Department of Aerospace Engineering

AERONAUTICAL DESIGN HANDBOOK

For Use With Undergraduate Design Projects

Issue: 7

Date: January 2001

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Introduction

This handbook contains material properties and design data for use with design projects on the aeronautical undergraduate degree course.

Most engineering companies will have a Design Handbook that contains the values it expects its engineers to use. These are often commercially confidential as they contain data obtained at company expense and represent the engineering expertise the company is selling. In addition to hard data values and equations they will also contain guidelines on desirable design practice.

This booklet is somewhat restricted to those features that will be met during the design exercises. Real Design Handbooks are much large than this, often running to several volumes. However the style and the type of content is realistic.

In addition to handbooks the other main source of engineering design data are the ESDU data sheets.

CONTENTS

HIGH FLIGHT

John Gillespie Magee, Jr.

Oh, I have slipped the surly bonds of earth
And danced the skies on laughter-silvered wings;
Sunward I've climbed, and joined the tumbling mirth
Of sun-split clouds - and done a hundred things
You have not dreamed of - wheeled and soared and swung
High in the sunlit silence. Hov'ring there,
I've chased the shouting wind along, and flung
My eager craft through footless halls of air.
Up, up the long, delirious, burning blue..
I've topped the windswept heights with easy grace
Where never lark, or even eagle flew
And, while with silent, lifting mind I've trod
The high untrespassed sanctity of space,
Put out my hand, and touched the face of God.

John Gillespie Magee, Jr. was an American Spitfire pilot in the Royal Canadian Air Force. He was killed at the age of nineteen during the Battle of Britain.

SOME NOTES ON DESIGN LOAD DEFINITIONS

LIMIT LOAD - is the maximum load anticipated in operation. It can be considered as the specification for the component. Normally any uncertainties in the loading will be included when establishing the **limit load.**

FACTORS OF SAFETY - (or **Safety Factor**) to establish a "design to" load, the **limit load** is then multiplied by a **Factor of Safety**. This makes allowance for:

- uncertainties in design, and in manufacture
- variation in strength due to deterioration in service
- continued structural integrity after design loads have been exceeded

There are general two factors of safety considered the **Proof factor** and the **Ultimate factor, giving Proof Load** and **Ultimate Load** respectively.

PROOF LOAD - is **Limit Load x Proof factor.** It is a load that is acceptable as a test load after which the component can still be used in service - hence the name proof load. This means the loading keeps distortion due to plastic deformation to an acceptable level.

What constitutes an acceptable level must be defined in terms of allowable permanent deformation. The aim is to ensure that the part is strong enough for service, without weakening it in the test.

Typical **proof factors** are 1.0 for Civil aircraft and 1.125 for Military. The higher proof factor for military aircraft is due to increased likelihood of frequent loadings close to the proof load in operations.

ULTIMATE LOAD - is **Limit Load x Ultimate factor.** It is the maximum loading the component must withstand without failure - although plastic deformation is permitted and the component may not be fit for further service after withstanding the ultimate load.

A typical Ultimate factor for aerospace applications is 1.5 although other factors may be used where greater uncertainty exists in material properties or the basis of the strength determination.

PROOF STRESS - is the stress induced in the component by the **Proof Load.**

ULTIMATE STRESS - is the stress induced in the component by the **Ultimate Load**.

RESERVE FACTOR (R.F.) - Is a comparison of the **Allowable Stress** (that is the stress that the material will carry with a specified probability of survival, under specified environmental or loading conditions, such as elevated temperature or fatigue), with the **Proof** and **Ultimate Stress**. It is defined by the ratios:

$$\textbf{Ultimate R.F.} = \frac{\textbf{Allowable stress}}{\textbf{Ultimate Stress}}$$

and similarly for proof loads:

$$\textbf{Proof R.F.} = \frac{\textbf{Allowable Stress}}{\textbf{Proof Stress}}$$

Alternatively the R. F. can be calculated by finding and comparing the load at which the component will fail with the Proof and Ultimate Loads. For many components, such as those that fail by fatigue, this is not a feasible practice.

When giving the R.F. in stress reports it is traditional to highlight the value by putting it in a small box; like this:

$\boxed{\text{RF } 1.2}$

The R.F. should never be less than 1. If it is significantly greater than 1 it implies the component is over designed and so the goal should generally be to get close to 1 without dipping below it.

In cases where the method of calculating the design loads and strengths is uncertain, R.F.s greater than 1 are sometimes specified and this practice is adopted for design exercises here. (Another practice is to only alter the factor of safety, which keeps all the RFs at a single level). The critical issue is to ensure that all design decisions are properly recorded and that an unambiguous appraisal of the safety of the design can be made by a third party..

MARGIN OF SAFETY - The USA practice is to quote the **Margin of Safety** rather than the **Reserve Factor**. It is defined as:

$$\frac{\textbf{Ultimate Allowable Stress - Ultimate Stress}}{\textbf{Ultimate Stress}}$$

(with a similar definition for Proof loads)

It will be readily seen that the Margin of Safety corresponds to the RF minus one.
To be acceptable a **Margin of Safety** must not be negative (i.e. below zero).
Do not confuse **Margin of Safety with Factor of Safety**.

SOME NOTES ON DESIGN DOCUMENTATION

SPECIFICATIONS

Specifications are the documents that formally define what the design job is (these are not shown on the figure).

Requirement Specification: This document defines what the object is supposed to do and is normally written by the customer.

Design Specification: When dealing with big complex systems the contractor answers the customer's Requirement Specification with a Design Specification. This defines the engineering approach that will be used to meet the requirements.

REPORTS AND RECORDS

During the process of design a number of different types of document are produced. The key ones are outlined in the diagram and discussed below.

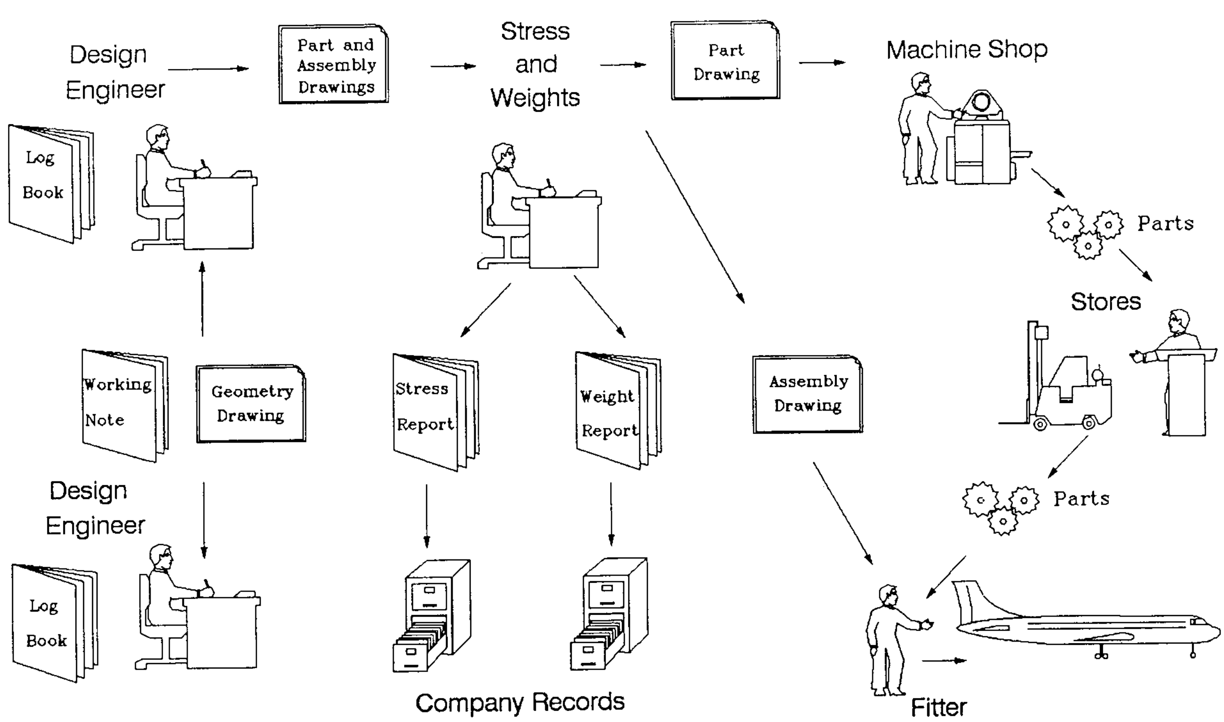
Log Book: This is kept by an engineer as a record of his work and calculations. It is not normally read by anyone other than himself. However Log Books are an official part of the design process, many companies issue Log Book numbers and control them like other documents.

Working Notes and Design Reports: Where material normally recorded in the Log Book needs wider distribution then Working Notes are used to formally report such design information. Different companies will have different terms and handling procedures for such documents. These would normally only be distributed between design engineers.

Stress Report: In the aerospace industry every drawing produced for manufacture must have been signed off by the Stress and Weights department. In the design exercises you will be expected to produce a stress and mass report to accompany your design drawings.

In industry the stress report would be produced by a separate specialist engineer who takes the manufacturing drawings produced by the design team and calculates the reserve factors. If judged strong enough a stress approval signature is added and the drawing can be given to manufacturing. The stress report is the company's official document proving the design is safe.

FLOW OF REPORTS AND DRAWINGS



The stress report should only give stressing sums for the parts on the drawing. DO NOT include calculations for all the options you rejected. In industry the engineer writing the report will not even know what these are. Generally no one is interested in how you decided on your design (although that information should be recorded in your log book) but everybody is very interested in whether it is strong enough. For each part you should look at the likely failure cases and report which cases you have examined, the formulae used, the arithmetic, and most important the reserve factor (in a box).

Weight Report: In the aerospace business weight or mass (the distinction is not closely observed in practical engineering) is a key part of the system's success. So in addition to calculating the strength of each drawing a calculation is made of the weight of each component. Again a record is kept for each drawing so that a calculation of the total mass properties of the aircraft or whatever can be made.

DRAWINGS

The Department specifies that engineering drawings should follow BSI drawing practice PP 7308.

A drawing should always define a job. It should contain all the information required to do that job unambiguously, <u>and nothing else</u>.

Geometry, Interface and Layout Drawings: These are used to exchange information between design engineers. What they contain is flexible, but should only contain information relevant to the purpose for which they are produced.

Part Drawings:
These are used to manufacture the parts. They should contain all the geometry and other information (like specifying the materials) needed to make the part.

It is not normal for a parts drawing to explain how to make the part. That is the job of a production engineer who will write a procedure for that purpose. However you should keep in your own mind possible ways of making it to keep your design practical.

It is not necessary to draw standard parts like bolts, nuts, rivets etc

> **WARNING:** 1:1 is always the preferred scale for a parts drawing. However, if it is not practical then move well away from 1:1. Never use half scale or double scale - long industrial experience shows you will inevitably get half size or double size parts.

Assembly Drawings:
Assembly drawings define how all the parts are to be put together. They are used in the assembly of the overall system as instructions for assembly.

An assembly drawing should contain a list of all the parts that are needed for the task it defines. Tradition has it that the list rises from the bottom of the drawing (where the title block normally is) and is numbered upwards (so you can always add more parts in later issues of the drawing).

Assembly drawing need only show how to put the thing together. They do not have to show what the parts look like - the person using the drawing should have the parts in his or her hand. Any dimensioning included should only be that needed to locate the parts on the overall system and other final assembly operations.

GAUGES FOR WIRES AND SHEET METAL

SWG (Standard Wire Gauge)

SWG	mm	SWG	mm	SWG	mm	SWG	mm	SWG	mm	SWG	mm	SWG	mm
0/7	12.7	0	8.23	10	3.25	20	0.914	30	0.315	40	0.122	50	0.025
6/0	11.8	1	7.62	11	2.95	21	0.813	31	0.295	41	0.112		
5/0	11.0	2	7.01	12	2.64	22	0.711	32	0.274	42	0.102		
4/0	10.2	3	6.40	13	2.34	23	0.610	33	0.254	43	0.091		
3/0	9.45	4	5.89	14	2.03	24	0.559	34	0.234	44	0.081		
00	8.84	5	5.38	15	1.83	25	0.508	35	0.213	45	0.071		
		6	4.88	16	1.63	26	0.457	36	0.193	46	0.061		
		7	4.47	17	1.42	27	0.417	37	0.173	47	0.051		
		8	4.06	18	1.22	28	0.376	38	0.152	48	0.041		
		9	3.66	19	1.02	29	0.345	39	0.132	49	0.030		

ISO (Metric) Sizes

Std.	Size in millimetres										
R10	0.020	0.040	0.080	0.160	0.315	0.63	1.25	2.50	5.0	10.0	20.0
R40	0.021	0.042	0.085	0.170	0.335	0.67	1.32	2.65	5.3	10.6	21.2
R20	0.022	0.045	0.090	0.180	0.355	0.71	1.40	2.80	5.6	11.2	22.4
R40	0.024	0.048	0.095	0.190	0.375	0.75	1.50	3.00	6.0	11.8	23.6
R10	0.025	0.050	0.100	0.200	0.400	0.80	1.60	3.15	6.3	12.5	25.0
R40	0.026	0.053	0.106	0.212	0.425	0.85	1.70	3.35	6.7	13.2	
R20	0.028	0.056	0.112	0.224	0.450	0.90	1.80	3.55	7.1	14.0	
R40	0.030	0.060	0.118	0.236	0.475	0.95	1.90	3.75	7.5	15.0	
R10	0.032	0.063	0.125	0.250	0.500	1.00	2.00	4.00	8.0	16.0	
R40	0.034	0.067	0.132	0.265	0.530	1.06	2.12	4.25	8.5	17.0	
R20	0.036	0.071	0.140	0.280	0.560	1.12	2.24	4.50	9.0	18.0	
R40	0.038	0.075	0.150	0.300	0.600	1.13	2.36	4.75	9.5	19.0	

Order of Preference R10, R20, R40

ALUMINIUM ALLOY 7075 EXTRUDED TUBES

OUTSIDE DIAMETER (mm)	WALL THICKNESS (mm)	OUTSIDE DIAMETER (mm)	WALL THICKNESS (mm)
20	R10: 0.50 to 2.50	35	R10: 0.80 to 5.00
22	R10: 0.50 to 2.50	40	R10: 1.00 to 5.00
25	R10: 0.50 to 5.00	45	R10: 1.00 to 5.00
28	R10: 0.50 to 5.00	50	R10: 1.00 to 8.00
30	R10: 0.80 to 5.00	55	R10: 1.00 to 8.00
32	R10: 0.80 to 5.00	60	R10: 1.00 to 8.00

STRENGTH OF SOLID RIVETS

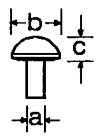
MADE FROM L37 ALUMINIUM ALLOY

Rivet Dia. (mm)			3	3.5	4	4.5	5
Max. Strength in single shear (kN)			1.8	2.4	3.1	3.9	4.9

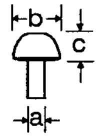
The above strengths apply only when the rivets are used to join aluminium alloy sheets, the rivet spacing $\geqslant$ 4 x rivet diameter (d), and when:

$$\frac{d}{t} \leqslant 2.85 \quad \text{where } t = \text{thickness of thinner sheet or plate.}$$

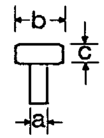
TYPES OF RIVET



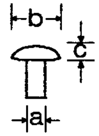
Mushroom Head
b = 2 a
c = 0.625 a



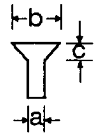
Round Head
b = 2 a
c = 0.75 a



Flat Head
b = 2 a
c = 0.4 a

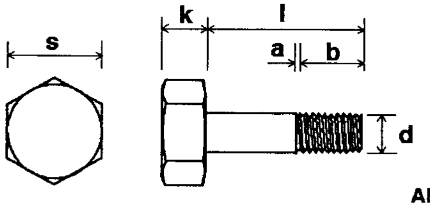


Brazier
b = 2 a
c = 0.5 a



Countersunk Head
b = 1.81 a
c = 0.5 a

HEXAGONAL HEAD BOLTS AND SCREWS



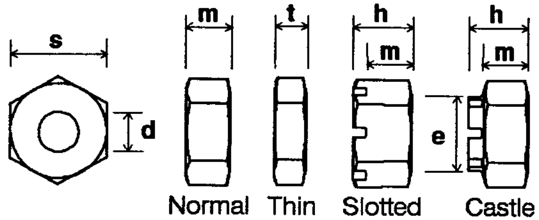
$l \leqslant 125$, $b = 2d + 6$
$125 \leqslant l \leqslant 200$, $b = 2d + 12$
$200 \leqslant l$, $b = 2d + 25$

$a \approx 2.5 \times$ pitch

NOTES
All dimensions in millimetres.
All sizes shown in brackets are non-prefered

SIZE	THREAD PITCH	DIAMETER (d) max.	DIAMETER (d) min.	FLATS WIDTH (s) max.	FLATS WIDTH (s) min.	HEAD HEIGHT (k) max.	HEAD HEIGHT (k) min.
M1.6	0.35	1.6	1.46	3.2	3.08	1.225	0.975
M2	0.4	2.0	1.86	4.0	3.88	1.525	1.275
M2.5	0.45	2.5	2.36	5.0	4.88	1.825	1.575
M3	0.5	3.0	2.86	5.5	5.38	2.125	1.875
M4	0.7	4.0	3.82	7.0	6.85	2.925	2.675
M5	0.8	5.0	4.82	8.0	7.85	3.650	3.35
M6	1	6.0	5.82	10.0	9.78	4.15	3.85
M8	1.25	8.0	7.78	13.0	12.73	5.65	5.35
M10	1.5	10.0	9.78	17.0	16.73	7.18	6.82
M12	1.75	12.0	11.73	19.0	18.67	8.18	7.82
(M14)	2	14.0	13.73	22.0	21.67	9.18	8.82
M16	2	16.0	15.73	24.0	23.67	10.18	9.82
(M18)	2.5	18.0	17.73	27.0	26.67	12.215	11.785
M20	2.5	20.0	19.67	30.0	29.67	13.215	12.785
(M22)	2.5	22.0	21.67	32.0	31.61	14.215	13.785
M24	3	24.0	23.67	36.0	35.38	15.215	14.785
(M27)	3	27.0	26.67	41.0	40.38	17.215	16.785
M30	3.5	30.0	29.67	46.0	45.38	19.26	18.74
(M33)	3.5	33.0	32.61	50.0	49.38	21.26	20.74
M36	4	36.0	35.61	55.0	54.26	23.26	22.74
(M39)	4	39.0	38.61	60.0	59.26	25.26	24.74
M42	4.5	42.0	41.61	65.0	64.26	26.26	25.74
(M45)	4.5	45.0	44.61	70.0	69.26	28.26	27.74
M48	5	48.0	47.61	75.0	74.26	30.36	29.74
(M52)	5	52.0	51.54	80.0	79.26	33.31	32.69
M56	5.5	56.0	55.54	85.0	84.13	35.31	34.69
(M60)	5.5	60.0	59.54	90.0	89.13	38.31	37.69
M64	6	64.0	63.54	95.0	94.13	40.31	39.69
(M68)	6	68.0	67.54	100.0	99.13	43.31	42.69

HEXAGONAL NUTS

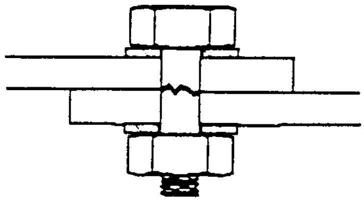


NOTES
All dimensions in millimetres.
All sizes in shown in brackets are non-prefered.
s & d as for equivalent bolt.
Slotted nut M4 to M39 only.
Castle nut (6 slots) M12 to M39 only.
Castle nut (8 slots) M42 to M68 only.

SIZE	NORMAL (m) max.	NORMAL (m) min.	THIN (t) max.	THIN (t) min.	SLOTTED (h) max.	SLOTTED (h) min.	DIAMETER (e) max.	DIAMETER (e) min.
M1.6	1.3	1.05	-	-	-	-	-	-
M2	1.6	1.35	-	-	-	-	-	-
M2.5	2.0	1.75	-	-	-	-	-	-
M3	2.4	2.15	-	-	-	-	-	-
M4	3.2	2.90	-	-	5.0	4.70	-	-
M5	4.0	3.70	-	-	6.0	5.70	-	-
M6	5.0	4.70	-	-	7.5	7.14	-	-
M8	6.5	6.14	5.0	4.70	9.5	9.14	-	-
M10	8.0	7.64	6.0	5.70	12.0	11.57	-	-
M12	10.0	9.64	7.0	6.64	15.0	14.57	17.0	16.57
(M14)	11.0	10.57	8.0	7.64	16.0	15.57	19.0	18.48
M16	13.0	12.57	8.0	7.64	19.0	18.48	22.0	21.48
(M18)	15.0	14.57	9.0	8.64	21.0	20.48	25.0	24.48
M20	16.0	15.57	9.0	8.64	22.0	21.48	28.0	27.48
(M22)	18.0	17.57	10.0	9.64	26.0	25.48	30.0	29.48
M24	19.0	18.48	10.0	9.64	27.0	26.48	34.0	33.38
(M27)	22.0	21.48	12.0	11.57	30.0	29.48	38.0	37.38
M30	24.0	23.48	12.0	11.57	33.0	32.38	42.0	41.38
(M33)	26.0	25.48	14.0	13.57	35.0	34.38	46.0	45.38
M36	29.0	28.48	14.0	13.57	38.0	37.38	50.0	49.38
(M39)	31.0	30.38	16.0	15.57	40.0	39.38	55.0	54.26
M42	34.0	33.38	16.0	15.57	46.0	45.38	58.0	57.26
(M45)	36.0	35.38	18.0	17.57	48.0	47.38	62.0	61.26
M48	38.0	37.38	18.0	15.57	50.0	49.38	65.0	64.26
(M52)	42.0	41.38	20.0	19.48	54.0	53.26	70.0	69.26
M56	45.0	44.38	-	-	57.0	56.26	75.0	74.26
(M60)	48.0	47.38	-	-	63.0	62.26	80.0	79.26
M64	54.0	53.26	-	-	66.0	65.26	85.0	84.13
(M68)	54.0	53.26	-	-	69.0	68.26	90.0	89.13

PIN/LUG FAILURE MODES

SINGLE SHEAR FAILURE

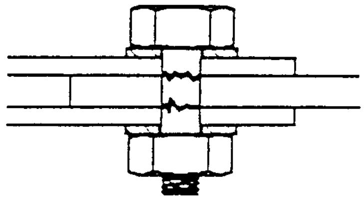


Design for $\tau \leq \tau_a$

Where $\tau = P/\pi(d/2)^2$ = shear stress at ultimate loading

and τ_a = allowable shear strength

DOUBLE SHEAR FAILURE

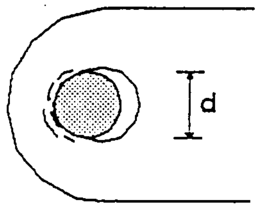


Design for $\tau \leq \tau_a$

Where $\tau = P/2\pi(d/2)^2$ = shear stress at ultimate loading

and τ_a = allowable shear strength

BEARING FAILURE

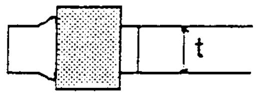


Design for $\sigma_{br} \leq \sigma_{bra}$

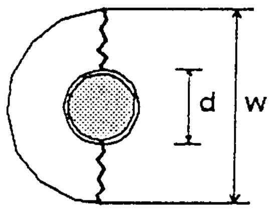
Where $\sigma_{br} = P/td$ = bearing stress at ultimate loading

and σ_{bra} = allowable bearing strength

Bearing failure can occur in either the pin or the lug depending upon which has the lower bearing strength.



TENSILE FAILURE

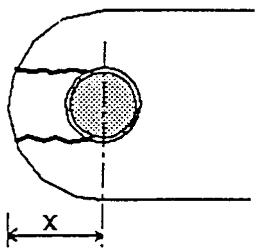


Design for $\sigma \leq \sigma_a$

Where $\sigma = P/(w-d)t$ = direct stress at ultimate loading

and σ_a = allowable tensile strength

TEAR-OUT FAILURE



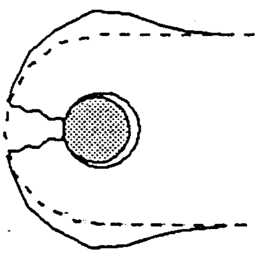
Design for $\tau \leq \tau_a$

Where $\tau = P/2xt$ = shear stress at ultimate loading

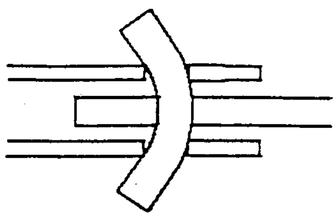
and and τ_a = allowable shear strength

This will be slightly lower than bulk material shear stress

ALSO POSSIBLE



BURSTING FAILURE



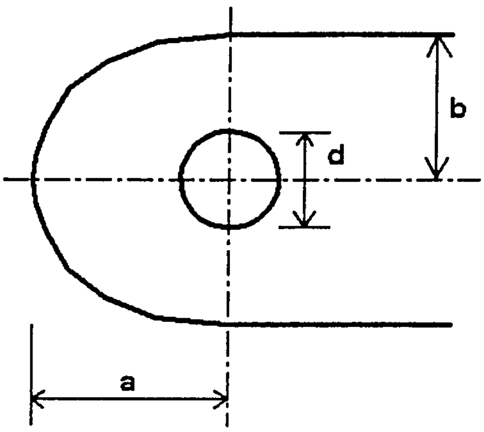
PIN BENDING FAILURE

GENERAL DESIGN GUIDELINE 1 - HOLE DIAMETER

Ideally a structure would be close to all its failure cases at the same time. So for the pin this would mean shear strength being roughly equal to the pin bearing strength. With typical metallic material properties this occurs (very roughly) when: $d \approx 3.5t$. Therefore design for:

$$d/t \leq 3.5$$ where t = thickness of thinner sheet or plate

GENERAL DESIGN GUIDELINE 2 - LUG SIZING



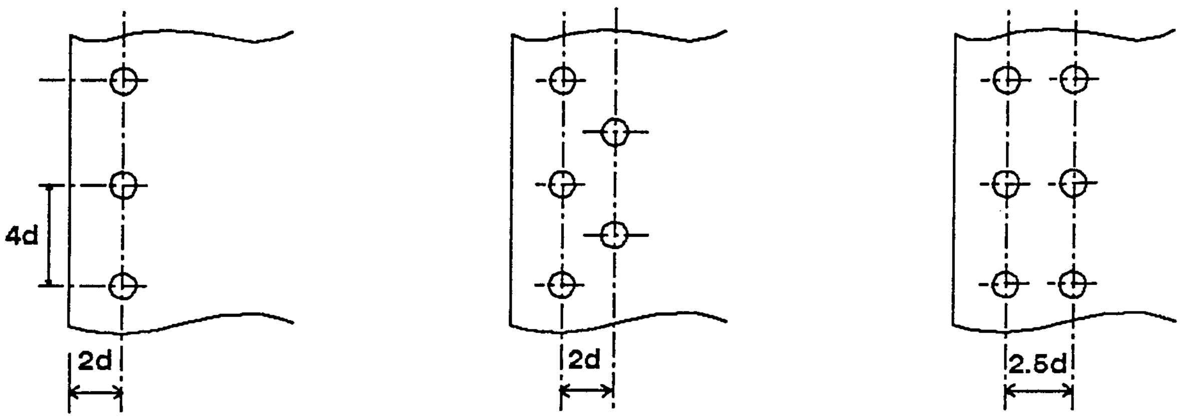
As with the pin we would like all the different strengths to be roughly equal for a good structure. Consider the three main types of lug failure, bearing, tensile and tear-out and Using typical metallic material properties this roughly occurs when: $a \approx 2d$ and $b \approx 1.5d$

Therefore design for:

$a/d \geq 2$ and $b/d \geq 1.5$

GENERAL DESIGN GUIDELINES 3 - CLUSTERS

Minimum spacings



Single Row Zig-Zag Stagger In Line Chain

Similar considerations to lug guidelines but generally we need to give more allowance to tensile failure.

These are suitable for protruding head rivets/bolts. Countersunk heads will need greater spacing to account for missing material typically:

Row spacing $\approx$ 5.5 d and Edge distance $\approx$ 2.5 d

Please note these are not rules! They are aids to quick design and are not always appropiate.

PROPERTIES OF METALLIC MATERIALS

Warning: Some simplifications and omissions are included in this table to aid its role to support teaching. Also the data is unconfirmed and may be inaccurate. The data should be not be used for any purpose beyond the Aeronautical design projects.

Where no Ultimate Bearing or Ultimate Compression data is provided the traditional method is to use the same value as the Ultimate Tensile Strength although this is normally a pessimistic assumption.

STEELS

Material: **Low Carbon and Alloy Steels**

Form:	Sheet, Strip, Plate, and Tubing (< 4.75 mm)	(> 4.75 mm)
Density Kg/m^3	7840	7840
Youngs Modulus (E) GPa:	200	200
Ultimate Tensile N/mm^2:	655	620
Ultimate Shear N/mm^2:	375	375
Ultimate Bearing N/mm^2:	965	965

Uses: Low stress applications, general structural use. Cost in mid 1980's £200/tonne

Material: **0.6%C Steel**

Form:	all
Density Kg/m^3	7640
Youngs Modulus (E) GPa:	200
Ultimate Tensile N/mm^2:	825
Ultimate Shear N/mm^2:	690

Uses: Medium stress applications including fasteners. Cost in mid 1980's £180/tonne

Material: **Stainless steel**

Form:	Sheet & Plate	Bars & Forgings
Density Kg/m^3	7640	7640
Youngs Modulus (E) GPa:	200	200
Ultimate Tensile N/mm^2:	1380	1275
Ultimate Shear N/mm^2:	895	825
Ultimate Bearing N/mm^2:	2275	2105

Uses: High temperature or high corrosion environments such as chemical or steam plants. Cost in mid 1980's £1100/tonne

ALUMINIUM

Material: **Pure Aluminium**

Density Kg/m^3:	2700
Youngs Modulus (E) GPa:	69
Ultimate Tensile N/mm^2:	70

Uses: Weak but ductile; non-structural material used for temperature or electrical conduction. Cost in mid 1980's £910/tonne.

Material: **Aluminium Alloy 2024 (T351)**

Form:	Plate, Extrusion, Forging
Density Kg/m^3:	2800
Youngs Modulus (E) GPa:	72
Shear Modulus (G) Gpa:	28
Poisson ratio (v)	0.3
Ultimate Tensile N/mm^2:	440
Ultimate Shear N/mm^2:	255
Bearing N/mm^2:	284
0.2% proof stress	280
% Elongation	10

Uses: A strong age-hardening alloy for aircraft skins, forgings, spars, lightweight structures. Suitable for use where fatigue is important and primary load is in tension. Cost in mid 1980's £1100/tonne.

Material: **Aluminium Alloy 7075**

Form:	Plate	Extrusion	Forging
Density Kg/m^3:	2800	2800	2800
Youngs Modulus (E) GPa:	71	71	71
Ultimate Tensile N/mm^2:	460	425	455
Ultimate Shear N/mm^2:	295	310	297
Ultimate Bearing N/mm^2:	720	670	587

Uses: A strong age-hardening alloy for aircraft forgings, spars, upper wing skins where fatigue is less critical, lightweight structures. Cost in mid 1980's £1100/tonne.

Material: **Aluminium Alloy L163**

Form:	Sheet
Density Kg/m^3:	2800
Youngs Modulus (E) GPa:	70
Ultimate Tensile N/mm^2:	385
Ultimate Shear N/mm^2:	230
Ultimate Bearing N/mm^2:	
0.2% Proof stress N/mm^2	245

Uses: Strong age-hardening alloy typically for aircraft skins. Cost in mid 1980's £1100/tonne.

Material: **Aluminium Alloy L72 (DTD 610)**

Form:	Sheet
Density Kg/m^3:	2800
Youngs Modulus (E) GPa:	70 (Also See Tangent Modulus Curve)
Shear Modulus (G) Gpa:	27
Ultimate Tensile N/mm^2:	345
Ultimate Shear N/mm^2:	200
Ultimate Bearing N/mm^2:	
0.1% Proof stress N/mm^2	230
0.1% Proof shear stress N/mm^2:	132

Notes: Standard Thickness SWG 10,12,14,16,20,22,24
Minimum Bend radius 2 x sheet thickness
Uses: Strong age-hardening alloy typically for aircraft skins. Cost in mid 1980's £1100/tonne.

Material: **Aluminium Alloy L65**

Form:	Forging
Density Kg/m^3:	2800
Youngs Modulus (E) GPa:	70 (Also See Tangent Modulus Curve)
Shear Modulus (G) Gpa:	27
Ultimate Tensile N/mm^2:	450
Ultimate Shear N/mm^2:	260
Ultimate Bearing N/mm^2:	
0.1% Proof stress N/mm^2	400
0.1% Proof shear stress N/mm^2:	230

Notes: Available in billets > 10 mm thick.
Minimum Machinable Thickness 0.75 mm.
Uses: Strong age-hardening alloy typically for lightweight forgings and rivets. Cost in mid 1980's £1100/tonne.

OTHER METALS

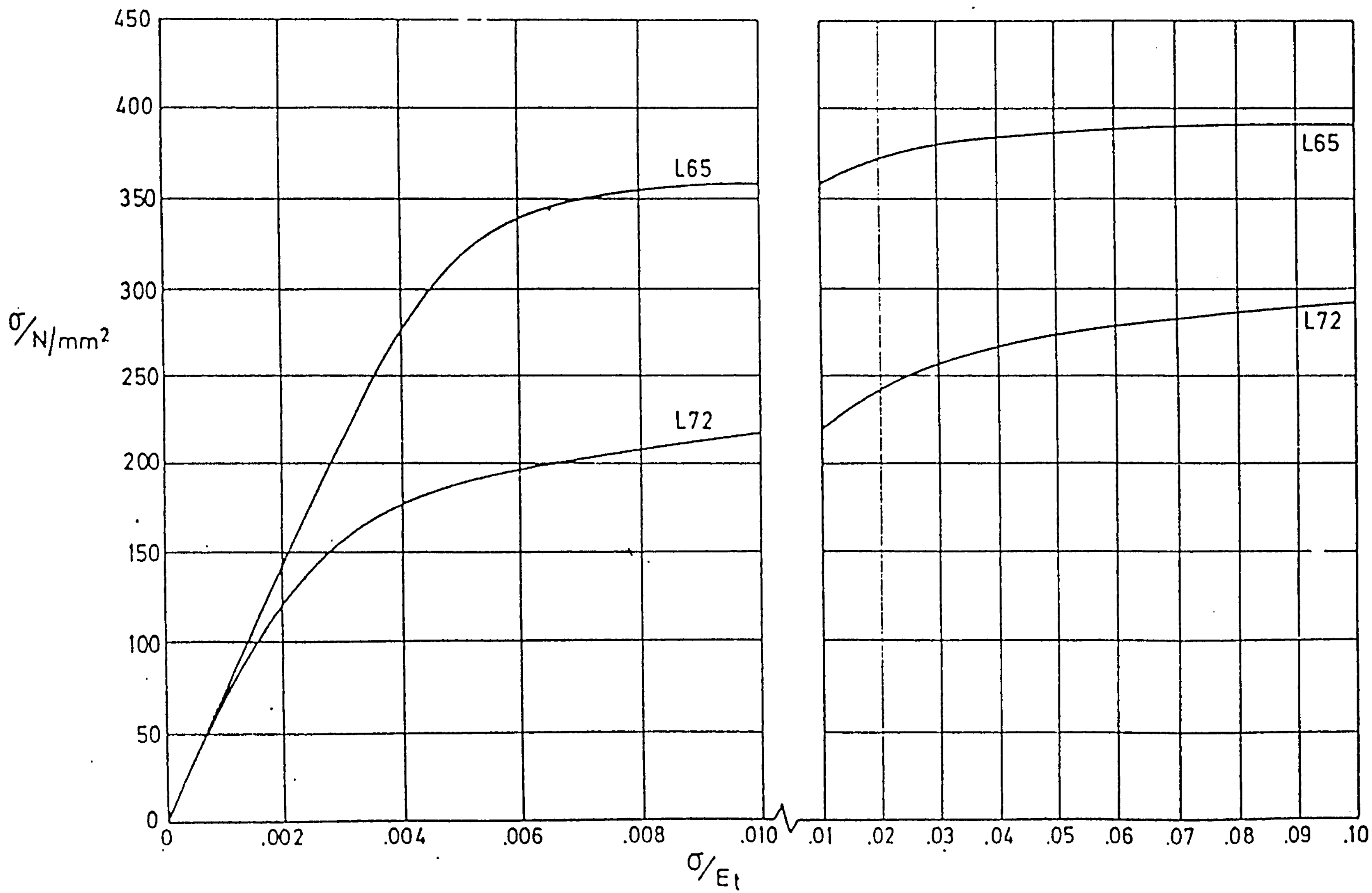
Material:	Titanium Alloy 6A14V	
Form:	Bar	Sheet
Density Kg/m^3	4430	4430
Youngs Modulus (E) GPa:	110	105
Ultimate Tensile N/mm^2:	895	895
Ultimate Shear N/mm^2:	550	522
Ultimate Bearing N/mm^2:	1200	1315

Uses: Light Strong, excellent corrosion resistance, high melting point, good creep resistance; used for turbofans, airframes, chemical plant, surgical implants. Cost in mid 1980's £4630/tonne

Material:	Cast Iron
Form:	All
Density Kg/m^3	7640
Youngs Modulus (E) GPa:	170
Ultimate Tensile N/mm^2:	370
Ultimate Shear N/mm^2:	330

Uses: Low stress uses, e.g. cylinder blocks, drain pipes. Cost in mid 1980's £120/tonne

Material:	Magnesium Alloy HK31A	
Form:	Sheet	Casting
Density Kg/m^3	1790	1790
Youngs Modulus (E) GPa:	45	45
Ultimate Tensile N/mm^2:	205	185
Ultimate Shear N/mm^2:	150	?
Ultimate Bearing N/mm^2:	295	?



TANGENT MODULUS (E_t) CURVES FOR L72 AND L65 MATERIALS

PROPERTIES OF COMPOSITE MATERIALS

Typical Generic Material Data

For preliminary design use only

For uni-directional tape and woven fabric composites

UD Tapes	**Stiffnesses**				**Strengths**				
	KN/mm^2				Nmm^2				
(60% Vf)	E_1	E_2	G_{12}	ν_{12}	σ_{1t}	σ_{1c}	σ_{2t}	σ_{2c}	τ_{12}
HSCFEP	140	10	5	0.3	1500	-1200	50	-250	70
HMCFEP	180	8	5	0.3	1000	-850	40	-200	60
EGFEP	40	8	4	0.25	1000	-600	30	-110	40
KFEP	75	6	2	0.34	1300	-280	30	-40	60

Typical Ply thickness: 0.125 - 0.2 mm

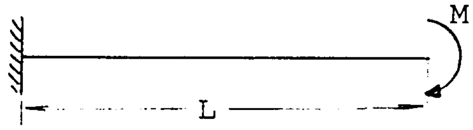
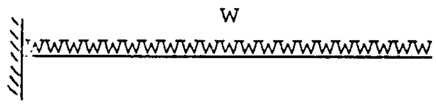
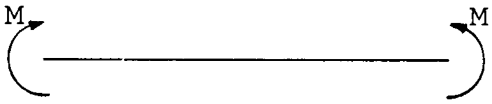
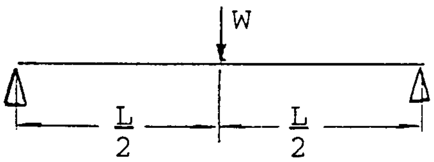
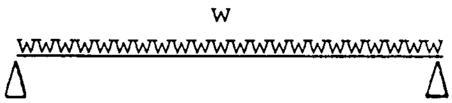
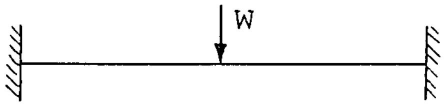
Fabrics	**Stiffnesses**				**Strengths**				
	KN/mm^2				Nmm^2				
(50% Vf)	E_1	E_2	G_{12}	ν_{12}	σ_{1t}	σ_{1c}	σ_{2t}	σ_{2c}	τ_{12}
HSCFEP	70	70	5	0.10	600	-570	600	-570	90
HMCFEP	85	85	5	0.10	350	-150	350	-150	35
EGFEP	25	25	4	0.20	440	-425	440	-425	40
KFEP	30	30	5	0.20	480	-190	480	-190	50

Typical Ply thickness: 0.25 - 0.4 mm

HSCFEP = High Strength Carbon Fibre Epoxy
HMCFEP = High Modulus Carbon Fibre Epoxy
EGFEP = E-Glass Fibre Epoxy
KFEP = Kevlar Fibre Epoxy

STANDARD BEAM BENDING EXPRESSIONS

Symmetric beams in bending about principal axes

	Max B.M	End slope	Max Deflection
		$\frac{ML}{EI}$	$\frac{ML^2}{2EI}$
	WL	$\frac{WL^2}{2EI}$	$\frac{WL^3}{3EI}$
	$\frac{wL^2}{2}$	$\frac{wL^3}{6EI}$	$\frac{wL^4}{8EI}$
		$\frac{ML}{2EI}$	$\frac{ML^2}{8EI}$
	$\frac{WL}{4}$	$\frac{WL^2}{16EI}$	$\frac{WL^3}{48EI}$
	$\frac{wl^2}{8}$	$\frac{wL^3}{24EI}$	$\frac{5wL^4}{384EI}$
	$\frac{WL}{8}$		$\frac{WL^3}{192EI}$

L = Beam length
I = Second moment of area about centroidal axes of beam cross section
W = Load (force)
w = Uniform load per unit length
M = Bending moment
σ = stress due to bending
y = distance from neutral axis
R = Radius of Curvature

Bending:

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$$

SMALL DEFLECTION (EULER) BUCKLING

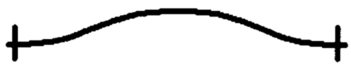
$$\text{Buckling Strength } F = \frac{a\,E\,I}{L^2}$$

Where: a = constant due to constraints; E = Young's Modulus;
I = Second Moment of Area: L = length between contraints.

Values of a

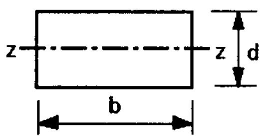
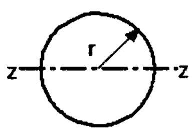
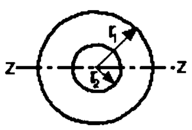
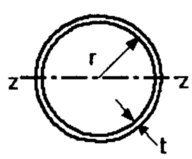
Pinned-pinned $a = \pi^2$

Pinned-fixed $a = 2.04\,\pi^2$



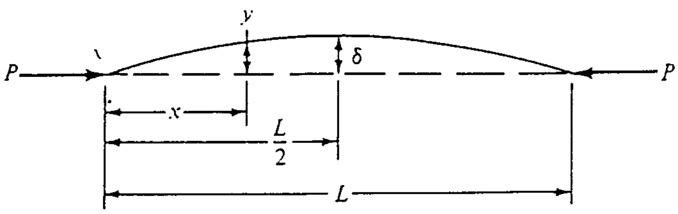
Fixed-fixed $a = 4\,\pi^2$

PROPERTIES OF CROSS SECTIONS

	Area	I_{zz} Second Moment of Area
	$a\,.\,b$	$\frac{b\,.\,d^3}{12}$
	πr^2	$\frac{\pi r^4}{4}$
	$\pi(r_1^2 - r_2^2)$	$\frac{\pi}{4}(r_1^4 - r_2^4)$
	$2\pi\, r\,.\,t$	$\pi\, r^3.\,t$

BUCKLING DESIGN OF STRUCTURAL MEMBERS

LONG COLUMNS



critical, or buckling, load P_{cr}

Euler load

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

Euler equation

buckling stress $\sigma_{cr} = P_{cr}/A$

can be written as :

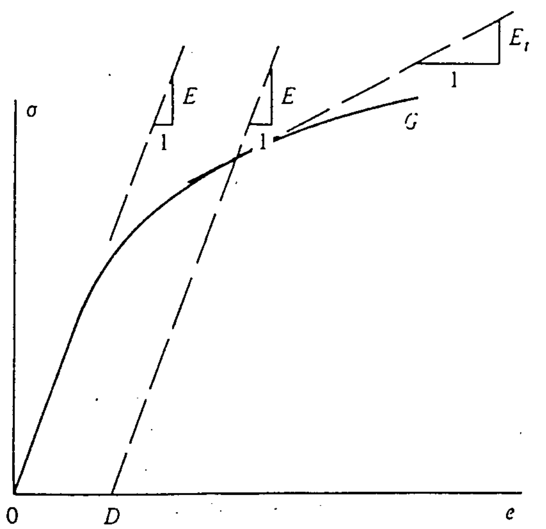
$$\sigma_{cr} = \frac{\pi^2 E}{(L/\rho)^2} \qquad = \pi^2 E \left(\frac{\rho}{L}\right)^2$$

where:

$\rho = \sqrt{I/A}$ radius of gyration of the cross-sectional area,

and L/ρ = "slenderness ratio"

SHORT COLUMNS



the *tangent modulus of elasticity* E_t.

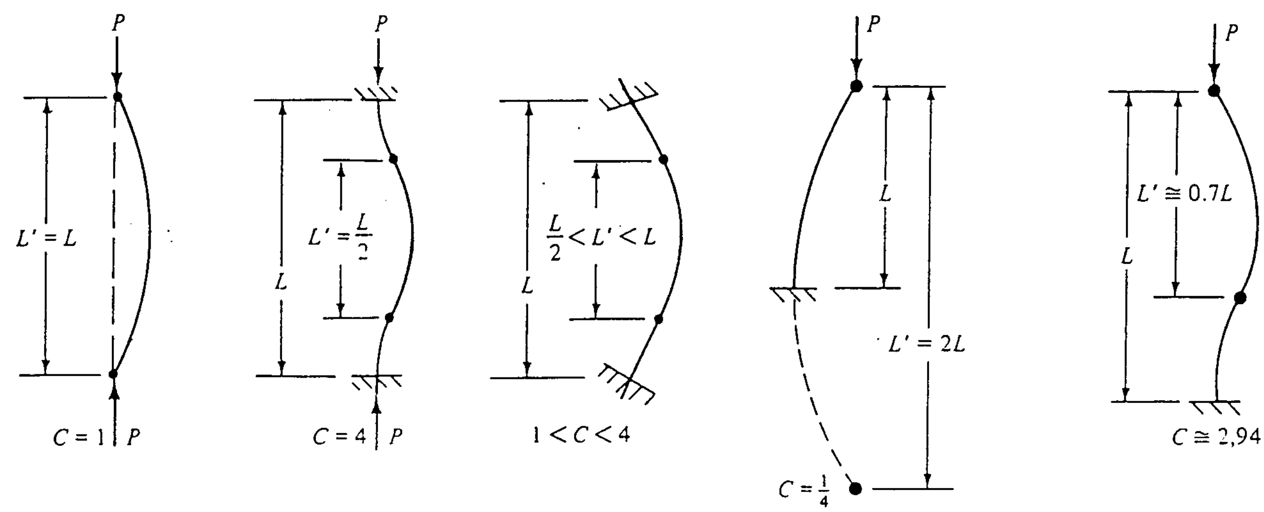
$$P_{cr} = \frac{\pi^2 E_t I}{L^2}$$

$$\sigma_{cr} = \frac{\pi^2 E_t}{(L/\rho)^2} \qquad = \pi^2 E_t \left(\frac{\rho}{L}\right)^2$$

Most flight vehicle materials have stress-strain curves similar to that shown

COLUMN END FIXITY

At points of contraflexure there is no curvature and hence no bending moment. The portion of the column between points of contraflexure thus may be treated as a pin-ended column. The length L' between the points of contraflexure is used in place of L in the column equations previously derived, and the slenderness ratio is defined as L'/ρ.



end-fixity term c

$$\sigma_{cr} = \frac{\pi^2 E}{(L'/\rho)^2} = \frac{c\pi^2 E}{(L/\rho)^2} \quad \text{for long columns} \qquad \text{ie.: } \sigma_{cr} = c\pi^2 E\left(\frac{\rho}{L}\right)^2$$

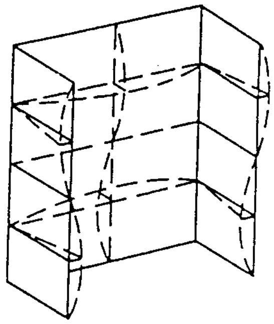
$$L' = \frac{L}{\sqrt{c}}$$

where $c = \left(\frac{L}{L'}\right)^2$ = "end fixity term"

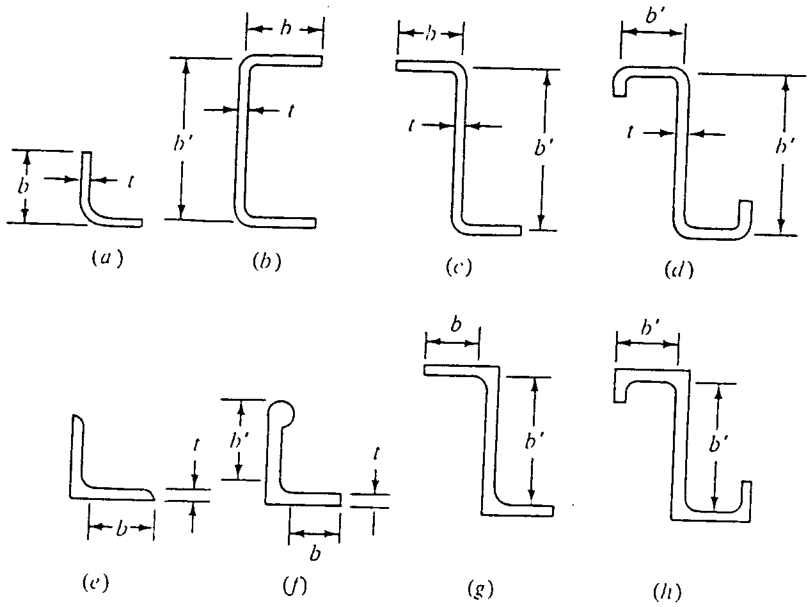
Note, the effect of end fixity on buckling strength is much less for short columns.

COLUMNS SUBJECT TO LOCAL CRIPPLING FAILURE

Columns of extrusions or bent sheets with thin walls.



Effective widths of column walls



Approximate total column crippling stress is given by the sum of plastic buckling strengths of the effective widths of the flat sheet elements of the cross section.

ie, total column crippling stress:

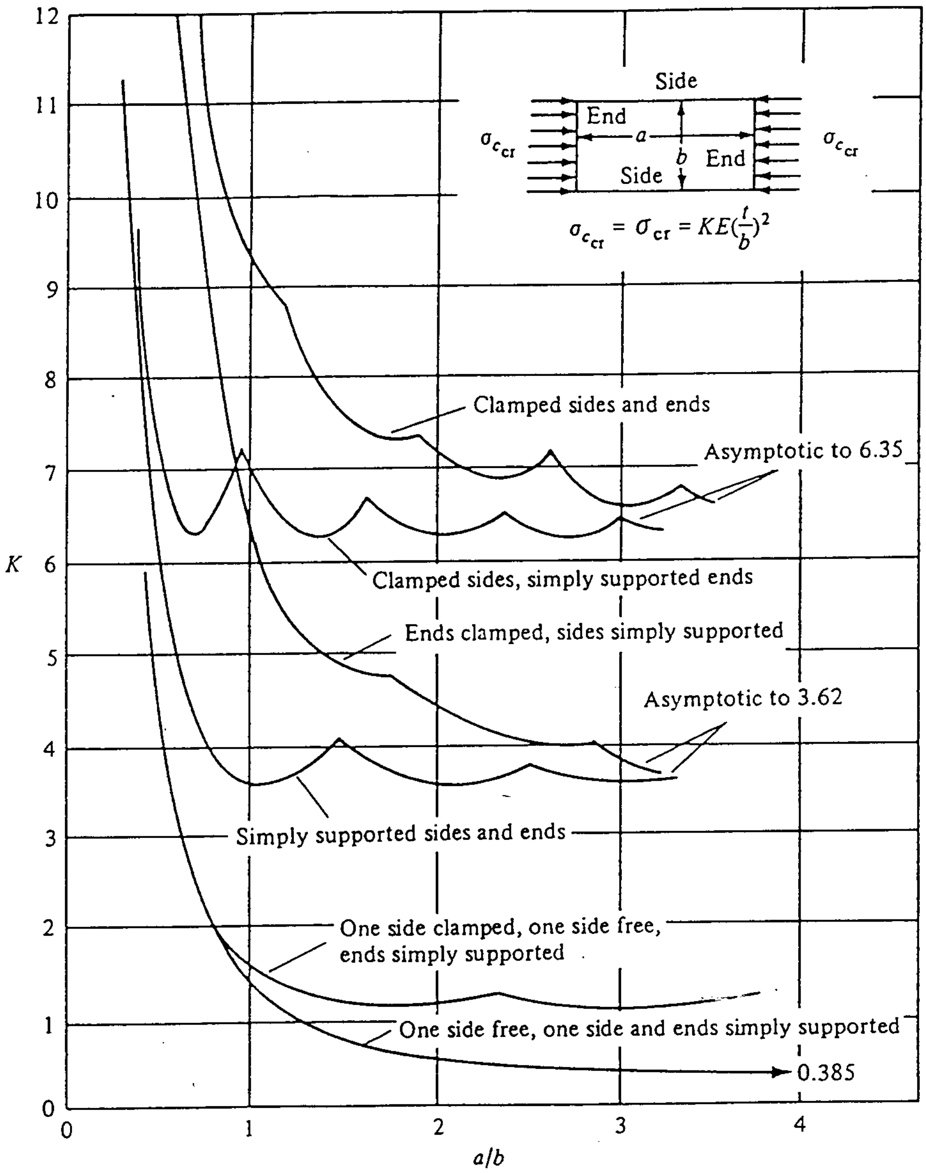
$$\sigma_{cc} = \frac{\sigma_1 b_1 t_1 + \sigma_2 b_2 t_2 + \sigma_3 b_3 t_3}{b_1 t_1 + b_2 t_2 + b_3 t_3} = \frac{\Sigma \sigma b t}{\Sigma b t}$$

- for rectangular areas with dimensions $b_1 t_1$, $b_2 t_2$, and $b_3 t_3$ and buckling stresses σ_1, σ_2, and σ_3,

Note, the denominator of the above expression may not be equal to the actual total area because the corners may not be included.

The total column crippling load may be obtained by multiplying the stress σ_{cc} by the actual total cross section area.

BUCKLING OF FLAT PLATES IN COMPRESSION



Where

for $\nu = 0.3$

K = Plate buckling constant, depending on plate aspect ratio and edge conditions (Here the value includes a poisson ratio of 0.3 for the plate material)

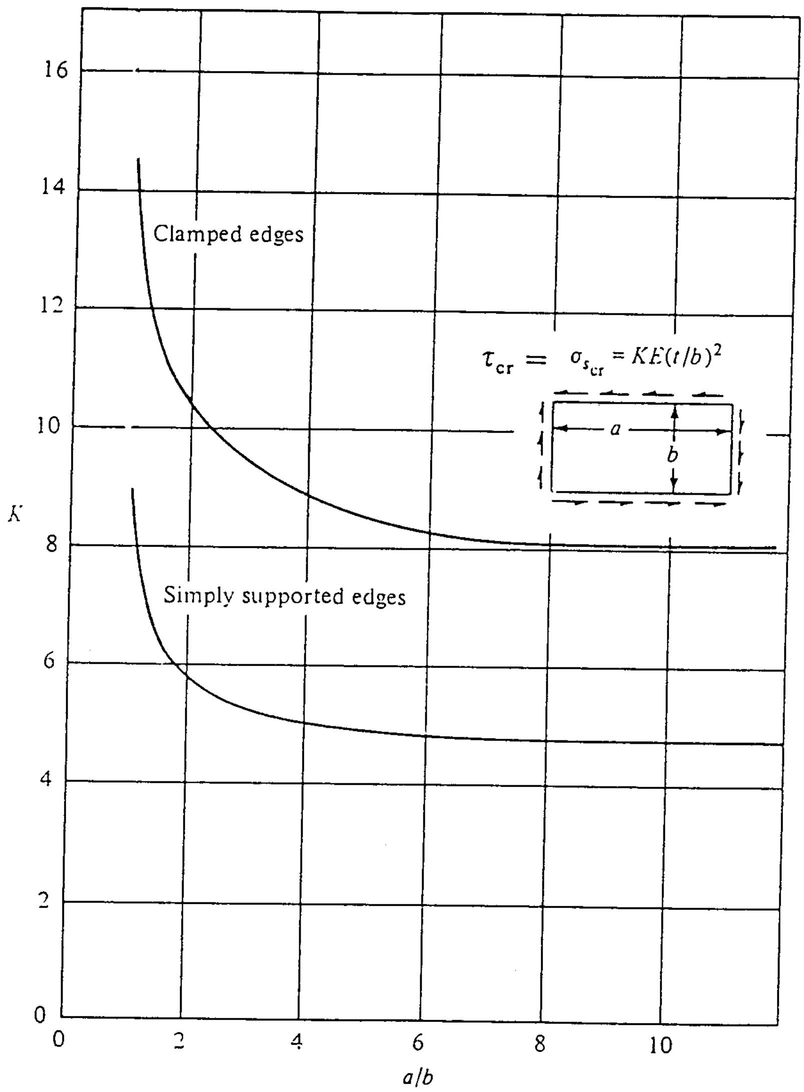
E = Youngs modulus of plate material for elastic buckling

E = Et = Tangent modulus of plate material for plastic buckling (Check tangent modulus curves for plate material)

t = plate thickness

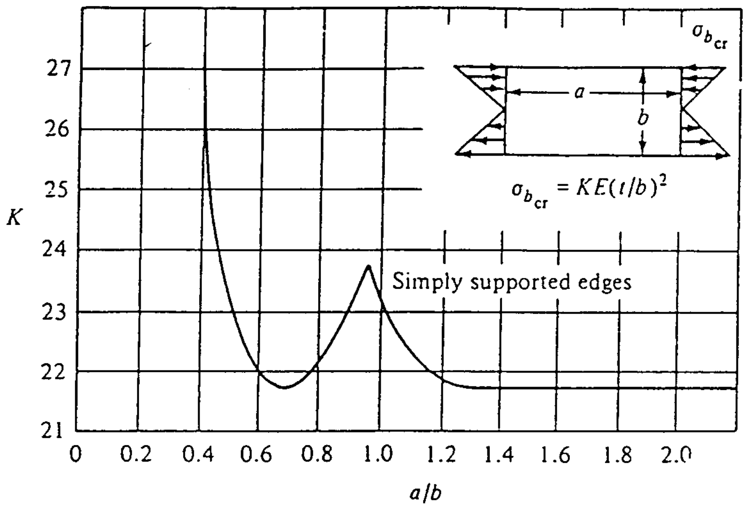
b = plate width

BUCKLING OF FLAT PLATES IN SHEAR



for $\nu = 0.3$

BUCKLING OF FLAT PLATES IN BENDING



for $\nu = 0.3$

BUCKLING OF FLAT PLATES UNDER COMBINED COMPRESSION, SHEAR, BENDING

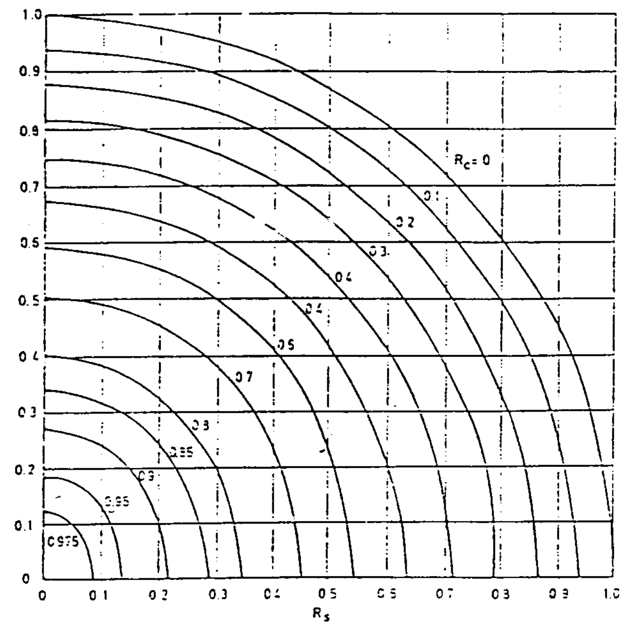


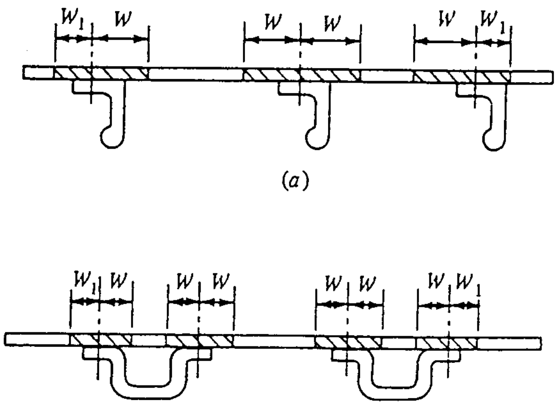
FIG 4 ELASTIC BUCKLING OF FLAT PLATES UNDER COMBINED LOADING

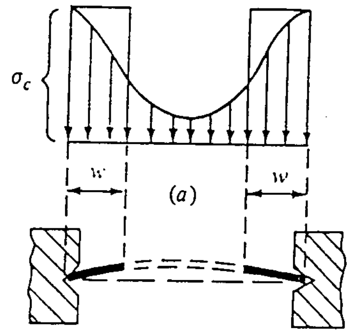
Combined stress buckling criteria: $R_c + R_s^2 + R_b^2 = 1$ at buckling failure

Where: $R_c = \dfrac{\sigma_c}{\sigma_{c_{cr}}}$ $R_s = \dfrac{\tau}{\tau_{cr}}$ $R_b = \dfrac{\sigma_b}{\sigma_{b_{cr}}}$

I.e. the ratios of the stresses in the plate to the critical buckling stresses

EFFECTIVE PLATE WIDTH CONTRIBUTION TO STIFFENER IN COMPRESSION





effective sheet widths w and w_1

$$w = 0.85t\sqrt{\frac{E}{\sigma_c}}$$

$$w_1 = 0.60t\sqrt{\frac{E}{\sigma_c}}$$

For σ_c use representative compressive stress in stiffener

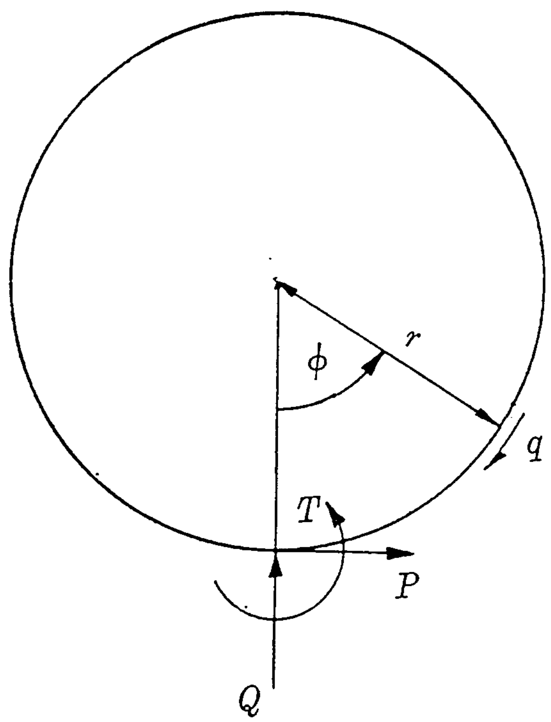
UNIFORM CIRCULAR RING
PERIPHERAL SHEAR FLOW REACTION
TO SINGLE POINT LOADING

Conservative estimate of a peripheral reacting shear flow q from engineers theory of bending:

For:

- A circular ring in which the material is uniformly distributed around the circumference.

- Subject to radial, tangential and moment point loadings, Q, P, T.



$$q = \frac{T + Pr}{2\pi r^2} + \frac{P \cos(\phi)}{\pi r} + \frac{Q \sin(\phi)}{\pi r}$$

UNIFORM CIRCULAR RING

INTERNAL DIRECT, SHEAR & MOMENT REACTIONS

TO SINGLE POINT LOADING WITH PERIPHERAL SHEAR REACTION

Conservative estimate of an internal system of direct, shear and bending moment reactions, N, S, M, from engineers theory of bending:

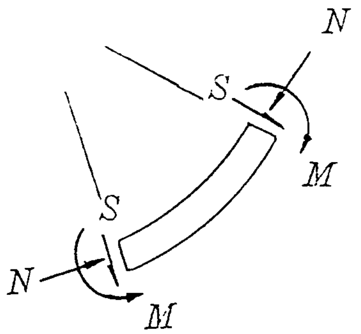
For:

- A circular ring in which the material is uniformly distributed around the circumference.

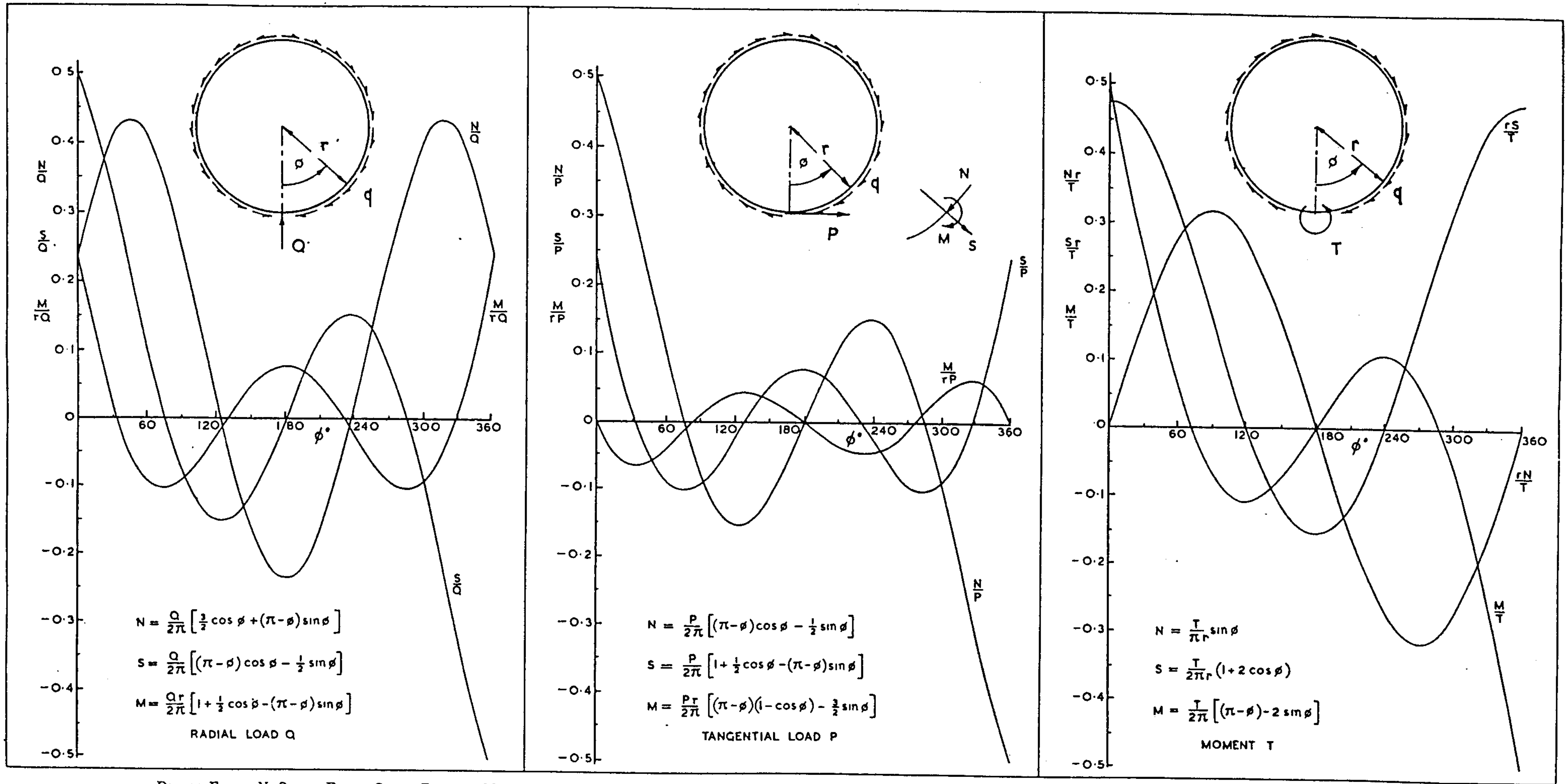
- Subject to radial, tangential and moment point loadings, Q, P, T and peripheral reacting shear flow q.

Using sign convention:

N is positive in compression.*
S is positive when acting outwards on the RHS of a section (viewed from outside the ring).
M is positive for compression in outer material.



See over



DIRECT FORCE N, SHEAR FORCE S AND BENDING MOMENT M IN UNIFORM CIRCULAR FUSELAGE RING DUE TO UNIT RADIAL, TANGENTIAL AND MOMENT LOADINGS, WHEN THE BALANCING SHEAR FLOW q IN THE FUSELAGE IS DISTRIBUTED ACCORDING TO THE ORDINARY ENGINEERS' THEORY OF BENDING

(See Equation (3))