العرض التوضيحي لمحتويات الكتاب المرجعي

VISUAL WALKTHROUGH

لمقرر / تصميم آلات زراعية

الفرقة الرابعة – هندسة زراعية

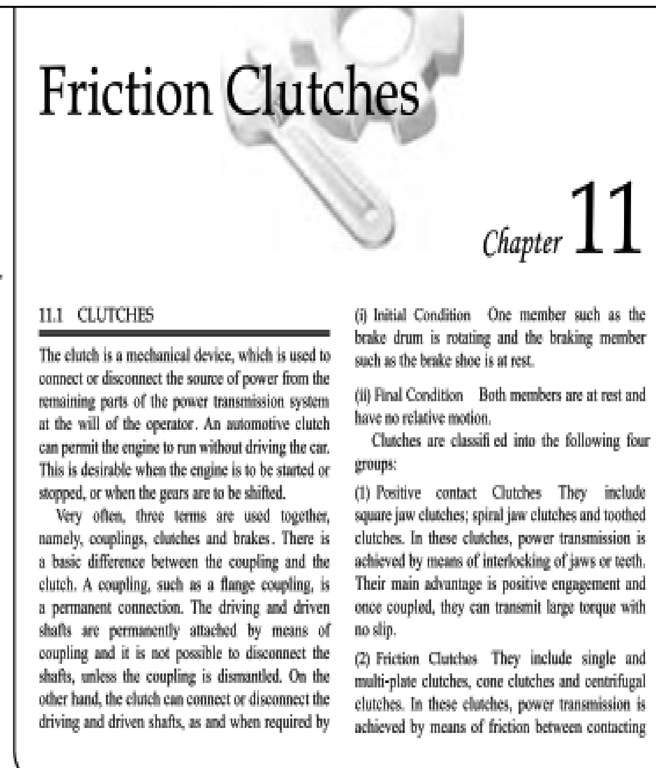
العام الجامعي ٢٠٢٠/ ٢٠٢١م.

المقدمة:

كل باب من ابواب الكتاب يحتوي على مقدمة عن العنصر التصميمي ووظائفه الاساسية. وهذا يساعد القارئ في اكتساب المعلومات الاساسية عن هذا العنصر.

Introduction

Each chapter begins with an Introduction of the Machine Element designed in the chapter and its functions. This helps the reader in gaining an overview of the machine element.



Friction Clutches

Chapter 11

11.1 CLUTCHES

The clutch is a mechanical device, which is used to connect or disconnect the source of power from the remaining parts of the power transmission system at the will of the operator. An automotive clutch can permit the engine to run without driving the car. This is desirable when the engine is to be started or stopped, or when the gears are to be shifted.

Very often, three terms are used together, namely, couplings, clutches and brakes. There is a basic difference between the coupling and the clutch. A coupling, such as a flange coupling, is a permanent connection. The driving and driven shafts are permanently attached by means of coupling and it is not possible to disconnect the shafts, unless the coupling is dismantled. On the other hand, the clutch can connect or disconnect the driving and driven shafts, as and when required by

(i) Initial Condition One member such as the brake drum is rotating and the braking member such as the brake shoe is at rest.

(ii) Final Condition Both members are at rest and have no relative motion.

Clutches are classifi ed into the following four groups:

(1) Positive contact Clutches They include square jaw clutches; spiral jaw clutches and toothed clutches. In these clutches, power transmission is achieved by means of interlocking of jaws or teeth. Their main advantage is positive engagement and once coupled, they can transmit large torque with no slip.

(2) Friction Clutches They include single and multi-plate clutches, cone clutches and centrifugal clutches. In these clutches, power transmission is achieved by means of friction between contacting

الاعتبارات النظرية:

يوضح هذا المحتوى المعادلات الاساسية المستخدمة في تصميم العنصر والتي يتم اشتقاقها من الاسس الهندسية بنظام النهج التدريجي.

$$= \frac{(R_o^2 - R_i^2)}{4} - \left(\frac{R_i^2}{2}\right)\log_e\left(\frac{R_o}{R_i}\right)$$

Substituting this value and Eq. (16.9) in (d), we have

$$W = \frac{\pi P_i}{2}\left[\frac{R_o^2 - R_i^2}{\log_e\left(\frac{R_o}{R_i}\right)}\right] \qquad (16.10)$$

The above equation can be used even if there is no recess, in which case, R_i will be the radius of the oil-supply pipe.

16.9 ENERGY LOSSES IN HYDROSTATIC BEARING

The total energy loss in a hydrostatic step bearing consists of two factors—the energy required to pump the lubricating oil and energy loss due to viscous friction. The energy E_P required to pump the oil is given by,

$$E_p = Q(P_i - P_o)\frac{\text{mm}^3}{s} \times \frac{N}{\text{mm}^2}$$

$$E_p = Q(P_i - P_o) \quad \text{N-mm/s}$$

$$= Q(P_i - P_o)(10^{-3}) \text{ N-m/s or W}$$

Therefore,

$$(\text{kW})_p = Q(P_i - P_o)(10^{-6}) \qquad (16.11)$$

where $(\text{kW})_p$ is the power loss in pumping (in kW). The frictional power loss is determined by considering the elemental ring of radius (r) and radial thickness (dr) illustrated in Fig. 16.14(a). The viscous resistance for this ring is (dF). It is determined by Newton's law of viscosity. According to this law:

$$d(M_t)_f = r \times dF = \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)r^3 dr$$

Integrating,

$$(M_t)_f = \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\int_{R_i}^{R_o} r^3 dr$$

$$= \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\left[\frac{r^4}{4}\right]_{R_i}^{R_o}$$

$$= \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\frac{(R_o^4 - R_i^4)}{4}$$

$$= \left(\frac{\pi^2}{60}\right)\frac{\mu n(R_o^4 - R_i^4)}{h_o}$$

The unit of $(M_t)_f$ is (N-mm).

$$(\text{kW})_f = \frac{2\pi n(M_t)_f}{60 \times 10^6}$$

$$= \frac{2\pi n}{60 \times 10^6}\left(\frac{\pi^2}{60}\right)\frac{\mu n(R_o^4 - R_i^4)}{h_o}$$

Therefore,

$$(\text{kW})_f = \left(\frac{2\pi^3}{3600 \times 10^6}\right)\frac{\mu n^2(R_o^4 - R_i^4)}{h_o}$$

or

$$(\text{kW})_f = \left(\frac{1}{58.05 \times 10^6}\right)\frac{\mu n^2(R_o^4 - R_i^4)}{h_o} \qquad (16.12)$$

The total power loss $(\text{kW})_t$ is given by

$$(\text{kW})_t = (\text{kW})_p + (\text{kW})_f \qquad (16.13)$$

Example 16.1 *The following data is given for a hydrostatic thrust bearing:*

thrust load = 500 kN

Theoretical Considerations

Basic equations for design are derived from first principle, with a step-by-step approach.

خواص المواد:

يوضح هذا المحتوى الجداول الخاصة بالخواص الميكانيكية للمواد الهندسية.

Properties of Materials

Exhaustive tables are provided from Indian Standards for Mechanical Properties of Engineering Materials.

Grade	Tensile strength (Min.) (N/mm^2)	Elongation (Min.) (%)	Hardness (HB)
(A) Grey cast iron			
FG 150	150	-	130–180
FG 200	200	-	160–220
FG 220	220	-	180–220
FG 260	260	-	180–230
FG 300	300	-	180–230
FG 350	350	-	207–241
FG 400	400	-	207–270
(B) Whiteheart malleable cast iron			
WM 400	400	5	220 (Max.)
WM 350	350	3	230 (Max.)
(C) Blackheart malleable cast iron			
BM 350	350	10	150 (Max.)
BM 320	320	12	150 (Max.)
BM 300	300	6	150 (Max.)
(D) Pearlitic malleable cast iron			
PM 700	700	2	240–290
PM 600	600	3	200–250
PM 550	550	4	180–230
PM 500	500	5	160–200
PM 450	450	6	150–200

المعايير القياسية:

يوضح هذا المحتوى المعايير القياسية القومية لعناصر الآلات مثل: (المسامير ذات الاسنان – السيور – اليايات – التروس – اسلاك الكابلات – اوعية الضغط).

Belt section	Pitch width W_p (mm)	Nominal top width W(mm)	Nominal Height T (mm)	Recommended Minimum pitch diameter of pulley (mm)	Permissible Minimum pitch diameter of pulley (mm)
Z	8.5	10	6	85	50
A	11	13	8	125	75
B	14	17	11	200	125
C	19	22	14	315	200
D	27	32	19	500	355
E	32	38	23	630	500

two factors, namely, the power to be transmitted and speed of the faster shaft. Figure 13.24 shows the range of speed and power for various cross-sections of the belt. Depending upon the power and speed of the faster pulley, a point can be plotted on this diagram and the corresponding cross-section selected. In borderline cases, alternative design calculations are made to determine the best solution.

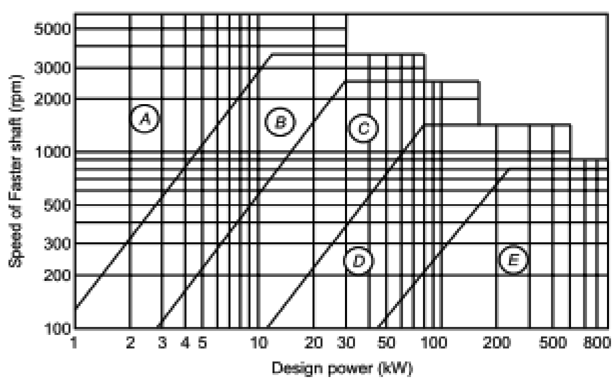


Fig. 13.24 *Selection of Cross-section of V belt*

The calculations of V-belts are based on preferred pitch diameters of pulleys and pitch lengths. The series of preferred values for pitch diameters and pitch lengths (in mm) are given in Tables 13.13 and 13.14 respectively.

The number of belts required for a given appli-

where,

P = drive power to be transmitted (kW)

F_a = correction factor for industrial service (Table 13.15)

P_r = power rating of single V-belt (from Table 13.16 to Table 13.20)

F_c = correction factor for belt length (Table 13.21)

Indian Standards

Indian Standards are used for Machine Elements like screw threads, belts, springs, gears, wire ropes and pressure vessels.

اختيار الاجراء:

يوضح هذا المحتوى انه عندما يتم اختيار عناصر اي آلة من كتالوجات المصنعين ، فإن اختيار الاجراءات يتم مناقشتها بناءا على مرجع محدد طبقا للجهة المنتجة.

operation (24 h per day) such as pumps, compressors and conveyors

The values given in the above tables are only for general guidance. For a particular application, the designer should consider the past experience, the difficulties faced by the customer in replacing the bearing and the economics of breakdown costs.

15.11 LOAD FACTOR

The forces acting on the bearing are calculated by considering the equilibrium of forces in vertical and horizontal planes. These elementary equations do not take into consideration the effect of dynamic load. The forces determined by these equations are multiplied by a *load factor* to determine the dynamic load carrying capacity of the bearing. Load factors are used in applications involving gear, chain and belt drives. In gear drives, there is an additional dynamic load due to inaccuracies of the tooth profile and the elastic deformation of teeth. In chain and

15.12 SELECTION OF BEARING FROM MANUFACTURER'S CATALOGUE

The basic procedure for the selection of a bearing from the manufacturer's catalogue consists of the following steps:

(i) Calculate the radial and axial forces acting on the bearing and determine the diameter of the shaft where the bearing is to be fitted.

(ii) Select the type of bearing for the given application.

(iii) Determine the values of X and Y, the radial and thrust factors, from the catalogue. The values of X and Y factors for single-row deep groove ball bearings are given in Table 15.4. The values depend upon two ratios, $\left(\frac{F_a}{F_r}\right)$ and $\left(\frac{F_a}{C_0}\right)$, where C_0 is the static load capacity. The selection of the bearing is, therefore, done by trial and error. The static and dynamic load capacities of

Selection Procedure

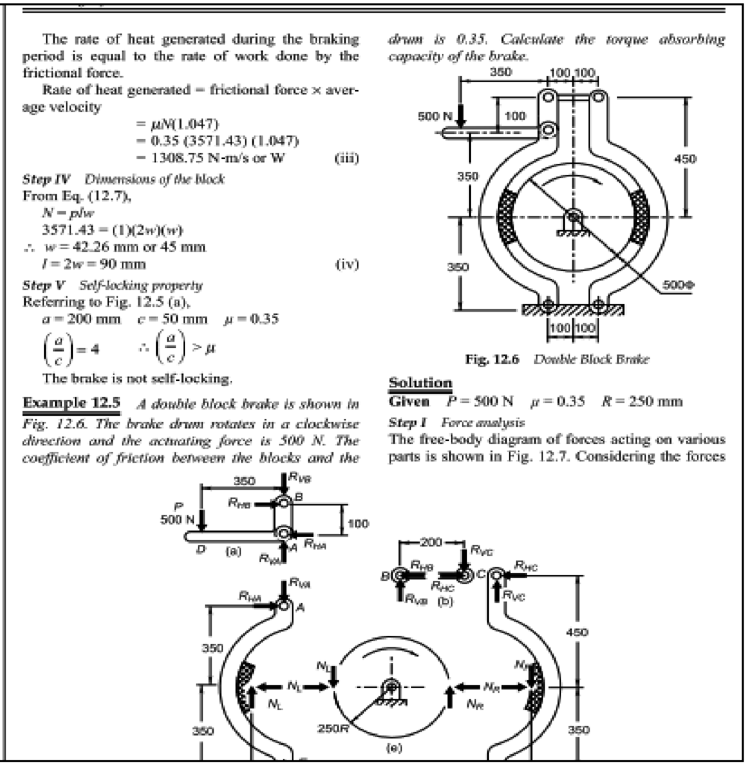
When a machine component is to be selected from manufacturer's catalogue, the selection processes are discussed with a particular reference to Indian products.

الرسم البيانى للجسم الحر:

يوضح هذا الجزء انه عندما يكون المطلوب رسم الجسم الحر لكي يساعد القارئ في فهم القوى التي تؤثر على المكونات الفردية لاجزاء الآلات.

Free-Body Diagram of Forces

Whenever required, free-body diagrams are constructed to help the reader understand the forces acting on individual components.



The rate of heat generated during the braking period is equal to the rate of work done by the frictional force.

Rate of heat generated = frictional force × average velocity

$$= \mu N(1.047)$$
$$= 0.35\,(3571.43)\,(1.047)$$
$$= 1308.75 \text{ N-m/s or W} \qquad \text{(iii)}$$

Step IV Dimensions of the block

From Eq. (12.7),

$$N = plw$$
$$3571.43 = (1)(2w)(w)$$
$$\therefore\ w = 42.26 \text{ mm or } 45 \text{ mm}$$
$$l = 2w = 90 \text{ mm} \qquad \text{(iv)}$$

Step V Self-locking property

Referring to Fig. 12.5 (a),

$a = 200$ mm $\quad c = 50$ mm $\quad \mu = 0.35$

$$\left(\frac{a}{c}\right) = 4 \qquad \therefore \left(\frac{a}{c}\right) > \mu$$

The brake is not self-locking.

Example 12.5 *A double block brake is shown in Fig. 12.6. The brake drum rotates in a clockwise direction and the actuating force is 500 N. The coefficient of friction between the blocks and the drum is 0.35. Calculate the torque absorbing capacity of the brake.*

Fig. 12.6 *Double Block Brake*

Solution

Given $P = 500$ N $\quad \mu = 0.35 \quad R = 250$ mm

Step I Force analysis

The free-body diagram of forces acting on various parts is shown in Fig. 12.7. Considering the forces

المنحنيات البيانية للارهاق او الكلل:

يوضح هذا المحتوى المنحنيات البيانية التي يتم رسمها لتصميم اجزاء الآلات المعرضة لاجهادات متغيرة.

$$\sigma_a = \sigma_m = \frac{1}{2}\sigma_{max.} = \frac{1}{2}\left(\frac{750}{t}\right) = \left(\frac{375}{t}\right) \text{N/mm}^2$$

$$\tan\theta = \frac{\sigma_a}{\sigma_m} = 1 \quad \text{or} \quad \theta = 45°$$

The modified Goodman diagram for this example is shown in Fig. 5.45.

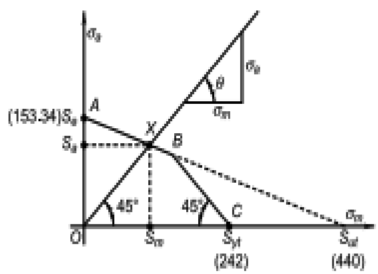


Fig. 5.45

Step III Permissible stress amplitude
Refer to Fig. 5.45. The coordinates of the point *X* are determined by solving the following two equations simultaneously.

(i) Equation of line *AB*

is 50%. Calculate the number of stress cycles likely to cause fatigue failure.

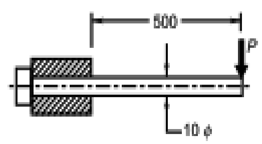


Fig. 5.46

Solution

Given For cantilever spring, $d = 10$ mm $l = 500$ mm $P = 75$ to 150 N $S_{ut} = 860$ N/mm² $S_{yt} = 690$ N/mm² $R = 50\%$

Step I Endurance limit stress for cantilever beam

$$S'_e = 0.5 S_{ut} = 0.5(860) = 430 \text{ N/mm}^2$$

From Fig. 5.24 (machined surface and $S_{ut} = 860$ N/mm²),

$K_a = 0.72$

For 10 mm diameter wire, $K_b = 0.85$

For 50% reliability,

$K_c = 1.0$

$$S_e = K_a K_b K_c S'_e = 0.72(0.85)(1.0)(430) = 263.16 \text{ N/mm}^2$$

Fatigue Diagrams

Fatigue diagrams are constructed for design of machine components subjected to fluctuating loads.

منظور الرؤية (المنظور):

يوضح هذا الجزء انه عندما يصعب فهم القوى التي تؤثر على عنصر من عناصر الالة خصوصا في الاتجاهات الثلاثة (الثلاثية لابعاد) ، فإن مقياس الرؤية (المنظور) هي المقياس الذي يوضح ذلك المفهوم.

$$P_p = \left(\frac{\pi D^2}{4}\right) p' \qquad (25.69a)$$

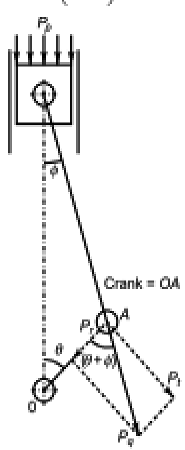


Fig. 25.23 *Force Acting on Crank*

The relationship between φ and θ is given by,

$\sin\theta$

Fig. 25.25. Due to the tangential component P_t, there are reactions $(R_1)_h$ and $(R_2)_h$ at bearings 1 and 2 respectively. Similarly, due to the radial component P_r, there are reactions $(R_1)_v$ and $(R_2)_v$ at bearings 1 and 2 respectively.

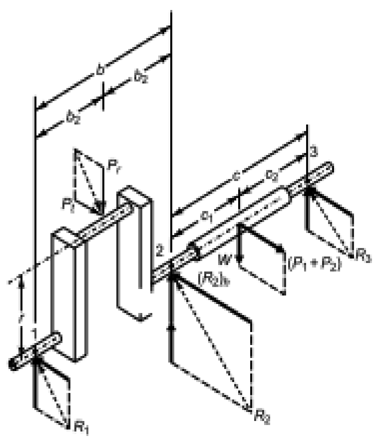


Fig. 25.24 *Centre Crankshaft at Angle of Maximum Torque*

Isometric Views

When it is difficult to understand the forces in three dimensions, isometric views are given for clear understanding.

الاعتبارات الاحصائية فى التصميم:

تم تخصيص باب منفصل للاعتبارات الاحصائية في التصميم ويحتوي هذا الباب على أمثلة توضيحية محلولة تعتمد على مفهوم الاعتمادية .

Statistical Considerations in Design

Chapter **24**

24.1 FREQUENCY DISTRIBUTION

Statistics deals with drawing conclusions from a given or observed data. Statistical techniques are used for collection, processing, analysis and interpretation of numerical data. On the basis of statistical analysis, valid conclusions are drawn and reasonable decisions are taken. Statistics enables the engineers to understand the phenomena of variations and how to effectively predict and control them. Statistics has made valuable contributions in the areas of product design and manufacture and effective use of material and labour.

The basic data consists of observations, such as the diameters of shafts manufactured in one shift. In this case, the group of all shafts is included in the entire population. Therefore, many times a sample is analysed instead of the entire population.

Let us consider an example of 100 shafts of hydrodynamic bearing, with recommended tolerance of 40e7. The shafts are manufactured on lathe and finished on grinding machine. Their diameters are measured by micrometer and the readings are tabulated in Table 24.1. The readings in Table 24.1 are called '*raw data*'. *A data is defined as the collection of numbers belonging to observations of one or more variables.* In this case, the diameter of shaft is a variable and one hundred numerical readings taken by micrometer are a data. Raw data is a data before it is arranged or analysed

Statistical Considerations in Design

A separate chapter on Statistical Considerations in Design is included and examples are solved on the basis of reliability.

أمثلة رقمية:

يوضح هذا الجزء امثلة رقمية تم حلها بنظام النهج التدريجي وذلك لكل باب من ابواب الكتاب وبعدد كافي حتى يتسنى للقارئ فهم الاجراءات التصميمية.

Numerical Examples

Numerical Examples solved by step by step approach are provided in sufficient number in each chapter to help the reader understand the design procedures.

370 N/mm². The permissible shear stress in the spring wire should be taken as 50% of the ultimate tensile strength. Design the spring and calculate
(i) wire diameter;
(ii) mean coil diameter;
(iii) number of active coils;
(iv) required spring rate; and
(v) actual spring rate.

Solution

Given $P = 1500$ N $C = 6$ $S_{ut} = 1360$ N/mm²
$G = 81\,370$ N/mm² $\tau = 0.5\, S_{ut}$

Step I *Wire Diameter*

The working principle of the spring balance is illustrated in Fig. 10.18. As the load acting on the spring varies from 0 to 1500 N, the pointer attached to the free end of the spring moves over a scale between highest and lowest positions. The length of the scale between these two positions of the pointer is 100 mm. In other words, the spring deflection is 100 mm when the force is 1500 N.

$$\therefore \qquad d = 6.5 \text{ or } 7 \text{ mm} \qquad \text{(i)}$$

Step II *Mean coil diameter*

$$D = Cd = 6(7) = 42 \text{ mm} \qquad \text{(ii)}$$

Step III *Number of active coils*

From Eq. (10.8),

$$\delta = \frac{8PD^3N}{Gd^4} \quad \text{or} \quad 100 = \frac{8(1500)(42)^3 N}{(81\,370)(7)^4}$$

$$\therefore \qquad N = 21.97 \text{ or } 22 \text{ coils} \qquad \text{(iii)}$$

Step IV *Required spring rate*

$$k = \frac{P}{\delta} = \frac{1500}{100} = 15 \text{ N/mm} \qquad \text{(iv)}$$

Step V *Actual spring rate*

$$k = \frac{Gd^4}{8D^3N} = \frac{(81\,370)(7)^4}{8(42)^3(22)} = 14.98 \text{ N/mm} \qquad \text{(v)}$$

اسئلة واجوبة قصيرة:

في نهاية كل باب من ابواب الكتاب تم اضافة مجموعة من الاسئلة والاجوبة القصيرة التي تساعد الطلاب وتعدهم للامتحانات الشفوية والنظرية.

Short Answer Questions

25.1 What are the functions of engine cylinder?
25.2 What are the cooling systems for engine cylinders? Where do you use them?
25.3 What are the advantages of cylinder liner?
25.4 What are dry and wet cylinder liners?
25.5 What are the desirable properties of cylinder materials?
25.6 Name the materials used for engine cylinder.
25.7 What do you understand by 'bore' of cylinder?
25.8 What are the functions of piston?
25.9 What are the design requirements of piston?
25.10 Name the materials used for engine piston.
25.11 What are the advantages and disadvantages of aluminium piston over cast iron piston?
25.12 Why is piston made lightweight?
25.13 Name two criteria for calculating the thickness of piston head.
25.14 Why is piston clearance necessary? What is
25.30 What is the force on bolts of of big end connecting rod?
25.31 What is the difference between centre and overhung crankshafts?
25.32 Where do you use overhung crankshafts?
25.33 Where do you use centre crankshafts?
25.34 What is the main advantage of overhung crankshafts?
25.35 Name the materials for crankshafts.
25.36 What is the manufacturing method for crankshaft?
25.37 When do you use push rod?
25.38 Why is the design of exhaust valve more critical than that of an inlet valve?
25.39 Why is the area of inlet valve port more than that of an exhaust valve?
25.40 Why do inlet and exhaust valves have conical heads and seats?
25.41 What is the function of rocker arm?
25.42 Why is rocker arm made of I section?
25.43 Name the materials for rocker arm.

Short-Answer Questions

At the end of each chapter, Short-Answer Questions are provided for the students for preparation of oral and theory examinations.

تمارين غير محلولة للتدريب:

في نهاية كل باب من ابواب الكتاب يوجد مجموعة من الامثلة مع ارشادات للاجوبة لها كتدريب للطلاب وايضا كمساعدة للمحاضرين في الاعمال الفصلية خلال المحاضرات وكذلك للواجبات غير الفصلية للطلاب.

Problems for Practice

At the end of each chapter, a set of examples with answers is given as exercise to students. It is also helpful to teachers in setting classwork and homework assignments.

Problems for Practice

11.1 A single plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 125 and 250 mm respectively. The coefficient of friction is 0.25 and the total axial force is 15 kN. Calculate the power transmitting capacity of the clutch at 500 rpm using :
(i) uniform wear theory; and
(ii) uniform pressure theory.

[(i) 18.41 kW (ii) 19.09 kW]

11.2 An automotive single plate clutch consists of two pairs of contacting surfaces. The outer diameter of the friction disk is 270 mm. The coefficient of friction is 0.3 and the maximum intensity of pressure is 0.3 N/mm^2. The clutch is transmitting a torque of 531 N-m. Assuming uniform wear theory, calculate:

($z = 1.1258$)]

11.5 A leather faced cone clutch transmits power at 500 rpm. The semi-cone angle α is 12.5°. The mean diameter of the clutch is 300 mm, while the face width of the contacting surface of the friction lining is 100 mm. The coefficient of friction is 0.2 and the maximum intensity of pressure is limited to 0.07 N/mm^2. Calculate the force to engage the clutch and the power transmitting capacity.

[1324.68 N and 9.61 kW]

11.6 A centrifugal clutch, transmitting 18.5 kW at 720 rpm, consists of four shoes. The clutch is to be engaged at 75% of the running speed. The inner radius of the drum is 165 mm, while the radius of the centre of gravity of each shoe, during engaged position, is 140 mm. The coefficient of friction is 0.25. Calculate the mass of each shoe.

[4.27 kg]

المراجع:

يوضح هذا الجزء قائمة باسماء المراجع من كتب ومجلات علمية وكذلك كتالوجات الشركات في نهاية الصفحات ذات الصلة لتساعد القارئ على المراجعة السريعة.

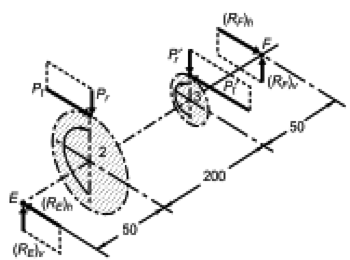


Fig. 17.30

Step II Reactions at bearings E and F
The forces acting in vertical and horizontal planes are shown in Fig. 17.31.

Considering vertical forces and taking moments about the bearing E,

$$P_r \times 50 + P_r' \times 250 = (R_F)_v \times 300$$
$$482.73 \times 50 + 1206.83 \times 250 = (R_F)_v \times 300$$
$$\therefore \quad (R_F)_V = 1086.15 \text{ N} \qquad \text{(i)}$$
$$P_r + P_r' = (R_F)_v + (R_E)_v$$
$$482.73 + 1206.83 = 1086.15 + (R_E)_v$$
$$\therefore \quad (R_E)_v = 603.41 \text{ N} \qquad \text{(ii)}$$

(b) Horizontal plane

Fig. 17.31

17.12 GEAR TOOTH FAILURES

There are two basic modes of gear tooth failure—breakage of the tooth due to static and dynamic loads and the surface destruction[4, 5]. The complete breakage of the tooth can be avoided by adjusting the parameters in the gear design, such as the module and the face width, so that the beam strength of the gear tooth is more than the sum of static and dynamic loads. The surface destruction or tooth wear is classified according to the basis of their primary causes. The principal types of gear tooth wear are as follows:

(i) Abrasive Wear Foreign particles in the lubricant, such as dirt, rust, weld spatter or metallic debris can scratch or brinell the tooth surface. Remedies against this type of wear are provision of oil filters, increasing surface hardness and use of high viscosity oils. A thick lubricating film developed by these oils allows fine particles to pass without scratching.

(ii) Corrosive Wear The corrosion of the tooth surface is caused by corrosive elements, such as

[4] RH Pearson–'Gear overdesign and how to avoid it'–Machine Design—May 9, 1968, vol. 40, no. 11, p. 153.
[5] Eugene E Shipley – 'Gear failures—how to recognize them, what causes them and how to avoid them?'—Machine Design—Dec 7, 1967, vol. 39, no. 28, p. 152.

References

The list of textbooks, journals and company catalogues is provided at the end of respective pages for quick reference.