Name: **Enrolment No:** **Roll No.**	UPES UNIVERSITY WITH A PURPOSE

UNIVERSITY OF PETROLEUM AND ENERGY STUDIES
End Semester/ Supplementary Examination, December 2020

Course: Automotive Engine Component Design **Semester: VIIth Sem**
Program: B.Tech. Automotive Design Engineering **Time 180 Minute**
Course Code: MEAD4001/ ADEG431 **Max. Marks: 100**

Instructions:
1. Assume the suitable data and mention in solution at start.
2. Draw the necessary diagrams.
3. Chapter of DDHB are pasted for use in Q. No. 2 of Section A & B at the end of QP.

Note:

1. Read the instruction carefully before attempting.

3. There are total of 4 questions in **Section A /B of scan and upload type.**

3. **Examination** will be conducted online on CODETANTRA platform.

4. Write the answer over A4 sheet and mention clearly the page number at the top. After the completion, scan and upload online through CODETANTRA platform.

S. No.	Statement of question	**Marks**	**CO**
	SECTION A **Attempt all the questions**		
Q 1	Write the answer in brief. (a) Compare the materials used for engine piston with their characteristics. (b) Differentiate between static and dynamic balancing. Discuss the points to be considered to obtain the firing order of a multi-cylinder engine. (c) Differentiate between the centre crank shaft and side crank shaft with help of diagrams. (d) Explain the controlling force diagram of Governor with its significance.	**20**	**CO1**

Q 2	The following data is given for a four-stroke diesel engine: Cylinder bore = 100 mm Length of stroke = 125 mm Speed = 2000 rpm Brake mean effective pressure = 0.65 MPa Maximum gas pressure = 5 MPa Fuel consumption = 0.25 kg per BP per hr Higher calorific value of fuel = 45000 kJ/kg Assume that 5% of the total heat developed in the cylinder is transmitted by the piston. The piston is made of grey cast iron and the permissible tensile stress is 37.5 N/mm^2 (k = 46.6 W/m/°C). The temperature difference between the Centre and edge of the piston head is 220°C. Bearing pressure at small end of connecting rod = 25 MPa Allowable bending stress for piston pin= 140 N/mm^2 Number of piston rings = 4. Design the piston completely. Assume the data required if not provided.	**20**	**CO2**
	SCTION B **Attempt all questions.**		
Q 1(a)	A governor of the Hartnell type has two balls , each ball of weight 15 N and the lengths of vertical and horizontal arms of the bell crank lever are 120 mm and 60 mm, respectively. The fulcrum of the bell crank lever is at a distance of 100 mm from the axis of rotation. The maximum and minimum radii of rotation of the balls are 120 mm and 80 mm and the corresponding equilibrium speeds are 325 and 300 rpm, respectively. Find the stiffness of the spring and the equilibrium speed when the radius of rotation is 100 mm.	**10**	**CO4**
Q 1(b)	The turning moment exerted on crankshaft of 2 stroke engine is given by T= 10000+1000 sin 2θ -3000 cos 2θ Where θ is inclination of crank to inner dead centre. The flywheel has mass of 600 kg with radius of gyration of 800 mm . The engine rotates at 360 rpm. Draw the **turning moment diagram** and determine ; (i) Power developed (ii) Fluctuation of speed (iii) Maximum acceleration of flywheel	**20**	**CO4**
Q 2 (a)	Four masses 150, 250, 200 and 300 kg are rotating in the same plane at radii of 0.25, 0.2, 0.3 and 0.35 m, respectively. Their angular location is 40°, 120° and 250° from mass 150 kg, respectively, measured in counter-clockwise direction. Find the position and magnitude of the balance mass required, if its radius of rotation is 0.25 m. Solve by using graphical method.	**10**	**CO3**

Q 2(b)	Data is given or the Centre crankshaft of a single cylinder vertical engine. Cylinder bore = 150mm L/r ratio = 4.75 Maximum gas pressure = 4 MPa Length of stroke = 200 mm Total belt pull = 1.8 kN Weight of the flywheel cum belt pulley = 3.5kN Allowable bending stress = 80 N/mm^2 Allowable compressive stress = 80 N/mm^2 Allowable shear stress = 40 N/mm^2 Allowable bearing pressure = 10 MPa The main bearings are 400 mm apart and the third bearing is 400 mm apart from the main bearing on its side. The belts are in horizontal direction. Consider the **case when the piston is at top dead center position**. Assume l/d =1 for crank pin . Design the crank shaft	**20**	**CO3**

Use the following chapters of DDHB for design problems as required.

18

Cylinders and Pistons of Steam and Internal Combustion Engines

Symbols	Description and Units
F_p	the maximum load on the piston, N (kgf)
h	the depth of the piston ring, mm
i	number of studs or bolts
L	length of the piston, mm
L_c	length of cylinder, mm
l_1	length of the gudgeon pin bearing, mm
n	speed of the engine, rev/s
p	working pressure, MN/m^2 (kgf/mm^2)
p_m	actual mean effective pressure, MN/m^2 (kgf/mm^2)
t	thickness of the cylinder, mm
t_f	thickness of the cylinder flange, mm
t_1	thickness of the piston head, mm
t_r	the radial thickness of the piston ring, mm
v	mean speed of the piston, m/s (Table 18.1)
W_R	weight of the reciprocating parts, N (kg)
σ_t	allowable stress, MN/m^2 (kgf/mm^2)

Particular	Equation	Eqn. No.
Steam Engine Cylinder:		
The diameter of a double acting eninge (steam engine)	$D = \left[\dfrac{1000(P)}{(\pi/4)p_m v}\right]^{\frac{1}{2}}$ SI units	18.1(a)
	$D = \left[\dfrac{75(P)}{(\pi/4)p_m v}\right]^{\frac{1}{2}}$ Metric units	18.1(b)

Particular	Equation	Eqn. No.
Diameter of piston Rod:		
The diameter of piston rod	$d = D\sqrt{p/\sigma_d}$	18.11

where p = unbalance pressure or difference between the steam inlet and the exhaust pressure, MN/m^2 (kgf/mm^2)

σ_d = allowable stress in the piston rod, (in calculating σ_d, take the factor of safety 10 for double acting engines and 8 for single acting engines on the ultimate strength)

The inertia force of the reciprocating parts on the piston, N (kgf)	$F = 0.004032rn^2 W_R\left(\cos\theta \pm \dfrac{\cos 2\theta}{n_1}\right)$	18.12

where r = crank radius, mm; W_R is the weight of reciprocating parts, N(kgf)

l = length of the connecting rod, mm; $n_1 = l/r$

θ = crank angle from the dead centre, positive sign is to be taken from inner dead centre while the negative sign from outer dead centre

Trunk Pistons:		
The thickness of piston head (Table 18.4)	$t_1 = 0.43D\sqrt{p/\sigma_d}$	18.13(a)

where p is the fluid pressure, MN/m^2 (kgf/mm^2); D is the diameter of piston, mm

σ_d = allowable tensile stress,

= 38 MN/m^2 (3.90 kgf/mm^2) for good close grained cast iron or aluminium alloys, with ultimate tensile strength of 137.0 MN/m^2 (14 kgf/mm^2)

= 55 MN/m^2 (5.6 kgf/mm^2) for Nickel Cast iron, semi-steel or special aluminium alloy having ultimate tensile strength of 206 MN/m^2 (21 kgf/mm^2)

= 82.5 MN/m^2 (8.4 kgf/mm^2) for forged steel

An empirical formula to determine the thickness of the cast iron automobile engine piston head, mm	$t_1 = 0.032D + 1.5\text{mm}$	18.13(b)
The thickness of the Crown from the consideration of heat flow, mm	$t_1 = \dfrac{D^2 q}{1600K(T_c - T_e)}$	18.13(c)

Particular	Equation	Eqn. No.
The diameter of the piston pin	$d = \dfrac{\pi D^2 p_{max}}{4 l_1 p_b}$	18.15(a)

where $l_1 = k_p d$, length of the gudgeon pin bearing, mm

$k_p = 1.5$ for petrol and gas engines and 2 for oil engines

p_b = bearing pressure

= 12.4 MN/m^2 (1.26 kgf/mm^2) for gas engines

= 15.0 MN/m^2 (1.54 kgf/mm^2) for oils engines

= 15.7 MN/m^2 (1.60 kgf/mm^2) for automotive engines

Another formula to determine the length of the gudgeon pin bearing	$l_1 = 0.45D$ to $0.5D$ if it oscillates in the connecting rod $= 0.62D$ if it oscillates in the piston bosses	18.15(b)

To check the strength of the piston pin:

The bending stress	$\sigma_b = \dfrac{F_p D}{18Z}$	18.16

where σ_b = bending stress

$\leq$ 82.0 MN/m^2 (8.4 kgf/mm^2) for case hardened carbon steel

$\leq$ 137.0 MN/m^2 (14.0 kgf/mm^2) for heat-treated alloy steel

F_p is the maximum load on the piston, N

Z is the section modulus, mm^3

Proportions of the Piston Rings:

The radial thickness of the cast iron snap ring	$t_r = D\sqrt{3p_r/\sigma}$	18.17(a)

where σ = allowable stress for Cast iron

= 82.0 to 110 MN/m^2 (8.4 to 11.2 kgf/mm^2)

p_r = magnitude of radial pressure on the piston rings, MN/m^2 (kgf/mm^2) (Table 18.9)

The depth 'h' of the piston ring	$h = 0.7t_r$ to t_r	18.17(b)
The minimum depth of the piston ring	$h = D/10i$ where i = number of piston rings	18.17(c)

The total depth of piston rings (according to Unwin)

$h_{total} = D/15 + 15.2$ mm for steam engines 18.17(d)

$= D/7 + 6$ mm for gas and oil engines; and $h_{total} = D/5.5$ for petrol engines

The distance from the top to the first groove	$t_g = t_1$ to $1.2t_1$	18.18
The lands between the ring grooves	$t_{land} = h$ or slightly less than h	18.19

Table 18.1 *Coefficients for Determining Head Thickness (Fig 18.5)*

Type of Head in Fig. 18.5	Coefficient	Remarks
A or A′...	0.162	Plate rigidly riveted or bolted to the shell flange.
B...	0.162	Integral flat head; $D_1 \leqq 600$ mm; $t \geqq 0.05D_1$.
C...	0.30	Flanged plate attached by a lap joint; $r \geqq 3t$.
D or E...	0.25	Plate butt-welded or forged integral; $r \geqq 3t_f$.
F...	0.50	Plate fusion-welded with fillet weld; throat $t_1 \geqq 1.25t_s$.
G or H...	$0.30 + K$	Bolts tend to dish the plate; K is found by the relation $K = 1.4Wt_o/HD_1$, where W = total bolt load in N; H = total pressure on area bounded by the outside diameter of the gasket, in pounds; and h_0 and d are as shown in Fig. 18.5.

Table 18.2 *Design Stresses for Bolted Flanged Heads*

Maximum Temperature (Deg C)	Minimum of Specified Range of Tensile Strength of Flange Material at Room Temperature N/mm² (kgf/mm²)					Alloy Bolt Steel N/mm²
	310	345	380	415	485	
370	74(7.55)	82(8.36)	90(9.18)	115(11.73)	98(10.00)	
400	65(6.63)	72(7.34)	80(8.16)	87(8.87)	101(10.30)	87(8.87)
425	56(5.71)	62(6.32)	68(6.93)	75(7.65)	87(8.87)	75(7.65)
450	47(4.80)	52(5.30)	57(5.81)	62(6.32)	73(7.44)	62(6.32)
480	37(3.77)	42(4.28)	46(4.70)	50(5.10)	58(5.91)	50(5.10)
510	28(2.86)	32(3.26)	34(3.47)	37(3.77)	44(4.50)	37(3.77)

Table 18.3 *Coefficients in Formulas for Cover Plates*

Material of Cover Plate	Method of Holding Edges	Circular Plate		Rectangular Plate		Elliptical Plate
		k_1	k_2	k_3	k_4	k_5
Cast iron...	Supported, free	0.54	0.038	0.75	1.73	1.5
	Fixed	0.44	0.010	0.62	1.4; 1.6*	1.2
Mild steel...	Supported, free	0.42	...	0.60	1.38	1.2
	Fixed	0.35	...	0.49	1.12; 1.28	0.9

* With gasket.

Table 18.4 *Thickness of Piston Head*

Type of Engine	Piston Material	Four Stroke	Two Stroke
Compression-ignition oil engines	Cast iron	0.11D – 0.13D	0.16D – 0.18D
-do-	Aluminium	0.13D – 0.16D	0.17D – 0.20D
Spark ignition gas engines	Cast iron	0.12D – 0.14D	0.20D – 0.23D

Table 18.5 *Recommended Piston Speed*

Class of Engine	Speed of Piston in m/s
Ordinary direct-acting pumping engines	0.45 – 0.65
Ordinary horizontal engines	1.00 – 2.00
Horizontal compound and Triple-expansion engines	2.00 – 4.00
Ordinary marine engines	2.00 – 3.25
Locomotive engines (mail)	4.00 – 5.00
Internal combustion engine	3.25 – 18.00

Table 18.6 *Stroke-Bore Ratio*

Class of Engine	Ration, L_s/D
Ordinary horizontal mill engines	1.50 to 2.00
Vertical quick-running engines	1.25 to 1.60
Locomotive engines	1.20 to 1.55
Internal combustion engines	0.90 to 1.90
Air-cooled air-craft engines	1.00

Table 18.7 *Values of Inertia Factor* ($\cos\theta + \cos 2\theta/n_1$)

Crank Angle θ from Top Dead Centre	Ratio of connecting rod length to crank length, l/r					
	3.75	4.0	4.25	4.50	4.75	5.30
0	1.267	1.253	1.235	1.222	1.210	1.200
10	1.236	1.219	1.206	1.194	1.183	1.175
20	1.144	1.131	1.120	1.110	1.101	1.093
30	0.999	0.991	0.984	0.977	0.971	0.966
40	0.812	0.809	0.807	0.804	0.803	0.801
50	0.597	0.599	0.602	0.604	0.606	0.608
60	0.367	0.375	0.382	0.389	0.395	0.400
70	0.138	0.150	0.162	0.172	0.181	0.189
80	−0.077	−0.061	−0.047	−0.350	−0.024	−0.014
90	−0.267	−0.250	−0.235	−0.222	−0.210	−0.200
100	−0.425	−0.408	−0.395	−0.382	−0.372	−0.361
110	−0.546	−0.543	−0.522	−0.512	−0.512	−0.495
120	−0.633	−0.625	−0.618	−0.611	−0.605	−0.660
130	−0.689	−0.686	−0.684	−0.681	−0.679	−0.677
140	−0.723	−0.723	−0.725	−0.727	−0.729	−0.731
150	−0.733	−0.741	−0.748	−0.755	−0.761	−0.766
160	−0.733	−0.748	−0.760	−0.769	−0.779	−0.786
170	−0.733	−0.750	−0.764	−0.776	−0.787	−0.797
180	−0.733	−0.750	−0.765	−0.778	−0.790	−0.800

Table 18.8 ***Weights of Reciprocating Parts***

Types of Engine		Weight W/A, MN/m²* of Piston Area	
		Spark Ignition	Compression Ignition
Single Acting Engine			
Air-craft	..	$1.50 \times 10^{-3} - 2.75 \times 10^{-3}$	$2.06 \times 10^{-3} - 3.10 \times 10^{-3}$
Automobile	..	$2.06 \times 10^{-3} - 3.40 \times 10^{-3}$	–
Truck and tractor	..	$2.40 \times 10^{-3} - 5.15 \times 10^{-3}$	$2.40 \times 10^{-3} - 5.15 \times 10^{-3}$
Medium speed stationary	..	$10.30 \times 10^{-3} - 27.50 \times 10^{-3}$	$10.30 \times 10^{-3} - 30.90 \times 10^{-3}$

*Values in Metric units (kgf/mm² of piston area) can be obtained by dividing the given values with 9.8067

Table 18.9 ***Magnitude of Radial Pressure of the Piston Rings on the Cylinder Barrel (according to Unwin)***

Type of engine	Steam Engines			Gas or Oil engines	Petrol engines
	H.p.	I.P.	L.P.		
p_r MN/m²	0.02746	0.02060	0.01372	0.02402–0.03090	0.02746–0.03432
(kgf/mm²)	(0.0028)	(0.0021)	(0.0014)	(0.00245–0.00315)	(0.0028–0.0035)

References

[1] Bhattacharya, S. C., Basu Mallik, J. R., *"Machine Design"*, Basu Mallik & Co., Calcutta, 1966.

[2] Vallance, A., Doughtle, V. L., *"Design of Machine Members"*, 3rd edition, McGraw Hill Book Company, 1951.

3 Design of Shafts

Symbols	Description and Units
c	distance from neutral axis to outer most fiber, mm
C_m, C_t	the numerical combined shock and fatigue factors to be applied to the computed bending moment and torsional moment respectively (Table 3.1)
d	diameter of solid shaft, mm (Table 3.5)
d_i, d_0	inside and outside diameters of the hollow shaft respectively, mm
F_c	Force on the connecting rod, N (kgf)
F_p	Force on the piston, N (kgf)
F_r	radial force along the crank, N (kgf)
F_t	tangential force perpendicular to the crank, N (kgf)
J	polar moment of inertia of cross-sectional area about axis of rotation, mm^4
K	ratio of inside to outside diameter of hollow shaft
L	length of the shaft, mm
M	bending moment, N mm (kgf-mm)
n	speed of the shaft, rpm
P	power, kW
p_{bc}	allowable crank pin bearing pressure, MN/m^2 (kgf/mm^2) (Table 3.6)
p_{bs}	main bearing pressure, MN/m^2 (kgf/mm^2) (Table 3.6)
T	torsional moment, N mm (kgf-mm)
σ_b	stress due to bending, MN/m^2 (kgf/mm^2)
σ_d	design stress, MN/m^2 (kgf/mm^2) (Table 3.5(b))
σ_e	stress at the elastic limit, MN/m^2 (kgf/mm^2)
τ	torsional shear stress, MN/m^2 (kgf/mm^2)
τ_d	design shear stress, MN/m^2 (kgf/mm^2) (Table 3.5(b))
τ_e	shear stress at the elastic limit, MN/m^2 (kgf/mm^2)
ω	angular velocity, rad/s
θ	angle of twist, deg

Particular	Equation	Eqn. No.
Torsion of circular shafts: The maximum torsional shear stress due to torsional loading	$\tau = \frac{Tc}{J} = \frac{16T}{\pi d^3}$ for solid shafts $= \frac{16T}{\pi d_o^3}\left(\frac{1}{1-K^4}\right)$ for hollow shafts	3.1
The angular deformation	$\theta = \frac{TL}{JG}$ rad $= \frac{584TL}{Gd^4}$ deg., for solid shaft $= \frac{584TL}{G(d_o^4 - d_i^4)}$ deg., for hollow shafts	3.2
The torque to be transmitted by the shaft N-mm	$T = \frac{10^6 P}{\omega}$ $T = \frac{9.55 \times 10^6 (P)}{n}$ SI Units	3.3(a)
The torque transmitted by the shaft, kgf mm	$T = \frac{75 \times 10^3 (MHP)}{\omega}$ $= \frac{7.16 \times 10^5 (MHP)}{n}$ $= \frac{9.74 \times 10^5 (P)}{n}$ Metric Units	3.3(b)
Hollow shaft:*		
The outside diameter of hollow shaft subjected to simple torsion	$d_o = \left[\frac{16T}{\pi\tau_d}\left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}}$	3.4(a)
Out side diameter of hollow shaft subjected to simple bending	$d_o = \left[\frac{32M}{\pi\sigma_d}\left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}}$ where $\tau_d = \frac{\tau_e B}{RK_s}$ and $\sigma_d = \frac{\sigma_e B}{RK_t}$	3.4(b)
The outside diameter of hollow shaft subjected to combined bending and torsion.	R is reliability factor Table 2.7 B is the size factor (Eqn. 2.9(b))	
a) According to maximum normal stress theory	$d_o = \left[\frac{16}{\pi\sigma_{\max}}\left(M + \sqrt{M^2 + T^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}}$	3.5(a)
b) According to maximum shear stress theory	$d_o = \left[\frac{16}{\pi\tau_{max}}\left(\sqrt{M^2 + T^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}}$	3.5(b)

*For solid shafts, substitute $K = 0$ and $d_0 = d$ in the above equaitons.

Particular	Equation	Eqn. No.

ASME Code for design of transmission shafting:

(a) According to maximum normal stress theory

$$d_o = \left[\frac{16}{\pi\sigma_{max}}\left(C_m M + \sqrt{(C_m M)^2 + (C_t T)^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.6(a)$$

(b) According to maximum shear stress theory

$$d_o = \left[\frac{16}{\pi\tau_{max}}\left(\sqrt{(C_m M)^2 + (C_t T)^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.6(b)$$

For fluctuating loads:

(a) According to maximum normal stress theory

$$d_o = \left[\frac{16}{\pi\sigma_{max}}\left(\left(M_m + \frac{K_t\sigma_{yp}}{\sigma_{en}}M_a\right) + \sqrt{\left(M_m + \frac{K_t\sigma_{yp}}{\sigma_{en}}M_a\right)^2 + \left(T_m + \frac{K_s\sigma_{yp}}{\sigma_{en}}T_a\right)^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.7(a)$$

(b) According to maximum shear stress theory

$$d_o = \left[\frac{16}{\pi\tau_{max}}\left(\sqrt{\left(M_m + \frac{K_t\sigma_{yp}}{\sigma_{en}}M_a\right)^2 + \left(T_m + \frac{K_s\sigma_{yp}}{\sigma_{en}}T_a\right)^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.7(b)$$

(c) According to Mises Hencky theory

$$d_o = \left[\frac{16}{\pi\sigma_{\max}}\left(\sqrt{4M_m^2 + 3T_m^2} + \frac{K_t\sigma_{yp}}{\sigma_{en}}\sqrt{4M_a^2 + 3T_a^2}\right) \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.7(c)$$

The out side diameter of hollow shaft subjected to an axial load F in addition to the torsional and bending loads.

(a) According to the maximum normal stress theory

$$d_o = \left[\frac{16}{\pi\sigma_{max}}\left\{\left(C_m M + \frac{\alpha F d_o(1+K^2)}{8}\right) + \sqrt{\left(C_m M + \frac{\alpha F d_o(1+K^2)}{8}\right)^2 + (C_t T)^2}\right\} \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.8(a)$$

(b) According to maximum shear stress theory

$$d_o = \left[\frac{16}{\pi\tau_{max}}\left\{\sqrt{\left(C_m M + \frac{\alpha F d_o(1+K^2)}{8}\right)^2 + (C_t T)^2}\right\} \times \left(\frac{1}{1-K^4}\right)\right]^{\frac{1}{3}} \qquad 3.8(b)$$

Particular	Equation	Eqn. No.

where α = ratio of the maximum intensity of stress resulting from the axial load, to the average axial stress from Table 3.2

$= \dfrac{1}{1 - 0.0044(L/k)}$ when $L/k < 115$

$= \dfrac{\sigma_{yp}}{n\pi^2 E}(L/k)^2$ when $L/k > 115$ (Euler's formula)

where L is the length between supporting bearings, mm

k is the radius of gyration of the shaft, mm

σ_{yp} is the yield stress in compression, MN/m^2 (kgf/mm^2)

E is the modulus of elasticity, MN/m^2 (kgf/mm^2)

n = constant for the type of column end support,

= 1.0, for free end supports or hinged ends

= 2.25, for fixed ends

= 1.60, for both ends pinned, guided and partly restrained

Particular	Equation	Eqn. No.
The relation between the diameter of a solid shaft to a hollow shaft if their torsional strengths are equal	$d_o^3 = \dfrac{d^3}{1 - K^4}$	3.9
Effect of keyways:		
H.F. Moore's formula for determining the shaft strength factor or the ratio of strength of shaft with keyway to the same shaft without keyway	$K_e = 1.0 - \dfrac{0.2b}{d} - \dfrac{1.1h}{d}$	3.10
Moore's formula for determining the ratio of the angular twist of the shaft with the key way to the same shaft without keyway	$K_e = 1.0 + 0.4\dfrac{b}{d} + 0.7\dfrac{h}{d}$ where b and h are width and depth of keyway, mm	3.11
Crank Shaft:		
Forces on crank arm: (Fig. 3.1)		
The force on the connecting rod	$F_c = \dfrac{F_p}{\cos\phi}$	3.12(a)
The tangential force or the rotative effort on the crank	$F_t = F_c \sin(\theta + \phi) = F_p \dfrac{\sin(\theta + \phi)}{\cos\phi}$	3.12(b)
The radial force along the crank	$F_r = F_c \cos(\theta + \phi) = F_p \dfrac{\cos(\theta + \phi)}{\cos\phi}$	3.12(c)
The diameter of the crank pin at the root (Fig. 3.2)	$d_{p1} = \sqrt[4]{\dfrac{16F_c^2}{\pi p_{bc}\sigma_b}}$	3.13(a)
The diameter of the crank pin or journal	$d_p = (d_{p1})$ to $(d_{p1} + 6.5\text{mm})$	3.13(b)

Particular	Equation	Eqn. No.
The length of the crank pin	$l_p = \dfrac{F_c}{d_{p1} p_{bc}}$	3.14
The diameter of the crank shaft at the root	$d_s = \sqrt[3]{\dfrac{16F_c r}{\pi \tau_d}}$	3.15(a)
The diameter of the crank shaft or journal	$(d_{s1}) = d_s$ to $(d_s + 6.5\text{mm})$	3.15(b)
The length of the main bearing	$l_s = \dfrac{1.2F_t}{d_s p_{bs}}$	3.15(c)
The diameter of the crank pin according to American Bureau of shipping method (Fig. 3.3)	$d_p = a\sqrt[3]{\dfrac{D^2 pc}{\sigma_d}}$	3.16

where a is the coefficient from Table 3.8
D is the diameter of cylinder bore, mm
p is the maximum gas pressure, MN/m^2 (kgf/mm^2)
c is the distance over the crank web plus 25 mm (Fig. 3.3)
σ_d is the allowable stress, MN/m^2 (kgf/mm^2)

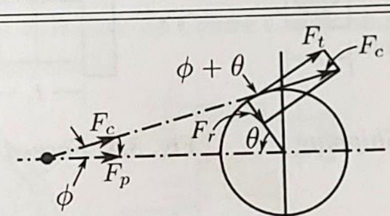


Fig. 3.1: ***Forces on crank arm***

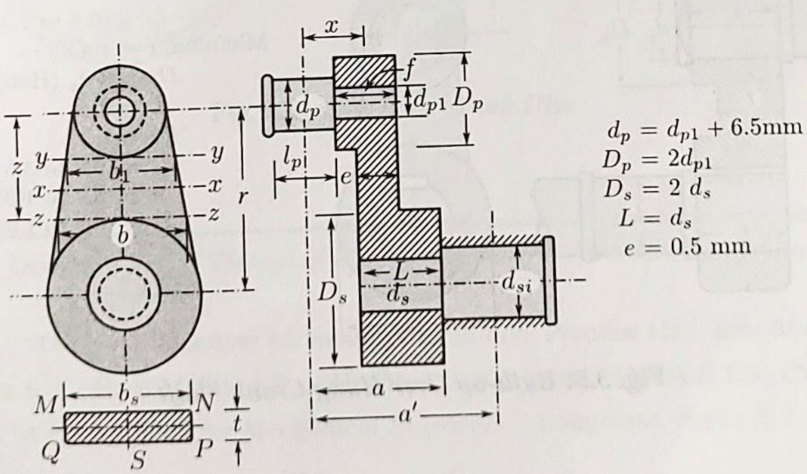


Fig. 3.2: ***Typical Built-up Over Hung Crank Shaft***

Particular	Equation	Eqn. No.
Torsion of rectangular bars:		
Shear stress at the point A_1 (Fig. 3.4)	$T_1 = \dfrac{T}{\alpha_1 b c^2}$	3.17(a)
Shear stress at the point A_2 (Fig. 3.4)	$T_2 = \dfrac{T}{\alpha_2 b c^2}$	3.17(b)
The angle of twist, θ radians per mm of length where α_1, α_2 and β are constants, from Table 3.7	$\theta = \dfrac{T}{\beta G b c^3}$	3.17(c)

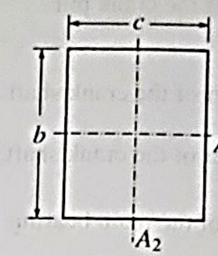


Fig. 3.4

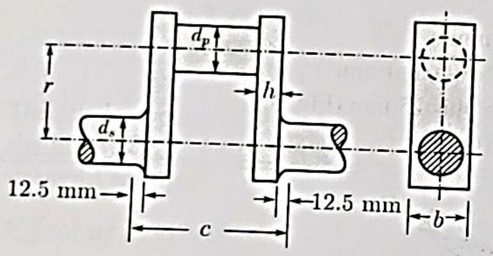
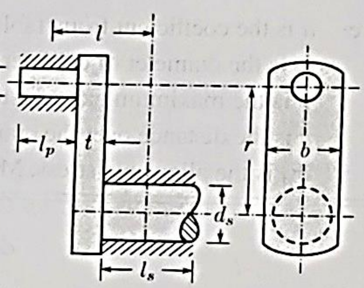
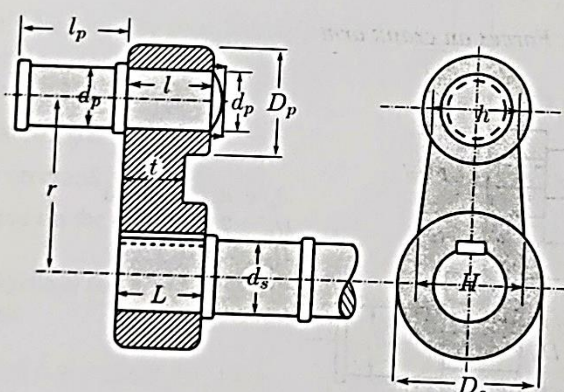


Fig. 3.3: ***American Bureau of Shipping Method***



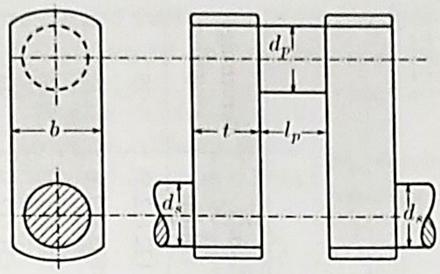
$l_p = (0.8 \text{ to } 1.1)\ d_p$
$t = (0.5 \text{ to } 0.9)\ d_p$
$b = (1.1 \text{ to } 1.2)\ d_s$
$l_s = (2 \text{ to } 2.5)\ d_s$

Fig. 3.4: ***Forged Over Hung Crank Shaft***



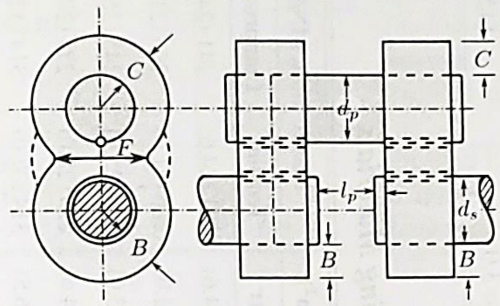
$l_p = (0.8 \text{ to } 1.3)\ d_p$
Minimum $t = 1.4\ d_p$
$D_p = 2\ d_p$ (Hub Diameter)
$L = d_s$
$D_s = 2d_s$
$l_s = (0.7 \text{ to } 3.0)\ d_s$
$h = (0.7 \text{ to } 1.0)\ D_p$
$H = (0.7 \text{ to } 1.0)\ D_s$

Fig. 3.5: ***Built-up Over Hung Crank Shaft***



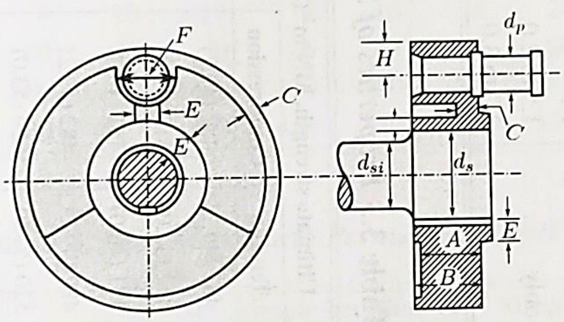
$l_p = d_p$
$t = (0.6 \text{ to } 0.75)\ d_p$
$b = (1.1 \text{ to } 1.2)\ d_p$
$d_s = d_p$

Fig. 3.6: ***Forged Centre Crank***



$d_p = d_s$
$l_p = (0.95 \text{ to } 1.1)\ d_p$
$t = 0.75\ d_p$
$B = 0.43\ d_p$
$C = 0.40\ d_p$
$F = 1.30\ d_p$
FILLET RADIUS $r \geqslant 0.2 d_p$

Fig. 3.7: ***Buit-up Centre Crank Shaft***



$A = 0.9\ d_{si}$
$B = 0.7\ d_{si} \text{ to } d_{si}$
$C = 0.25\ d_{si}$
$d_s = d_{si} \text{ to } 1.2\ d_{si}$
$E = 0.4\ d_{si}$
$F = 2\ d_p$
$H = d_p \text{ to } 1.5\ d_p$
$L = d_{si} \text{ to } 1.25\ d_{si}$

Fig. 3.8: ***Cast Iron Crank Disc***

References

[1] Vallance A., Doughtie V. L., *"Design of Machine Members"*, 3rd edition, McGraw Hill Book Co. 1951.

[2] Spotts M. E., *"Design of Machine Elements"*, 3rd edition, Prentice Hall, Inc., Maruzen Co., Ltd.

[3] Bhattacharya S. C., Basu Mallik, J.R., *"Machine Design"*, basu Mallik & Co., Calcutta, 1966

[4] Low D. A., *"A Pocket Book for Mechanical Engineers"*, Longmans, Freen & Co., Ltd., London, 1955.

[5] Siegel M. J., Maleev V. L., Hartman J. B., *"Mechanical Design of Machines"*, 4th edition, International Text Book Company, 1965.

Table 3.1 *Constants for ASME Code (Shock and Endurance Factors)*

Nature of Loading	Value for C_m	Value for C_t
Stationary Shafts :		
Gradually applied load	1.0	1.0
Suddenly applied load	1.5 to 2.0	1.5 to 2.0
Rotating Shafts :		
Steady or gradually applied loads	1.5	1.0
Suddenly applied loads, minor shocks only	1.5 to 2.0	1.0 to 1.5
Suddenly applied loads, heavy shocks	2.0 to 3.0	1.5 to 3.0

Table 3.2 *Value of α to use in Equation 3.8(a) and 3.8(b)*

Slenderness ratio (L/k)	Factor, α
0	1.00
25	1.12
50	1.28
75	1.49
100	1.78
115	2.02

Table 3.3 *Properties of Shafting Materials*

Material	Percentage carbon	Ultimate strength, MN/m² (kgf/mm²)			Elastic limit, MN/m² (kgf/mm²)			Percentage Elongation
		Tension	Compression	Shear	Tension	Compression	Shear	
Commercial Cold rolled ...	0.10-0.25	482 (49.2)	482 (49.2)	241 (24.6)	241 (24.6)	241 (24.6)	122 (12.5)	35
Commercial turned ...	0.10-0.25	412 (42.0)	412 (42.0)	206 (21.0)	206 (21.0)	206 (21.0)	103 (10.5)	35
Hot rolled or forged ...	0.15-0.25	451 (46.0)	451 (46.0)	225 (23.0)	245 (25.0)	245 (25.0)	113 (11.5)	26
	0.25-0.35	482 (49.2)	482 (49.2)	241 (24.6)	275 (28.0)	275 (28.0)	121 (12.3)	24
	0.35-0.45	520 (53.0)	520 (53.0)	260 (26.5)	314 (32.0)	314 (32.0)	130 (13.2)	22
	0.45-0.55	553 (56.4)	553 (56.4)	276 (28.2)	345 (35.2)	345 (35.2)	138 (14.1)	20
$3\frac{1}{2}$% Nickel ...	0.15-0.25	588 (60.0)	588 (60.0)	294 (30.0)	392 (39.0)	382 (39.0)	147 (15.0)	26
Chrome Vandium ...	0.25-0.35	620 (63.2)	620 (63.2)	310 (31.6)	414 (42.2)	414 (42.2)	155 (15.8)	25

Table 3.4 ***Fatigue Stress concentration Factors in bending for shafts with keyways based on section modulus of full area***

Steel	Tensile strength, MN/m² (kgf/mm²)	Yield strength (Plastic deformation 0.2%), MN/m² (kgf/mm²)	For reversed bending stress	Chrome Nikel Heat treated σ_{en} MN/m² (kgf/mm²)	Chrome Nikel Heat treated K_{tf}	Medium Carbon normalised σ_{en} MN/m² (kgf/mm²)	Medium Carbon normalised K_{tf}
Carbon Nickel (about SAE 3 140)	714.0 (72.8)	482.5 (49.2)	No Keyway ordinary tapered specimen	400.0 (40.0)	..	255.0 (26.0)	..
	–	–	Sledrunner keyway	248.0 (25.3)	1.61	193.0 (19.7)	1.32
	–	–	Profiled keyway	193.0 (19.7)	2.07	159.0 (16.2)	1.61
Medium Carbon (about SAE 1045)	552.0 (56.3)	310.0 (31.6)	Profiled Keyway 6.25mm transverse hole	–	–	83.0 (8.5)	3.06

Table 3.5 ***(a) Standard shaft sizes in mm***

6	8	10	12	14	16	18	20	22	25	28	32
36	40	45	50	56	63	71	80	90	100	110	125
140	160	180	200	220	240	260	280	300	320	340	360
380	400	420	440	450	480	500	530	560	600	–	–

Table 3.5 ***(b) Maximum Permissible Working Stresses for Shafts***

Grade of shafting	Simple bending MN/m² (kgf/mm²)	Simple torsion MN/m² (kgf/mm²)	Combined stress MN/m² (kgf/mm²)
"Commercial Steel" shafting without allowance for keyways	110 (11.2)	55(5.6)	55(5.6)
"Commercial Steel" shafting with allowance for keyways	83 (8.5)	41 (4.2)	41(4.2)
Steel purchased under definite specification (without keyways)*	60% of the elastic limit but not over 36% of the ultimate in tension.	30% of the elastic limit but not over 18% of the ultimate in tension.	30% of the elastic limit but not over 18% of the ultimate in tension.

*The values are to be reduced by 25% if keyways are present.

Table 3.6 *Allowable bearing pressure*

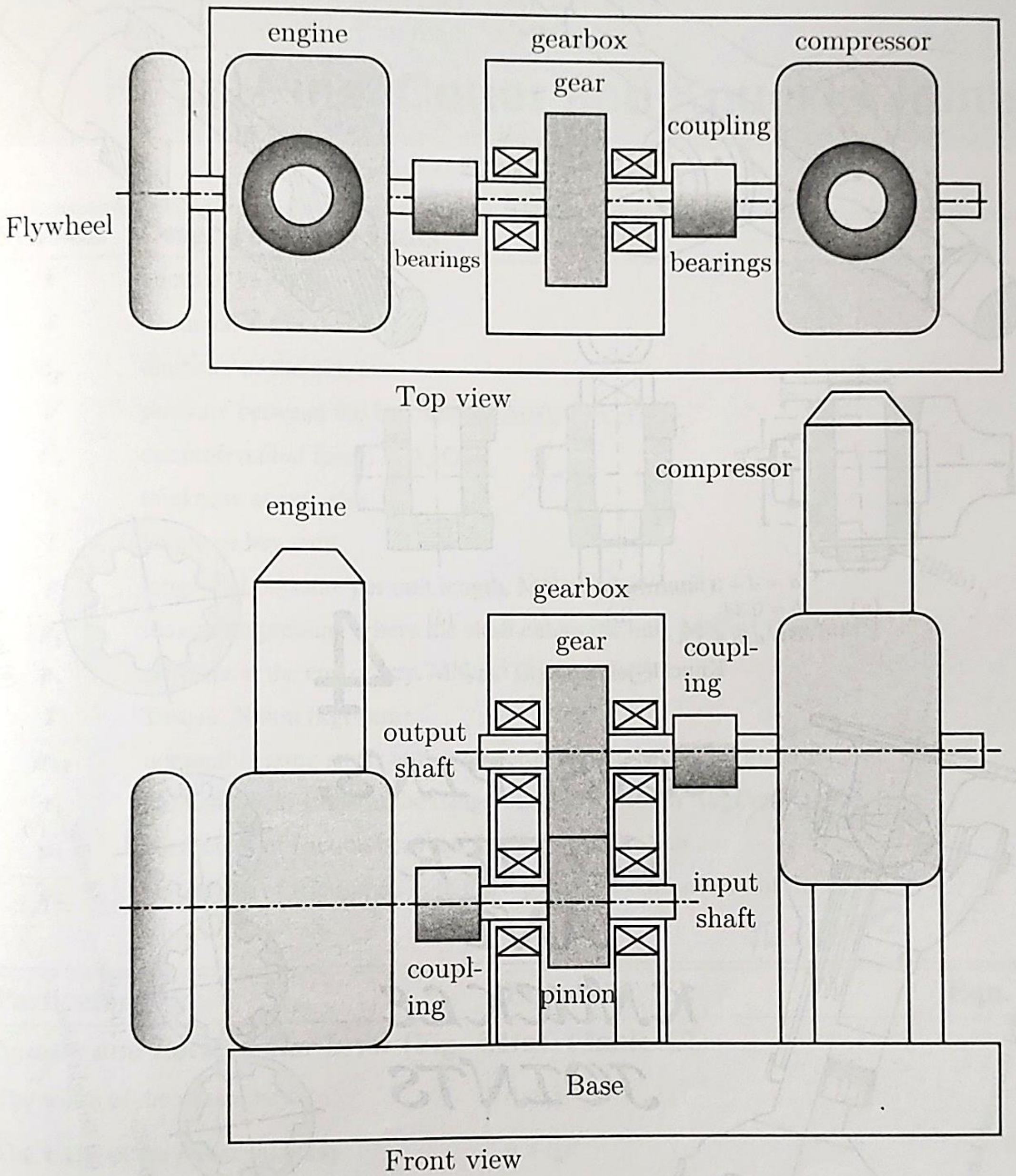
Class of work	Main bearing pressure p_{bs}, MN/m^2 (kgf/mm^2)	Crank pin pressure p_{bc}, MN/m^2 (kgf/mm^2)
Automobile Engines	10.0—13.5 (1.0-1.40)	2.5—2.75 (0.25-0.28)
Diesel Engines	5.5—7.0 (0.56-0.70)	7.0—8.5 (0.70-0.85)
Marine Diesel Engines	2.75—3.5 (0.28-0.35)	7.0—10.0 (0.70-1.00)
Rail road, locomotive	1.2—1.4 (0.12-0.14)	10.0—12.0 (1.05-1.20)
Shear and punches, slow speed	20.5—27.5 (2.10-2.80)	34.0—54.0 (3.50-5.50)

Table 3.7 *Constant for torsion of rectangular bars*

b/c	1.00	1.20	1.50	1.75	2.00	2.50	3.00
α_1	0.208	0.210	0.231	0.239	0.246	0.258	0.267
α_2	0.208	0.235	0.269	0.291	0.309	0.336	0.355
β	0.141	0.116	0.196	0.214	0.229	0.249	0.263
b/c	4.00	5.00	6.00	8.00	10.00	∞	–
α_1	0.282	0.291	0.299	0.307	0.312	0.333	–
α_2	0.378	0.392	0.402	0.414	0.421	–	–
β	0.281	0.291	0.299	0.307	0.312	0.333	–

Table 3.8 *Coefficient 'a' in the American Bureau of Shipping Formula 3.16*

Type	Number of *cylinder*		Ratio of stroke to distance over crank webs = L/c							
	Four stroke	Two stroke	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
Explosion engines	1,2,4	1,2	1.17	1.17	1.17	1.17	1.17	1.17	1.17	1.17
	3,5,6	3	1.17	1.17	1.17	1.17	1.19	1.20	1.22	1.24
	8	4	1.17	1.19	1.21	1.23	1.25	1.28	1.30	1.32
	10,11,12	5,6	1.18	1.20	1.23	1.25	1.28	1.31	1.33	1.35
Air-injection diesel engines	1,2,4	1,2	1.17	1.19	1.22	1.25	1.28	1.31	1.34	1.36
	3,5,6,	3	1.19	1.22	1.25	1.28	1.32	1.35	1.38	1.41
	8	4	1.20	1.24	1.27	1.30	1.33	1.37	1.40	1.43
	12	5,6	1.22	1.25	1.29	1.32	1.36	1.39	1.42	1.45
	16	8	1.25	1.29	1.33	1.36	1.40	1.44	1.47	1.50



Schematic diagram of Gasoline-Engine-Powered Portable Air Compressor, Gearbox, Couplings, Shafts and Bearings.

Note: Excerpted from Norton, Machine Design Pearson Education (Singapore) Pvt. Ltd., Delhi.