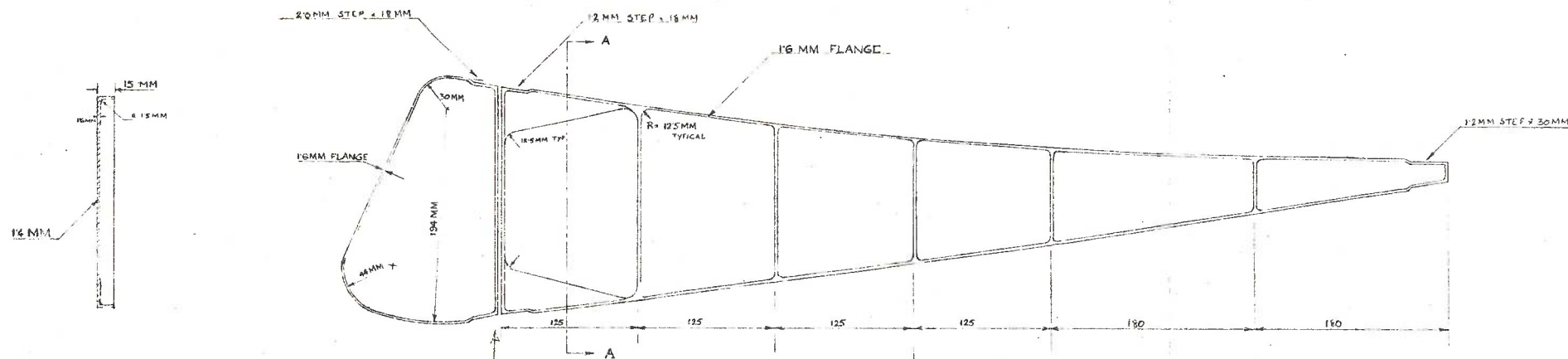


SPAR INBOARD ELEVATOR



SPAR OUTBOARD ELEVATOR

DRAWN	CHKD	APPR	SCALE	TITLE:-
2/11/20			1:20	ELEVATOR SPAR
COLLEGE OF AERONAUTICS CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD.				DRAWING NO. A 74-E4
SHEET				OF
				SHEETS



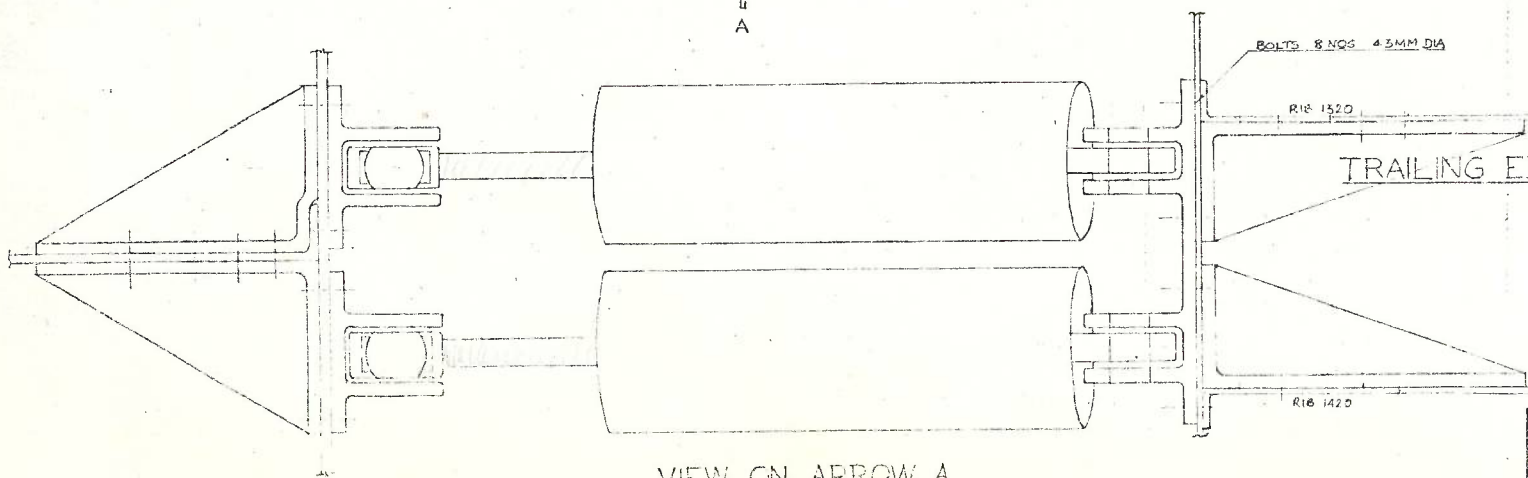
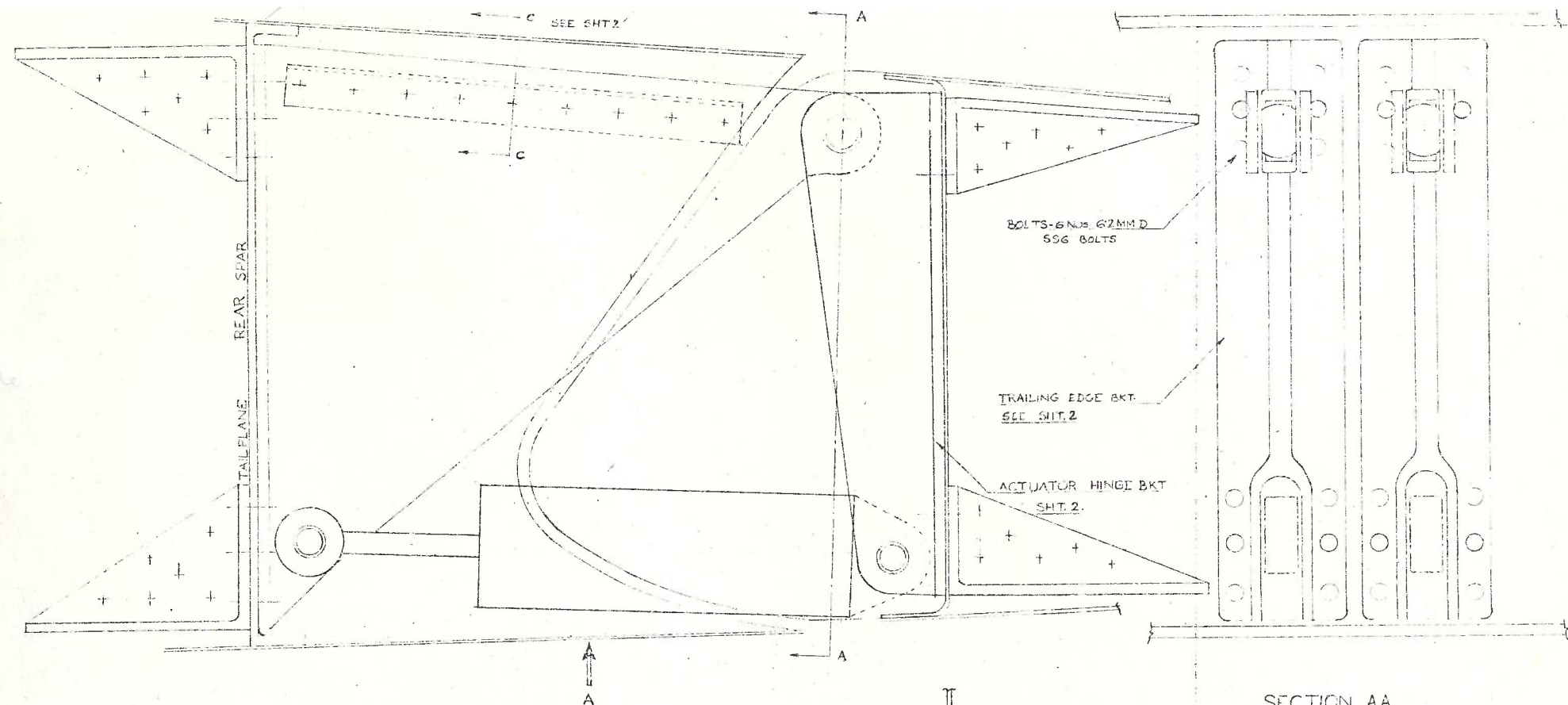
SECTION AA

HINGE RIB AT 1420 FROM € (INBOARD HINGE RIB-O/B SEGMENT)

NOTES

1. ALL DIMENSIONS IN MM
2. MATL. DTD 5020
3. ONLY MAJOR DIMENSIONS SHOWN
TOP AND BOTTOM CONTOUR AS
PER LOFT DRAWING
4. CORNER RADIUS 12.5 MM

DRAWN	CHK	APP'D	SCALE	TITLE: INBOARD ELEVATOR RIB AT STATION 1420
EM 12/1/68			1:1	
COLLEGE OF AERONAUTICS				DRAWN BY
CRANFIELD INSTITUTE OF TECHNOLOGY				A 74-E6
CRANFIELD				CUT OF SHEET

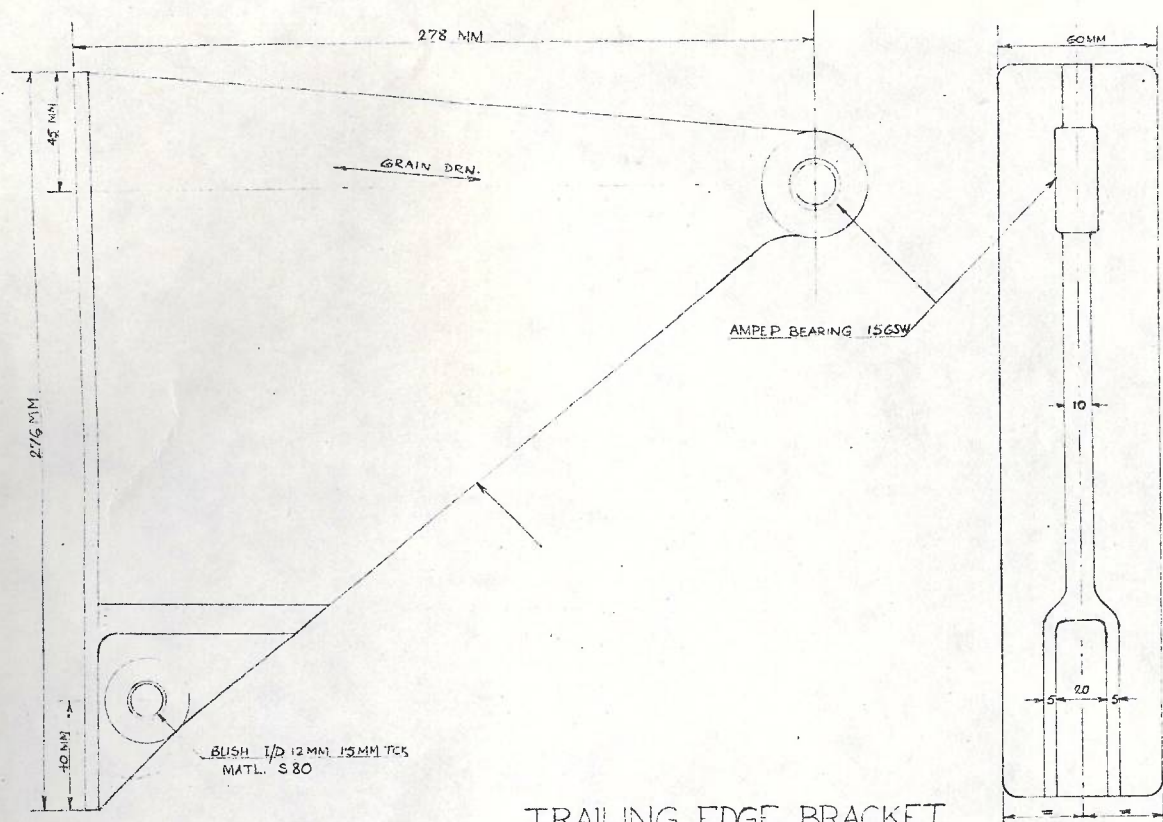


VIEW ON ARROW A

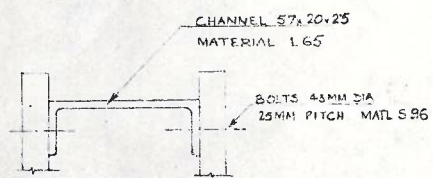
SECTION AA
ONLY TAIL PLANE BRACKET SHOWN

TRAILING EDGE FITTINGS AT STN. 1370

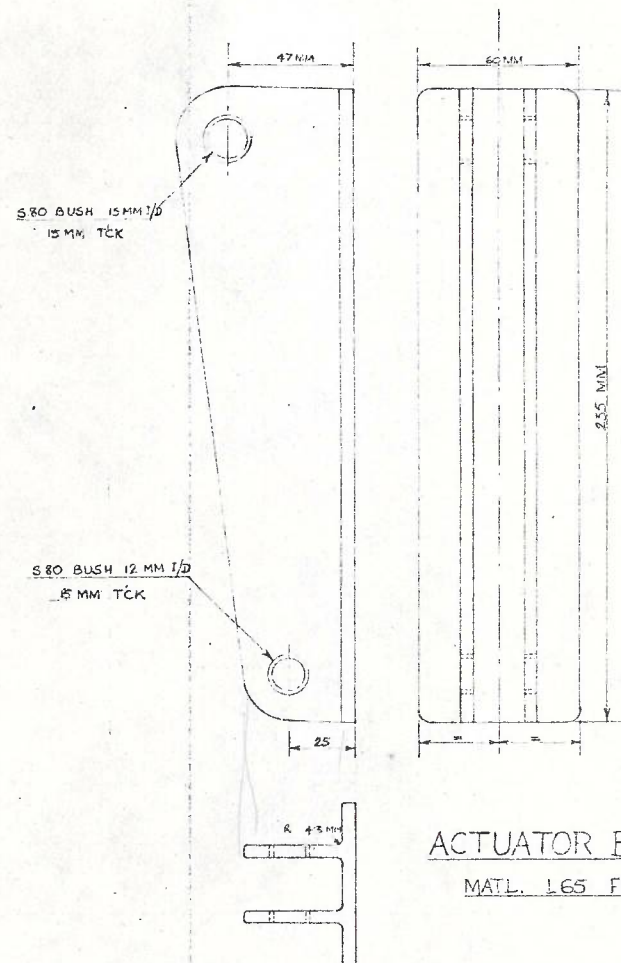
DRAWN	CHKD	APPROV	SCALE	TITLE
FOR STRAIGHT			1:1	GA OF ACTUATOR CRANFIELD
COLLEGE OF AERONAUTICS CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD.				DRAWING No. A 74-E7
SHEET 1 OF 1				REVISED



TRAILING EDGE BRACKET
MATERIAL L 65 FORGING



SECTION 'CC' (SEE DWG E 7)



ACTUATOR BRACKET
MATL. L 65 FORGING

DRAWN	CHKD	APPVD	SCALE	TITLE
			1:1	BRACKETS
COLLEGE OF AERONAUTICS CRANFIELD INSTITUTE OF TECHNOLOGY CRANFIELD.				DRAWING NO. A74-E8
SHEET 1 OF 1				SHEET 1