Chapter - 01

Introduction to Design

Marks - 16

General Procedure in Machine Design

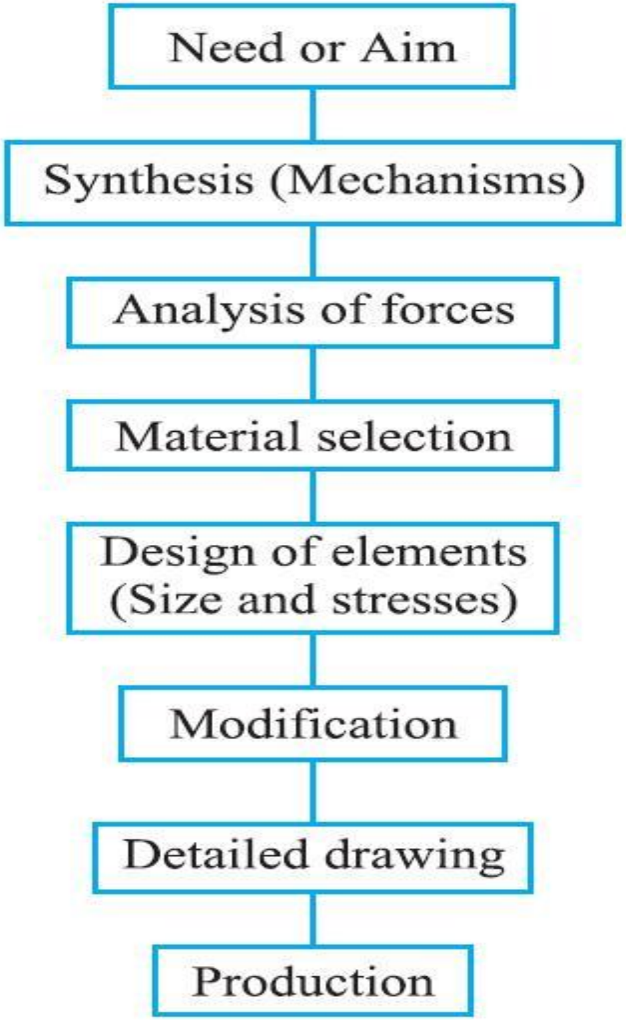


Fig. 1.1. General procedure in Machine Design.

1. **Recognition of need** - First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.
2. **Synthesis (Mechanisms)** - Select the possible mechanism or group of mechanisms which will give the desired motion.
3. **Analysis of forces** - Find the forces acting on each member of the machine and the energy transmitted by each member.
4. **Material selection** - Select the material best suited for each member of the machine.

5. **Design of elements (Size and Stresses)** - Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.
6. **Modification** - Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost.
7. **Detailed drawing** - Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested.
8. **Production** - The component, as per the drawing, is manufactured in the workshop.

General consideration in Machine Design

01)Type of Load and Stresses caused by the Load

The load on the Machine Component may be act in several ways due to which the Internal Stresses are set up.

02)Motion of Parts

The successful operation of any Machine depends largely upon the simplest arrangements of the Parts, which will give the required motion.

The Motion of the Part may be Rectilinear Reciprocating Motion. Curvilinear, Rotary, Oscillatory Simple Harmonic.

03)Selection of Material

Every Machine Design Engineer should have a thorough Knowledge of the Properties of Material and their behavior under working conditions.

04)Form and Size of the Parts

In order to Design any Machine Part for form and size, it is necessary to know the Forces which the Part must sustain. Any suddenly applied or impact load must be taken into consideration which may cause failure.

The smallest Practicable Cross-Section may be used, but it may be checked that the Stresses induced in the Designed Cross-Section are reasonably safe.

05)Frictional Resistance and Lubrication

There is always a Loss of Power due to Frictional Resistance. Careful attention must be given to the matter of Lubrication of all surfaces which moves in contact with others.

06)Safety of Operator

A Machine Designer should always provide safety device for the safety of the operator. The Safety Appliances should in no way interfere with the operation of the Machine.

07)Use of Standard Parts

The use of Standard Parts are closely related to the Cost of Machine, because the Cost of Standard Parts is only a fraction of the cost of similar parts made to order.

08)Convenient and Economical Features

The operating feature of the Machine should be carefully studied. The Starting, Controlling and Stopping Levers should be located on the basis of convenient handling.

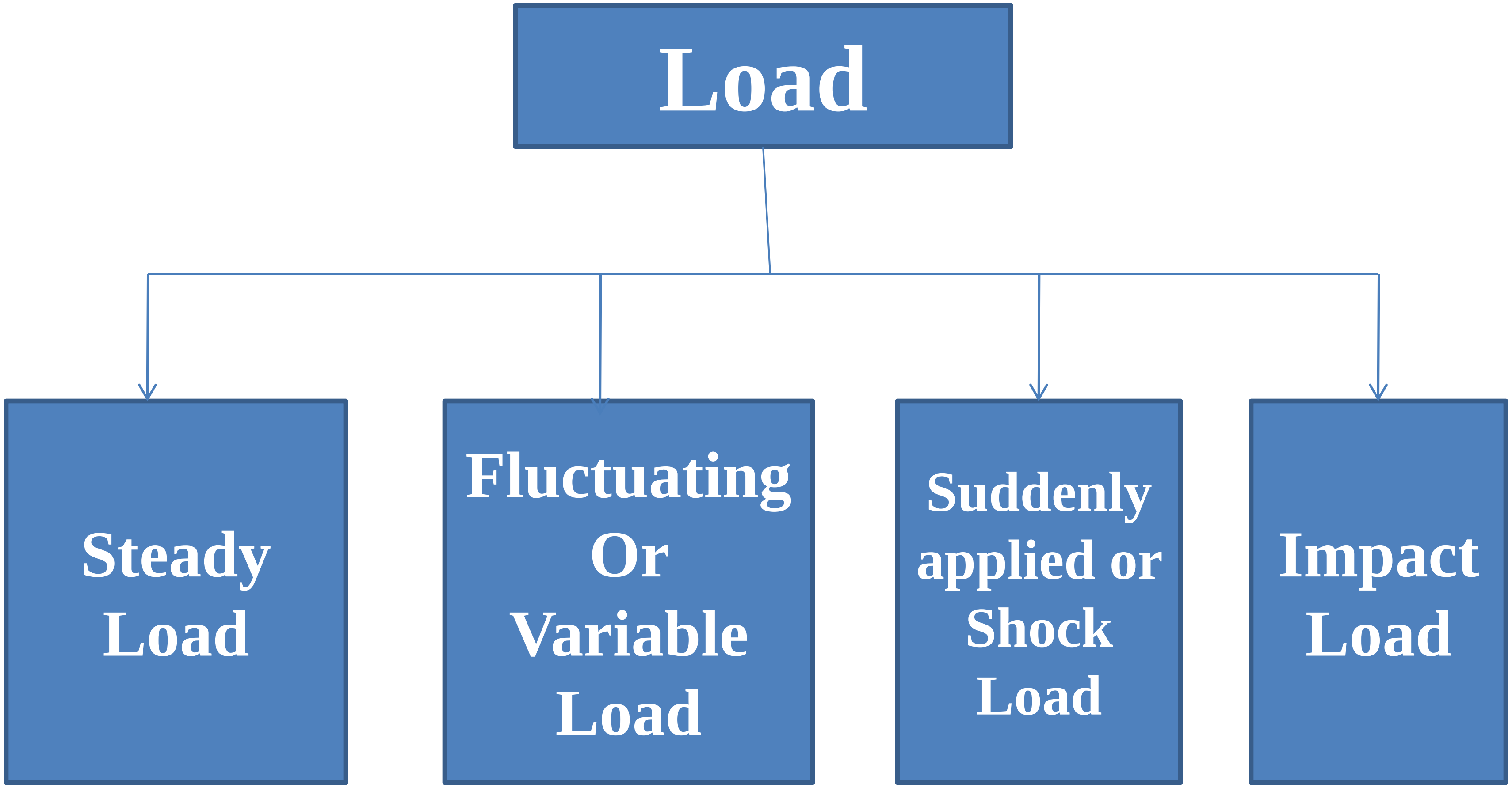
09) Workshop Facilities

A Design Engineer should be familiar with limitation of his Employer's Workshop, in order to avoid the necessity of having work-done in some other Workshop.

10)Assembling

Every Machine must be Assembled as a unit before it can function. The final Location of any Machine is important and the Design Engineer must anticipate the exact location and the local facilities for erection.

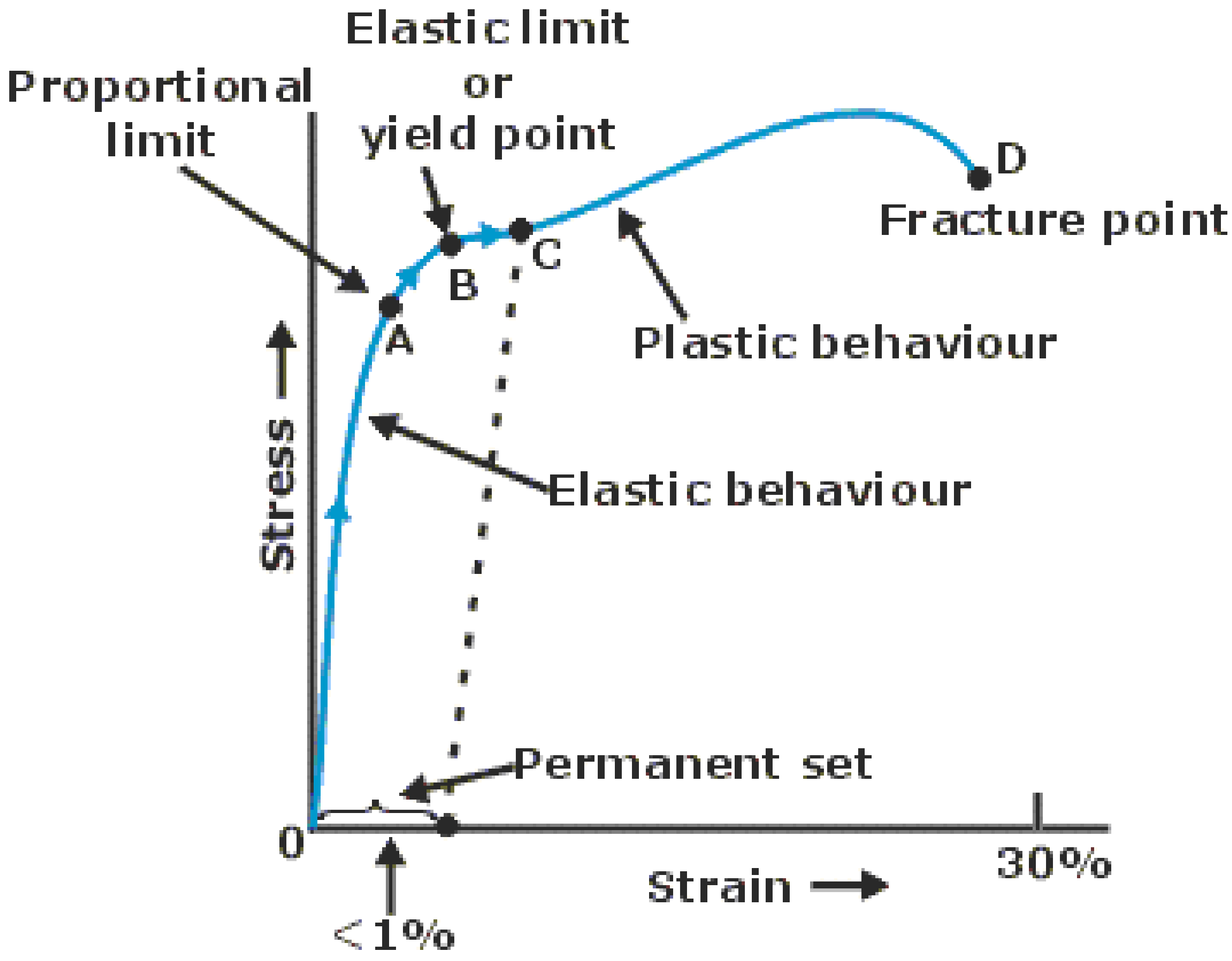
Types of Loads



1) **Steady Load** – It is the type of load which do not changes its intensity and direction of applied load.
2) **Fluctuating or Variable Load** – A load is said to be a variable load when it changes continuously.
3) **Suddenly Applied or Shock Load** – it is the type of load which suddenly applied and rapidly build up stress in the component.
4) **Impact Load** – The instantaneous arrest of a falling of mass or shock meeting two mating parts.

- Stress – When an external force applied on a body or component, some internal forces are set up in the body which resist the external force is known as stress.
- Stress = Force / Area
- Strain – whenever a force is acted on a body or component, it undergoes deformation per unit length called as strain.
- Strain = change in length / original length

Stress – Strain Diagram for Ductile Material



A typical stress-strain curve for a ductile metal

- **Proportional Limit -** From the origin O to the point called proportional limit, the stress-strain curve is a straight line. This linear relation between elongation and the axial force causing was first noticed by **Sir Robert Hooke** in 1678 and is called Hooke's Law that within the proportional limit, the stress is directly proportional to strain
- **Elastic Limit -** The elastic limit is the limit beyond which the material will no longer go back to its original shape when the load is removed, or it is the maximum stress that may e developed such that there is no permanent or residual deformation when the load is entirely removed.

- **Elastic and Plastic Ranges -** The region in stress-strain diagram from O to P is called the elastic range. The region from P to R is called the plastic range.
- **Yield Point -** Yield point is the point at which the material will have an appreciable elongation or yielding without any increase in load.
- **Ultimate Strength -** The maximum ordinate in the stress-strain diagram is the ultimate strength or tensile strength.

Stress – Strain Diagram for Brittle Material

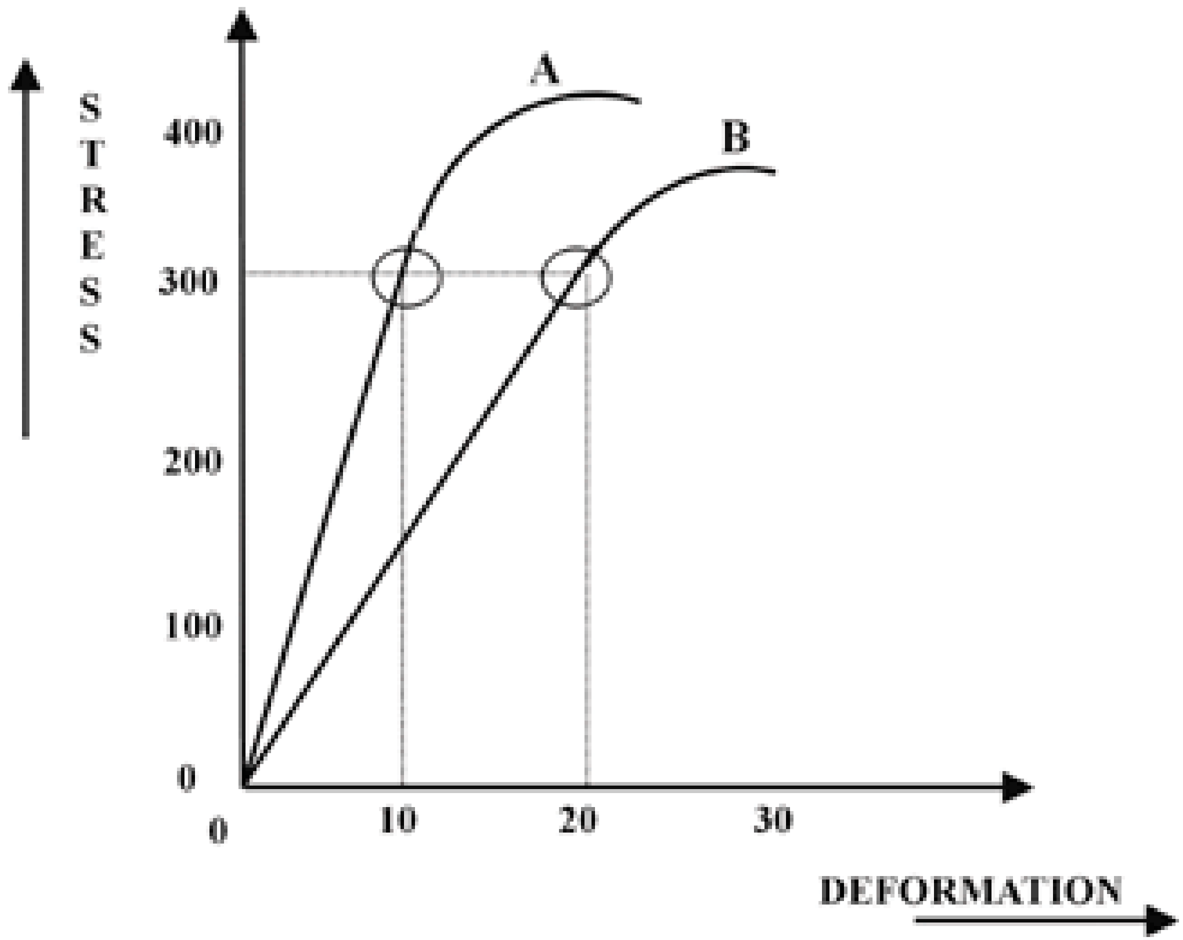
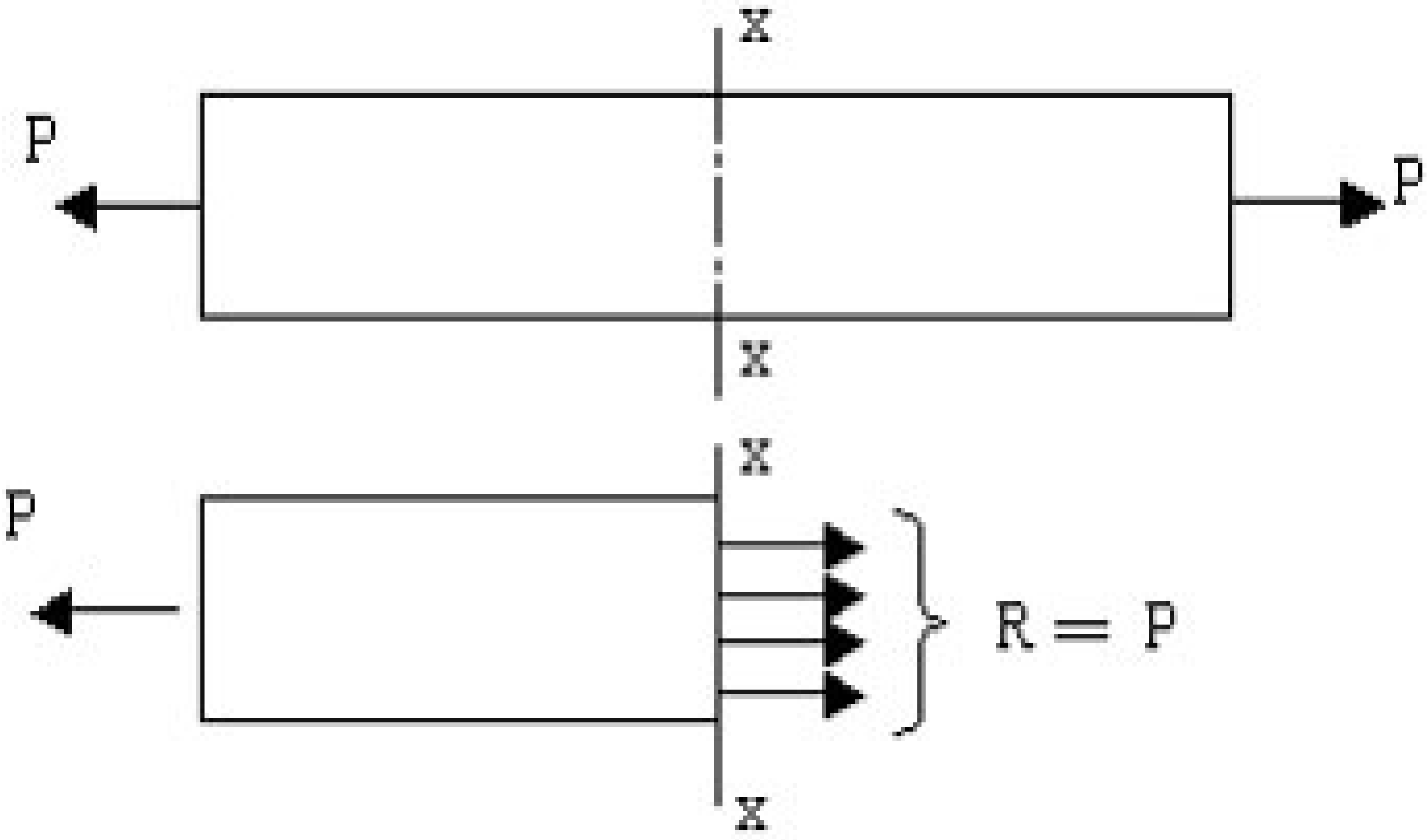


FIGURE 2- Stress-strain curve for hypothetical materials (A e B) showing that A (30GPa) presents higher elastic modulus than B (15 GPa)

Types of Stresses

- **Tensile Stress -** Consider a uniform bar of cross sectional area A subjected to an axial tensile force P. The stress at any section x-x normal to the line of action of the tensile force P is specifically called tensile stress pt . Since internal resistance R at x-x is equal to the applied force P, we have,
  6_t = (internal resistance at x-x)/(resisting area at x-x)
  $6_t = R/A$
  $6_t = P/A.$


x
P
P
x
x
P
R = P
x

- **Compressive Stress -** If the bar is subjected to axial compression instead of axial tension, the stress developed at x-x is specifically called compressive stress 6_c.
  6_c =R/A

  6_c = P/A.

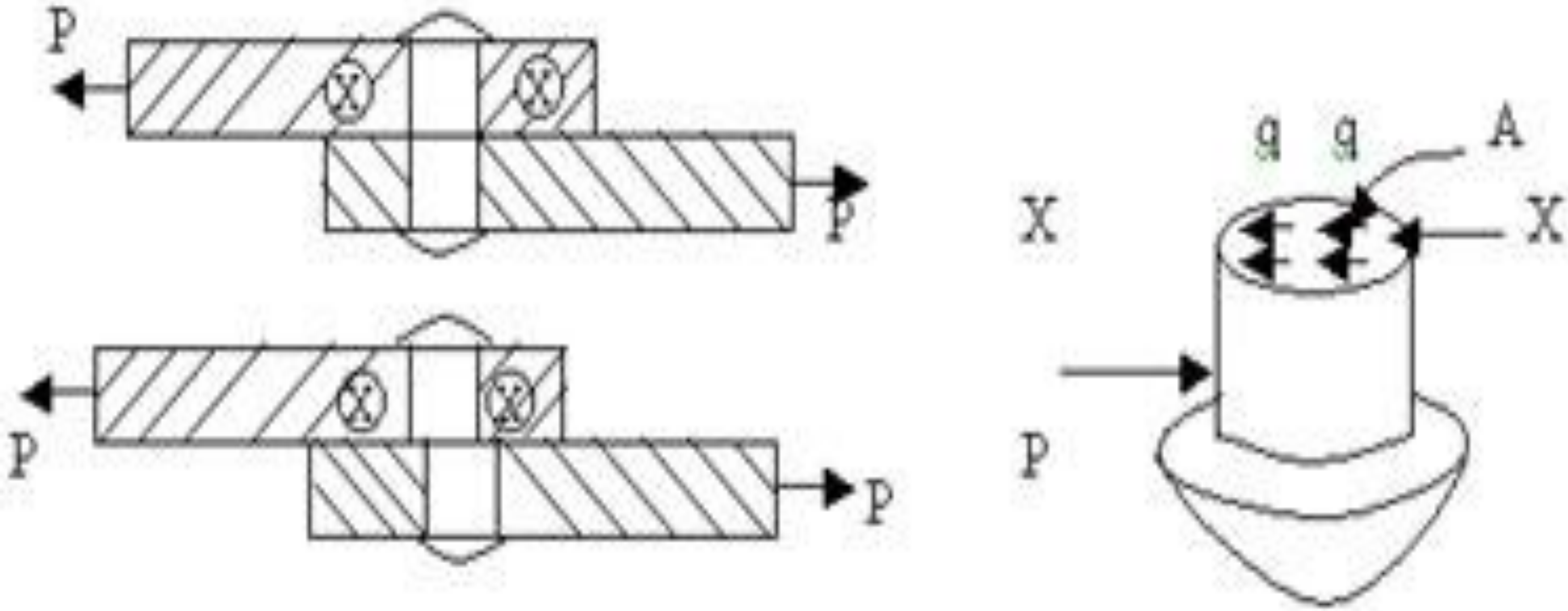
- **Bending Stress** – When a body or mechanical component is subjected to bending moment (M), the stress induced in it is known as bending stress. (6_b)
- $(M/I) = (6_b/y) = (E/R)$
- $6_b = (M \times y) / I = M / (I / y) = M / Z$
- Where $Z = I/y$ = Section Modulus
- M = Bending moment acting on section X-X.
- 6_b = Bending stress at a distance y from neutral axis.
- I = Moment of inertia of C/S about the neutral axis.

- **Shear Stress**

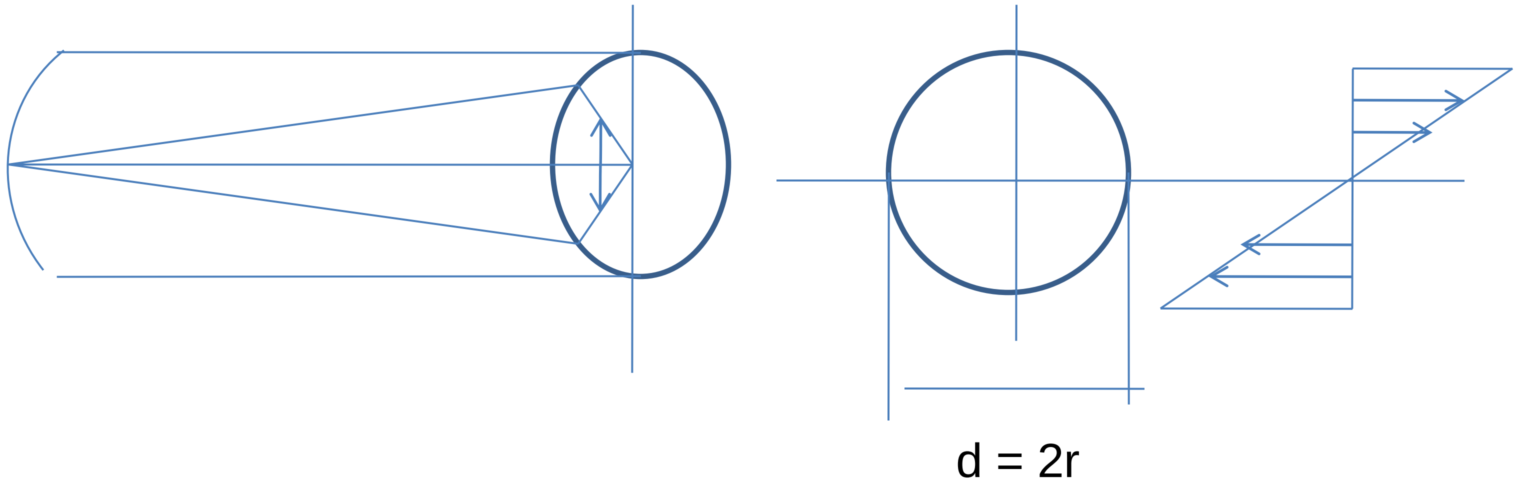
a) Direct or Transverse shear stress – When a mechanical component is subjected to two equal and opposite forces acting tangentially across the resisting area results to shear off the section. The stress induced in such section is known as direct or transverse shear stress.

i) Single Shear – $q = P/A$

ii) Double Shear – $q = P/2A$


P
P
P
P
X
X
q
q
A
X
X
P

b) Tensional Shear stress – When a machine member is subjected to the action of two equal and opposite couple acting in parallel plane (torque or twisting) then the machine component is subjected to torsion and the stress developed in element is torsional shear stress.



- The torsional shear stress induced at any fibre is given by torsional equation of rigidity.
- $(\tau/r) = (T/I_p) = (G\theta/l)$ ---------------(1)
- Where r = Radius of shaft in mm = (d/2)

T = Torque or twisting moment in N-mm = F x l

I_p = Polar moment of inertia in mm^4

d = diameter of shaft in mm

G = modulus of rigidity in N/mm^2

Θ = angle of twist in radians

l = length of shaft in mm.

- From above equation no.1

$$\boxed{\tau = (T \times r) / I_p}$$

- For Solid Circular section polar moment of inertia is
- Ip = (

- **Bearing pressure or Bearing stress** –

A localized compressive stress at the area of contact between two components having relative motion between them is known as bearing pressure or bearing stress.

- **Crushing Stress** - A localized compressive stress at the area of contact between two components having no relative motion between them is known as bearing pressure or bearing stress.

- Bearing pressure and the crushing stress is calculated as
- Bearing Pressure = P_b = (Force / Projected area of contact)
- $P_b = P/A = P/dl$
- P = Force acting
- A = Projected area of contact in mm^2
- D = diameter of pin in mm
- l= length of pin in eye in mm

Factor of safety

- While designing any mechanical system or its components, always there are certain areas of uncertainties such as variation in dimensions, variation and non uniformity in the material strength etc. Hence in order to prevent the failure of the component, designer assuming a value of design stress which is very less as compared to the yield stress or ultimate stress. So factor of safety is designed as maximum stress to working stress or design stress.

- The **factor of safety** is defined as the ratio of yield stress or ultimate stress to the design stress.(Nf)
- F.S. = Yield or Ultimate stress / Working Stress
- F.S. = 6_{yt} or 6_{ut} / 6_{design}
- Factor of safety for shear stress condition
- F.S. = 6_{ys} / 6_{design} and
- $6_{ys} = 6_{yt} / 2$

Selection of Factor of Safety

- The reliability of the properties of the material and change of these properties during service.
- The reliability of applied load.
- The extent of localized stresses.
- The certainty as to exact mode of failure.
- The extent of simplifying assumptions.
- The reliability of test result and accuracy of application of these results to actual machine parts.
- The extent of initial stress set up during manufacturing.
- The extent of loss of life if failure occurs.

Stress Concentration

- In design of machine elements, there are several stress acts on the elements such as bending, twisting, shear stress, tensile or compressive stress.
- These stress equations are developed with assumption that there is no discontinuities in the cross section of the machine element and no abrupt change in c/s of component.
- In actual practice discontinuities in the any machine part are there due to key way, threaded holes, grooves and steps to perform their functions.

- Such discontinuities alter the stress distribution in the vicinity of the discontinuity and elementary stress equation no longer describe the state of stress in the component.
- The stress induced in the neighborhood of the discontinuity are much higher than the stresses in the other part of the component.
- This concentration of high stresses due to discontinuities or abrupt changes of cross section is called as stress concentration.

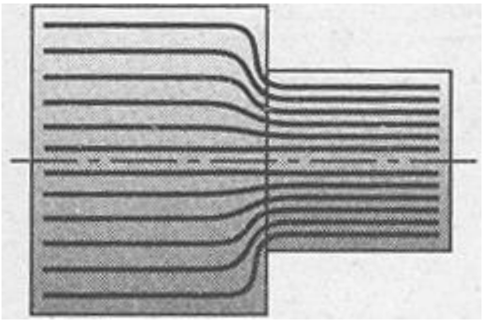
Causes of stress concentration

1) Abrupt change of c/s – the abrupt change in the c/s like key way, steps, grooves, threaded holes etc results in stress concentration.
2) Poor surface finish – The surface irregularities also act as the source of stress concentration.
3) Localized loading – When the load is transmitted through a small area contact the stress concentration takes place in the vicinity of loaded area.
4) Variation in material properties – The material variations, like internal cracks and flaws, air holes, non-metallic inclusions and cavities in welds act as the source of stress concentration.

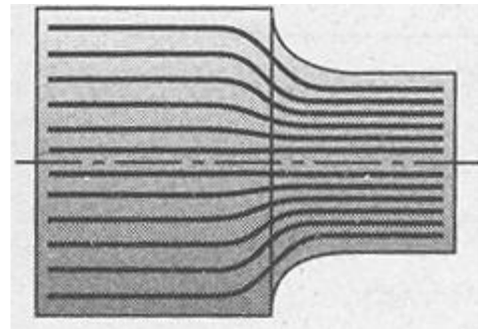
Methods of reducing stress concentration

A number of methods are available to reduce stress concentration in machine parts. Some of them are as follows:

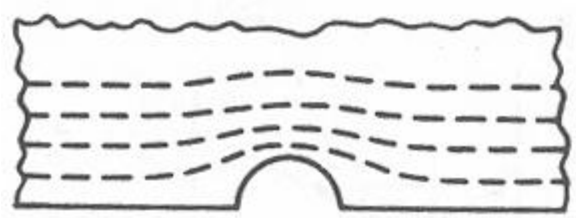
1. Provide a fillet radius so that the cross-section may change gradually.
2. Sometimes an elliptical fillet is also used.
3. If a notch is unavoidable it is better to provide a number of small notches rather than a long one. This reduces the stress concentration to a large extent.
4. If a projection is unavoidable from design considerations it is preferable to provide a narrow notch than a wide notch.
5. Stress relieving groove are sometimes provided.



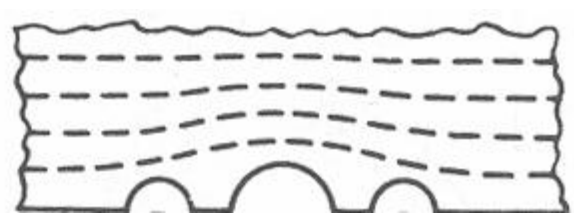
Force flow around a sharp corner



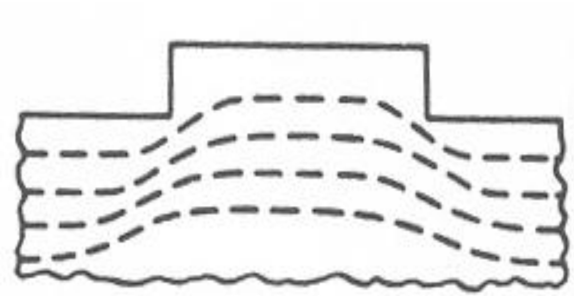
Force flow around a corner with fillet: Low stress concentration.



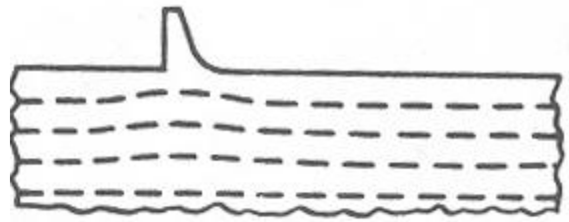
Force flow around a large notch



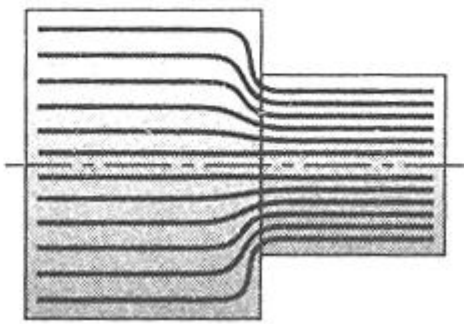
Force flow around a number of small notches: Low stress concentration.



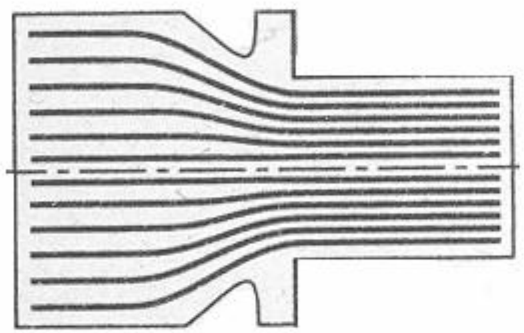
Force flow around a wide projection



Force flow around a narrow projection: Low stress concentration.



Force flow around a sudden change in diameter in a shaft



Force flow around a stress relieving groove

Theoretical Stress Concentration Factor

- The stress concentration factor is used to rotate the maximum stress at discontinuity to the nominal stress.
- The theoretical or geometric stress concentration factor (K_t) is defined as the ratio of the maximum stress in a component at discontinuity to the nominal stress at the same section.

$$K_t = \frac{Max.stress}{Nominal\ stress} = \frac{\sigma_{\max}}{\sigma_o} \ or \ \frac{\tau_{\max}}{\tau_o}$$

- The theoretical stress concentration factor depends only upon the geometry of the component and the material has no effect on its value.

Fatigue Stress concentration Factor

- In actual loading especially fatigue loading the effect of stress concentration is usually less than that predicted by the theoretical stress concentration factor.
- Hence in the design, the stress concentration factor is incorporated by the factor known as fatigue (actual) stress concentration factor (K_f) instead of theoretical stress concentration factor (K_t).
- The value of fatigue stress concentration factor is lower than the theoretical stress concentration factor.

- **The fatigue stress concentration factor is defined as the ratio of the actual maximum stress in a component at discontinuity to the nominal stress at the same section.**

$$K_f = \frac{Actual\ \max.stress\ in\ notched\ specimen}{No\min al\ stress\ in\ notch - free\ specimen}$$

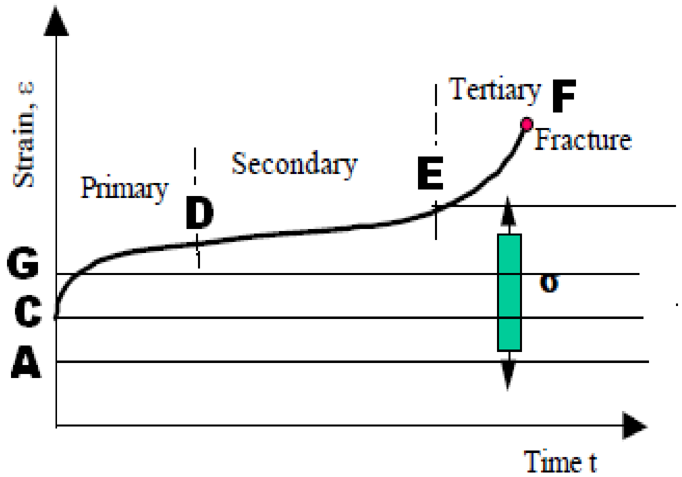
Notch Sensitivity

- The term notch sensitivity is used to relate the fatigue stress concentration factor (K_f)with theoretical stress concentration factor (K_t).
- It indicates the sensitivity of the material to notches or discontinuities.
- **The notch sensitivity is defined as the degree to which the theoretical expected effect of stress concentration is actually reached.**

Creep Strain & Creep Curve

- **Creep** can be defined as the slow and progressive deformation of a material with time under a constant stress.
- The phenomenon of creep is observed in metal, non metals as well as amorphous materials like glasses and high polymers.

Creep Curve and Its Stages



In first stage, creep continues at a decreasing rate, that becomes approximately constant during the second stage, the beginning of the third stage is marked by a rapid increase in creep rate which continues until fracture occurs.

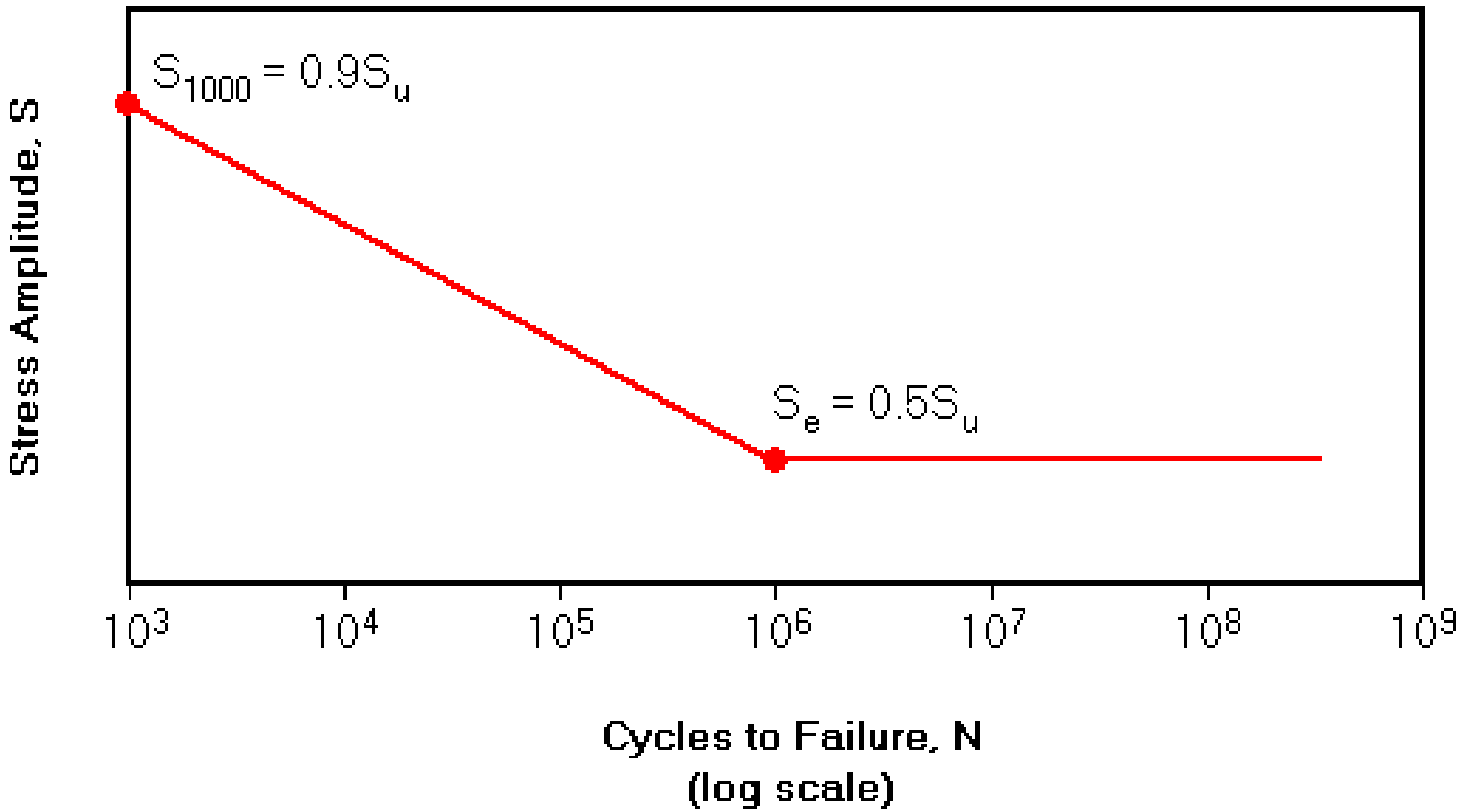
Fatigue Or Endurance

- When a machine component is subjected to repeated stress, the component may fail at stress below yield point stress. Such type of stress is known as fatigue.
- The failure is caused by means of progressive crack formation and the phenomenon of failure or fracture is known as fatigue failure.

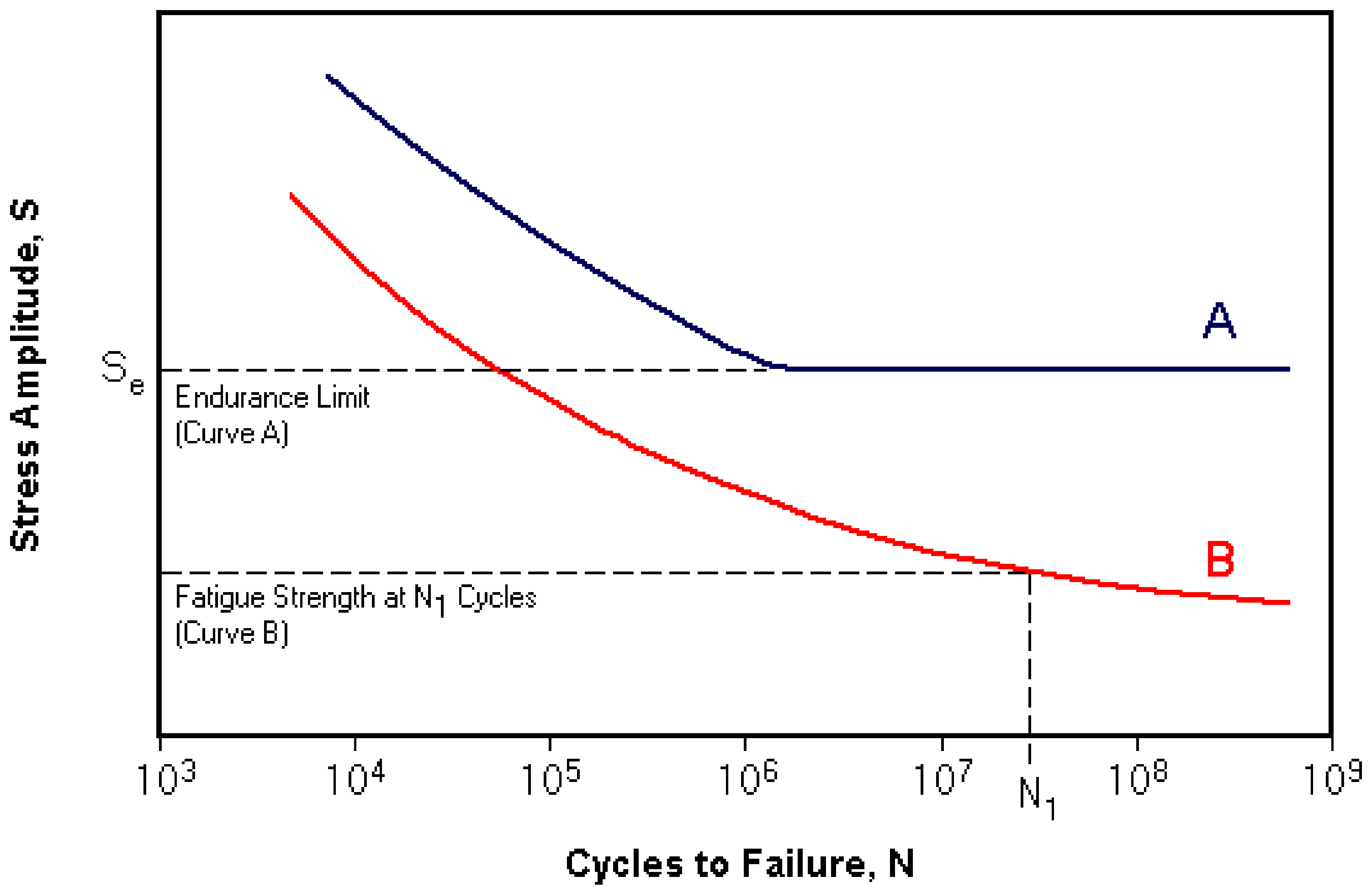
S-N Diagram and Endurance Limit

- The fatigue strength or endurance strength of a material is defined as the value of a completely reversed stress that the standard test specimen can withstand without failure for the given number of cycles.
- It is denoted by S_f.
- The endurance limit or fatigue limit of a material is defined as the max. value of a completely reversed stress that the standard test specimen can withstand without failure for an infinite number of cycles.
- It is denoted by S_e.

- It has been found that the magnitude of the stress at which fatigue occurs decreases as the number of stress cycles increases.
- The plot of fatigue strength(S_f) versus stress cycles (N) on log-log paper is known as S-N diagram.
- For ferrous materials like steel, the graph becomes horizontal at 10^6 cycles, indicating that the fatigue failure will not occur below this stress, whatever may be the number of cycles.



Stress Amplitude, S
S1000 = 0.9Su
Se = 0.5Su
10^3
10^4
10^5
10^6
10^7
10^8
10^9
Cycles to Failure, N
(log scale)


Stress Amplitude, S
S_e
Endurance Limit
(Curve A)
A
B
Fatigue Strength at N_1 Cycles
(Curve B)
10^3
10^4
10^5
10^6
10^7
N_1
10^8
10^9
Cycles to Failure, N

Theories of elastic failures

- When a component is subject to increasing loads it eventually fails.
- It is comparatively easy to determine the point of failure of a component subject to a single tensile force.
- The strength data on the material identifies this strength.
- However when the material is subject to a number of loads in different directions some of which are tensile and some of which are shear, then the determination of the point of failure is more complicated.

- Metals can be broadly separated into DUCTILE metals and BRITTLE metals. Examples of ductile metals include mild steel, copper etc.
- Cast iron is a typical brittle metal.
- Ductile metals under high stress levels initially deform plastically at a definite yield point or progressively yield.
- Brittle metals experience little ultimate elongation prior to failure and failure is generally sudden.

- A ductile metal is considered to have failed when it has suffered elastic failure, that is when a marked plastic deformation has begun.
- A number of theories of elastic failure are recognized including the following:

1) Maximum principal stress theory(Rankine theory)

2) Maximum shear stress theory (Guest’s or Tresca’s theory)

3) Maximum principal strain theory (Saint Venant's theory)

4) Maximum strain energy theory (Heigh's theory)

5) Maximum distortion energy theory (Hencky and Von mises theory)

Maximum principal stress theory

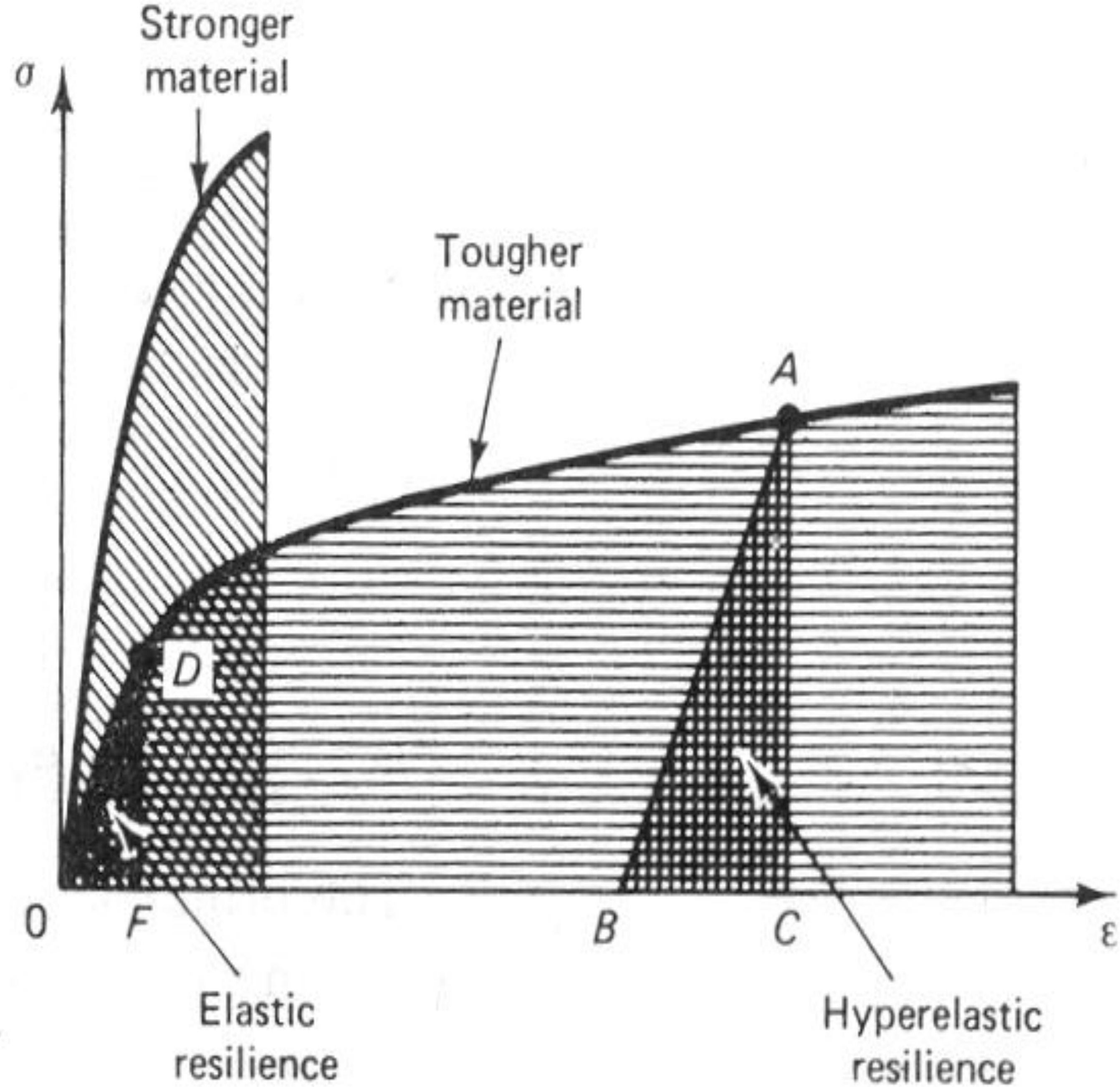
- The theory associated with Rankine.
- This theory is approximately correct for cast iron and brittle materials generally.
- According to this theory failure will occur when the maximum principal stress in a system reaches the value of the maximum strength at elastic limit in simple tension.
- $6_t = (6_{yt} / F.S.)$ – For ductile material
- $6_t = (6_{ut} / F.S.)$ – For brittle material

Maximum shear stress theory

- The theory associated with Tresca and Guest.
- This is very relevant to ductile metals.
- It is conservative and relatively easy to apply.
- It assumes that failure occurs when a maximum shear stress attains a certain value.
- This value being the value of shear strength at failure in the tensile test.
- $q_{max} = (q_{yt} / F.S.)$ or
- $q_{max} = (6_{yt} / 2\ F.S.)$

Mechanical Properties of Material

- **Resilience:** Amount of energy stored in material *up to elastic limit* per unit volume
- **Toughness:** Amount of energy stored in material *up to fracture per* unit volume.



Stronger material
σ
Tougher material
A
D
0
F
B
C
ε
Elastic resilience
Hyperelastic resilience

- **Hardness -** Resistance to permanent indentation.
- Good hardness generally mean that the material is resistant to scratching and wear. It is also an indication of strength.

- **Ductility:** Extent to which material can sustain plastic deformation before rupture. **Gold is most ductile** material.

- **Brittleness:** undergoes a very little plastics deformation before rupture.
- **Malleability:** Ability to be flattened into thin sheets without cracking.
- **Stiffness:** Ability of material to resist deformation.

Material for Machine Parts

Sr. No.	Machine Part	Material
1.	Crank Shaft	Alloy Steel (35 Mn 2 Mo 28)
2.	Helical Spring	Chromium Vanadium Alloy Steel
3.	Bushes for Knuckle Pin	Gray C.I. (FG200)
4.	Lathe Bed	Gray C.I. (FG 150)
5.	Hydraulic Cylinder	Plain Carbon Steel (FG260)
6.	Turbine Blade	Chromium Steel Containing 12 to 14% chromium 7 Cr 13
7.	Bearing Bush	Bronze (Alloy Steel)

Sr. No.	Machine Part	Material
8	Clutch Spring	Plain Carbon steel with Manganese (60 C 4)
9	Closed Coil Helical Spring	Low Alloy Steel (55 Si 2 Mn 90)
10	Locomotive Carriage And Wagon wheels	Plain Carbon Steel (Fe310 or FeE230)
11	Connecting Rod	35 Mn 2 Mo 28 Or 40 Cr 1
12	Machine Tool Spindle	20 Mn 2
13	Heavy Duty Gear	40 Ni 1 Cr 1 Mo 15

Material Specification & Application

Sr. No.	Material	Specification	Application
1	FG300	Grey C. I.	
2	FeE230	Plain Carbon steel	Locomotive carriage, car structure, screw stock
3	SG800/2	Nodular C.I.	Heavy duty gears, automobile door hinges, furnace doors, steam plants components
4	SG450/10	Nodular C.I.	
5	35C8	Plain Carbon steel	Cycle and motor cycle tubes fish plates for rail and fasteners, low stressed parts in machine structure.
6	60C4	Plain Carbon steel	Spike bolts, gear shaft, cylinder liners, clutch spring, hardened screws and nuts, machine tool spindle, coupling crankshaft, axles, pinions.

Sr. No.	Material	Specification	Application
7	X20Cr18 Ni	High Alloy Steel	Sheet or strips for cold forming and press operations.
8	35Mn2Mo28	Low Alloy Steel	Crank shaft, connecting rods, levers, etc.
9	20Mn2	Low Alloy Steel	Transmission Shaft, spindles, crank shaft, levers, welded structure.

Standardization

- Modern system are increasingly become more and more complex.
- A large number of mechanical components are found interconnected in a complex system such a complex system can have many sources of errors.
- The main purpose of standardization is to establish mandatory norms for the design and production of machine components so as to reduce variation in their types and grades and to achieve quality characteristics in raw materials.

- The code is defined as “a set of specification or procedure for the design analysis, testing and manufacturing of a system component or product to achieve uniformity, quality, interchangeability and safety and to put reasonable limit on the variety is established by standardization.

Standards used in Mechanical Engineering

1) Standards for size and shape of component like nuts and bolts, bearing, keys, gear, chain, belt, etc.
2) Standards for product like electric motors, engines, gear boxes, pressure vessels, etc.
3) Standards for fits, tolerances and surface finish grades of components.
4) Standards for conventional representation of component on the drawing.

Advantages of Standardization

- Interchangeability of the machine component is possible.
- Better product quality, reliability and longer service life.
- Mass production of components at low overall cost.
- Some time it ensures the safety.
- Easy & quick replacement of the component is possible.
- Less time and effort required for manufacturing.
- Reduction in various size and grades of an article.

Use of Design Data Book

- During the process of designing the machine element, system or product, the design engineer needs variety of information such as ---

1) Available material and their properties.
2) Design procedures as per various national and international standards and codes.
3) Standard sizes and shapes of components like screws, bolts, nuts, circlips, etc.
4) Standard sizes and load ratings of standard components like rolling contact bearing, chains, belts, ropes, etc.

5) Types of fits and tolerances and values of various types of stresses

6) Surface finish, etc.

- It is really difficult task for design engineers and design office to get latest information of data required during the design process.
- The advantage of using design data book lies in the fact that consolidated information is available at one place so the design engineer is not to run frequently for necessary information.

- The various sources of design data are as follows.

1) **Textbook and references books –**

   J.E.Shigley and C.R. Mischke, “Mechanical Engineering Design”, McGraw-Hill Book Company, 1989.

2) **Handbooks** –

   Dudley D.W. “Handbook of Practical Gear Design”, McGraw-Hill International Book Company, 1984.

3) **National and international Standards and codes –**

   IS 2825-1960 code for unfired pressure vessels.

4) Manufacture’s Catalogue –

SKF catalogue of ball and roller bearings.

5) Charts –

Charts of theoretical stress concentration factors $K_{t.}$

6) Technical journals –

ASME journals.

Types of Standards

1) **Company standards –**

   Company standards are defined or set by a company or a group of companies for their particular use.

2) **National Standards –**

   These standards are defined or set by a national apex body and normally followed throughout the country.

   The examples are standards prepared by

1) Bureau of Indian Standards (BIS).
2) American society of Mechanical Engineers (ASME).
3) American Gear Manufactures Association (AGMA).
4) American Welding Society (AWS).
5) American National Standards Institute (ANSI).

3) International Standards –

These standards are defined or set by an international apex body and are normally followed all over the world.

The examples are

1) International Standards Organization (ISO).
2) International Bureau of Weight and Measures. (IBWM).

Preferred Number Series

- In standardization, the concept of preferred number help to reduce unnecessary variations in sizes and grades of the product or articles.
- French balloonist and engineer Charles Renard first introduced preferred number in the 19^{th} century.
- The system is based on the use of geometric progression to develop a set of numbers.
- These are basic five series are given IS:1076-1985, denoted as R_5, R_{10}, R_{20}, R_{40} and R_{80} series which increases in step of 58%, 26%, 12%, 6% and 3%.
- Each series has its own series factor.

The series factor are given in table.

$$R_5\ series \qquad \sqrt[5]{10} = 1.58$$

$$R_{10}\ series \qquad \sqrt[10]{10} = 1.26$$

$$R_{20}\ series \qquad \sqrt[20]{10} = 1.12$$

$$R_{40}\ series \qquad \sqrt[40]{10} = 1.06$$

$$R_{80}\ series \qquad \sqrt[80]{10} = 1.03$$

- The series is established by taking the first number and multiplying it by a series factor to get second number and so continue to find next number.
- These are two terms namely ‘basic series’ and derived series.
- Which are frequently used in relation to preferred numbers.
- R_5, R_{10}, R_{20}, R_{40} & R_{80} are called basic series and any series formed from the five basic series called derived series.

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