Production & Industrial Engineering

General Engineering

Vol. III : Machine Design

Comprehensive Theory

with Solved Examples and Practice Questions

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Contents

Machine Design

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VOLUME III

General Engineering

Machine Design

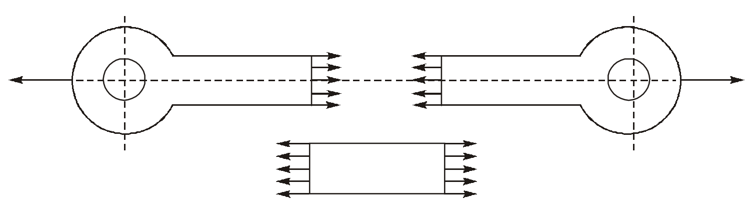
INTRODUCTION

Machine Design or mechanical design is primarily concerned with the systems by which the energy is converted into useful mechanical forms and of mechanisms required to convert the output of the machine to the desired form. The design may lead to an entirely new machine or an improvement on an existing one.

Thus machine design is the production or creation of the right combination of correctly proportioned moving and stationary components so constructed and joined as to enable the liberation, transformation, and utilization of energy.

3.1 Loading of Machine Elements

Machine parts fail when the stresses induced by external forces exceed their strength. The external loads cause internal stresses in the elements and the component size depends on the stresses developed. Stresses developed in a link subjected to uniaxial loading is shown in figure. Loading may be due to: (a) The energy transmitted by a machine element; (b) Dead weight; (c) Inertial forces; (d) Thermal loading; (e) Frictional forces.

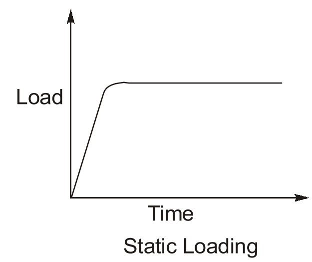


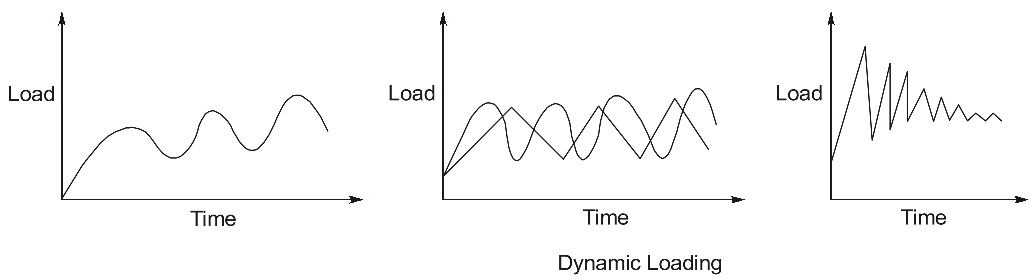
Stresses developed in a link subjected to uniaxial loading

In another way, load may be classified as:

(a) Static load : Load does not change in magnitude and direction and normally increases gradually to a steady value.

(b) Dynamic load : Load may change in magnitude for example, traffic of varying weight passing a bridge. Load may change in direction, for example, load on piston rod of a double acting cylinder. Vibration and shock are types of dynamic loading. Figure shows load vs time characteristics for both static and dynamic loading of machine elements.



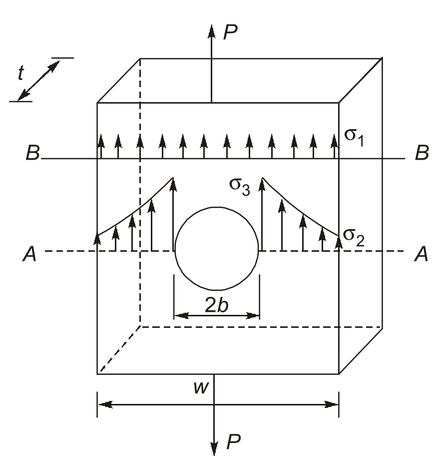


Types of loading on machine elements

3.2 Concept of Stress Concentration

In developing a machine it is impossible to avoid changes in cross-section, holes, notches, shoulders etc. Any such discontinuity in a member affects the stress distribution in the neighbourhood and the discontinuity acts as a stress raiser. Consider a plate with a centrally located hole and the plate is subjected to uniform tensile load at the ends. Stress distribution at a section A-A passing through the hole and another section BB away from the hole are shown in figure below.

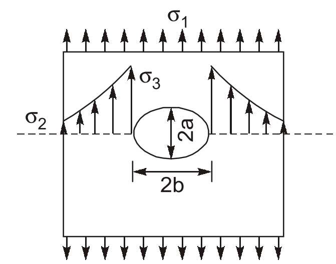
Stress distribution away from the hole is uniform but at AA there is a sharp rise in stress in the vicinity of the hole. Stress concentration factor k_1 is defined as $k_1 = \dfrac{\sigma_3}{\sigma_{av}}$, where σ_{av} at section AA is simply $\dfrac{P}{t(w-2b)}$ and $\sigma_1 = \dfrac{P}{tw}$. This is the theoretical or geometric stress concentration factor and the factor is not affected by the material properties.



Stress concentration due to a central hole in a plate subjected to an uni-axial loading.

It is possible to predict the stress concentration factors for certain geometric shapes using theory of elasticity approach. For example, for an elliptical hole in an infinite plate, subjected to a uniform tensile stress σ_1 as shown in figure, stress distribution around the discontinuity is disturbed and at points remote from the discontinuity the effect is insignificant. According to such an analysis

$$\sigma_3 = \sigma_1\left(1+\frac{2b}{a}\right)$$



Stress concentration due to a central elliptical hole in a plate subjected to a uni-axial loading

If $a = b$ the hole reduces to a circular one and therefore $\sigma_3 = 3\sigma$ which gives $k_t = 3$. If, however '*b*' is large compared to '*a*' then the stress at the edge of transverse crack is very large and consequently *k* is also very large. If '*b*' is small compared to a then the stress at the edge of a longitudinal crack does not rise and $k_t = 1$.

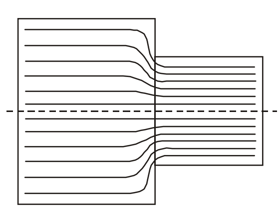
Stress concentration factors may also be obtained using any one of the following experimental techniques:

1. Strain gage method
2. Photoelasticity method
3. Brittle coating technique
4. Grid method

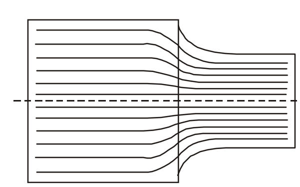
3.2.1 Methods to Reduce Stress Concentration

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

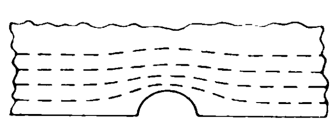
1. Provide a fillet radius so that the cross-section may change gradually.
2. Sometimes an elliptical fillet is also used.
3. If a notch is unavoidable it is better to provide a number of small notches rather than a long one. This reduces the stress concentration to a large extent.
4. If a projection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



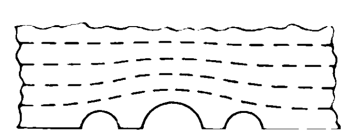
(a) Force flow around a sharp corner



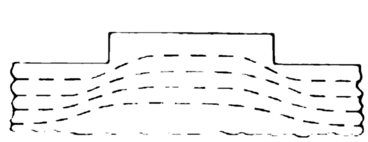
Force flow around a corner with fillet: Low stress concentration.



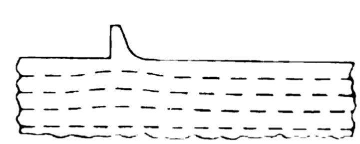
(b) Force flow around a large notch



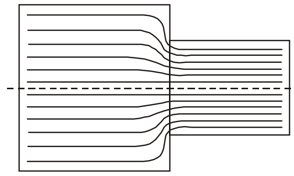
Force flow around a number of small notches: Low stress concentration.



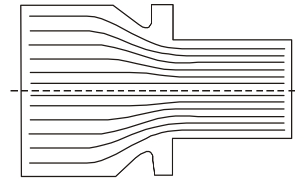
(c) Force flow around a wide projection



Force flow around a narrow projection: Low stress concentration.



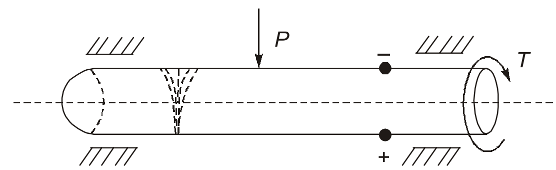
(d) Force flow around a sudden change in diameter in a shaft



Force flow around a stress relieving groove

Illustrations of different methods to reduce stress concentration

3.3 Dynamic Loading



Stresses developed in a rotating shaft subjected to a bending load

Conditions often arise in machines and mechanisms when stresses fluctuate between a upper and a lower limit. For example in figure, the fiber on the surface of a rotating shaft subjected to a bending load, undergoes both tension and compression for each revolution of the shaft.

Any fiber on the shaft is therefore subjected to fluctuating stresses. Machine elements subjected to fluctuating stresses usually fail at stress levels much below their ultimate strength and in many cases below the yield point of the material too. These failures occur due to very large number of stress cycle and are known as fatigue failure. These failures usually begin with a small crack which may develop at the points of discontinuity, an existing subsurface crack or surface faults. Once a crack is developed it propagates with the increase in stress cycle finally leading to failure of the component by fracture. There are mainly two characteristics of this kind of failures:

(a) Progressive development of crack.

(b) Sudden fracture without any warning since yielding is practically absent.

Fatigue failures are influenced by :

(i) Nature and magnitude of the stress cycle.

(ii) Endurance limit.

(iii) Stress concentration.

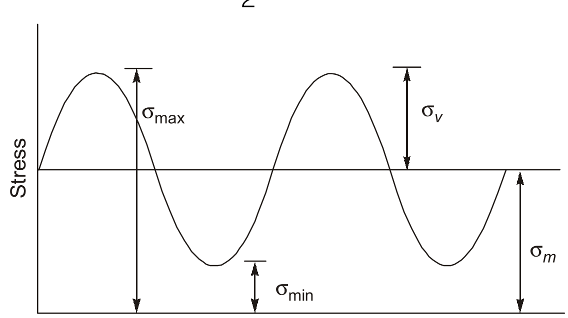
(iv) Surface characteristics

3.3.1 Stress Cycle

A typical stress cycle is shown in figure where the maximum, minimum, mean and variable stresses are indicated. The mean and variable stresses are given by

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2}$$

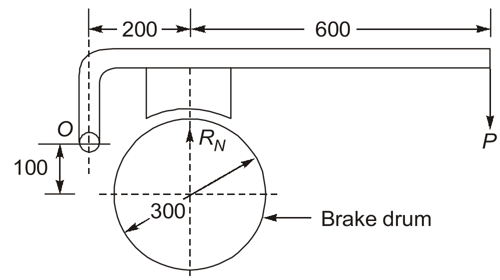
$$\sigma_{variable} = \frac{\sigma_{max} - \sigma_{min}}{2}$$



A typical stress cycle showing maximum, mean and variable stresses

Example 3.33 The block brake, as shown in Figure, provides a braking torque of 360 N-m. The diameter of the brake drum is 300 mm. The coefficient of friction is 0.3. Find :

1. The force (P) to be applied at the end of the lever for the clockwise and counter clockwise rotation of the brake drum; and
2. The location of the pivot or fulcrum to make the brake self locking for the clockwise rotation of the brake drum

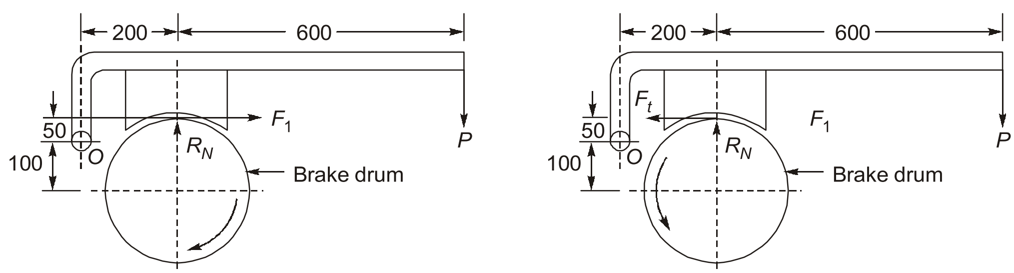


Solution :

Given : T_B = 360 N-m = 360 × 10^3 N-mm; d = 300 mm or r = 150 mm = 0 15 m; μ = 0.3

1. Force (P) for the clockwise and counter clockwise rotation of the brake drum

For the clockwise rotation of the brake drum, the frictional force or the tangential force (F_t) acting at the contact surfaces is shown in figure.



(a) Clockwise rotation (b) Anticlockwise rotation

We know that braking torque (T_B),

$$360 = F_t \times r = F_t \times 0.15 \text{ or } F_t = \frac{360}{0.15} = 2400 \text{ N}$$

and normal force,
$$R_N = \frac{F_t}{\mu} = \frac{2400}{0.3} = 8000 \text{ N}$$

Now taking moments about the fulcrum O, we have

$$P(600 + 200) + F_t \times 50 = R_N \times 200$$
$$P \times 800 + 2400 \times 50 = 8000 \times 200$$
$$P \times 800 = 8000 \times 200 - 2400 \times 50 = 1480 \times 10^3$$

$$\therefore \quad P = 1480 \times \frac{10^3}{800} = 1850 \text{ N}$$

For the counter clockwise rotation of the drum, the l fictional force or the tangential force (F_t) acting at the contact surfaces is shown in figure.

Taking moments about the fulcrum O, we have

$$P(600 + 200) = F_t \times 50 + R_N \times 200$$
$$P \times 800 = 2400 \times 50 + 8000 \times 200 = 1720 \times 10^3$$

$$\therefore \qquad P = 1720 \times \frac{10^3}{800} = 2150 \text{ N}$$

2. **Location of the pivot or fulcrum to make the brake self-locking**

   The clockwise rotation of the brake drum is shown in figure. Let x be the distance of the pivot or fulcrum O from the line of action of the tangential force (F_t). Taking moments about the fulcrum O, we have

   $P(600 + 200) + F_t \times x - R_N \times 200 = 0$

   In order to make the brake self-locking, $F_t \times x$ must be equal to $R_N \times 200$ so that the force P is zero.

$$\therefore \qquad F_t \times x = R_N \times 200$$

$$2400 \times x = 8000 \times 200 \quad \text{or} \quad x = \frac{8000 \times 200}{2400} = 667 \text{ mm}$$

Example 3.34 A rope drum of an elevator having 650 mm diameter is fitted with a brake drum of 1 m diameter. The brake drum is provided with four cast iron brake shoes each subtending an angle of 45°. The mass of the elevator when loaded is 2000 kg and moves with a speed of 2.5 m/s. The brake has a sufficient capacity to stop the elevator in 2.75 metres. Assuming the coefficient of friction between the brake drum and shoes as 0.2. Find :

1. width of the shoe, if the allowable pressure on the brake shoe is limited to 0.3 N/mm^2; and
2. heat generated in stopping the elevator.

Solution :

Given : d_e = 650 mm or r_e = 325 mm = 0.325 m; d = 1 m or r = 0.5 m = 500 mm; n = 4; 2θ = 45° or θ = 22.5°; m = 2000 kg; v = 2.5 m/s; h = 2.75 m; μ = 0.2; p_b = 0.3 N/mm^2

1. Width of the shoe

Let $\qquad w$ = Width of the shoe in mm.

First of all, let us find out the acceleration of the rope (a). We know that

$$v^2 - u^2 = 2a.h \quad \text{or} \quad (2.5)^2 - 0 = 2a \times 2.75 = 5.5a$$

$$\therefore \qquad a = \frac{(2.5)^2}{5.5} = 1.136 \text{ m/s}^3$$

and $\qquad$ accelerating force = Mass × Acceleration

$$= m \times a = 2000 \times 1.136 = 2272 \text{ N}$$

∴ Total load acting on the rope while moving,

$$W = \text{Load on the elevator in Newton + Accelerating force}$$
$$= 2000 \times 9.81 + 2272 = 21892 \text{ N}$$

We know that torque acting on the shaft,

$$T = W \times r_e = 21892 \times 0.325 = 7115 \text{ N-m}$$

∴ Tangential force acting on the drum

$$= \frac{T}{r} = \frac{7115}{0.5} = 14230 \text{ N}$$

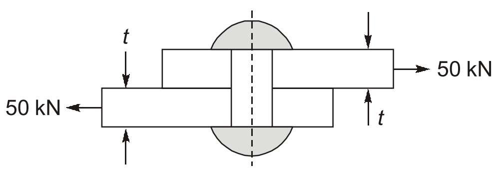
The brake drum is provided with four cast iron shoes, therefore tangential force acting on each shoe,

$$F_t = \frac{14230}{4} = 3557.5 \text{ N}$$

Since the angle of contact of each shoe is 45°, therefore we need not to calculate the equivalent coefficient of friction (μ').

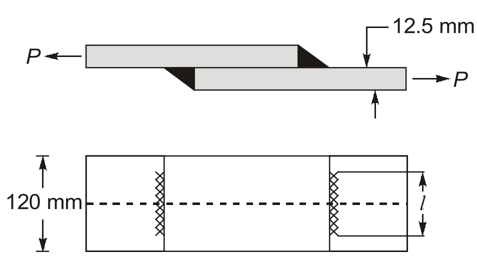
Student's Assignments 2

Q.16 Two plates, subjected to a tensile force of 50 kN, are fixed together by means of three rivets as shown in figure. The plates and rivets are made of plain carbon steel with a tensile yield strength of 250 N/mm^2. The yield strength in shear is 50% of the tensile yield strength, and the factor of safety is 2.5. Width of plate is 200mm. Neglecting stress concentration, the ratio of diameter of the rivets to the thickness of plates is



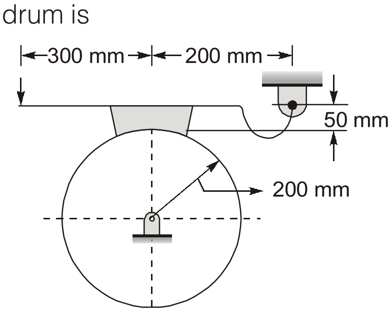
(a) 3.21 (b) 5.69
(c) 7.82 (d) 9.21

Q.17 Two steel plates, 120 mm wide and 12.5 mm thick, are jointed together by means of double transverse fillet weld as shown in figure. What is the required length of the weld (l) if the permissible shear stress in the weld is 55 N/mm^2?



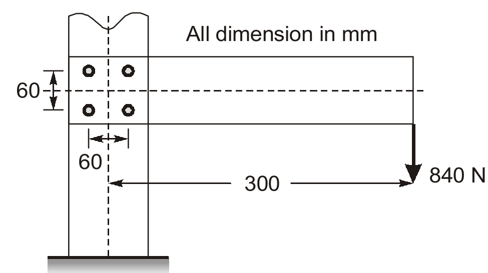
(a) 70.21 mm (b) 84.87 mm
(c) 96.21 mm (d) 102.28 mm

Q.18 A single block brake with a torque capacity of 250 Nm is shown in figure. The brake drum rotates at 100 rpm and the friction coefficient is 0.35. The hinge-pin reaction for clockwise rotation of the drum is

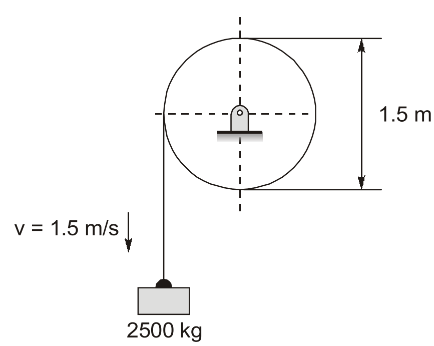


(a) 2375 N (b) 2575 N
(c) 3220 N (d) 3730 N

Q.19 A rectangular steel plate is joined to a vertical post using four identical rivets arranged as shown in below in the figure. The shear load on the worst loaded rivet approximately is ________ N.



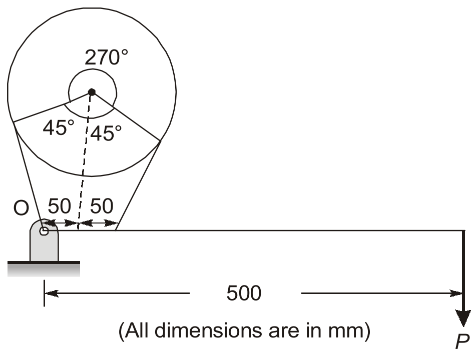
Q.20 A mass of 2500 kg is lowered at a velocity of 1.5 m/s from the drum as shown in figure. The mass of the drum is 50 kg and its radius of gyration can be taken as 0.7 m. On applying the brake, the mass is brought to rest in a distance of 0.5 m. The energy absorbed by brake is __ J.



Q.21 The standard cross-section for a flat key is 14 × 9 mm which is used to connect a pulley to a 45 mm diameter shaft. The key is made of commercial steel ($S_{yt} = S_{yc}$ = 230 N/mm^2) and factor of safety is 3. Find the length of key (in mm) if 15 kW power at 360 rpm is transmitted through the keyed joint.

(a) 51.26 (b) 33
(c) 512.6 (d) 20

Q.22 A simple band brake as shown below, has a drum diameter of 500 mm. The drum rotates at 300 rpm. The angle of wrap is 270°.



(All dimensions are in mm)

Find, the minimum force (in N) for its direction required at the end of the brake lever to stop the drum, if 50 kW power is to absorbed
[Take, μ = 0.2]
(a) 1475 N in clockwise direction
(b) 575 N in anticlockwise direction
(c) 1475 N in anticlockwise direction
(d) 575 N in clockwise direction

Q.23 A belt is required to transmit 9 kW from a pulley 120 cm diameter running at 200 rpm. The angle embraced is 165° and the coefficient of friction 0.3. If the safe working stress for a leather is 1.4 N/mm^2, the weight of 1 cm^3 of leather = 0.01 N and the thickness of belt = 10 mm, then the value of width of belt will be required taking into account the centrifugal force is ________ cm.

Q.24 A journal bearing is loaded with a radial load of 30 kN. The journal diameter and length both are 100 mm and it rotates at 1200 rpm. The heat is dissipated from the surface at the rate of 96 J/m^2/sec./°C. The bearing housing is 20 times the projected area. If coefficient of friction is 0.003 and room temperature is 35°C, then the surface temperature of the bearings is _________°C.

Q.25 A spherical vessel of a1000 mm inner diameter is subjected to an internal pressure which varies from 5 MPa to 10 MPa. The material of the vessel is cold drawn steel having ultimate strength 450 MPa and yield strength 240 MPa. If the reliability of the vessel is 90% and required factor of safety is 2, the thickness of pressure vessel for an infinite life period is ________ mm. [Assume, $\sigma_e = 0.5\,\sigma_u$]

ANSWERS

1. (c)	2. (d)	3. (b)	4. (c)
5. (a)	6. (a)	7. (70)	8. (360)
9. (1.75)	10. (0.95)	11. (b)	12. (151)
13. (c)	14. (19)	15. (63.33)	16. (b)
17. (b)	18. (b)	19. (1640.23)	20. (15124)
21. (a)	22. (d)	23. (9.967)	24. (64.45)
25. (21.8)			

HINTS

4. (c)

(i) The stiffness of solid shaft is less than the hollow shaft with same weight.

(ii) Due to manufacturing constraints, hollow shaft is difficult to make and hence is costlier.

5. (a)

We know that endurance or bending strength is approximately one-third of the ultimate tensile strength.

$$\sigma_b = \frac{S_{ut}}{3} = \frac{600}{3} = 200 \text{ N/mm}^2$$

$$\therefore \quad S_b = \sigma_b bmy = 200 \times 3 \times 40 \times 0.4 = 9600 \text{ N}$$

6. (a)

$$r_m = \frac{1}{3}\left(\frac{D^3 - d^3}{D^2 - d^2}\right)$$

$$= \frac{1}{3}\left(\frac{50^3 - 30^3}{50^2 - 30^2}\right) = 20.42 \text{ mm}$$

7. 70 (69 to 71)

Since the key is wider than its depth or thickness, it fail due to compression, before it will fail due to shear.

$$T = F \times \frac{d}{2}$$

$$= \left(l \times \frac{t}{2}\right) \times \sigma_c \times \frac{d}{2}$$

$$\Rightarrow 700 = \left(l \times \frac{0.009}{2}\right) \times 110 \times 10^6 \times \frac{0.04}{2}$$

$$\Rightarrow \quad l = 0.07 \text{ m} = 70 \text{ mm}$$

8. (360)

$$P = \frac{2\pi NT}{60}$$

and, $$T = \frac{\pi}{16}d^3\tau$$

So, $$P \propto Nd^3$$

$$\frac{P_2}{P_1} = \frac{N_2 d_2^3}{N_1 d_1^3} = \frac{1}{2}\times 2^3 = 4$$

$\therefore$ $$P_2 = 4P_1 = 4 \times 90 = 360 \text{ kW}$$

9. 1.75 (1.73 to 1.77)

Given: $p = 4$ kN

$k_b = 2.5$ N/m

$k_m = 3.2$ N/m

$$p_b = \frac{k_b}{k_b + k_m}P$$

$$= \frac{2.5}{2.5+3.2}\times 4000$$

$$= \frac{2.5}{5.7}\times 4000$$

$$= 1754.385 \text{ N} = 1.75 \text{ kN}$$

10. (0.95)

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}\left[1 - \frac{S_1}{100} - \frac{S_2}{100}\right]$$

$$\frac{V_2}{V_1} = 1 - \frac{S_1}{100} - \frac{S_2}{100}$$

$$= 1 - \frac{2}{100} - \frac{3}{100} = 0.95$$

12. (151) (145 to 160)

$P = 0.828 \times t \times l \times \tau_{max} \times 2$

t = thickness of leg

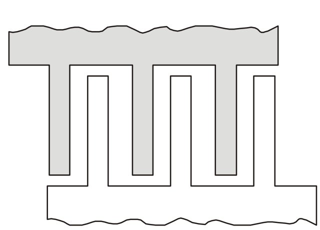
l = Length of leg

$P = 0.828 \times 10 \times 200 \times 2 \times \tau_{max}$

$P = 500$ kN

$\tau_{max} = 150.96 \approx 151$ N/mm^2

13. (c)



Here $n_1 = 3$ and $n_2 = 3$

But surfaces are 5, i.e., $n_1 + n_2 - 1$

14. 19 (18.9 to 19.3)

$$t = \frac{PD}{2\sigma_t\eta} = \frac{2.75\times 1000}{2\times 90\times 0.8}$$

$$= 19.09 \simeq 19 \text{ mm}$$

15. 63.33 (63 to 64)

$$\text{Efficiency} = \frac{\text{Least of } P_1, P_2 \ldots}{P}$$

$$= \frac{19}{30} = 0.633 = 63.33\%$$

16. (b)

Permissible shear stress for rivets,

$$\tau = \frac{S_{ys}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5\times 250}{2.5}$$

$$= 50 \text{ N/mm}^2$$

Since there are three rivets,

$\therefore$ $$3\left(\frac{\pi d^2}{4}\right)\tau = P \text{ or } 3\left(\frac{\pi d^2}{4}\right)50 = 50 \times 10^3$$

$\therefore$ $d = 20.60$ mm

Permissible tensile stress for plates,

$$\sigma_t = \frac{S_{yt}}{(f_s)} = \frac{250}{2.5} = 100 \text{ N/m}^2$$

$\therefore \sigma_t(200 - 3d) \times t = P$

or, $100(200 - 3 \times 20.6) \times t = 50 \times 10^3$

$\therefore$ $t = 3.62$ mm

So, $$\frac{d}{t} = \frac{20.6}{3.62} = 5.69$$

17. (b)

The plates are subjected to tensile stress, so the maximum tensile force is given by,

$$P = (Wt)\sigma_t = (120 \times 12.5) \times \frac{55}{0.5}$$

$$= 165000 \text{ N}$$

So, length of the weld,

$$l = \frac{P}{1.414hl\sigma_t} = \frac{165000}{(1.414\times 12.5\times 110)}$$

18. (b)