Machine Design II

Q: 1 A) *What are the materials used for worm and worm wheel. State the advantages of worm and Worm wheel over other types of gears. Also explain self locking property of worm and wormWheel.*

Ans: Materials used for worm and worm wheel.

Worms are made of case hardened steel with a surface hardness of 60 HRC and a case depth of 0.75 to 4.5 mm . The following varieties of steel are used for the worm.

Normalized carbon steels - 40C8, 55C8
Case- hardened carbon steels- 10C4, 14C8
Case- hardened alloy steels
Nickel-Chromium steels

Wormwheel material should be soft & confirmable. Phosphor bronze with a surface hardness of 90 to 120 BHN, is widely used for the worm wheel.

Advantages of worm gear drives & self locking property:

- The most important characteristics of wprm gear drives is their high speed reduction. A speed reduction as high as 100: 1 can be obtained with a single pair of worm wheel.
- The worm gear drives are compact with small overall dimensions , compared with equivalent spur or helical gear drives having same speed reduction.
- The operation is smooth & silent.
- Provision can be made for self locking operation, where the motion is transmitted only from the worm to the worm wheel. This is advantageous in applicxation like crains & lifting devices.
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D) *list different types of rolling contact bearings. Define static & dynamic capacity of bearing .* 05

ANS: Types of rolling contact bearing :

1. Deep groove ball bearing
2. Cylindrical roller bearing
3. Angular contact bearing
4. Self aligning bearing
5. Taper roller bearing
6. thrust ball bearing

- static & dynamic capacity of bearing

  Static load carrying capacity of a bearing is defined as the static load which corresponds to a total permanent deformation of balls & races, at the most heavily stressed point of contact, equal to 0.0001 of the ball diameter.

  Dynamic load carrying capacity of a bearing is defined asd the radial load in radial bearings (or thrust load in thrust bearing) can be carried for a minimum life of 1million revolutions.

C) *Explain static and dynamic seals with examples.* 05

Ans: Seals are divided in to two main classes: static and dynamic. The static seals is used to prevent the Loss of fluid in a pressure vessel. A dynamic seal is used to prevent loss of fluid in a sliding or rotating

Joint. Numerous types of materials are used such as organic and mineral fibers, natural and synthetic

Rubber, cork, paper and soft metals.

For static seals, a flat rigid surface and a thin gasket are preferable, and a minimum amount of Packing surface should be exposed to the fluid. Sometimes sealing is created by friction. Sometimes Gasket is confined to a chamber and distorted to affect a seal by jamming the packing across the leakage path. Fir a dynamic seal, it is important that sliding member have a very smooth hard surface.

The mechanical seal is very effective for sealing a rotating shaft against gas or liquid under pressure. The sealing member may be either stationary or rotating.

B) *Write short notes on materials used for sliding contact bearing.* *05*

Ans: When the journal and the bearings are having proper lubrication i.e there is a film of clean, non-Corrosive lubricant in between, separating the two surfaces in contact, the only requirement of of the bearing material is that they should have sufficient strength and rigidity. The materials used for sliding contact bearings should posses the following properties: compressive strength, fatigue strength, confirmability, corrosion resistance,etc.

The materials commonly used for sliding contact bearings are given below:

Babbit metal: tin base babbit, lead base babbit

Bronzes: gun metal, phosphor bronze

Cast iron, siliver, non- metallic bearings.

Q:2 *Two stage spur and helical gear box is used to transmit 10 KW power from an electric motor rotating at 1440 rpm to a machine with approximately overall reduction ratio of 10. Design spur gear pair based on bending and and wear strength considerations. Check the gears for dynamic load using Buckingham's equation. Sketch the gear with all constructional features. Show main dimensions.* *20*

Solution:

given data:

Two stage gear box

Design based on bending and wear strength

Check the gears for dynamic load

Power: p=10 KW

Speed: N= 1440 rpm

Transmission ratio: i'=10

In a multistage gear box consisting of two or three stages, the velocity ratio at each stage should not exceed 6:1. In this case, the intermediate speeds are arranged in geometric progression.

If I' is the total transmission ratio, then the speed reduction at each stage (i) is obtained in the following way.

Step:1 Calculation of velocity ratio in each stage.

For two stage, i=√i' = √10 =3.16 (velocity ratio)

2 Selection of standard speed reduction ratio (PSG 8.12)

Velocity ratio(i)= 3.16

Step:3. Selection of standard tooth profile

20^0- Full depth Involute system

4. Selection of No. of teeth on pinion.

Zp= 18 (in order to avoid interference)

5. Calculation of No. of teeth on gear.

We know that,

Velocity ratio (i) = Zg/Zp (here Zp=18)

Zg= 57 (one hunting tooth)

6. Selection of material for pinion and gear. (PSG 8.4)

When speed ratio less than 4

Pinion- 40Ni2Cr1Mo28

Gear - 15Ni2Cr1Mo15

7. Selection of Design bending stress $[\sigma_b]$, and Design surface stress $[\sigma_c]$, Kgf/cm^2 (PSG 8.5)

By considering module up to 6,

$[\sigma_b]_p$ = 4000 Kgf/cm^2 $[\sigma_c]_p$ = 11000 Kgf/cm^2

$[\sigma_b]_g$ = 3200 Kgf/cm^2 $[\sigma_c]_g$ = 9500 Kgf/cm^2

8. Selection of Lewis form factor for pinion y_p & for gear y_g (PSG 8.18, T-18)

Zp= 18 y_p = 0.377

Zg= 57 y_g = 0.480 (by using interpolation formula)

9. Decide the weaker element.

Beam strength of gear tooth (Lewis equation)

Sb= m. b. σ_b. y

Beam strength for pinion, $[\sigma_b \times y]_p$ = 4000 x 0.377 = 1507 Kgf/cm^2

Beam strength for pinion, $[\sigma_b \times y]_g$ = 3200 x 0.480 = 1536 Kgf/cm^2

From above calculation pinion is weaker than gear.

10. Calculation of Nominal twisting moment Mt, Kgf-cm (PSG 8.15)

Mt= 97420 KW/n = 97420 x 10/1440 = 676.52 Kgf-cm

11. Calculation of Design twisting moment [Mt], Kgf-cm (PSG 8.15)

[Mt]= Mt. Kd.K for symmetric scheme Kd.K = 1.3

= 676.52 x 1.3

= 879.48 Kgf-cm

12. Selection of $\vartheta m = b/m$ (face width to module ratio) PSG 8.14

$\vartheta m = 10$

13. Calculation of module (m) PSG 8.13

Formula based on beam (beanding) strength of pinion tooth.

$$m \geq 1.26 \sqrt[3]{\frac{[Mt]}{y.[\sigma b]\psi_m Z_p}}$$

$$m \geq 1.26\sqrt[3]{\frac{879.84}{0.377\times 3200\times 10\times 18}}$$

m= 2.008 mm

14. Selection of standard module PSG 8.2

m= 2.00 mm

15. Calculation of equivalent young's modulus (Eequ)

For steel Eequ= 2.15 x 10^6 Kgf/cm^2

16. Calculation of center distance (a) PSG 8.22

$$a = m\left(\frac{Z_p + Z_g}{2}\right)$$

$$a = 2\left(\frac{18+57}{2}\right)$$ = 75 mm

17. Calculation of face width (b)

b= 10m = 10 x 2 = 20 mm

18. Calculation of surface (contact, compressive) strength. PSG 8.13

Check the module for wear

$$\sigma_c = 0.74\frac{i+1}{a}\sqrt{\frac{i+1}{i.b}Eequ.[Mt]}$$

$$\sigma_c = 0.74\frac{3.16+1}{7.5}\sqrt{\frac{3.16+1}{3.16\times 2}2.15\times 10^6\times 879.84}$$

$\sigma_c = 14483.48$ Kgf/cm^2

$[\sigma_c]_p$ = 11000 Kgf/cm^2

From the above calculation, it is observed that the design surface stress is higher than the actual surface stress. So that gear tooth failure occur due to wear. Therefore, it becomes necessary to increases the module of gear tooth, in order to avoid failure of tooth.

19. Change the module by using center distance formula. PSG 8.13

a $\geq i + 1\sqrt[3]{\left(\frac{0.74}{[\sigma c]}\right)\frac{E\times[Mt]}{i\times\varphi}}$ Type equation here.

here, i=3.16, [σc]= 10250 Kgf/cm^2, E= 2.15X 10^6 Kgf/cm^2, φ = b/a =0.266

from above equation, m=5.00 mm

20. Calculation of dynamic load by using Buckingham's equation.

$$Fd = Ft + \left[\frac{0.164V_m(cb+Ft)}{0.164V_m + 1.485\sqrt{cb+Ft}}\right]$$

$Ft = \frac{2Mt}{d1}$ = 2x $\frac{676.52}{9.0}$ = 150.33 Kgf , d1= Z1x m= 90 mm

Vm= π d1 N1 = 3.14 x 0.9 x 1440 = 4071.50 m/s,

B= 10 x 5 = 50 mm = 5.0 cm

C = 11860 x e = 11860 x 0.025= 256.5 mm= 25.65 cm

By putting thease values in Buckingham's equation, we get

Fd= 418.93 Kgf

21. Calculation of beam strength of gear tooth.

Beam strength of gear tooth is to be calculated by using Lewis equation.

Sb = m. b. σb. Y where y is form factor

= 0.5 x 5.0 x 4000 x 0.377

= 3770 Kgf

From the above calculations, beam strength of gear tooth is greater than dynamic load acting On gear tooth, failure of the tooth is not occur. Hence gear design is safe.

22. Gear proportions.

	Pinion	Gear
1. module (m)	5.0 mm	5.0 mm
2. center distance (a)	187.5 mm	
3. height factor (fo)	1	1
4. bottom clearance (c)	1.25 mm	1.25mm
5. tooth depth (h)	11.25 mm	11.25 mm
6. Pitch dia. (d)	90 mm	285 mm
7. Tip dia. (da)	100 mm	295 mm
8. Root dia. (df)	77.5 mm	272.5 mm
9. No. of teeth	18	57

Q: 3) *A deep groove ball bearing having SKF No. 6207 subjected to load cycles as below which is repeated.*

Phase	*Radial load (KN)*	*Axial load (KN)*	*N (rpm)*	*%*
I	*3*	*1*	*600*	*15*
II	*3.5*	*1*	*800*	*20*
III	*5.0*	*0.1*	*900*	*30*
IV	*0.5*	*2*	*1500*	*35*

under each phase of loads average with high shocks. Determine the expected life of bearing in hrs with probability of survival 95%. *20*

Solution:

Phase	N (rpm)	%	
I	600	15	N1= 600 X 0.15 = 90 rev
II	800	20	N2 = 800 X 0.20 = 160 rev
III	900	30	N3 = 900 X 0.30 = 270 rev
IV	1500	35	N4 = 1500 X 0.25 = 525 rev

N= 1045 rev

- Calculation of mean cubic load.

PSG 4.2

$$F_m = \sqrt[3]{\frac{F_1 N_1^3 + F_2 N_2^3 + F_3 N_3^3 + F_4 N_4^3}{N}}$$

First find F1, F2, F3 and F4 (Equivalent bearing load)

F = (X V Fr + Y Fa) S

Where X – radial factor
Y - axial factor
V - race rotation factor, V= 1, when inner race rotates;
Fa – axial load, KN
Fr - radial load, KN
S – Service factor = 1.2

Given bearing size: SKF 6207 PSG 4.13
Static load carrying capacity, Co= 1370 Kgf
Dynamic load carrying capacity, C= 2000 Kgf

For phase I
Fa= 1 KN, Fr= 3 KN, Fa/Fr= 0.333, Fa/Co= 0.0729
for Fa/Co= 0.0729, e = 0.27, ∴ Fa/Fr > e
X= 0.56 and Y=1.6
F1= (X Fr + Y Fa) S
= (0.56 x 3 + 1.6 x 1) 1.2
= 3.936 KN

For phase II
Fa= 1 KN, Fr= 3.5 KN, Fa/Fr= 0.285, Fa/Co= 0.0729
for Fa/Co= 0.0729, e = 0.27, ∴ Fa/Fr > e
X= 0.56 and Y=1.6
F2= (X Fr + Y Fa) S
= (0.56 X 3.5 + 1.6 X 1) 1.2
= 4.272 KN

For phase III
Fa=0. 1 KN, Fr= 5.0 KN, Fa/Fr= 0.02, Fa/Co= 0.00729
for Fa/Co= 0.00729, Therefore e is less than 0.22
∴ Fa/Fr > e
X= 0.56 and Y=2.0
F3= (X Fr + Y Fa) S
= (0.56 X 5.0 + 2.0 X0. 1) 1.2
= 3.6 KN

For phase IV
Fa=2.0 KN, Fr= 0.5 KN, Fa/Fr=4, Fa/Co= 0.145
for Fa/Co= 0.145, e = 0.385, ∴ Fa/Fr > e
X= 0.56 and Y=1.3
F4= (X Fr + Y Fa) S
= (0.56 X 0.5 + 1.3 X 2.01) 1.2
= 3.456 KN

From the above calculations,
F1= 3.936 KN
F2 = 4.272 KN
F3 = 3.6 KN
F4 = 3.456 KN

- Calculation of mean cubic load.

PSG 4.2

$$F_m = \sqrt[3]{\frac{F_1 N_1^3 + F_2 N_2^3 + F_3 N_3^3 + F_4 N_4^3}{N}}$$

$$F_m = \sqrt[3]{\frac{3.936 \times 90^3 + 4.272 \times 160^3 + 3.6 \times 270^3 + 3.456 \times 525^3}{1045}}$$

Fm = 82.712 KN = 8271 Kgf

Let us find the life of bearing (SKF 6207)

C = Fm $(L)^{1/k}$ k = 3 for ball bearing

By putting the values of mean cubic load and the dynamic load, we get

L_{10mr} = 156 mr

Life of bearing for 95% of survival

$$\frac{L_{95}}{L_{10}} = \left[\frac{In\left(\frac{1}{0.95}\right)}{In\left(\frac{1}{0.90}\right)}\right]^{1/b}$$ where b=1.34, and L_{10}= 156 mr

L_{95} = 88.38 mr

Life of bearing in hrs.

$L_{mr} = 60 \times N \times Lh / 10^6$

$88.38 = 60 \times 1045 \times Lh / 10^6$

Lh= 1409.56 hrs

Q: 4 *A pair of bevel gears is required to transmit 5 KN power from pinion shaft rotating at 400 rpm with a reduction ratio of 3.4 approximately. The shaft angle is 80^0 and the drive is subjected to moderate shock and operating for 12 hrs per day. Determine pitch cone angle of pinion and gear. Select suitable materials and design stresses, determine module and face width. Determine pitch Circle diameters.* *20*

Solution: Given data:

P= 5 KN

N = 400 rpm

i = 3.4

$\delta = \delta 1 + \delta 2 = 80^0$

step:1 To find the No. of teeth on pinion and gear.

Let us consider 20^0 Full depth involute system.

Here, Zmin $\geq$ 17 in order to avoid interference and undercutting.

Take Zp = 18, as i= 3.4 Zg= 61

Actual reduction ratio, i= 3.38,

Step:2 To find pitch cone angles δ1, and δ2

$\delta = \delta1 + \delta2 = 80^0$ given

$i = \frac{\sin\delta 2}{\cos\delta 2} = \tan\delta 2$

$3.38 = \frac{\sin\delta 2}{\sin(80-\delta 2)}$

Sinδ2 = 3.38 [sin80. Cosδ2 – cos80. Sinδ2]

Sinδ2 = 3.38 [0.984. Cosδ2 – 0.173. Sinδ2]

Sinδ2 = 3.3259. Cosδ2 – 0.584. Sinδ2

1.584 Sinδ2 = 3.325 Cosδ2

$i = \frac{\sin\delta 2}{\cos\delta 2} = \tan\delta 2 = 2.099$

$\delta 2 = \tan^{-1} 2.0999 = 64.520$

δ1 = 15.470

step:3 calculation of rated torque.

$Mt = 97420 \times \frac{KW}{n}$

$Mt = 97420 \times \frac{5}{400}$

Mt = 1217.75 Kgf-cm

Step:4 Selection of material for pinion and gear.

As rated torque Mt is higher we will select alloy steel for pinion and gear.

Pinion- 40Ni2Cr1Mo28 PSG 8.5

Gear - 15Ni2Cr1Mo15

Step:5 Calculation of virtual No. of teeth on pinion and gear. PSG 8.39

$Zv1 = \frac{Z1}{COS\delta 1}$ and $Zv2 = \frac{Z2}{\cos\delta 2}$

$Zv1 = \frac{18}{COS(15.47)}$ $Zv2 = \frac{61}{\cos(64.52)}$

Step: 6 Calculation of Lewis form factor for bevel pinion and bevel gear based on virtual No. of teeth.

For Zv1 = 18.67 Yv1 = 0.379 PSG 8.18

Zv2 = 141.79 Yv2 = 0.520 by interpolation

Step:7 Selection of permissible bending stresses for pinion and gear material:

By considering module up to 6 psg 8.5

[σb]p = 4000 Kgf/cm^2

[σb]p = 3200 Kgf/cm^2

Step:8 Decide the weaker element.

$[\sigma b \times Y]p = 4000 \times 0.379 = 1516$ Kgf/cm^2

$[\sigma b \times Y]p = 3200 \times 0.520 = 1664$ Kgf/cm^2

From the above calculation pinion is weaker than gear.

Step:9 Calculation of Design twisting moment [Mt]

[Mt] = Mt Kd.k for overhang system Kd.k = 1.5 PSG 8.15

$[Mt] = 1217.75 \times 1.5$

[Mt] = 1826.625 Kgf-cm

Step:10 Calculation of module for pinion and gear.δ

Design based on beam strength of pinion.

$m_{ave} \geq 1.28 \sqrt[3]{\frac{[Mt]}{Yv.[\sigma b]\vartheta m.Z1}}$ PSG 8.13

$m_{ave} \geq 1.28 \sqrt[3]{\frac{1826.625}{1516\times10\times18}}$

$m_{ave} \geq 0.241$ cm

$m_t = m_{ave} + \frac{b}{z1}\sin\delta1$ PSG 8.13

$= 0.241 + \frac{10\times0.241}{18}\sin 15.47^0$

$m_t = 2.76$ mm

Selection of standard module PSG 8.2

$m_t = 3.00$ mm

Step:11 To find pitch cone radius (R)Type equation here.

$sin\delta1 = \frac{d1/2}{R} = \frac{d1}{2R} = \frac{Mt.Z}{2R}$

$\therefore$ $R = \frac{3\times18}{2\times sin\delta1} = \frac{54}{2\times sin15.47}$

R = 101.225 mm

Step:12 To find face width (b)

$b = \varphi m \times m_t$

b = 10 x 3

b = 30 mm

Step:13 To check for wear.

$$\sigma_C = \frac{0.72}{(R-0.5b)}\sqrt{\frac{\sqrt{(i^2+1)^3}}{ib}E[Mt]}$$

$$\sigma_C = \frac{0.72}{(10.12-0.5b)}\sqrt{\frac{\sqrt{(3.38^2+1)^3}}{3.38\times3}2.15\times10^6\times1826.625}$$

= 0.0835 x 130236.142

= 10874.71 Kgf/cm^2

Step:14 Calculation of Design Surface Stress [σc]

[σc]p = 11000 Kgf/cm^2

[σc]g = 9500 Kgf/cm^2

$[\sigma c]_{ave}$ = 10250 Kgf/cm^2

As σc > [σc], design is not safe.

For that increase Mt to satisfy condition.

Select Mt = 4.00 mm

Step:15 Bevel pinion & gear dimensions. PSG 8.39

Pinion: Z1 = 18

P. C. D. = mt x Z1 = 4 X 18 72 mm

b = 10 x 4 = 40 mm

addendum =ha = mt = 4 mm

dedendum = hf = 1.236 x mt = 4.944 mm

Gear: Z2 = 61

P. C. D. = 244 mm

ha = 4.00 mm

hf = 4.944 mm

Q:5 *A 360^0 hydrodynamically lubricated bearings supports a load of 20 KN when operating at 1000 rpm for steam turbine. Assume bearing pressure of 1.6 N/mm^2. Find*

1. Diameter and length of bearing
2. Clearance ratio
3. Minimum film thickness
4. Viscosity of oil
5. Coefficient of friction
6. Friction power loss
7. operating temp. of oil *20*

Solution: given data:

360^0 hydrodynamically lubricated bearing

Load acting on bearing : W= 20 KN

Speed of journal: N = 1000 rpm

Application: steam turbine

Allowable bearing pressure: P = 1.6 N/mm^2

Step:1 Selection of L/D .

Let L/D = 1 i.e Square bearing

Step:2 Calculation of length and diameter of bearing

We know that,

Bearing Pressure(P) = $\frac{LOAD(W)}{Projected\ area\ of\ bearing\ (L\times D)}$

$1.6 = \frac{20000}{L\times D}$

$D^2 = 12500$

L= D = 111.80

L = D = 112 mm

Step:3 Calculation of clearance

In this problem fit is not given, selection of proper fit for steam turbine application. PSG 3.4

Selected fit H9 e9 (normal running fit)

For Hole tolerance PSG 3.9

$\Phi\ 112^{+87}_{000}$

Upper limit = 112.087 mm

Lower limit = 112 mm

For shaft tolerance

$\Phi\, 112^{-0.072}_{-0.159}$

Maximum clearance between shaft and bearing.

= 112.087- 111.841

= 0.246 mm

Minimum clearance between shaft and bearing.

= 112.00 – 111.928

= 0.072 mm

Average clearance between shaft and bearing.

$= \frac{0.246+0.072}{2}$ = 0.159 mm

Average clearance = 0.159 mm

Step:4 Selection of Viscosity of oil (Z) PSG 7.34

For steam turbine application

Z = 2 – 16

Select Z = 10 centipoise

Step:5 calculation of Sommerfeld Number (S) PSG 7.34

$S = \frac{Z'n'}{P}\left(\frac{D}{C}\right)^2$ where, $z' = \frac{Z}{9.81\times10^7}$ Kgf-sec/cm^2

$S = \frac{10}{9.81\times10^7}\times\frac{1000}{60}\times\frac{1}{160}\left(\frac{112}{0.159}\right)^2$ $z' = \frac{10}{9.81\times10^7}$ Kgf-sec/cm^2

S = 0.00526 $n' = \frac{1000}{60}$ rps

Step:6 Calculation of minimum film thickness PSG 7.36

For full journal bearing When S= 0.00474, $\frac{2\,ho}{C} = 0.03$

S = 0.0188, $\frac{2\,ho}{C} = 0.1$

By interpolation,

$$\frac{0.00526-0.00474}{0.0188-0.00474} = \frac{2\,ho/c-0.03}{0.1-0.03}$$

$$\frac{0.00052}{0.01406} = \frac{2\,ho/c-0.03}{0.07}$$

2 ho/c = 0.0325

∴ ho= 0.00259 mm ---- minimum oil film thickness

Step:7 Calculation of coefficient of friction (μ) PSG 7.36

For S= 0.00526 and $\frac{L}{D} = 1$

S lies between 0.00474 - 0.0188

$\frac{\mu D}{C} =$ lies between 0.514 – 1.05

$$\frac{0.00526-0.00474}{0.0188-0.00474} = \frac{\frac{\mu D}{C}-1.05}{1.05-0.514}$$

$$\frac{5.2\times10^{-4}}{0.01406} = \frac{\frac{\mu D}{C}-1.05}{0.536}$$

$\frac{\mu D}{c} = 0.555$

$\mu = 7.89 \times 10^{-4} = 0.0007$

Step:8 Calculation of increase in temp of oil ($\Delta$to)

L/D = 1, and S = 0.00526

S lies between 0.00474 – 0.0188

$\frac{\rho C' \Delta to}{P}$ = (2.61 -- 5.16)

By interpolation method,

$\Delta$to = 3.047^{0}c rise in temp of oil.

Q:6 *The following data refers to centrifugal pump for pumping water:*

Static suction head = 2.5 m
Length of suction pipe = 6 m
Static delivery head = 15 m
Length of delivery pipe = 30 m
Discharge = 1200 LPM
Design completely a centrifugal pump for given application which include the design of impeller, Shaft, bearing and casing. Also draw suitable layout for this pump. 20

Solution: Assumptions

- Single stage as Hstatic $<$ 40 m

Hstatic = hs + hd = 2.5 + 15 = 17.5 m

- Single stage as Q is less.
- Horizontal layout – ease of assembly and lubrication.
- Direction of flow : Radial (to control cavitation)

Step:1 suction and delivery pipe sizes.

Material: cast steel or mild steel

- Suction pipe size

Actual discharge (Qa) = 1200 LPM

(Qa) = 0.02 m^3/ sec

Assume, Volumetric efficiency η = 95 %

Volumetric efficiency $\eta = \frac{Qa}{Qt}$

$\therefore$ Therotical discharge, Qt = 0.02 / 0.95 = 0.0210 m3/sec

Velocity of liquid in suction pipe (Vs) = 1.5 to 2 m/ sec

Vs= 2 m/sec as per pump manufacturer

Qt = As x Vs As = 0.0210 / 2 = 0.0105 m^2

0.0210 = As x 2

ds^2 = 0.0134 m

ds = 130 mm dia of suction pipe

- Delivery pipe size

Velocity of liquid in delivery pipe = 5 m/sec

Qt = Vd x Ad

0.0210 = 5 x Ad

Ad = 4.2 x 10^{-3}

dd = 75 mm diameter of delivery pipe.

Step:2 Estimation of Losses.

A) At suction side

ΣH_{Losses} = hfs + hb + hv

According to Darcy- Weisbatch formula

hfs = $\frac{4\,f\,ls\,Vs^2}{(ds)2\,g}$ assume 4f= 0.01

= $\frac{0.01 \times 6 \times (4)}{0.130 \times 2 \times 9.81}$

= 0.078 head loss in friction

hb = hv = $\frac{kv^2}{2g}$

we assume directly

hb = 0.035m, and hv = 0.2m as per pump manufacturer

ΣH_{Losses} = 0.078 + 0.035 + 0.2

ΣH_{Losses} = 0.313m head losses in suction pipe

Check it for cavitation

NPSH = Ha – Hv – hs – hfs assume hv = 2.3m for water at 27^0c

= 10.3 – 2.3 – 2.0 – 0.313

= 5.687 m available NPSH

AS NPSH > 2.5 m no cavitation on suction side.

2.5 m is the required NPSH.

B) At Delivery side

ΣH_{Losses} = hfd +2 x hb + hv + $h_{K.E}$

hfd = $\frac{4\,f\,ls\,Vs^2}{(ds)2\,g}$

hfd = $\frac{0.01 \times 30 \times (4)^2}{(0.075)2 \times 9.81}$

hfd = 3.26 m

assume hb = 0.35 m 2 bend hb = 0.07 m

hv = 0.2 m

$h_{K.E} = \frac{Vd^2}{2g} = \frac{16}{2\,(9.81)} = 0.815\ m$

ΣH_{Losses} = 3.26 + 0.07 + 0.2 + 0.815

ΣH_{Losses} = 4.345 m head losses in delivery pipe

Manometric Head (Hm) : is defined as the head against which a centrifugal pump has to work.

Hm = hs + hd + $\Sigma H_{Losses\ in\ suction}$ + $\Sigma H_{Losses\ in\ delivery}$

Hm = 2.5 + 15 + 0.313 + 4.345

Hm = 22.158 m

Hm = 23 m

Step:3 Drive Unit

Motor type : 3ϕ – Induction moter

Zero maintenance, cheap, robust construction, high starting torque.

Efficiencies: volumetric = 95 %

Mechanical = 95 %

Pump = 95 %

Overall = 85 %

Moter capacity or power of moter.

[P] = $\frac{Qa \times w \times Hmano \times 10^{-3}}{Overall\ efficiency}$ KW

[P] = $\frac{0.02 \times 1000 \times 9.81 \times 23 \times 10^{-3}}{0.85}$

[P] = 5.3 KW

Selection of standard moter from PSG 5.124

P = 5.5 KW

N = 1440 rpm

Step:4 Design of pump unit (Shaft)

Material : C35 / C40

Shear stress: τ = 30 to 35 N / mm^2

τ = 35 N / mm^2

Failure due to torsional shear,

[Mt] = $\frac{[P] \times 10^3}{\omega}$ $\omega = \frac{2\pi N}{60}$

[Mt] = 36.47 N-m

[Mt] = service factor x 36.47 S. F. = 1.5

[Mt] = 54.70 N-m

[Mt] = 54.70 x 10^3 N-mm

[Mt] = $\frac{\pi}{16} \times ds^3 \times \tau$

ds = 1.5 $\sqrt[3]{\frac{[Mt] \times 16}{\pi \times \tau}}$

ds = 1.5 $\sqrt[3]{\frac{[54.7 \times 10^3] \times 16}{\pi \times 35}}$

ds = 30 mm dia. of shaft.

Step:5 Design of Impeller

Function: The rotating part of a centrifugal pump is called impeller. It consist of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to shaft of an electric motor. The main function of an impeller is to impart kinetic energy to liquid.

Types of Impeller : selection based on viscosity of fluid.

Closed (shrouded) / Semi- closed / Open

Here the fluid is water, which is non-viscous so select closed type of impeller. This type of impeller is sutable for pumping pure liquids.

Material selection : desirable properties

Corrosion resistance, Abrasive resistance, good weldability, stability, cheap and Availability.

Material : carbon steel (C35 / C50)

Elements of impeller: Hub, Blades, Shroudes

Outersize of Impeller (D2) :

U2 = K $\sqrt{2} \times g \times Hmano$

U2 = 0.95 $\sqrt{2} \times 9.81 \times 23$

U2 = 20.18 m/sec

U2 = $\frac{\pi D_2 N}{60}$

D_2 = $\frac{60\times 20.18}{\pi \times 1440} = 0.2676\ m$

D_2 = 260 mm ----- outersize of impeller

D_1 = 130 mm ----- Innersize of impeller

Step:6 Design of casing

Function: The casing of a centrifugal pump is similar to casing of a reaction turbine. It is an air tight passage surrounding the impeller and is designed in such a way that kinetic energy of water discharged at the outlet of the impeller is converted in to pressure energy before the water leaves the casing and enters the delivery pipe.

Type of Casing : Volute

Clearance between impellar and casing:

C = 1 to 2 % D_2

C = 0.01 X 260

C = 2.6 mm

Material of Casing:

Requirments: Anti-corrosive, abrasive, good weldability, castability, cheap.

Mtl: GCI- 25

$\sigma u = 250\ N/mm2$

Factor of safety (n) = 10

Working stress $[\sigma t]$ = 25 N/mm^2

Failure : due to hoop stress

Casing thickness : 10 to 12 mm

Assume thickness tc = 10 mm

$\sigma t = Pmax \frac{D2}{2tc}$ --- based on thin cylinder

Working or normal pressure = Hmano x Wwater

Pd = 23 x 1000 x 9.81 N/m^2 = 0.225 N/mm^2

Pmax = Pd x 1.2 = 0.270 N/mm^2

σt = Pmax x D2/ 2 tc

= 0.270 $\times$ 260/ 2 x 10

σt = 3.51 N/mm2

Diameter of throat.

Qth= Vth x Ath

dth = 73.86 mm

Q:7 *The following specification refers to an EOT Crane.* *20*

Application: ClassII
Load to be lifted: 250 KN
Hosting speed: 4.5 m/min
Maximum lift : 10 m

1. Select sutaible type and size of wire rope for an expected life of 10 months.
2. Select Standard hook, material and design stresses. Check the induced stresses at the most critical sections.
3. Design pulley axel.
4. Design the cross piece.
5. Design the shackle plates.

Solution:

Step:1 Design of Hook

Duty factor = 1.2 ----- for application class II PSG 9.2

Safe load [P] = 250 X 1.2 = 300 KN

[P] = 30 Tonnes

Proof load = 60 tonnes PSG 9.11

We select mild steel, C = 207 mm

Various diamensions of hook are as follows:

A = 2.75C = 570 mm
B= 1.31C =271 mm
D= 1.44 C= 298 mm
E = 1.25 C = 259 mm
F = 207 mm
H = 0.93C = 193 mm
J = 0.75C = 155 mm
K = 0.92C = 190 mm
L = 0.7 C= 145 mm
M = 0.6 C = 124 mm
N = 1.2C = 248 mm
P= 0.5C = 104 mm
R = 0.5C = 104 mm
U = 0.3C = 62 mm
Z = 0.12C = 25 mm
G= 110 mm
M= 110 mm

Checking the hook at the most critical section (2-2)

Area of trapezium (A) = $\frac{1}{2}(bi + bo) \times h$

$(A) = \frac{1}{2}(124 + 50) \times 193$

$(A) = 16.791 \times 10^3 \text{ mm}^2$

Direct tensile stress

$\sigma t = [P] / A = 300 \times 10^3 / 16.791 \times 10^3$

$\sigma t = 17.89 \text{ N/mm}^2$

bi = M = 124 mm

bo = 2 Z = 25 X 2 = 50 mm

h = H = 193 mm

ri = C/2 = 207 / 2 = 103.5 mm

h = ro – ri

ro = ri + h = 103.5 + 193 = 296.5 mm

rn = distance from the center of curvature to the neutral axis.

R = distance from the center of curvature to the centroidal axis.

$$r_n = \frac{\frac{1}{2}(bi+bo)h}{\left[\frac{bi\,ro-bo\,ri}{h}\right]\ln\left(\frac{ro}{ri}\right)-(bi-bo)}$$ PSG 6.3

rn = 171.57

$$R = ri + \frac{h}{3}\left(\frac{bi+2\,bo}{bi+bo}\right)$$

R = 186.31 mm

e = R – rn

e = 186.31- 171.51

e = 14.80 mm

hi = rn – ri

hi = 171.51 – 103.5

hi = 68.01 mm distance of innermost fiber from NA.

ho = ro – rn

ho = 296.5 – 171.51

ho = 124. 99 mm distance of outermost fiber from NA.

Bending Moment (Mb) = [P] X R

= 300 X 10^3 X 186.31

= 55.893 X 10^6 N- mm

Bending stress maximum at outer fiber PSG 6.2

(σb max) outer fiber = $\frac{Mb \times ho}{a \times e \times ro}$

$$= \frac{55.893 \times 10^6 \times 124.99}{16.791 \times 10^3 \times 14.80 \times 296.5}$$

(σb max) outer fiber = 94.813 N/ mm^2

Bending stress maximum at inner fiber PSG 6.2

(σb max) inner fiber = $\frac{Mb \times hi}{a \times e \times ri}$

$$= \frac{55.893 \times 10^6 \times 68.01}{16.791 \times 10^3 \times 14.80 \times 103.5}$$

(σb max) inner fiber = 147.79 N/mm^2

Maximum tensile stress at inner fibre

= Direct Tensile Stress + (σb max) inner fiber

= 17.86 + 147.79 N/mm^2

$(\sigma t\ max)$ inner fiber = 165.65 N/mm^2

Maximum tensile stress at outer fibre

$(\sigma t\ max)$ outer fiber = $(\sigma b\ max)$ inner fiber - Direct Tensile Stress

= 147.79 – 17.86

= 129.93

$(\sigma t\ max)$ outer fiber = 129.93 N/mm^2

Material Selection for Hook

Selecting C-45 as hook material

Yield stress = 36 Kgf / mm^2

σyt = 360 N/mm^2

F. O. S. = 2

$[\sigma t]$ = 180 N/mm^2

as $(\sigma t) < [\sigma t]$

Design is safe in tension.

Checking the Hook at section 3-3

Bending Moment = [P] X R X cos 45^0

= 300 X 10^3 X 186.31 X 0.707

= 39.522 X 10^6 N – mm

$(\sigma b\ max)$ inner fiber = $\frac{Mb \times hi}{a \times e \times ri} \times \cos 45^0$

= 147.79 $\times$ 0.707

= 104.48 N/mm^2

Direct tensile stress

$\sigma t = [P] / A \times \cos 45^0$

= 17.86 $\times$ 0.707

= 12.627 N/mm^2

Total tensile load at inner fiber

= Direct tensile stress + $(\sigma b\ max)$ inner fiber

= 12.627 + 104.48

= 117.107 N/mm^2

Shear stress (τ) = $[P] / A \times \cos 45^0$

= 12.627 N/mm^2

Principal Stress = $\frac{\sigma}{2} \sqrt{(\frac{\sigma}{2})^2 + (\tau)^2}$

= $\frac{104.48}{2} \sqrt{(\frac{104.48}{2})^2 + (12.627)^2}$

= 105.98 N/mm^2

Maximum shear stress = $\sqrt{(\frac{\sigma}{2})^2 + (\tau)^2}$

= $\sqrt{(\frac{104.48}{2})^2 + (12.627)^2}$

= 53.744 N/mm^2

Design shear Stress $[\tau]$ = 0.577 $\times$ σt

= 0.577 $\times$ 180

= 103.86 N/mm^2

As design shear stress is greater than induced shear stress design is safe.

Step:2 Rope Design

Steel wire ropes are extensively used in hoisting machinery as flexible lifting appliances.

Material: Steel

σt = 130 to 200 Kgf/mm^2

Assume σt = 180 Kgf/cm^2

Type of rope : Ordinary

Size: 6 X 37

Design criteria: Finnding rope size based on strength and check for life.

A = $\frac{F}{\frac{\sigma u}{n} - \frac{d}{Dmin}\left(\frac{dw}{d}\right)E'}$ PSG 9:1

F = [P]/ No of falls = 7500 Kgf

A = 5.227 cm^2

Effective area = 0.4 X $\frac{\pi}{4}$ d^2

d = 4.07 cm = 40.79 mm

Selection of standard wire dia. PSG 9.4

d = 41 mm

Check it for life

N = $\frac{0.4 \times Z}{a \times \beta \times z_2}$

N = $\frac{0.4 \times 214 \times 1000}{1000 \times 0.5 \times 4}$

N = 42.8 months

As actual life is greater than expected life, design of wire rope is safe.

Step:3 Design of pulley axle.

For wire rope dia. =41

L1 = A + spacer thickness (ts)

= 155 mm

L2 = $\frac{A}{2} + ts + \frac{(ts+tc)}{2}$

= 96.50 mm

Axle Material : C-45

σyt = 360 N/mm^2

f. o. s. = 4

[σt] = 90 N/mm^2

[τ] = 45 N/mm^2

Selection of axle dia. based on bending failure and check it for shear failure.

Maximum bending moment = 2F $\times$ L2

= 2 $\times$ 75 $\times$ 10^3 $\times$ 96.50

= 14.475 $\times$ 10^6 N-mm

(B. M.)max = Z $\times$ [σt]

14.475 $\times$ 10^6 = π/32 $\times$ d^3 $\times$ 90

d = 117.88 mm

Take d = 120 mm

$(\tau) = \frac{2F}{Shear\ area}$

$(\tau) = \frac{2 \times 75 \times 10^3}{\frac{\pi}{4}(120^2)}$

$(\tau) = 13.26 \quad N/mm^2$

As $(\tau) < [\tau]$

Design is safe.

Step:4 Design of Cross- Piece

Material: C-40

$\sigma yt = 360 \quad N/mm^2$

F. O. S. = 4

$[\sigma t] = 90 \quad N/mm^2$

$[\tau] = 45 \quad N/mm^2$

di = G = 110 mm

do = 115 mm

L1 = 2A + 3ts

= 173 mm

h= 1.75 X 110

= 192.5 thickness of cross- piece

$\sigma b = Mb / Z \leq [\sigma t]$

$80 = \frac{4.116 \times 10^6}{\frac{1}{6}(b - 95) \times (95)^2}$

b = 150 mm width of cross-piece

Step:5 Design of Shackle Plate

Material: C-40

$\sigma yt = 360 \quad N/mm^2$

F. O. S. = 4

$[\sigma t] = 90 \quad N/mm^2$

$[\tau] = 45 \quad N/mm^2$

H2 = h/2 + margin

= 192.5/2 + 5

= 101.25 mm

H1 = movable sheave dia./2 + h/2 + cover height + margin

= 1000/2 + 192/2 + 90 + 5

= 691.25 mm

Width of plate = 1.5 X d

= 157.5 mm