Theories of failure

Introduction

Theories of failure are those theories which help us to determine the safe dimensions of a machine component when it is subjected to combined stresses due to various loads acting on it during its functionality.

Some examples of such components are as follows:

1 . I.C. engine crankshaft

2. Shaft used in power transmission

3. Spindle of a screw jaw

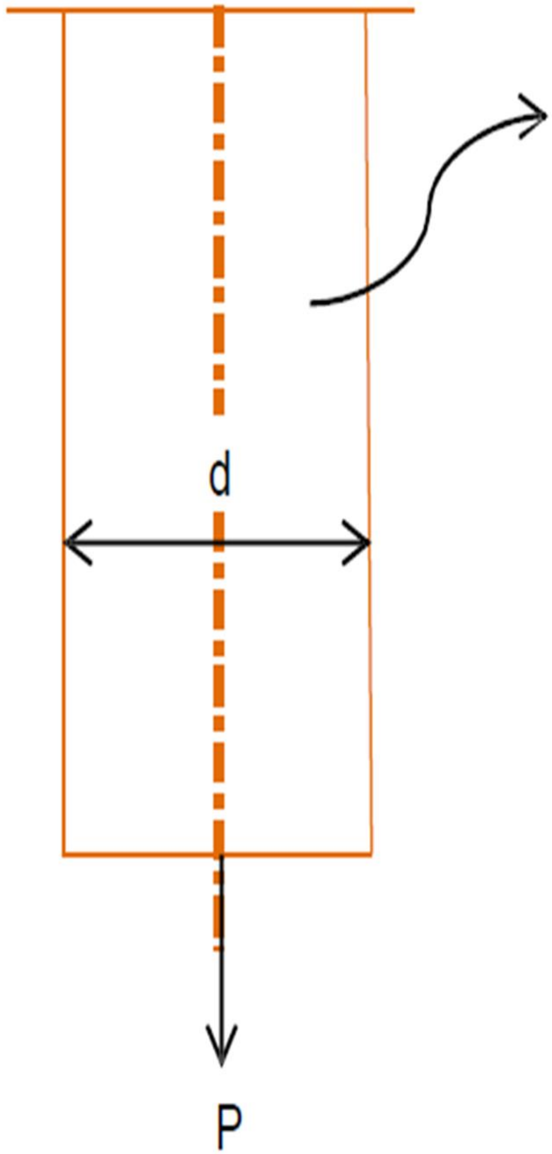
4. Bolted and welded joints used under eccentric loading

5. Ceiling fan rod

- Theories of failure are employed in the design of a machine component due to the unavailability of failure stresses under combined loading conditions. Theories of failure play a key role in establishing the relationship between stresses induced under combined loading conditions and properties obtained from tension test like **ultimate tensile strength (Sut) and yield strength (Syt).**

Examples:

1.



$S_{yt} = 200$ MPa

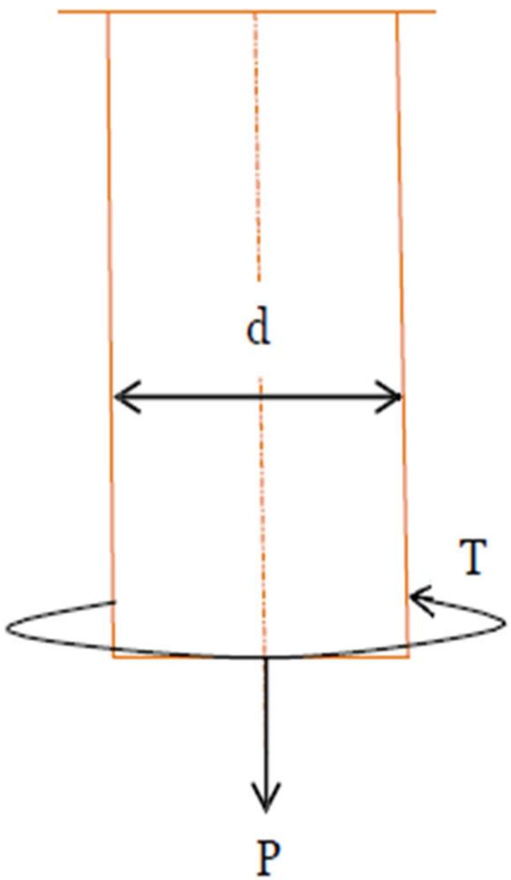
$S_{ut} = 300$ MPa

Directly we can get (d) without using any failure theory because only uniaxial load (P)

$\sigma_1 \leq S_{yt}$

$\frac{4P}{\pi d^2} \leq S_{yt}$

2.



Member is subjected to both Twisting moment and uniaxial load, hence combined loading conditions.

We cannot determine (d) directly in this case because failure stresses under combined loading conditions are unknown.

So, different scientists give relationships between

Stresses induced under combined loading conditions and (S_{yt} and S_{ut}) obtained using tension test which are called **theories of failure.**

Various Theories of Failure

1. Maximum Principal Stress theory also known as RANKINE'S THEORY
2. Maximum Shear Stress theory or GUEST AND TRESCA'S THEORY
3. Maximum Principal Strain theory also known as St. VENANT'S THEORY
4. Total Strain Energy theory or HAIGH'S THEORY
5. Maximum Distortion Energy theory or VONMISES AND HENCKY'S THEORY

Maximum Principal Stress theory (M.P.S.T)

The failure of mechanical component subjected to bi-axial or tri-axial stress occurs when the max. Principal stress reaches the yield or ultimate strength of material .

Condition for safe design,

Factor of safety (F.O.S) > 1

Maximum principal stress ($_1$) ≤ Permissible stress ($_{per}$)

where permissible stress $= \dfrac{\text{Failure stress}}{\text{Factor of safety}} = \dfrac{S_{yt}}{N}$ or $\dfrac{S_{ut}}{N}$

$$_1 \leq \frac{S_{yt}}{N} \text{ or } \frac{S_{ut}}{N} \qquad \text{——————} \qquad \textbf{Eqn (1)}$$

1. This theory is suitable for the safe design of machine components made of brittle materials under all loading conditions (tri-axial, biaxial etc.) because brittle materials are weak in tension.
2. This theory is not suitable for the safe design of machine components made of ductile materials because ductile materials are weak in shear.
3. This theory can be suitable for the safe design of machine components made of ductile materials under following state of stress conditions.

Factor of safety (f.o.s**):**

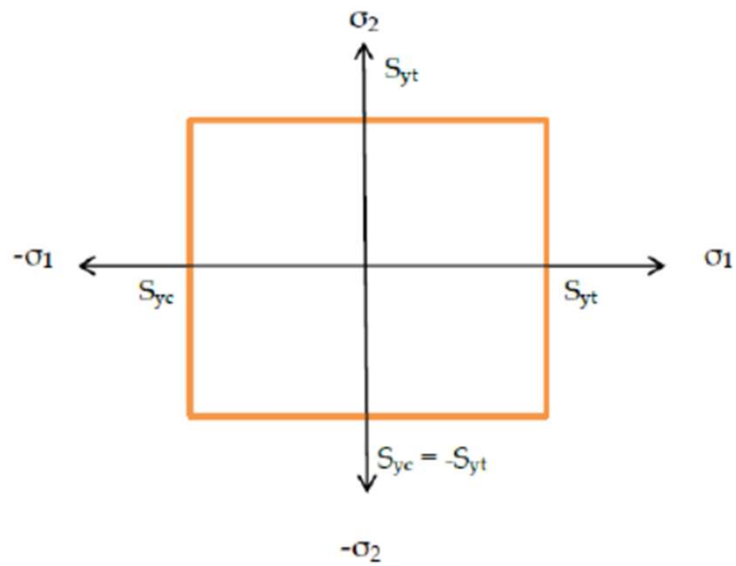
The ratio of ultimate to allowable load or stress is known as factor of safety i.e.

The factor of safety can be defined as the ratio of the material strength or failure stress to the allowable or working stress.

The factor of safety must be always greater than unity.

f.o.s = failure stress / working or allowable stress

M.P.S.T :- Square



In a two dimensional loading situation for a ductile material where tensile and compressive yield stress are nearly of same magnitude: