DESIGN OF CRANKSHAFT

PROJECT REPORT

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CERTIFICATE

This is to certify that the report entitled

" DESIGN OF CRANKSHAFT "

has been submitted by

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in partial fulfilment for the award of Bachelor of Engineering in the **MECHANICAL ENGINEERING** branch of the BHARATHIAR UNIVERSITY, COIMBATORE during the academic year 1989 – 90.

Guide **Head of the Dept.**

Certified that the candidate was examined by us in the Project work viva – voce examination held on ___________ and the university register number was___________

Internal Examiner **External Examiner**

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION:

The factors which are of importance in design of a crankshaft are

a) Strength and stiffness to resist bending and twisting moments.

b) Adequate projected area of journals and crankpins to give reasonable bearing pressures.

c) Stiffness to minimise and strength to resist stresses due to torsional vibrations of the cranckshaft.

Nowadays the main problem in the fields of development and improvement of motor vehicle and tractor engines are concerned with wider use of diesel engines, reducing fuel consumption and weight per horse power of the engines and cutting down the costs of their product on and service. The engine pollution control, we sell as the engine noise contol in service have been raised to a new level. For more emphasis is given to the use of computers in designing and testing engines. Ways have been outlined to utilize computers directly in the design and analysis of engine parts.

The computation of engine parts with a ivew to determining stresses and strains occuring in an operating engine are performed by formulas dealing with strength of materials and machine parts.

Forces caused by gas pressure in the cylinders and inertia of reciprocating and rotating masses and also loading produced by elastic vibrations and heat stresses are the main loads on the engine parts.

The loading caused by gas pressure continuously varies during the working cycle and reaches its maximum within a comparatively small portion of the piston stroke. Loading due to inertia forces varies periodically and sometimes reaches in highspped engines the values exceeding the load due to gas pressure. The above loads are sources of various eleastic osciallations dangerous during resonance.

The crankshaft is a most complicated and strained engine part subjected to cyclic loads due to gas pressure, inertia forces and their couples. The effect of these forces and their moments cause considerable stresses of torsion, bending and tension-compression in the crankshaft material. Apart from this, periodically varying moments cause torsional vibration of the shaft with resultant additi;onal torsional stresses.

Therefore, for the most complicated and severe operating conditions of the crankshaft, high and diverse requirements are imposed on the materials utilized for manufacturing crankshafts. The crankshaft material has to feature high strength and toughness, high resistance to wear and fatigue stresses, resistance to impact loads and hardness. Such properties are possessed by propely machined carbon and alloyed steels and also high duty cast iron.

The intricate shape of crankshaft, a variety of forces and moments loading it, changes in which are dependent on the rigidity of the crankshaft and its bearings, and some other causes do not allow the crankshaft strength to be computed precisely. In view of this, various approximate methods are used which allow us to obtain conventional stresses and safety factors for individual elements fo a crankshaft.

When designing a crankshaft, we assume that : the cranks are freely supported as a simply supported beam the supports and force points are in the centre planes of the crankpins and journals the entire span between supports represents an ideally rigid beam.

1.2 COMPUTER AIDED DESIGN:

Computer Aided Design (CAD) can be defined as the use of computer system to assist in the creation, modification, analysis or optimization of a design. The computer systems consists of the hardware and software to perform the specialised design functions required by the particular user firm. The CAD hardware typically includes the computer, one or more graphics display terminals, keyboards and other peripheral equipment. Computer graphics on the system plus application programmes to facilitate the engineering functions of the user company. Expamples of these application programme include stress-strain analysis, dynamic response, heat transfer calculation and Numerical control part programming.

1.2.1 DESIGN PROCESS:

The various design related tasks which are performed by a modern computer aided design system can be grouped into four areas namely:

1. Geometric modelling
2. Engineering Analysis
3. Design Review and evaluation
4. Automated drafting

1.2.2. BENEFITS OF COMPUTER AIDED DESIGN:

There are many benefits of computer aided design, only some of which can be easily measured. Some of the benefits are intangible, reflected in improved work quality, more pertaint and usable information and improved control, all of which are difficult to quantify.

Increased productivity translates into a more competitive position for the firm because it will reduce staff requirements on a given project. This leads to lower costs in addition to improving response time on project with tight schedules. Productivity imporvement in computer aided design as compared to the traditional design process is dependent on such factors as

1. Complexity of the engineering drawing.
2. Level of detail required in the drawing.
3. Degree of repetitiveness in the designed parts.
4. Degree of symmetry in the parts.
5. Extensiveness of library of commonly used entitles.

As each of these factors is increased, the productivity advantage of CAD will tend to increase.

CHAPTER 2

DESIGN OF CRANKSHAFTS

2.1 KINEMATICS

The piston travel, mm

$$S = R\,(1 - \cos \Phi) + \lambda/4\,(1 - \mathrm{Cost}\ \Phi) \qquad (2.1)$$

The angular velocity of crankshaft revolution, rad/Sec.

$$W = \pi N/30 \qquad (2.2)$$

The piston Speed

$$V_P = WR\,(\mathrm{Cost}\ \Phi + \mathrm{cist}\ 2\Phi) \qquad (2.3)$$

2.2 DYNAMICS

2.2.1 GAS PRESSURE FORCE

$$\text{Gas pressue force } P, = \frac{\pi d_{cy}^{2} \times Pex}{4} \qquad (2.4)$$

2.2.2 EXTERNAL FORCES ACTING ON SINGLE THROW - CRANKSHAFT:

The gas and mass forces active on a single throw as well as the reaction forces active in the bearing can be illustrated in Figure 2.1. They result in bearing and torsional moments in the web. As the individual throw is a part of an array of throws, for a multicylinder engine reaction forces from throws active in cross planes in the middle of the main journal must also be considered.

In Figure 2.1.

A, E	=	Centre line of main jouranls
B, D	-	Centre line ow web
C	-	Centre line of crankpin
P_2	-	Gas force Newton
P_r	-	Radial component of P_2 Newton
P_t	-	Tangential component of P_2 Newton
h_r	-	Bearing reactions in the horizontal direction
h_v	-	Bearing reacation in the vertical direction
$\emptyset_v$	-	Crank angle

Force on the piston due to torque pressure

$$P_2 = (\pi \times d_{cy}^2 \times p_t)/(4 \times \cos(\Phi)) \text{ Newton} \qquad (2.5)$$

Tangential component of the force P

$$P_t = P_2 \times \sin(\theta+\emptyset) \text{ Newton} \qquad (2.6)$$

Radial component of the force P

$$P_r = P_2 \times \cos(\theta+\emptyset) \text{ Newton} \qquad (2.7)$$

Reaction due to radial force

$$rr1 = P_r/2 \qquad (2.8)$$

Horizontal component of radial reaction

$$rr1h = rr1 \sin \Phi \qquad (2.9)$$

Vertical component of radial force

$$rr1v = rr1 \cos \Phi \qquad (2.10)$$

Reaction due to tangential force

$$r_t = P_t/2 \qquad (2.11)$$

Horizontal component of tangential force

$$rth = rt \sin \Phi \qquad (2.12)$$

Vertical component of tangential force

$$rtv = rt \times \text{Cos } \Phi \text{ Newton} \qquad (2.13)$$

Vertical force due to weight of flywheel

$$V_2 = S_1 \times wfw/SS \text{ Newton} \qquad (2.14)$$

Resultant vertical reacation of flywheel

$$V_3 = wfw - V_2 \text{ Newton} \qquad (2.15)$$

Torque transmitted

$$\text{Torque} = hp \times 71620/N \text{ Newton mm} \qquad (2.16)$$

Tension in the pulley belt due to this torque

$$t_2 = 2 \times (\text{torq}) / (d_p \times e^{u\Phi}-1) \qquad (2.17)$$

$$t_1 = t_2 e^{u\Phi} \qquad (2.18)$$

Total force due to the tension in the Pulley

$$h_2 = t_1 + t_2 \times {}^{S}pul/SS \text{ N} \qquad (2.19)$$

Resultant force in the Pulley

$$h_3 = t_1 + t_2 - h_2 \qquad (2.20)$$

Vertical Reaction in the journal Bearing

$$vrb = v_2 + rr1v - rtv \ N \tag{2.21}$$

Horizontal reaction in the journal bearing

$$hrb = h_2 + rr1h + rth \ N \tag{2.22}$$

Resultant reaction in the Journal bearing

$$R_j = vrb^2 + hrb^2 \tag{2.23}$$

2.2.3 CRANKSHAFT REACTION

The complex overall load of a single throw of crankshaft can be reduced into the following partial load as shown in figure 2.2.

1. Radial force
2. Tangential force
3. Torsional moment
4. Bending moment
5. Shear force
6. Normal force

In Fig. 2.2.

N1, N2	-	Axial force
Q1, Q3	-	Shear force normal to crank plane
Q2, Q4	-	Shear force in the crank plane
MB1, MB3	-	B.M. normal to crank plane
MB2, MB4	-	B.M in the crank plane
MT1, MT2	-	Torsional moments.

2.3 DESIGN PROCEDURE:

1. Load on the crankpin for maximum explosion pressure is determined considering the crank as a simple supported beam supported at the bearings, maximum value of Bending Moment Computed.

Load due to explosion pressure P_{ex}

$$P_1 = \pi \times d_{cy}^2 \times P_{ex} / 4 \text{ Newton} \qquad (2.24)$$

Bending moment

$$bm_1 = (P_1 \times S)/4 \text{ Newton mm} \qquad (2.25)$$

The dia of crankpin

$$d_o \quad \sqrt[3]{\frac{32 \times bm_1 \times fos}{\pi \times \sigma y}} \text{ mm} \qquad (2.26)$$

2. Similary, for the maximum torque pressure, in the angle at which maximum torque occurs. maximum load (equation 2.4) and hence maximum bending moment computed

$$bm_2 = \frac{P_2 \times S}{4} \text{ N mm} \qquad (2.27)$$

dia of crankpin

$$d_o = \sqrt[3]{\frac{32 \times bm_2 \times fos}{\pi \times \sigma y}} \text{ mm} \qquad (2.28)$$

3. Choose bigger one of the value calculated above.

4. Length of crankpin generally varies between 1.25 to 1.5 times crankpin diameter. Hence fix the length of crankpin.

5. Thickness of crankweb generally varies between 0.45 to 0.75 times the diameter of crankpin. Width of crankweb generally varies between 1.1 to 1.2 times the diameter of crankpin. So calculate the width and thickness of crankweb.

6. The crank shaft webs are loaded by complex alternating stresses.

   a. direct stress due to the radial force
   b. bending stress due to radial force and
   c. shear stress due to tangential force

   Bending stress due to radial force

$$bst = \frac{rr1 \times (1_{3} + t)\ 3}{(w \times t^{2})} \qquad (2.29)$$

direct stress

$$dst = rr1/wt \qquad (2.30)$$

$$dst = \frac{rr1}{wt}\ \text{N/sq.mm} \qquad (2.31)$$

Bending stress due to tangential force

$$bst = \frac{P_t \times R \times 1.5}{wt^2} \quad (2.32)$$

total stress,

$$tst = bst + dst + bst1 \quad (2.33)$$

Tosional moment

$$tmt = \frac{P_t (1_o + t)}{4} \quad (2.34)$$

Therefore, shear stress

$$sst = \frac{tmt \times 4.5}{wt^2} \quad (2.35)$$

Combined stress

$$Comst = \left(\frac{tst^2}{2}\right) + \sqrt{\left(\frac{tst^2}{2}\right) + sst^2} \quad (2.36)$$

The combined stress is checked to be within allowable stress.

7. Journal is subjected to bending moment due to radial and tangential force and twisting moment due to tangential force. From the twisting moment and bending moment, equivalent bending moment is calculated

$$M = P_2 \times \frac{S}{4} \quad (2.37)$$

$$P_t = P_2 = \sin (2 \Phi + \theta) \quad (2.38)$$

Twisting moment

$$T = P_t \times R \text{ N mm} \quad (2.39)$$

equivalent bending moment

$$M_3 = 0.5 (M_2 + \sqrt{M^2 + T^2}) \text{ N mm} \quad (2.40)$$

Then, diameter of journal

$$d_3 = \sqrt[3]{\frac{32 \times M \times fos}{\pi c \sigma y}} \text{ mm} \quad (2.41)$$

8. Length of journal fixed by space consideration

$$1_3 = S-1_1 - 2t \quad (2.42)$$

9. The value of unit area pressure on the working surface of a crankpin or a main bearing determines the conidtion under which the bearing operates and its service life in the long run. With the bearings in operation measures are taken to prevent the lubrication oil film from being squeexed out, damage to the white metal and premature wear of crankshaft journals and crankpins. The designed crankshaft journals and crankpins are checked for induced bearing stress. Highest allowable pressure for various engines are given below:(7)

Type of Engine	Kind of Bearing	Highest allowable Pressure N/sq. mm
Automotive and Aircraft Engine	Journal	5.50 to 12.00
	Crankpin	10.00 to 25.00
	Pistonpin	15.00 to 35.00
Gas and oil engine (4-stroke)	Journal	5.50 to 8.50
	Crankpin	10.00 to 15.00
	Pistonpin	12.50 to 17.00
Gas and oil engine (2-stroke)	Journal	3.50 to 5.50
	Crankpin	7.00 to 10.00
	Pistonpin	8.50 to 12.50

Bearing pressure in the journal

$$P_j = \frac{R_j}{d_3 \times l_3} \text{ N/sq.mm} \qquad (2.43)$$

Bearing pressure in the crankpin

$$P_c = \frac{P_2}{l_o \times d_o} \text{ N/sq.mm} \qquad (2.44)$$

10. Draw the designed crankshaft

11. If an existing crankshaft demensions has to be checked, check for combined stress in the web to be less than allowable stress, check for bearing stress in the crankpin and journal within limits and shear stress within allowable stress.

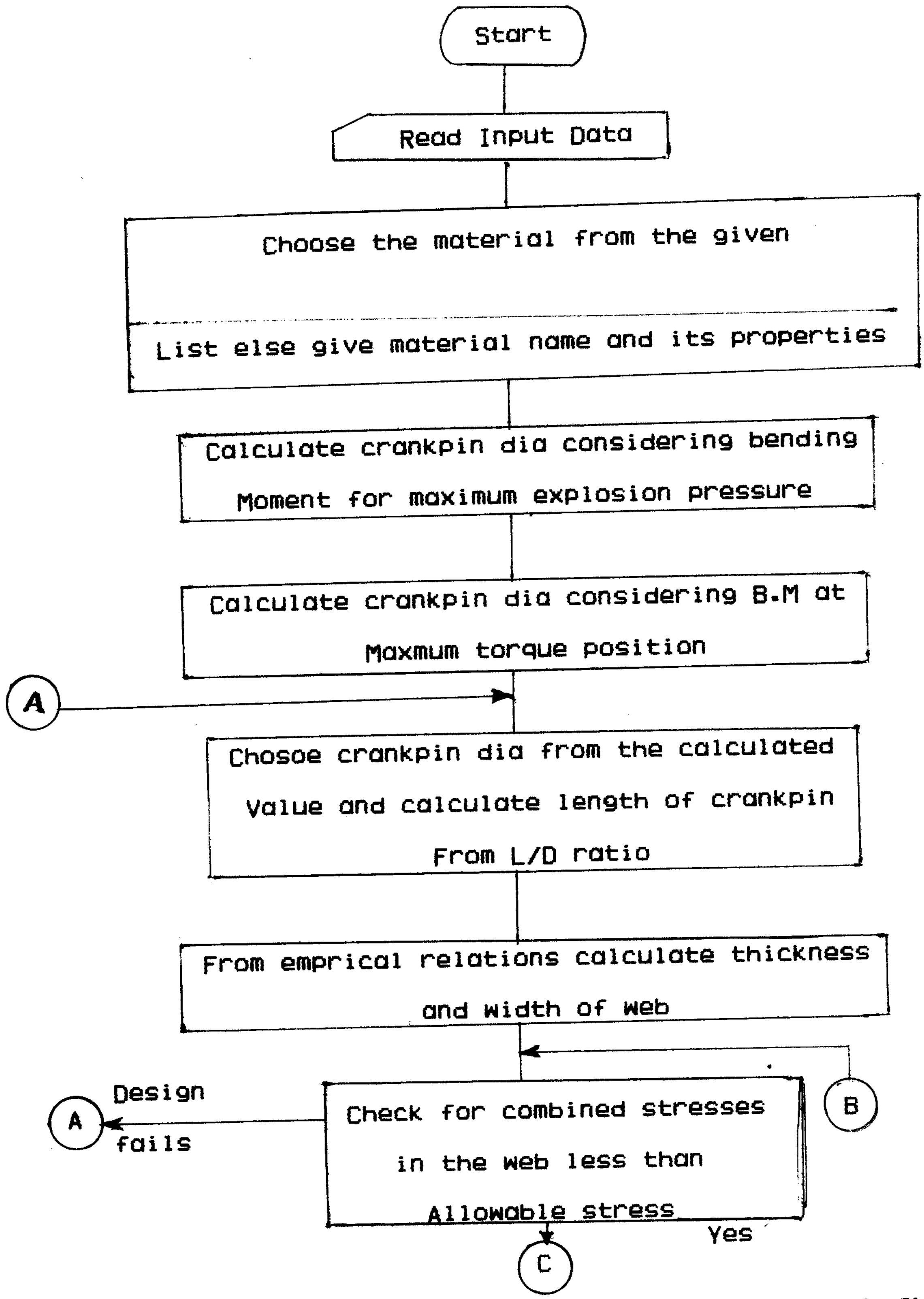
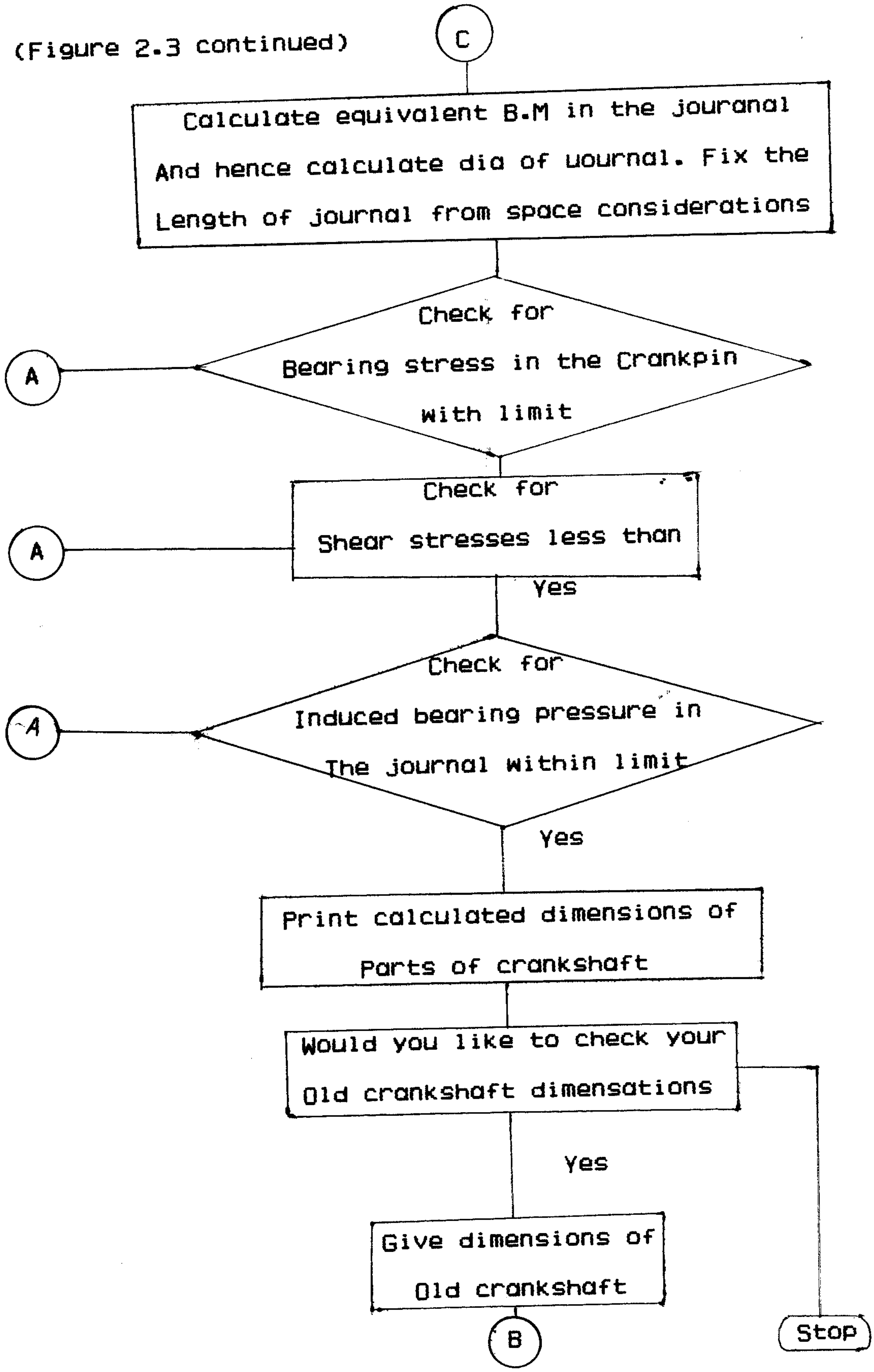


Figure 2.3 Flowchart for the design of crankshafts

(Figure 2.3 continued)



CHAPTER 3

STRENGTH OF CRANKSHAFTS AND METHODS OF IMPROVING STRUCTURAL DURABLITY

3.1 STRENGTH OF CRANKSHAFTS

The crankshaft is subjected to the following forces: bending moments due to gas pressure and inertial forces, torque due to the power delivered and torsional vibrations, and axial thrust due to unbalanced forces and couples. The presence of small flexural and axial vibrations increases the loading on the crankshaft in bending and compression. Consequently the continued reliable service of a crankshaft is dependent on its ability to withstand these loads without failure for long time. The repeated application of the stresses reduce the maximum stress that the shaft can carry without fatigue failure. Therefore we can define, the maximum bending fatigue stress the crankshaft can be subjected without fatigue failure s the bending fatigue strngth. Similarly the torsional fatigue strength and axial strength can be defined. In the case of torsional fatigue strength the operating speed and the critical speeds of the crankshaft system which give rise to torsional vibrations must be taken into account.

The production techniques which affect the structural durability in service are

a) casting or forging process adopted

b) machining operation and finish and

c) surface treatments like

i) cold rolling

ii) Surface hardening like carbursing or nitriding

iii) high frequency induction hardening and

iv) chill casting.

3.2 SHAPE OF THE CRANKSHAFT

The shape of the crankshaft is dependent upon three requirements namely (a) the length of the shaft must remain within the limits laid down by the cylinder distance (b) in spite of the short intervals between the bearings, the bearing surfaces must be as large as possible and (c) the shaft must be capable of withstanding not only mechanical stresses prevailing but also the effects of critical speeds. Crtical speeds are speeds at which teh crankshaft system goes into resonance. The variation of the turning moment diagram gives rise to torque hormonics of the first and higher orders, these forces give rise to resonance if the engine is operated at that particular speed so as to cause vibrations to be built up. The magnitude of these vibrations will be very large but the damping present prevents very high raise in the amount of vibration at critical speeds.

3.3 METHODS OF IMPROVING STRUCTURAL DURABILITY

3.3.1 Bored Webs

A web which is thicker around the pin allows the force lines to enter more uniformly. This effect is increased if the web is bored along with the crankpin since, the lines of force then are compelled to pass to a greater extent from the circumference of the pin into the web and to distribute itself uniformly.

3.3.2 OVERLAP OF CRANKPIN AND JOURNAL

Very thick crankpin which overlaps the journals has two effects i) its stiffens the whole shaft and ii) it flattens the curve followed by lines of force and thus increases the durability of the crankshaft. Figure 3.1 shows the effect of overlap; from which it is clear that even slight overlap makes considerable increase in bending fatigue strenght.(8)

3.3.3. WEB SHAPE

Figure 3.2 shows the vrious shapes and their effective durability in alternating torsion. The figures written above represent the maximum alternative twisting stress the shaft can withstand in kgf/sq.mm. The shaping is progressively from the lowest strength normal crank to the crank element which has oval webs which is broader than normal crank, bored crankpins and webs but the bore is barrel shaped so that the bore is narrow at the web end. With these alterations the durability is increased by about 3.5 times which is very astonishing when no change is made in the essential dimensions like the diameter of crankpin and journals and the thickness of web. (8)

3.3.4 OIL HOLES

The problem of favourably arranging oil hole confronts the designer. Detailed investigations have shown that even small bores usually reduce the durability of a steel shaft by more than half. Failure of crankshafts after long service can infact be mostly attributed to the oil bore, or atleast influenced by this. If the use of oil bore is unavoidable an attempt must be made to reduce the danger of failure to a minimum by the adoption of suitable measures. For this purpose the edges of the bore are provided with large, rounded and preferably polished lips, as fracture almost always begins at the mouth of the bore. Whenever possible, an endeavour must be made to place the oil bore in a region of low stressing. This can be done for instance by locating it in the fibres which are neutral in respect of the bending stresses at the moment of highest combustion pressure. For highly stressed hollow crankshaft it may preferable to chamber the edge of the oil bore inside the pin bore. If it should not be possible for a straight bore to be kept at a sufficient distance from the fillet., the course followed by the bore will have to be altered.

3.3.5 BEARING CLEARANCE

This is the radial distance between the journal and bearing which allows the movement of crankshaft in bearings. The influence of bearing displacement on the maximum bending stress is given by the folowing results of experiments.(11)

Bearing Clearance mm	Increase of load at fillet Percent
2.032	8
4.064	16
6.096	24
8.128	33
10.160	41

It is intresting to note that stress at fillets which are caused by bending loads are increased both by increase in distance between supports and by greater clearances.

3.3.6 SURFACE HARDENING

Usual methods of surface hardening are (a.) carburzing (b.) nitriding and (c.) high frequency inducation hardening. Nitriding gives very high degree of harness compared with carburzing and requires a lower temperature to perform. Nitriding also gives rise to high fatigue strength. Induction hardening requires lesser time but needes quenching where as for nitriding no quench is needed. The hardening is necessary only to bearing loads specify the amount of hardness necessary for crankpin and journals which are obtained by one of these two methods.

3.3.7 COLD ROLLING

To improve the fatigue resistances of large crankshafts the fillets are cold rolled and thus stregnthen the point of peak stresses. Cold rolling produces net residual compressive stresses at the surface leading to improved fatigue resistance. The effect of cold rolling is influenced by number of factors such as roll form, specific pressure, feed, speed, number of passes and depth of rolled surface layer.

3.4 FORGED STEEL CRANKSHAFTS

The usual procedure in manufacturing crank shafts is to forge it. The process is to heat the metal andbeat it is a die so as to get the required shape. The forging operations can be done only on materials which lend themselves to plastic deformation at high temperature. Open hearth carbon steel or nickel alloy steel can be used for forged shafts. The metal is heated to a temperature sufficiently greater than the critical tmperature and beating on the drop stamp gives the required shape. Hot work refines the structure of steel by smashing up present. The forging must be carried out with the strength requirements of crankshafts in mind. The shape of grain flow of the metal must be so controlled that the direction of the flow of the metal may be in the same direction as the lines of force thus adding strength. The forged blank must be properly machined so as to give the required shape. The machining possibilities is limited to the extent to which it could be carried out with;out very costly machines and very special processes. The machining of the j;ournal bearings and their proper allingment should be done with great care. Fatigue strength of the forged steel crankshaft is much higher than that of cast iron or cast steel crankshafts of the same ultimate strength. The forged steel crankshafts can be heat treated to very great extent to give the proper harness required for bearings. Use can be made of the various modifications mentioned in the beginning to increase the strength of forged crankshafts.

3.5 CAST STEEL CRANKSHAFTS

The use of cast steel for crankshafts is also possible and in many cases can be used instead of a forged steel crankshaft. The ratio of yield point to ultimate tensile strength and also the figures for elongation and reduction of area are approximately same for cast and forged steels. The results of bending tests and of notched-bar impact tests may also be described as good, while the micro structure is 5to be regarded as normal. In forged steel the fatigue strength is about one half the tensile strength: in this cast steel it is about one-third the tensile strength. The use of cast steel can be done if proper manufacturing process is used. Usually copper is added to increase fluidity and to reduce shrinkage. Chromium is added to improve wearing properties.

3.6 CAST IRON CRANKSHAFTS

The modern cast iron crankshafts on the whole be regarded as equal to the noraml steel crankshafts in durability. This is so because the casting technique offers far wider possibilities with regard to shaping. The high-quality cast iron possesses properties which make it suitable for use in crankshafts. These can be listed as follows:

a. Cast iron is less sensitive to notching and therefore a fillet or a stress raiser does not give rise to heavy stresses.

b. It is naturally shock absorbing so can withstand shock loading due to vibrations.

c. It has a high hardness figure which can give a good bearing surface without special hardening process.

d. It has a great resistance to wear and good running qualities.

e. It is much easier to shape the cast-iron crankshafts so that it has great durability.

f. Cast iron crankshafts are cheaper in cost and requires less time for maching and fewer working processes than the corresponding steel shafts.

3.6.1 SHAPING OF CAST IRON CRANKSHAFTS

Starting from the normal shape first the pins can be bored to the barrle shape. Webs can be made of oval form and use can be made either of relief notch proposed by Klose or by adopting Fraimot's incision. In the latter case the outer edges of the webs are made slightly bulding to make the force lines as evenly as possible. The ideal fom of crankshaft is difficult to realise in forged steel as machining possibilities are limited in practice. The positiion of cast crankshafts is quite different, the possibilities in the way of shaping are here almost unlimited, and the ideal form of crankshafts proposed can be realised without economically inadmissible work. This makes it clear that, as a result of very limited possibilities of shaping crankshafts by machining after forging the great material strength of forged sateel can be poorly used. On the other hand, the extensive possibilities of shaping available i;n the case of cast iron crankshafts allows much smaller strength of cast iron to be utilized to the greatest possible extent. Smooth changes in the cross secation enables cast iron crankshafts to be very durable. Further the hardening of crankpin and j;ournals can be easily done by chill casting without costly hardening processes.(11)

3.6.2 FATIGUE STRENGTH OF CAST IRON CRANKSHAFTS

The nominal limiting bending fatigue strength for the cast iron crankshafts are very approximately one-third of the limiting stress of material. The presence of fillets considerably affects fatigue strength unfavourably. Increase of crankweb thickness or breadth gives an increse in fatigue strength. Introduction of lightening holes does increase the bending fatigue strength as in the case of forged steel shaft if the holes is within 0.4 times the crankpin diameter. Similarly the journal and crankpin overlap increases the fatigue strength. There is some variation of the results in different batches of castings which gives a warning to see that care is exercised in production.

3.7 BUILT-UP OR SEMI BUILT-UP CRANKSHAFTS

Large crankshafts differ from the rest of the crankshafts not only in appearance but also in design. Generally a U - shaped forged and ready machined crnak is shrunk on to a smooth shaft (Semi built-up crankshaft), if dimensions are very large it may be necessary even for the web to be shrunk on to the crankpin (fully built-up crankshaft). The large crankshaft has thicker webs so that the shrunk fit of the web on the pin or journal is sufficient so that torque will be transmitted with certainity. Here too the crankpins can be bored to the shape of a barrel and polished with good rounding at its meeting with the web, ususally the fillet is undercut inside the web. Rightly dimensioned shrunk-on connections are capable of transmitting alternating torques which exceeding 95 per cent of those permissible for a smooth crankpin.

Ker wilson states the following general conclusions for the effect of shrink fits.

a) The effect of shrinkage may reduce the fatigue resistance by 50 per cent.

b) The fatigue resistance for light shrinkage pressure i.e., pressure less than 31.50 N/sq.mm2 is always greater for higher pressures. In the case of larger crankshafts this pressure may be about 94 to 142 Newton per sq.mm.

c) The fatigue resistance is reduced quite apreciably by the effect of fretting corrosion in some cases to the extent of from 25 to 30 percent.

d) Stress reliving grooves provide an appreciable improvement in fatigue resistance.

The zone adjacent to the plane of the shrink fit is usually subjected to combined static and dynamic stresses. At this point fracture in torsion of semit-built crankshafts have occured.(11)

The above factors can be applied practically to improve structural durability of crankshafts.

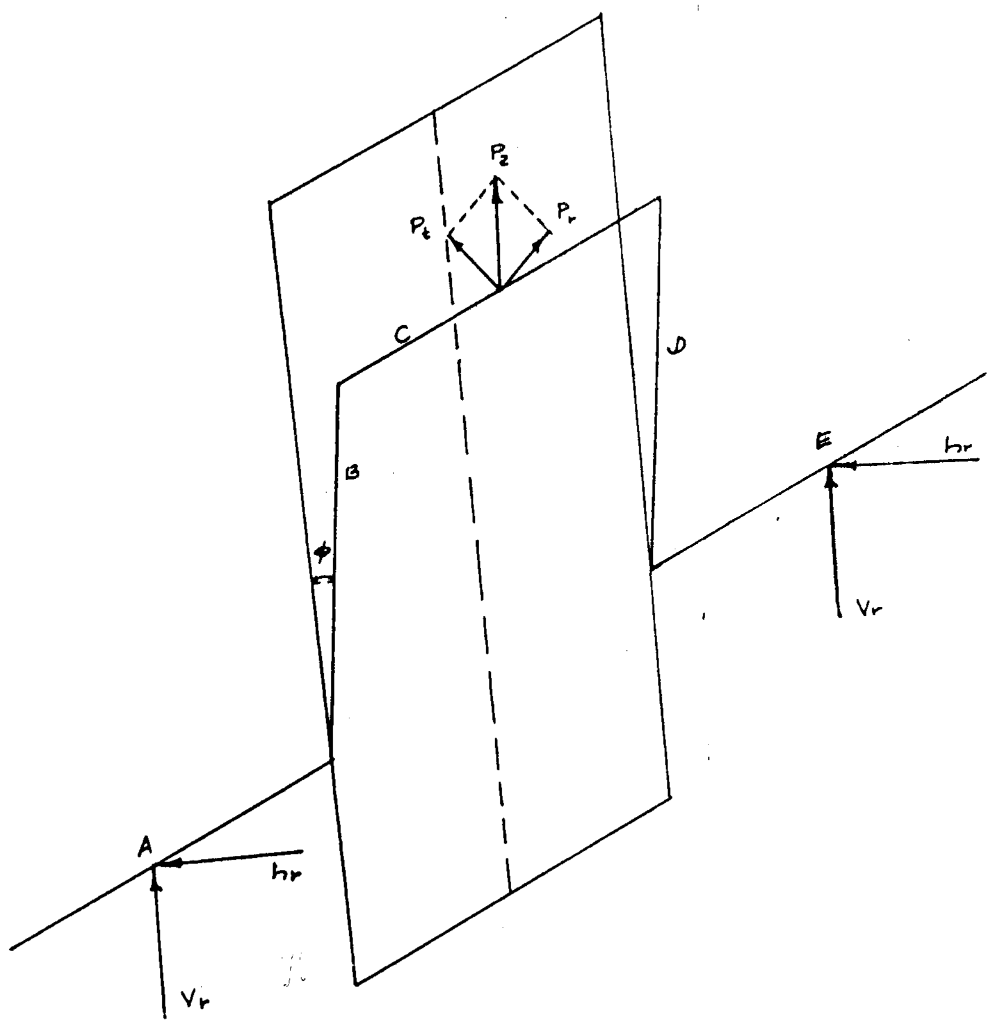


Figure 2.1 Forces Acting on a single Throw-Crankshaft

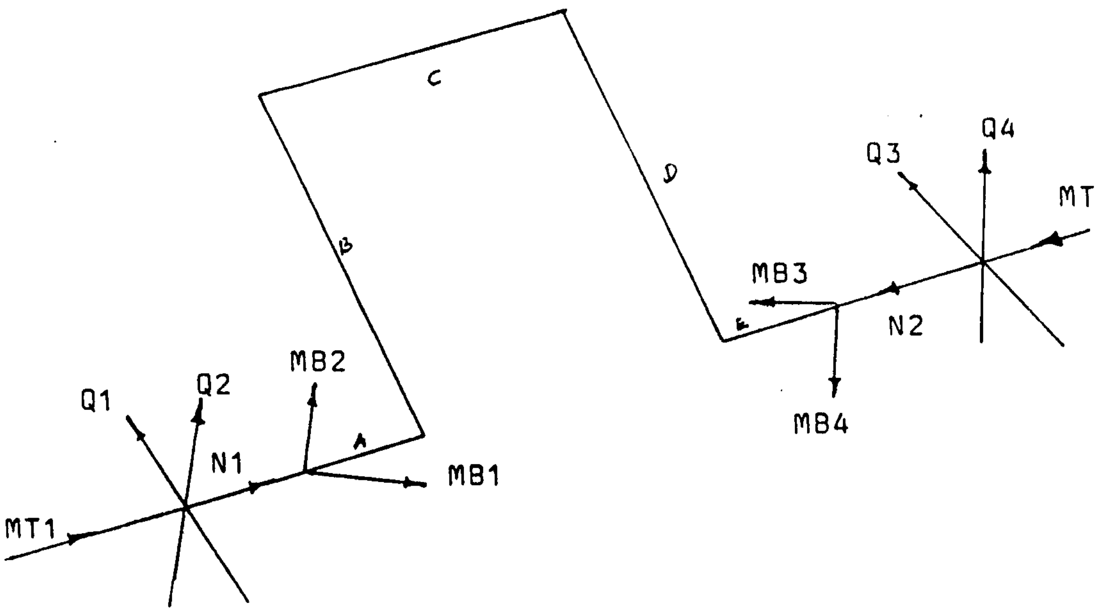


Figure 2.2 Crankshaft Reactions

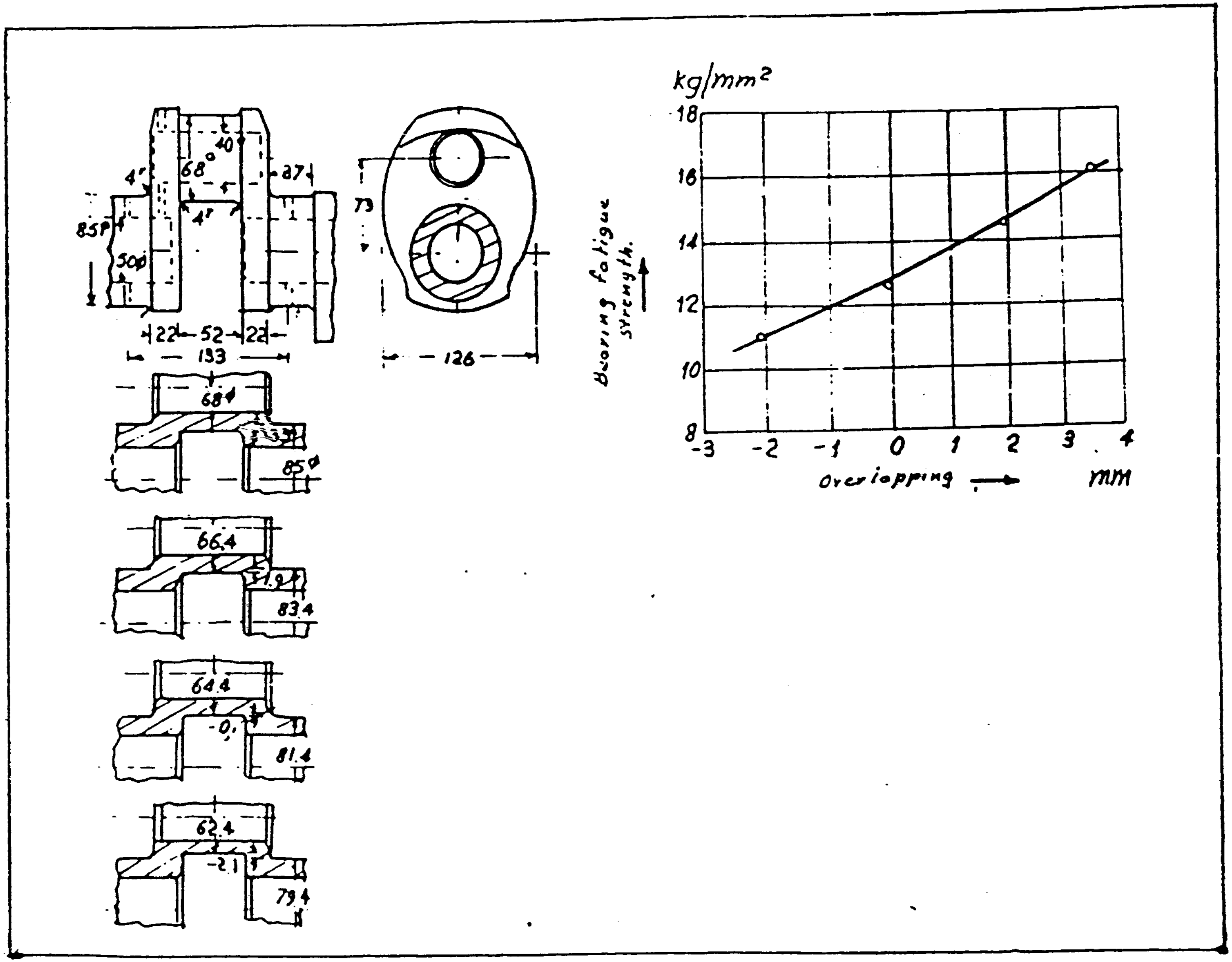


Figure 3.1 Effect of various degrees of overlapping on the durability of a Crankshaft [11]

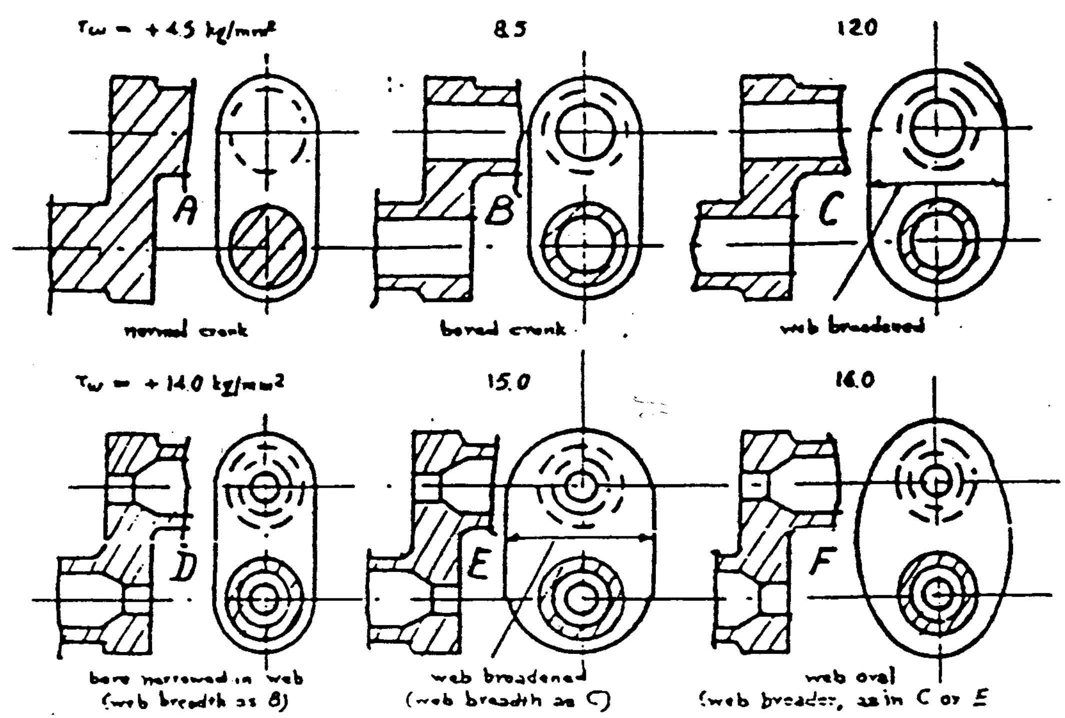


Figure 3.2 Structural durability trails with full size crankshafts carried out on an alternating torsion machine, S_t - 60, τ_B = 65 Kgf per sq.mm. [11]

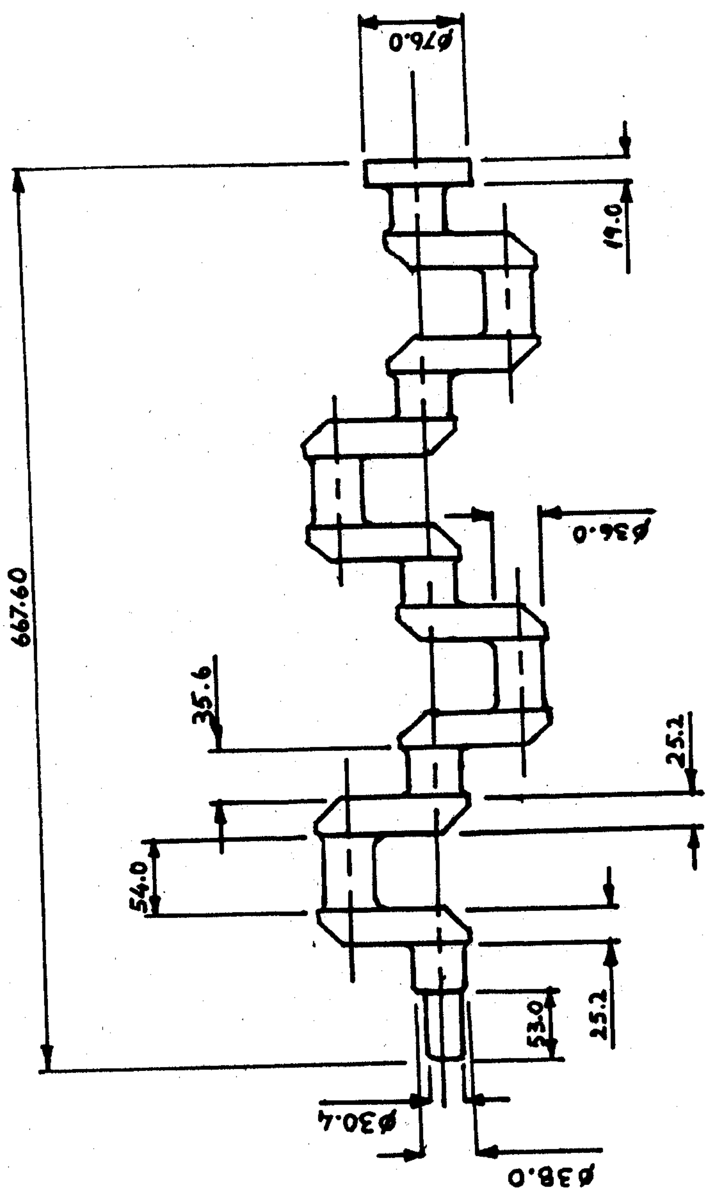


Fig 4.4.1

Drawing of Designed Crankshaft for four cylinder engine.

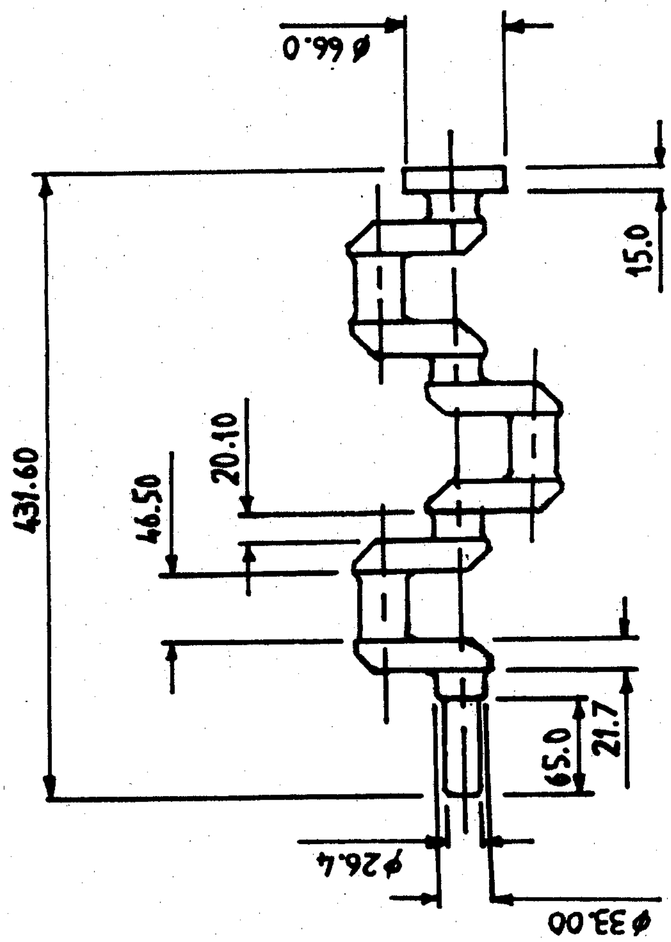


Fig 4.4.2

Drawing of Designed Crankshaft for three cylinder engine.

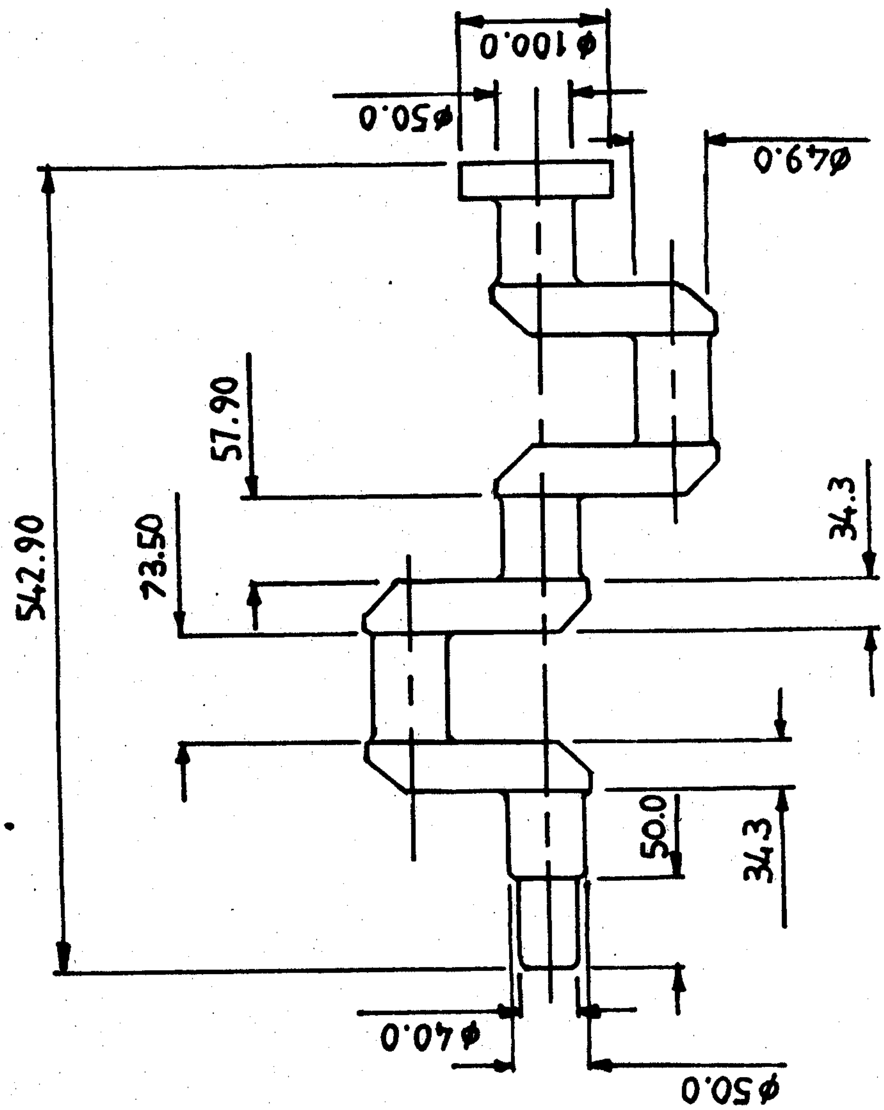


Fig 4.4.3

Drawing of Designed Crankshaft for two cylinder engine.

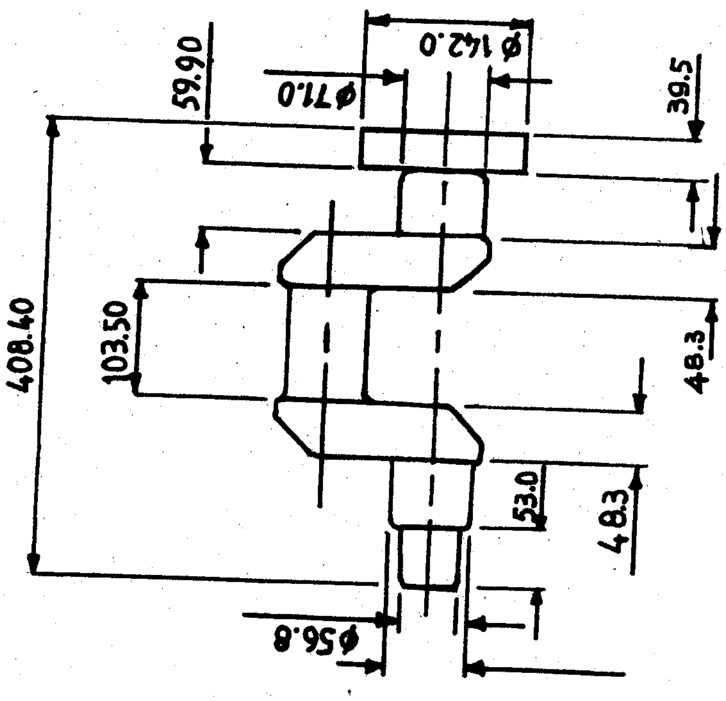


Fig 4.44

Drawing of Designed Crankshaft for single cylinder engine.

LIST OF SYMBOLS

D_c	-	Outer diameter of crankpin, mm
d_c	-	Inner diameter of carankpin, mm
L_e	-	Length of crankpin, mm
L_j	-	Length of journal, mm
D_j	-	Outer diameter of journal, mm
d_j	-	Inner diameter of journal, mm
L_w	-	Thickness of web, mm
B	-	Width of web, mm
R_{web}	-	Radius of web, mm
D_e	-	Equivalent diameter of crankshafat, mm
L_e	-	Equivalent length of crankshaft, mm
fos	-	Factor of safety
oy	-	Yield stress of the crankshaft material, N/mm^2
S	-	Distance between main bearing centres, mm
S_{pul}	-	Distance between left main bearing centre and pulley centre, mm
S_1	-	Distance between right main bearing centre and flywheel centre, mm
u	-	Coefficient of friction between belt and pulley
wt_{con}	-	Weight of connecting rod, Newton
wt_{pis}	-	Weight of piston, Newton
wt_{cb}	-	Weight of crankpin, Newton
wtrv	-	Weight of revolving parts, Newton
wtrp	-	Weight of reciprocating parets, Newton
Φ	-	Angle at which maximum torque pressure occurs, degrees

Ø	-	Crank angle, degress
R	-	Crank radius
	-	Crank radius/length of connecting rod
d_{cy}	-	Cylinder diameter, mm
P_{ex}	-	Exposion pressure, N/mm^2
	-	Density of the crankshaft material, N/mm^2
a	-	Major axis of web, mm
wfw	-	Weight of flywheel, Newton
R_k	-	Radius of gyration of flywheel, mm
d_3	-	Diameter of journal, mm
l_3	-	Length of crankpin, mm
d_0	-	Diameter of crankpin, mm
l_0	-	Length of crankpin, mm
w	-	Width of web, mm
t	-	Thickness of web, mm
g	-	Acceleration due to gravity, mm/sec^2

CHAPTER 4

RESULTS

4.1 DESIGN AND ANALYSIS OF FOUR CYLINDER ENGINE CRANKSHAFT

INPUT AND OUTPUT ARE AS FOLLOWS

DATA GIVEN

H.P POWER TO BE TRANSMITTED	=	55.00
NUMBER OF CYCLINDERS	=	4
SPEED OF THE ENGINE	=	2400.00 RPM
CYLINDER BORE DIAMETER	=	91.000 mm
STROKE	=	127.000 mm
EXPLOSION PRESSURE AT I.D.C.	=	3,600 NEWTON/SQ MM
ANGLE AT MAXIMUM TORQUE	=	23.000 DEGREES
L by R RATIO	=	3.000
MAXIMUM TORQUE PRESSURE	=	3.600N/sq.mm
DISTANCE BETWEEN MAIN BEARINGAS	=	140.00mm
DISTANCE OF PULLEY LEFT MAIN BEARING	=	23.000mm
DISTANCE OF FLYWHEEL FROM RIGHT MAIN BEARING	=	12.000mm
RADIUS OF FLYWHEEL	=	127.000mm
WEIGHT OF FLYWHEEL	=	920.000 NEWTONS
ANGLE OF LAP OF PULLEY BELT	=	180.0000 DEGREES
CO-EFFICIENT OF FRICTION BETWEEN BELT & PULLEY	=	.3000
MATERIAL CHOOSEN IS	=	C55 Mn75
YEILD STRESS OF THE MATERIAL	=	460.0000 N/SQ.MM

FACTOR OF SAFETY	=	2.500
WEIGHT OF PISTON	=	12.00 NEWTONS
WEIGHT OF CONNECTING ROD	=	20.00 NEWTONS

OUTPUT RESULTS

CRANK PIN DIAMTER	=	36.000 MM
LENGTH OF CRANK PIN	=	54.000 MM
JOURNAL DIAMETER	=	38.000 MM
LENGTH OF JOURNAL	=	35.600 MM
CRANK WEB THICKNESS	=	25.200 MM
CRANK WEB WIDTH	=	43.200 MM

4.2 DESIGN AND ANALYSIS OF THREE CYLINDER ENGINE CRANKSHAFT

DATA GIVEN

HORSE POWER TO BE TRANMITTED	= 43.00
NUMBER OF CYLINDERS	= 3
SPEED OF THE ENGINE	= 2000.00 RPM
CYLINDER BORE DIAMETER	= 78.00 mm
STORKE	= 100.000 mm
EXPLOSION PRESURE AT I.D.C	= 3.6000 NEWTON/SQ MM
ANGLE AT MAXIUM TORQUE	= 23.000 DEGREES
EXPLOSION PRESSURE AT MAX TORQUE	= 4.000N/Sq. mm
DISTANCE BETWEEN MAIN BEARINGS	= 110.000 mm
DISTANCE OF PULLEY FROM LEFT MAIN BEARINGS	= 23.000 mm
DISTANCE OF FLYWHEEL FROM RIGHT MAIN BEARING	= 12.000 mm
L by R RATIO	= 4.000
WEIGHT OF FLYWHEEL	= 650.000NEWTONS
RADIUS OF FLYWHEEL	= 650.000NEWTONS
CO-EFFICIENT OF FRICTION BETWEEN BELT & PULLEY	= .3000
ANGLE OF LAP PULLEY BELT	= 190.0000 DEGREES
WEIGHT OF CONNECTING ROD	= 20.00 NEWTONS
WEIGHT OF PISTON	= 12.00 NEWTONS
MATERIAL CHOOSEN IS	C60
YIELD STRESS OF THE MATERIAL	= 420.0000N/SQ.MM

OUTPUT RESULTS

CRANK PPIN DIAMETER	= 31.000 MM
LENGTH OF CRANK PIN	= 46.500 MM
JOURNAL DIAMETER	= 33.000 MM
LENGTH OF JOURNAL	= 20.100 MM
CRANK WEB THICKNESS	= 21.700 MM
CRANK WEB WIDTH	= 37.200 MM

4.3 DESIGN AND ANALYSIS OF TWO CYLINDER ENGINE CRANKSHAFT

DATA GIVEN	
HORSE POWER TO BE TRANSMITTED	= 30.00
NUMBER OF CYLINDERS	= 2
SPEED OF THE ENGINE	= 2200.00 RPM
CYLINDER BORE DIAMETER	= 127.000 mm
STORKE	= 175.000 mm
EXPLOSION PRESSURE AT I.D.C	= 3.000 NEWTON/SQ MM
ANGLE AT MAXIUM TORQUE	= 19.000 DEGGREES
L by R RATIO	= 4.000
MAX TORQUE PRESSURE	= 2.190N/Sq mm
DISTANCE BETWEEN MAIN BEARINGS	= 200.000 mm
DISTANCE OF PULLEY FROM LEFT MAIN BEARING	= 30.000 MM
DISTANCE OF FLYWHEEL FROM RIGHT MAIN BEARING	= 20.000 MM
RADIUS OF FLYWHEEL	= 120.000 MM
WEIGHT OF FLYWHEEL	= 675.000 NEWTONS
ANGLE OF LAP OF PULLEY BELT	= 180.0000 DEGREE

CO-EFFICIENT OF FRICTION BETWEEN BELT & PULLEY	= .3000
MATERIAL CHOOSEN IS	C60
YEILD STRESS OF THE MATERIAL	= 420.0000 N/SQ. MM
FACTOR OF SAFETY	= 2.500
WEIGHT OF PISTON	=21.00 NEWTONS
WEIGHT OF CONNECTING ROD	= 30.00 NEWTONS

OUTPUT RESULT

CRANK PIN DIAMETER	= 49.000 MM
LENGTH OF CRANK PIN	= 73.500 MM
JOURNAL DIAMETER	= 50.000 MM
LENGTH OF JOURNAL	= 57.900 MM
CRANK WEB THICKNESS	= 34.300 MM
CRANK WEB WIDTH	= 58.800 MM

4.4 DESIGN AND ANALYSIS OF SINGLE CYLINDER ENGINE CRANKSHAFT

DATA GIVEN

HORSE POWER TO BE TRANSMITTED	= 30.00
NUMBER OF CYLINDERS	= 1
SPEED OF THE ENGINE	= 1250.00 RPM
CYLINDER BORE DIAMETER	= 150.00 mm
STORKE	= 190.000 mm
EXPLOSION PRESSURE AT I.D.C	= 4.250 NEWTON/SQ MM
ANGLE AT MAXIUM TORQUE=.646 DEGREES	
L by R RATIO	= 4.000
EXPLOSION PRESSURE AT MAX TORQUE	= 4.123 N /Sq.MM
DISTANCE OF PULLEY FROM LEFT MAIN BEARING	= 20.000 MM
RADIUS OF FLYWHEEL	= 150.000 MM
WEIGHT OF FLYWHEEL	= 1200.000 NEWTONS
ANGLE OF LAP OF PULLEY BELT	= 2.8798 DEGREES
CO-EFFICIENT OF FRICTION BETWEEN BELT & PULLEY	= .3000
MATERIAL CHOOSEN IS	= C40
YEILD STRESS OF THE MATERIAL	= 380.0000 N/SQ MM
FACTORY SAFETY	= 2.500
WEIGHT OF PISTON	= 27.00 NEWTONS
WEIGHT OF CONNECTING ROD	= 55.00 NEWTONS

OUTPUT RESULT

CRANK PIN DIANETER	= 64.500 MM
LENGTH OF CRANK PIN	= 96.750 MM
JOURNAL DIAMETER	= 63.600 MM
LENGTH OF JOURNAL	= 22.950 MM
CRANK WEB THICKNESS	= 45.150 MM
CRANK WEB WIDTH	= 77.400 MM

```
C                          PROGRAM

C            COMPUTER AIDED DESIGN OF CRANKSHAFT.

C      Done by : B.V.B.REDDYI,               / /
C                R.SATISH,                  / /  FINAL YEAR PROJECT
C                S.PRABHU,                 / /   WORK
C                N.VENKI.                 / /    1986 - 90  BATCH

      dimension matl(7,2),string(3),dim(2)
      dimension le(10),aj(10),ak(10),j(10),x(10),ajppth(10),sjppth(10)
      dimension brgpr(18),yldst(4),q(10),thet(10)
      dimension engvar(3,8),matnam(4,2),brgnam(3,4)
      real le ,yldst,fos
      data engvar(1,1),engvar(1,2),engvar(1,3),engvar(1,4),
     1engvar(1,5),engvar(1,6),engvar(1,7),engvar(1,8) /
     1'Auto','moti','ve a','nd  ','Airc','raft',' eng','ines'/
      data engvar(2,1),engvar(2,2),engvar(2,3),engvar(2,4),
     1engvar(2,5),engvar(2,6),engvar(2,7),engvar(2,8) /
     1'Gas ','and ','oil ','engi','nes ','four',' str','oke '/
      data engvar(3,1),engvar(3,2),engvar(3,3),engvar(3,4),
     1engvar(3,5),engvar(3,6),engvar(3,7),engvar(3,8) /
     1'Gas ','and ','oil ','engi','nes ','two ','stro','ke  '/
      data brgnam(1,1),brgnam(2,1),brgnam(2,2),brgnam(2,3),
     1brgnam(3,1),brgnam(3,2),brgnam(3,3) /
     1'Main','Cran','k pi','n   ','Pist','on P','in  '/
      data brgpr(1),brgpr(2),brgpr(3),brgpr(4),brgpr(5),brgpr(6),
     1brgpr(7),brgpr(8),brgpr(9),brgpr(10),brgpr(11),brgpr(12),
     2brgpr(13),brgpr(14),brgpr(15),brgpr(16),brgpr(17),brgpr(18)/
     35.5,12.,10.,25.,15.,35.,5.5,8.5,10.,15.,12.5,17.,
     4 3.5,5.5,7.,10.,8.5,12.5/
      data matnam(1,1),matnam(2,1),matnam(2,2),matnam(3,1),
     1matnam(4,1),matnam(4,2) /'C40','C55 ','Mn75','C60',
     1'othe','rs  '/
      data yldst(1),yldst(2),yldst(3),yldst(4),fos /
     1380.,460.,420.,0.,2.5/
      iedm = 0
      WRITE (*,910)
 910  FORMAT(1X ,'GIVE VALUE FOR HP TO BE TRANSMITTED'/)
      READ(*,911) hp
 911  FORMAT(F15.7)
      WRITE(*,912)
 912  FORMAT(1X ,'GIVE NO OF CYLINDERS'/)
      READ(*,913) ncy
 913  FORMAT(I3)
      hp = hp / float(ncy)
      WRITE(*,914)
 914  FORMAT(1X ,'GIVE VALUE FOR SPEED OF THE ENGINE IN RPM'/)
      READ(*,915) an
 915  FORMAT(F15.8)
      WRITE(*,916)
 916  FORMAT(1X ,'GIVE VALUE FOR CYLINDER BORE DIA. IN MM '/)
      READ(*,917) cyldia
```

```
917   FORMAT(F15.8)
      WRITE(*,918)
918   FORMAT(1X ,'GIVE VALUE FOR STROKE LENGTH IN MM '/)
      READ(*,919) strok
919   FORMAT(F15.8)
      WRITE(*,920)
920   FORMAT(1X,'GIVE VALUE FOR EXPLOSION PRESSURE AT IDC IN N/SQMM '/)
      READ(*,921) exppr
921   FORMAT(F15.8)
      WRITE(*,922)
922   FORMAT(1X,'GIVE VALUE FOR ANGLE AT WHICH MAX. TORQUE OCCURS IN DEG
     1REES '/)
      READ(*,923) theta
923   FORMAT(F15.8)
      WRITE(*,924)
924   FORMAT(1X,'GIVE THE VALUE OF L/R RATIO ' /)
      READ(*,925) albyr
925   FORMAT(F15.8)
      WRITE(*,926)
926   FORMAT(1X,'GIVE VALUE FOR PRESSURE IN PISTON AT MAX. TORQUE IN N/S
     1QMM '/)
      READ(*,927) tpr
927   FORMAT(F15.8)
      WRITE(*,928)
928   FORMAT(1X,'GIVE VALUE FOR DISTANCE BETWEEN MAIN BRGS IN MM '/)
      READ(*,929) dmbrg
929   FORMAT(F15.8)
      WRITE(*,930)
930   FORMAT(1X,'GIVE VALUE FOR DISTANCE OF FLYWHEEL FROM LEFT MAIN BRG.
     1IN MM '/)
      READ(*,931) disfw
931   FORMAT(F15.8)
      WRITE(*,932)
932   FORMAT(1X,'GIVE VALUE FOR DISTANCE OF PULLEY FROM RIGHT MAIN BEARI
     1NG IN MM '/)
      READ(*,933) dispul
933   FORMAT(F15.8)
      WRITE(*,934)
934   FORMAT(1X,'GIVE WEIGHT OF FLYWHEEL IN NEWTONS'/)
      READ(*,935) wfw
935   FORMAT(F15.8)
      WRITE(*,936)
936   FORMAT(1X,'GIVE RAIDIUS OF FLYWHEEL IN MM '/)
      READ(*,937) rk
937   FORMAT(F15.8)
      WRITE(*,938)
938   FORMAT(1X,'GIVE VALUES FOR CO-EFF OF FRICTION AND ANG. OF LAP IN D
     1EG '/)
      READ(*,939) amu , anglap
939   FORMAT(F15.8)
      WRITE(*,940)
940   FORMAT(1X,'GIVE VALUE FOR DIAMETER OF PULLEY IN MM '/)
      READ(*,941) dpul
941   FORMAT(F15.8)
      WRITE(*,942)
```

```
942   FORMAT(1X,'GIVE WEIGHT OF CONNECTING ROD IN NEWTONS '/)
      READ(*,943) wtcon
943   FORMAT(F15.8)
      WRITE(*,944)
944   FORMAT(1X,'GIVE WEIGHT OF PISTON IN NEWTONS '/)
      READ(*,945) wtpis
945   FORMAT(F15.8)
      totdis = dmbrg + disfw + dispul
      distpu = dispul
500   WRITE(*,1000)
1000  FORMAT(1X,'Slno',5x,'Material',5X,'Yield strees in N/sqmm'//)
      DO 2000 k = 1,4
      WRITE(*,111) k,(matnam(k,I),I=1,2),yldst(k)
111   FORMAT(1h1,I4,5x,2A4,5x,F10.5)
2000  CONTINUE
      WRITE(*,112)
112   FORMAT(1X,'GIVE Sl.No. OF CHOSEN MATERIAL '/)
      READ(*,113) matno
113   FORMAT(I3)
      IF(matno.eq.4) THEN
      WRITE(*,114)
114   FORMAT(1X,'GIVE NAME OF NEW MATERIAL ' /)
      READ(*,115) matnam(4,1),matnam(4,2)
115   FORMAT(2A4)
      WRITE(*,116)
116   FORMAT(1X,'GIVE YLD STRESS OF NEW MATERIAL IN N/SQMM '/)
      READ(*,117) yldst(4)
117   FORMAT(E15.8)
      END IF
C     CALCULATIONS FOR CRANK PIN DIAMETER
      pi   = 3.1415927
      theta= theta * pi/180.0
      p1   = pi * cyldia ** 2 * exppr * 0.25
      h1   = 0.5 * p1
      h2   = h1
      bm1  = h1 * 0.5 * dmbrg
      WRITE(*,118)
118   FORMAT(1X ,'ASSUMED FACTOR OF SAFETY IS 2.5. PRINT Y/N')
      READ(*,119) ans
119   FORMAT(A1)
      IF((ans.eq.'y').or. (ans.eq.'Y')) GO TO 4000
      WRITE(*,121)
121   FORMAT(1X,'GIVE NEW VALUE FOR FACTOR OF SAFETY '/)
      READ(*,122) fos
122   FORMAT(F10.5)
4000  powr = 1.0 / 3.0
      dO   =(32.0 * bm1 * fos / (pi * yldst(matno))) ** powr
C     CALCULATIONS FOR CRANK AT MAX. TORQUE POSSITION
      phi  = asin (sin(theta) / albyr)
      p2   = pi * cyldia ** 2 * tpr/(4.0 * cos(phi))
      pt   = p2 * sin(theta + phi)
      pr   = p2 * cos(theta + phi)
      rr1  = pr * 0.5
      rr2  = rr1
      rt1  = pt * 0.5
```

```fortran
      rt2  = rt1
      v2   = disfw * wfw / totdis
      v3   = wfw - v2
      tork = hp * 71620.0 / an
      anglap = anglap * pi / 180.0
      t2   = 2.0 * tork / (dpul * (exp (amu * anglap) - 1.0))
      t1   = t2 * exp (amu * anglap)
      h2   = (t1 + t2) * distpu / totdis
      h3   = t1 + t2 - h2
      rr2h = rr2 * cos(theta)
      rr2v = rr2 * sin(theta)
      rt2h = rt2 * sin(theta)
      rt2v = rt2 * cos(theta)
      vrb2 = v2 + rr2v - rt2v
      hrb2 = h2 + rr2h + rt2h
      resrct = sqrt (vrb2 ** 2 + hrb2 ** 2)
      ar3 = sqrt(v3 ** 2+h3 ** 2)
      bm1 = p2 * dmbrg * 0.25
      d1 = (bm1 * 32.0 * fos / (pi * yldst(matno))) ** powr
      WRITE(*,123) d0
123   FORMAT(1X, 'DIA OF CRANK PIN CAL. ON MAX. EXPPPR =',F8.2)
      WRITE(*,124) d1
124   FORMAT(1X,'DIA OF CRANK PIN CAL. FOR MAX.TORQUE =',F8.2)
      WRITE(*,125)
125   FORMAT(1X,'CHOOSE DIA. OF CRANK PIN ROUND OF VALUE AND PRINT'/)
      READ(*,126) d0
126   FORMAT(F10.5)
      WRITE(*,127)
127   FORMAT(1X,'GIVE EMPERICAL RELATION BETWEEN LENGTH AND DIAMETER
     1OF CRANK PIN GENERALLY   al0 VARIES BETWEEN 1.25d0 to 1.5d0'/)
      READ(*,128) a
128   FORMAT(F10.5)
      aal0 = a * d0
      WRITE(*,129)
129   FORMAT(1X,'GIVE EMPERICAL RELATION BETWEEN CRANK WEB THICKNESS
     1AND CRANK PIN DIA.GENERALLY t VARIES BETWEEN .45d0 to .75d0 '/)
      READ(*,131) b
131   FORMAT(F10.5)
      t = b * d0
      WRITE(*,132)
132   FORMAT(1X,'GIVE EMPERICAL RELATION BETWEEN CRANK WEB WITH AND
     1CRANK PIN DIA.GENERALLY w VARIES BETWEEN 1.1d0 to 1.2d0'/)
      READ(*,133) c
133   FORMAT(F10.5)
      w = c * d0
C     CHECK FOR INDUCED STREESES IN THE WEB
36    CALL webst(h1,dmbrg,aal0,w,t,yldst,fos,matno)
C     DESIGN OF JOURNAL
39    bmj2 = p2 * dmbrg * 0.25
      twm  = pt * strok * 0.5
      eqbm = 0.5 * (bmj2 + sqrt (bmj2 ** 2+twm ** 2))
      d3   = (eqbm * 32.0 * fos / (pi * yldst(matno))) ** powr
      WRITE(*,134) d3
134   FORMAT(1X,'VALUE OF JOURNAL DIA. = ',F8.3)
      WRITE(*,135)
```

```
135    FORMAT(1X,'ROUND OFF THE VALUE AND PRINT '/)
       READ(*,136) d3
136    FORMAT(F10.5)
       ajnlln = (dmbrg - aal0 -2.0 * t)
       WRITE(*,50)
50     FORMAT(1X,80('~'))
       WRITE(*,60)
60     FORMAT(2X,'KIND OF BEARING',9X,'HIGHEST ALLOW. PRESSURE IN N/SQ
      1MM '//)
       DO 120 I=1,3
       WRITE(*,80) (engvar (I,k) , k=1,8)
80     FORMAT(1h1,2X,4A4)
       WRITE(*,70)
70     FORMAT(1X,80('-'))
       DO 90 m1 =1,3
       IF ((i.eq.1).and.(m1.eq.1)) k = m1
       WRITE(*,85) (brgnam(m1,i),i=1,4),brgpr(k),brgpr(k+1)
85     FORMAT(5x,4A4,10x,f8.2,'to',f8.2)
90     k = k + 2
120    CONTINUE
       WRITE(*,50)
       WRITE(*,137)
137    FORMAT(1X,'GIVE THE TYPE OF ENGINE FOR CRANK SHAFT BEING DESIGNED'
      1 /' TYPE 1 FOR Automotive and oil engines'
      1 /' TYPE 2 FOR Gas and oil engines and so on'/)
       READ(*,141) m
141    FORMAT(I3)
C      CHECK FOR INDUCED BEARING STRESSES
37     pb = p2 / (aal0 * d0)
       IF (m.eq.1) ij = 3
       IF (m.eq.2) ij = 9
       IF (m.eq.3) ij = 15
       IF (pb.lt.brgpr(ij).and.pb.gt.brgpr(ij+1)) GO TO 100
C      CHECK FOR SHEAR STRESSES
       shrst = p2 * 0.5 / (pi * d0 ** 2)
       allst = yldst (matno) * 0.5/fos
       IF(shrst.gt.allst) GO TO 100
C      CHECKING FOR STRESSES IN CRANK WEB AT POWER END
       bst   = rr2 * (ajnlln + t) * 3.0/(w * t ** 2)
       cst   = rr2 / (w * t)
       bst1  = pt * strok * 1.5/ (w * t ** 2)
       totst = bst + cst + bst1
       tmmt  = pt * (aal0 + t) * 0.25
       torst = tmmt * 4.5/ (w * t ** 2)
       pplst = totst * 0.5 + sqrt((totst/2.0) ** 2 + torst ** 2)
       IF (pplst.gt.(2.0 * allst)) GO TO 100
C      CHECK FOR INDUCED BEARING PRESSURES IN THE JOURNAL
       pj = resrct / (d3 * ajnlln)
       IF (m.eq.1) ik = 1
       IF (m.eq.2) ik = 7
       IF (m.eq.3) ik = 13
       IF (pj.lt.brgpr(ik).and.pj.gt.brgpr(ik+1)) GO TO 100
       IF(iedm.eq.1) GO TO 755
       GO TO 200
100    WRITE(*,142)
```

```
142   FORMAT(1X,'DESIGN FAILS RUN THE PROGRAM WITH ALTERNATE VAL. OF
     1INPUT'/)
      WRITE(*,150)
150   FORMAT(5x,'PRINT $ TO GO TO BEGINING OF THE PROGRAM '/)
      READ(*,143) choice
143   FORMAT(A4)
      IF (choice.eq.'$') GO TO 500
      WRITE(*,144)
144   FORMAT(1X,'YOU ARE COMING OUT OF THE PROG. WITH A FAILED DESIGN')
      GO TO 300
200   WRITE(*,146)
146   FORMAT(1X,'INPUT AND OUTPUT ARE AS FOLLOWS')
      WRITE(*,50)
      WRITE(*,147)
147   FORMAT(1X,70('-'))
      WRITE(*,148)
148   FORMAT(1X,'  ')
      WRITE(*,149)
149   FORMAT(1X,'                    DATA GIVEN    ')
      WRITE(*,151)
151   FORMAT(1X,'  ')
      WRITE(*,147)
      WRITE(*,148)
      WRITE(*,148)
      xxx = hp * float(ncy)
      WRITE(*,215) xxx , ncy , an
215   FORMAT(5x,'HORSE POWER TO BE TRANSMITTED =',F15.7,//
     15X,'NUMBER OF CYLINDERS                =',I3,//
     15X,'SPEED OF THE ENGINE                =',F15.8,2X,'RPM'//)
      WRITE(*,210) cyldia,strok,exppr,theta*180./pi
210   FORMAT(5x,'CYLINDER BORE DIAMETER          =',2x,F15.8,2X,'MM'//
     15x,'STROKE                             =',2x,F15.8,2x,'MM'//
     15x,'EXPLOSION PRESSURE AT I.D.C    =',2x,F15.8,2x,'NEWTON/SQ MM'//
     15x,'ANGLE AT MAXIUM TORQUE         =',2x,F15.8,2x,'DEGRESS'//)
      WRITE(*,220)albyr,tpr,dmbrg,disfw
220   FORMAT(5x,'L / R RATIO                               =',F15.8//
     15x,'EXPLOSION PRESSURE AT MAX TORQUE          =',F15.8,'N/SQMM'//
     15x,'DISTANCE BETWEEN MAIN BEARINGS            =',F15.8,2x,'MM'//
     15x,'DISTANCE OF PULLEY FROM LEFT MAIN BEARING =',F15.8,2X,'MM'//)
      WRITE(*,230) dispul,rk,wfw
230   FORMAT(5X,'DISTANCE OF FLYWHEEL FROM RIGHT MAIN BEARING =',
     1F15.8,'MM'//
     15X,'RADIUS OF FLYWHEEL                       =',F15.8,'MM'//
     15X,'WEIGHT OF FLYWHEEL                       =',F15.8,'NEWTONS'//)
      WRITE(*,240) anglap * 180./pi,amu,(matnam(matno,I),I=1,20)
240   FORMAT(5X,'ANGLE OF LAP OF PULLEY BELT =',F15.8,2X,'DEGREES'//
     15X,'CO-EFFICIENT FRICTION BETWEEN BELT &PULLEY =',F15.8 //
     15X,'MATERIAL CHOOSEN IS =  ',2A4 //)
      WRITE(*,11) yldst(matno),fos
11    FORMAT(5X,'YIELD STRESS OF THE MATERIAL = ',E15.8,2X,'N/SQ.
     1MM'//5X,'FACTOR OF SAFETY = ',F15.8//)
      WRITE(*,241) wtpis,wtcon
241   FORMAT(5x,'WEIGHT OF PISTON          =',F15.8,2X,'NEWTONS'//
     15X,'WEIGHT OF CONNECTING ROD       =',F15.8,2X,'NEWTONS'//)
755   WRITE(*,147)
```

```
      WRITE(*,148)
      WRITE(*,156)
156   FORMAT(1X,'                    OUTPUT RESULT         '//)
      WRITE(*,148)
      WRITE(*,147)
      WRITE(*,250) d0 , aal0 , d3
250   FORMAT(10x,'CRANK PIN DIAMETER    =',3X,F10.3,2X,'MM'//
     110x,'LENGTH OF CRANK PIN                =',3X,F10.3,2X,'MM'//
     110x,'JOURNAL DIAMETER                   =',3X,F10.3,2X,'MM'//)
      WRITE(*,2220) ajnlln , t , w
2220  FORMAT(10x,'LENGTH OF JOURNAL    =',3X,F10.3,2X,'MM'//
     110x,'CRANK WEB THICKNESS                =',3X,F10.3,2X,'MM'//
     110x,'CRANK WEB WIDTH                    =',3X,F10.3,2X,'MM'//)
      WRITE(*,158)
158   FORMAT(1X,'DO YOU LIKE TO COMPARE THE DESIGNED VALUES WITH THE
     1EXISTING CRANKSHAFT DIMENSIONS.PRINT y/n ')
      READ(*,159) yeyno
159   FORMAT(A2)
      IF ((yeyno.eq.'y').or.(yesno.eq.'Y')) then
      iedm=1
      WRITE(*,161)
161   FORMAT(1X,'GIVE  CRANK WEB WIDTH AND THICKNESS '/)
      READ(*,162) w , t
162   FORMAT(F10.5)
      CALL webst(h1,dmbrg,aal0,w,t,yldst,fos,matno)
38    WRITE(*,163)
163   FORMAT(1X,'GIVE CRANK PIN DIAMETER AND LENGTH IN MM'/)
      READ(*,164) ad0 , aal0
164   FORMAT(F10.5)
      IF(ad0.lt.d0) THEN
      WRITE(*,166)
166   FORMAT(1X,'CRANK PIN DIAMTER IS NOT WITHIN SAFE LIMITS '/)
      ELSE
      d0 = ad0
      ENDIF
      WRITE(*,167)
167   FORMAT(1X,'GIVE JOURNAL DIAMETER AND LENGTH OF EXISTING
     1CRANK SHAFT IN MM'/)
      READ(*,168) ad3 , ajnlln
168   FORMAT(F10.5)
      IF(ad3.lt.d3) THEN
      WRITE(*,169)
169   FORMAT(1X,'JOURNAL DIAMETER IS NOT WITHIN SAFE LIMITS '/)
      ELSE
      d3 = ad3
      GO TO 37
      ENDIF
      ELSE
      END IF
300   STOP
      END


      subroutine webst(h1 , dmbrg , aal0 , w ,t , yldst , fos , matno )
      dimension yldst(4)
```

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