

Module-03

Design of a General Purpose Industrial Helical Gear Reduction Unit- Part I

Lecture-11

Gear Unit Design – Selection of Stage Ratios, Pinion & Gear Teeth Numbers

Outline of the Lecture

- **Description of a Gear Reduction Unit**
- **A typical Design Problem**
- **Constructional features of Gear Units**
- **Steps in Design**
- **Selection of Stage Ratios, Pinion and Gear Teeth Numbers**

A Typical General Purpose Industrial Gear Box (Speed Reducer):

Intermediate Shafts

Output Shaft

One Input Shaft

One Output Shaft

One Intermediate Shafts

Bottom Housing

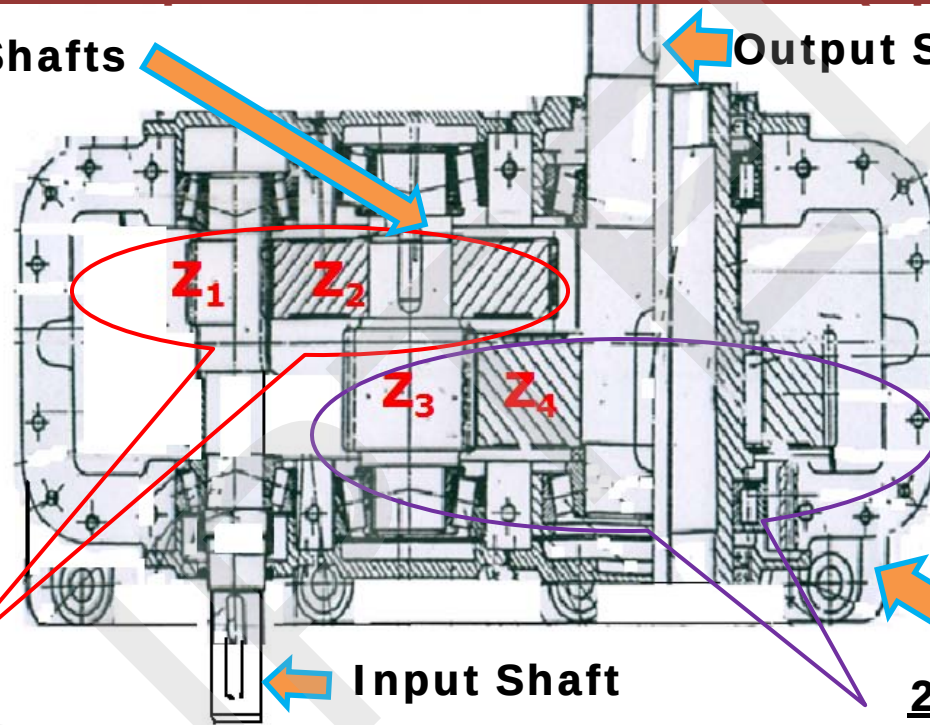
1st. Stage

Input, Helical Gear Stage

Input Shaft

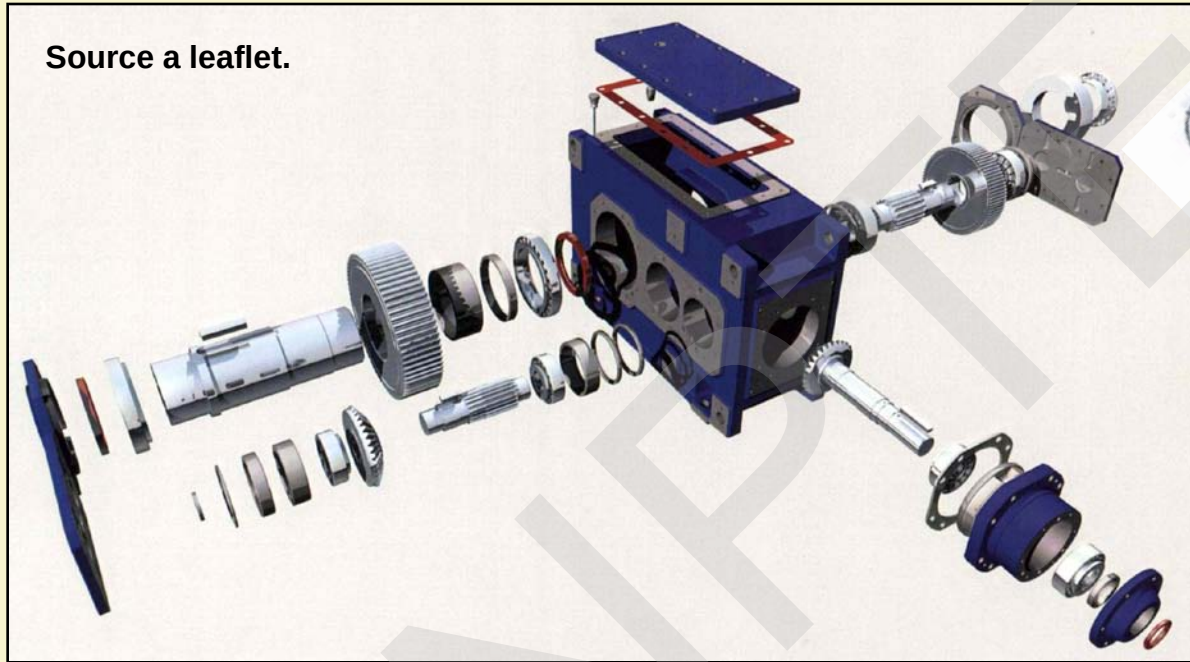
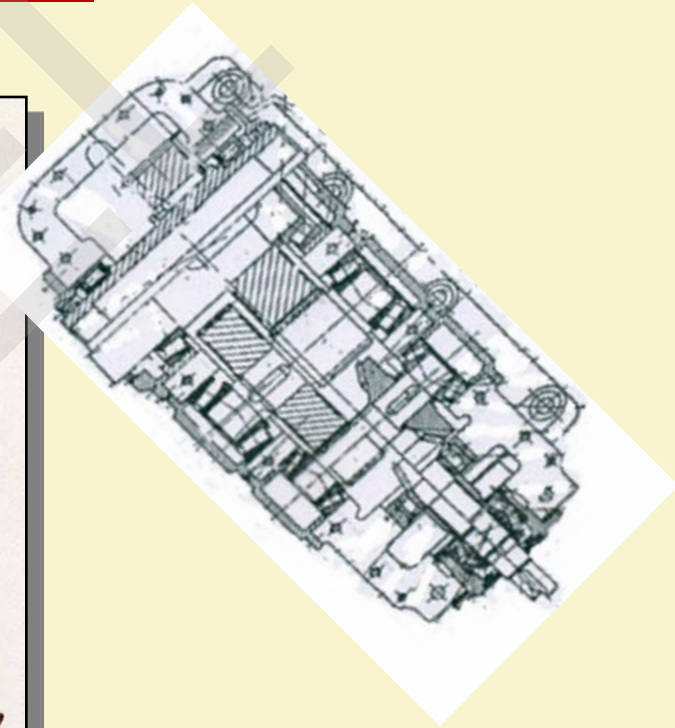
2nd. Stage

Output, Helical Gear Stage

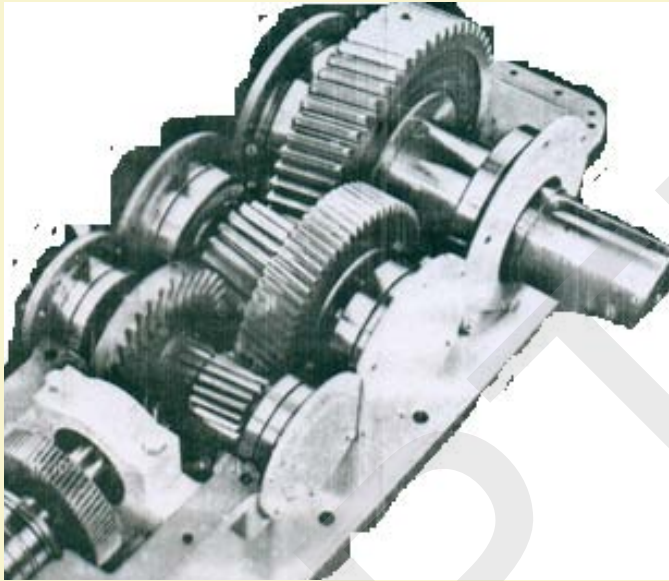


Welded Housing- (Single Piece) – GB Exploded View (3-Stage- Bevel & Helical)

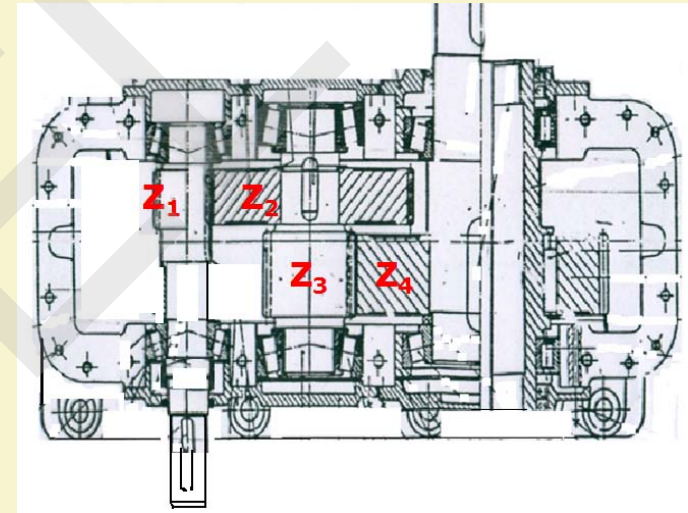
Source a leaflet.



A Typical Gear Box Design Problem



Photographic plan view is of 4-stage gear box
(Top cover removed).



Assembled plan view is of 2-stage gear box.
(Top Cover removed).

Problem : A helical gear reduction unit has to transmit 30 Nm input torque at 1500 rpm with a total reduction of about 37 to 40.

A Typical Gear Box Design Problem (Contd....) :

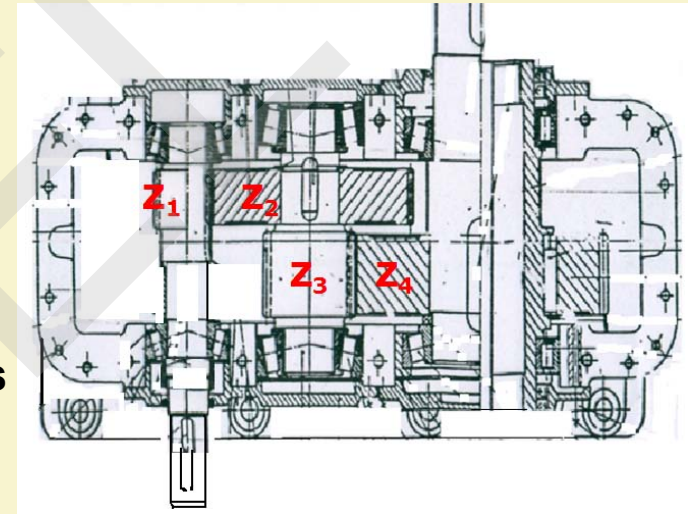
At starting the torque may go as high as 200 % and also there is medium shock loads during operation.

The material for pinion is EN 19A and for gear wheel it is EN 18A.

The gear box may be an ordinary industrial class unit preferably with uncorrected gears.

It is of continuous duty (with medium shock) and overhauling time is Two years.

(Alternatively -the bearing life should not be below 10,000 hours).



Assembled plan view is of Two-stage gear box.

Essential Steps in Designing a Gear Box :

Step-1

Selection of stages, Gear Ratios
& Teeth numbers.

Step-2

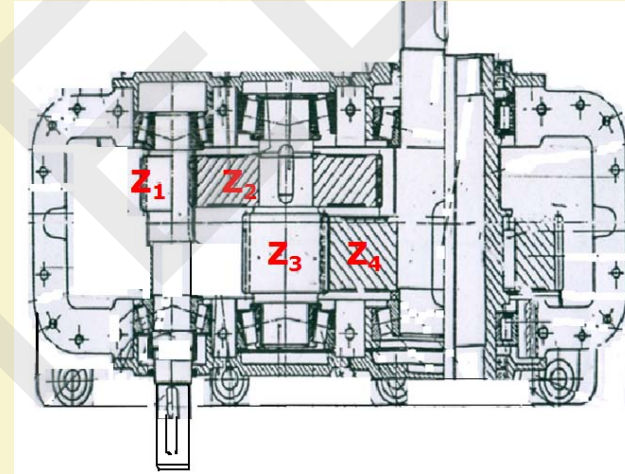
Design of Gears.

Step-3

First layout is made.

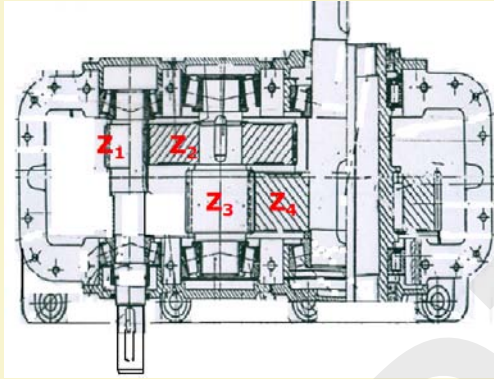
Step-4

Rough shape to the shafts
are given.



Assembled plan view is of
a Two stage gear box.

Essential Steps in Designing a Gear Box (Contd...):



Assembled plan view is of a Two stage gear box.

Step-5 Bearings are selected preliminarily.

Step-6 After putting the bearings in layout load calculations become possible.

Step-7 Lives of all bearings are estimated.

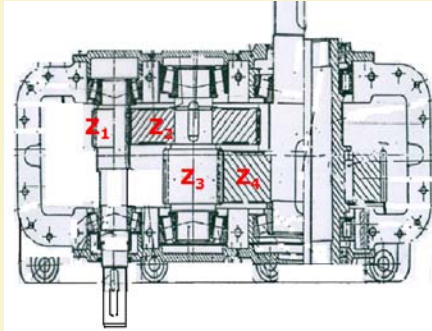
Step-8 If estimated lives are not satisfactory then a new set of bearings are chosen.

Last Step If necessary shaft(s) dimensions are also altered .

Further, gear design may need to be altered.

Design Steps:

Step-1



Assembled plan view is of
a Two stage gear box.

Knowing the direction of Input & Output Transmission ratios, type of Gears & teeth numbers are selected.

Rules and Limitations:

Stage ratio not more than 6.

Number of teeth, $Z_{cr-min} \geq 2 / \sin^2 (\text{Pressure Angle})$.

Note: Normally helical gears are chosen rather than straight tooth spur gears.

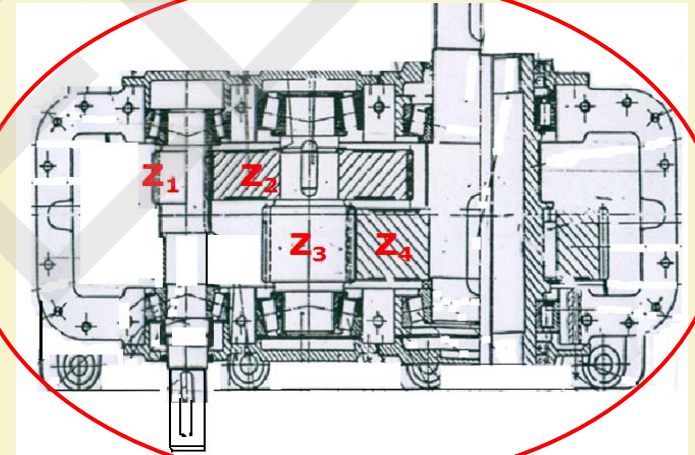
Therefore, we consider formative number of teeth: $Z' = Z / \cos^3 \beta$,
 Z being actual number of teeth.

Where, β is helix angle which is commonly between 12° to 20° for single helical gear .

For which considering $Z' = 17$, Z may be taken as 16 to 14 respectively.

Considering cutter geometry, speed of gears and manufacturing methods this number may further be lowered.

A Typical General Purpose Industrial Gear Box (Speed Reducer):



Z_1 - Number of Teeth of 1st. Stage Pinion.

Z_2 - Number of Teeth of 1st. Stage Gear.

Z_3 - Number of Teeth of 2nd. Stage Pinion.

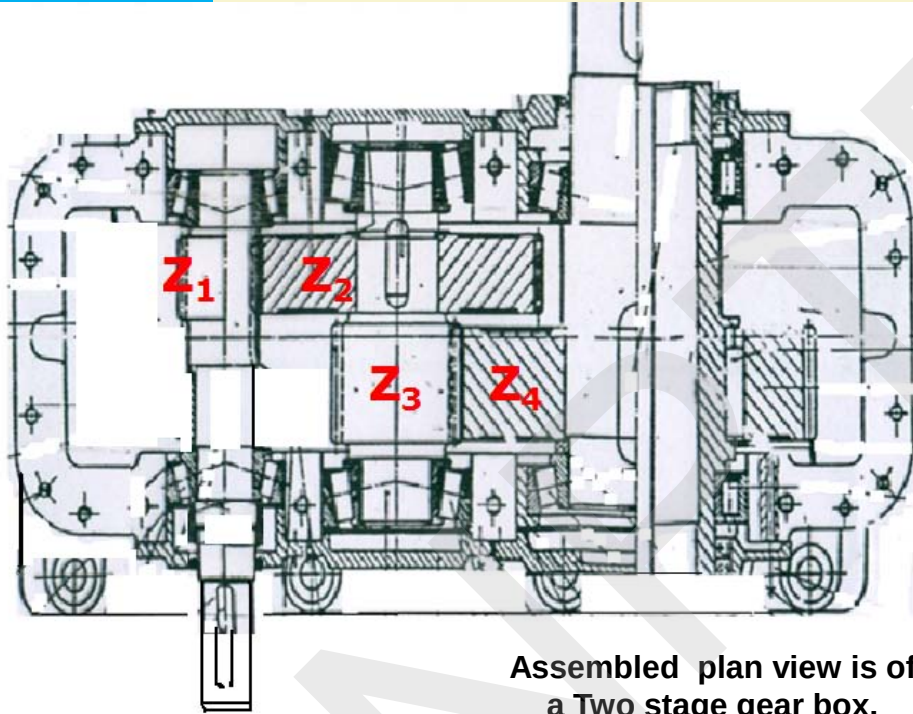
Z_4 - Number of Teeth of 2nd. Stage Gear.

Design Steps:

1st. Step (Contd).

- Selection of number of stages for a
Total Transmission Ratio
 $i_t = 37 \text{ to } 40$.

In choosing the numbers of teeth and stage ratios, not only the size optimization is considered, but also the roundness in centre distances with uncorrected gears is taken care* of:



Assembled plan view is of
a Two stage gear box.

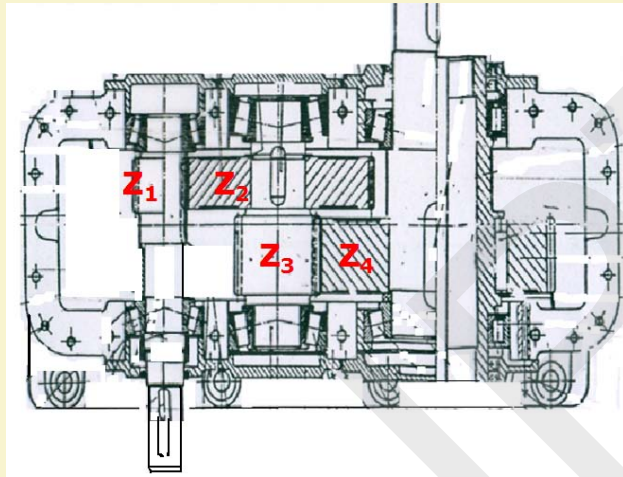


Design Steps:

1st. Step (Contd).

➤ Selection of number of stages for a Total Transmission Ratio $i_t = 37$ to 40.

Considering two stage reduction the numbers of teeth of pinions and gears were selected as follows:



Assembled plan view is of a Two stage gear box.

1st. Stage:
$$i_1 = \frac{Z_2}{Z_1} = \frac{81}{17} = 4.76$$

2nd. Stage:
$$i_2 = \frac{Z_4}{Z_3} = \frac{131}{16} = 8.19$$

Therefore, total ratio becomes:

$$i_t = i_1 \times i_2 = \frac{Z_2}{Z_1} \times \frac{Z_4}{Z_3} = \frac{81}{17} \times \frac{131}{16} = 4.76 \times 8.19 = 39.01$$

This is acceptable.

Thank You



IIT KHARAGPUR



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Module-03

Design of a General Purpose Industrial Helical Gear Reduction Unit- Part I

Lecture-12

Gear Unit Design – First Stage Pinion & Gear Design- I (Module on Beam Strength Basis)

Outline of the Lecture

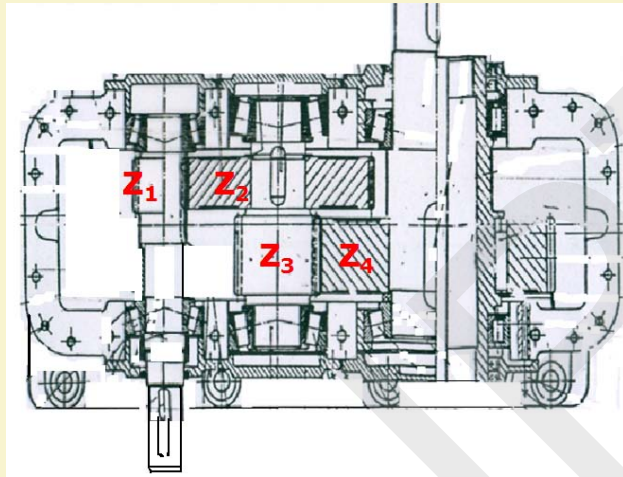
- **Selection of Stage Ratios , Gear and Pinion Teeth Numbers**
 - **Determination of Design Torque**
 - **Material properties and material selection**
 - **In a gear and pinion set which one is weaker and to be designed**
 - **Choosing other parameters of Lewis formula**
 - **Module Selection (Preliminary / 1st Attempt).**

Design Steps:

1st. Step (Recapitulation).

➤ Selection of number of stages for a Total Transmission Ratio $i_t = 37$ to 40.

Considering two stage reduction the numbers of teeth of pinions and gears were selected as follows:



Assembled plan view is of a Two stage gear box.

1st. Stage:
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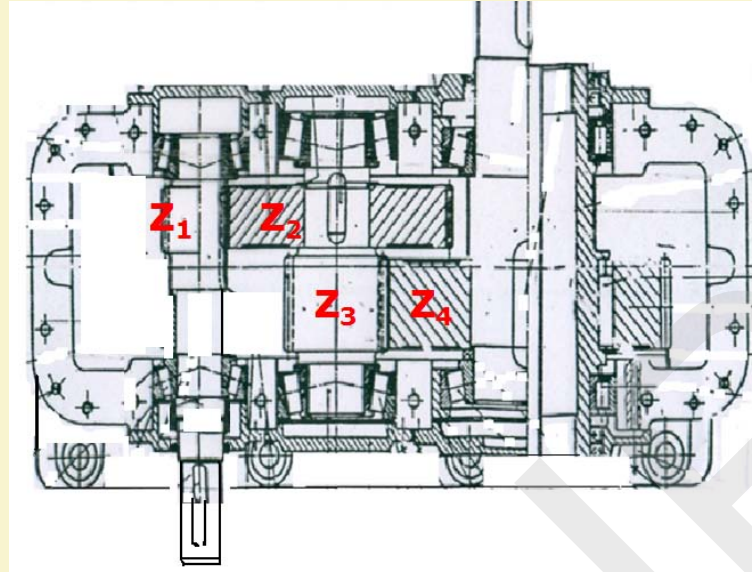
$$i_t = i_1 \times i_2 = \frac{Z_2}{Z_1} \times \frac{Z_4}{Z_3} = \frac{81}{17} \times \frac{131}{16} = 4.76 \times 8.19 = 39.01$$

This is acceptable.

Gear Design (1st. Stage):

Step-2

In next step gears are designed:



Assembled plan view is of 2-stage gear box.

1st. Stage

Module Estimation on Strength Basis:

$$m_n = \sqrt[3]{\frac{2T \cos \beta}{S_o C_v \psi Y Z C_w}}$$

Input Torque (Nominal):

$$T_i = \frac{Power}{2\pi\omega_i}$$

Gear Design (1st. Stage):

Step-2

In next step gears are designed:

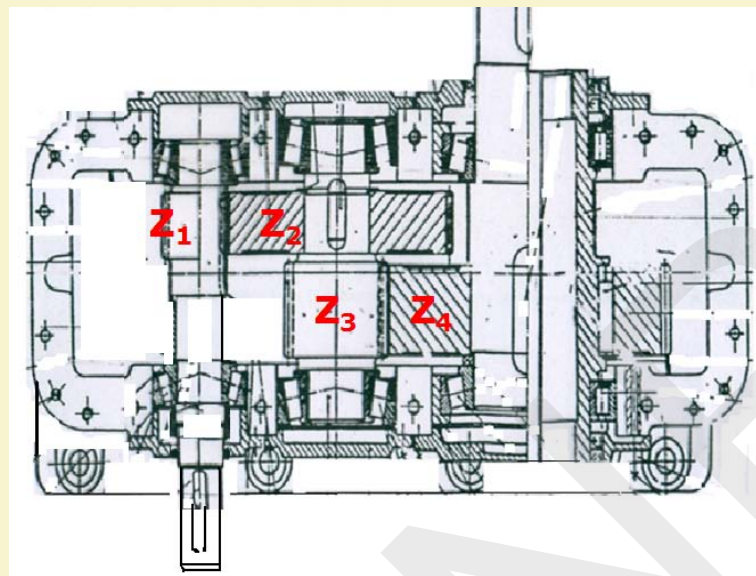
1st. Stage

In this problem input continuous torque = 30 Nm.

$$\text{i.e. } T_i = 30 \text{ Nm}$$

As starting torque is 200%
the design torque may be taken as:

$$T_{id} = 2 \times T_i = 60 \text{ Nm}$$



Assembled plan view is of 2-stage gear box.

Gear Design (1st. Stage):

Step-2

- Suggested Materials for Pinion , Gear & Shaft

	<u>Pinion</u>	<u>Gear</u>	<u>Shaft</u>
	EN 19	EN 18A	EN 8
Ultimate Strength :	$S_u = 940 \text{ MPa}$	$S_u = 860 \text{ MPa}$	$S_u = 570 \text{ MPa}$
Yield Strength :	$S_y = 600 \text{ MPa}$	$S_y = 550 \text{ MPa}$	$S_y = 280 \text{ MPa}$
BHN (Hardened and Tempered) :	300-340	250-300	

Alloy Steel Equivalent Grades

International Standard	BS	DIN	IS	EN	SAE/AISI
EN18A	530A40	37Cr4	40Cr1	EN18	5140
EN18	530A40	37Cr4	40Cr1	EN18A	5140
EN19	709M40	42Cr4Mo2	40Cr4Mo3	EN19	4140,4142
EN19C	709M40	-	40Cr4Mo3	EN19C	4140,4142
EN24	817M40	34CrNiMo6	40NiCr4Mo3	EN24	4340
Carbon Steel					
EN8	080A40	CK45	45C8	EN8	1040,1045

Gear Design (1st. Stage):

Step-2

Materials &- Chemical Composition

EN 19

C(%)	0.35~ 0.45	Si(%)	0.10~0.35	Mn(%)	0.50~ 0.80	P(%)	≤0.035
S(%)	≤0.050	Cr(%)	0.90~1.50	Mo(%)	0.20~ 0.40		

	C	Mn	Si	S	P	Cr	Ni
EN18	.35 – .45	.60 – 1.00	.10 – .35	.050 MAX	.050 MAX	-	-

Standard	Grade	C	Mn	P	S	Si
BS 970	EN8/08 0M40	0.36- 0.44	0.60- 1.00	0.05	0.005	0.10- 0.40

Gear Design (1st. Stage):

Step-2 **Materials Heat Treatment**

Consider **EN19**

which is equivalent material with **AISI 4140**, **DIN 42CrMo4**, **IS 42CrMo4**,
EN 1.7225, **GB 42CrMo**, **JIS SCM440C** alloy engineering steel

Thermal processing temperature / °C | Forging: 1050 ~ 850

The heat treatment temperature / °C | Annealing: 680 to 720

The heat treatment temperature / °C | normalizing: 840 to 880

The heat treatment temperature / °C | quenching: 820 ~ 850 830 ~ 860 oil water

The heat treatment temperature / °C | Tempering: 540 to 680

Hardness after annealing $\leq$ HBS: 241

Gear Design (1st. Stage):

Step-2

Design of Gears – Module estimation (Contd.):

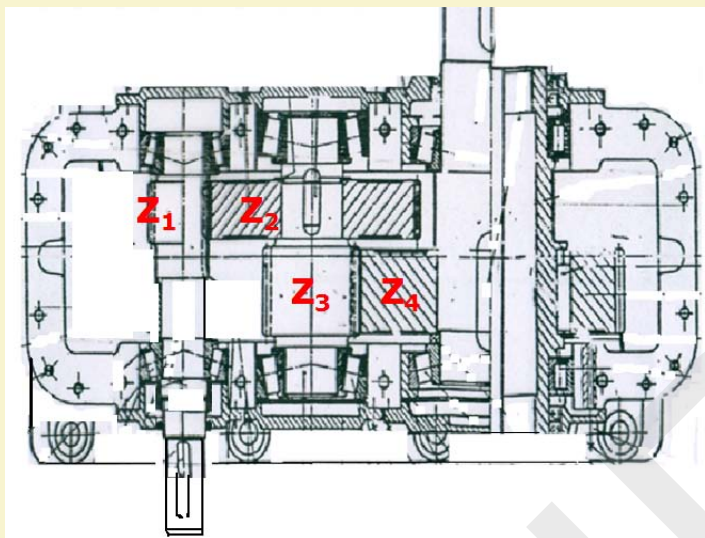
1st. Stage (Cont.)

In a gear pair which one to be designed?

$$m_n = \sqrt[3]{\frac{2T \cos \beta}{S_o c_v \psi Y Z} \frac{1}{c_w}}$$

Compare $(S_o Y)_{gear}$ with $(S_o Y)_{pinion}$

$$Z_{Pinion} = 17 \qquad Z_{Gear} = 81$$



Assembled plan view is of 2-stage gear box.

Gear Design (1st. Stage):

Step-2

Allowable strength of selected material in gear design.

It is to be noted that in the following Lewis formula for bending strength of gear tooth.

$$m_n = \sqrt[3]{\frac{2T \cos \beta}{S_o C_v \psi Y Z C_w}}$$

$$\frac{S_o C_v}{C_w}$$

Is the ultimate allowable strength for the particular gear considering its dynamic and lubrication conditions in operations,

Now S_o is taken as certain proportion of either ultimate strength or yield strength of the selected material.

S_o = Yield strength/2.5 for gear design is often considered.

Gear Design (1st. Stage):

Step-2

$$(S_o)_{pinion} = (S_y)_{EN19A} / 2.5 = 240 \text{ MPa} = 240 \times 10^6 \text{ Pa}$$
$$(S_o)_{gear} = (S_y)_{EN18A} / 2.5 = 220 \text{ MPa} = 220 \times 10^6 \text{ Pa}$$

Considering nominal helix angle $\beta = 12^\circ$

$$Y_{Pinion} = 0.484 - 3.28 / Z' = 0.484 - 3.28 / (17 / \cos^3 12^\circ) = 0.303$$

$$Y_{Gear} = 0.484 - 3.28 / Z' = 0.484 - 3.28 / (81 / \cos^3 12^\circ) = 0.446$$

$$(S_o Y)_{gear} = 220 \times 0.446 = 98.12$$

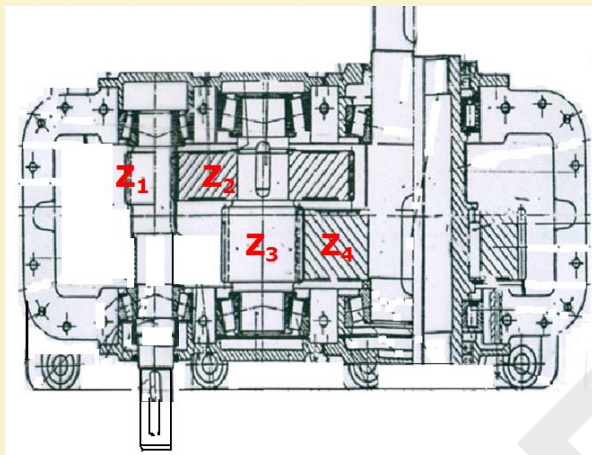
$$(S_o Y)_{pinion} = 240 \times 0.303 = 72.72$$

$(S_o Y)_{pinion} < (S_o Y)_{gear}$ means pinion is weaker and it is to be designed.

Gear Design (1st. Stage):

Step-2

Design of Gears – Module estimation (Contd.):



Assembled plan view is of 2-stage gear box.

1st. Stage (Cont.)

Pinion shaft is the input shaft. Therefore:

$$T = 60 \text{ Nm} \quad \beta = 12^\circ \quad \psi = 20$$

Considering accurately cut gear and assuming module may be 2.5 mm.

$$V = \frac{\pi(17 \times 2.5) \times 1500}{\cos 12^\circ \times 1000 \times 60} = 3.41 \text{ m/sec}$$

$$c_v = \frac{4.5}{4.5 + V} = \frac{4.5}{4.5 + 3.41} = 0.57$$

Gear Design (1st. Stage):

Step-2

Substituting the values in-

$$m_n = \sqrt[3]{\frac{2T \cos \beta}{S_o C_v \psi YZ C_w}}$$

C_w Being 1.25 for oil lubrication with regular inspection.

$$= \sqrt[3]{\frac{2 \times 60 \times 0.978}{\frac{240 \times 10^6 \times 0.57}{1.25} \times 20 \times 0.303 \times 17}} = 0.218 \times 10^{-2} \text{ m} = 2.18 \text{ mm}$$

Therefore, 1st stage normal module (in 1st Attempt / round) may be taken as **2.5 mm**

Thank You



Module-03

Design of a General Purpose Industrial Helical Gear Reduction Unit- Part I

Lecture-13

Gear Unit Design – Failure of Gear Tooth

(Probable Dynamic Load and Wear Load Capacity)



Outline of the Lecture

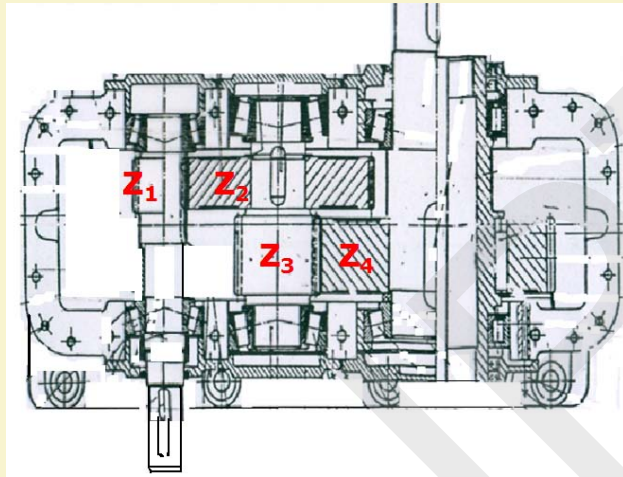
- **Failure of gear tooth.**
 - **Wear load capacity at tooth contacts.**
 - **Estimation of Probable Dynamic Load .**
 - **Comparison of Wear load capacity with probable dynamic load for infinite life of gears.**

Design Steps:

1st. Step (Recapitulation).

➤ Selection of number of stages for a Total Transmission Ratio $i_t = 37$ to 40.

Considering two stage reduction, with input rpm 1500, the numbers of teeth of pinions and gears were selected as follows:



Assembled plan view is of a Two stage gear box.

1st. Stage:
$$i_1 = \frac{Z_2}{Z_1} = \frac{81}{17} = 4.76$$

2nd. Stage:
$$i_2 = \frac{Z_4}{Z_3} = \frac{131}{16} = 8.19$$

Therefore, total ratio becomes:

$$i_t = i_1 \times i_2 = \frac{Z_2}{Z_1} \times \frac{Z_4}{Z_3} = \frac{81}{17} \times \frac{131}{16} = 4.76 \times 8.19 = 39.01$$

This is acceptable.

Gear Design (1st. Stage):

Step-2

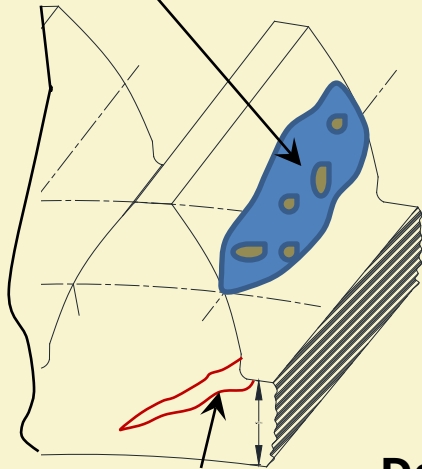
Suggested Materials for Pinion & Gear (Recapitulation)

For General Purpose Industrial Gear unit recommended materials are:

	<u>Pinion</u>	<u>Gear</u>	<u>Shaft</u>
	EN 19A	EN 18A	EN 8
Ultimate Strength :	$S_u = 940 \text{ MPa}$	$S_u = 860 \text{ MPa}$	$S_u = 570 \text{ MPa}$
Yield Strength :	$S_y = 600 \text{ MPa}$	$S_y = 550 \text{ MPa}$	$S_u = 280 \text{ MPa}$
BHN (Hardened and Tempered) :	300-340	250-300	

Failure of Gear Teeth

Failure- Contact Surface



Failure- Crack

Gear Tooth may fail at root due to excessive bending stress.

The bending stress is exaggerated due to dynamic load.

Ultimately fatigue failure may occur.

Next tooth surface may fail due to wear and pitting.

Therefore, surface hardness usually increased to withstand at high contact pressure.

In designing gear tooth, all these issues are considered carefully.

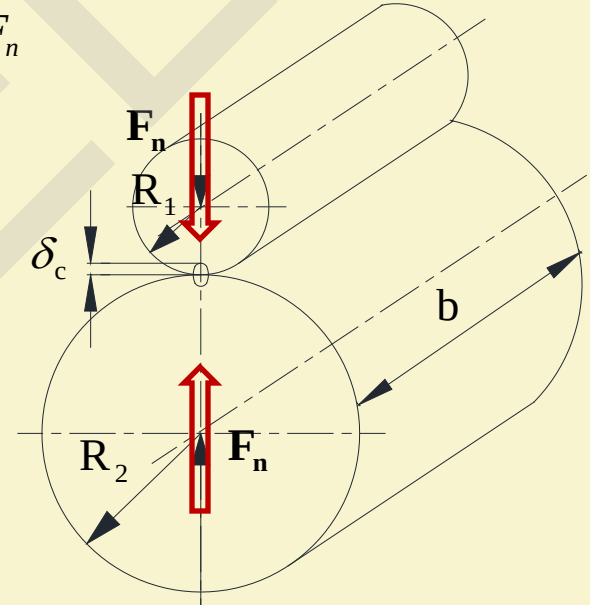
Depending on loads, loading pattern and severity in operation 'through hardening' or 'case carburizing' steel is selected.

Wear Load Capacity of Teeth Pair

When two cylindrical bodies are in contact with force F_n contact deformation (δ_c) occurs:

According to Hertz Contact theory,
Developed contact stress (σ_c) is expressed as:

$$\sigma_c = \sqrt{\frac{F_n \left(\frac{1}{R_1} + \frac{1}{R_2} \right)}{\pi (1 - \nu^2) b \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}}$$



Where, ν Poisson's ratio, And E_1 & E_2 Modulus of Elasticity.

Wear Load Capacity of Teeth Pair

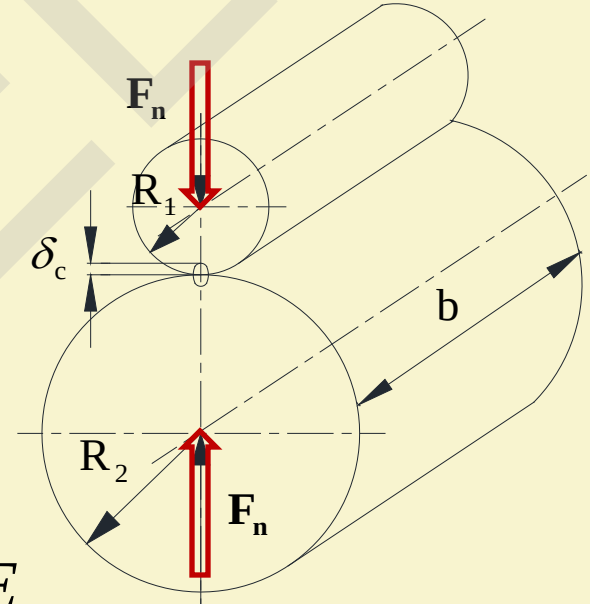
Considering $\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$

And as both materials are steel

$$\frac{1}{E} = \frac{1}{E_1} + \frac{1}{E_2}$$

Also, Poisson's ratio (ν) for steel = 0.3

$$\sigma_c = \sqrt{\frac{F_n \left(\frac{1}{R_1} + \frac{1}{R_2} \right)}{\pi (1 - \nu^2) b \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}} \Rightarrow \sigma_c^2 = 0.35 \frac{F_n E}{b R}$$



Wear Load Capacity of Teeth Pair

In involute tooth pair contact

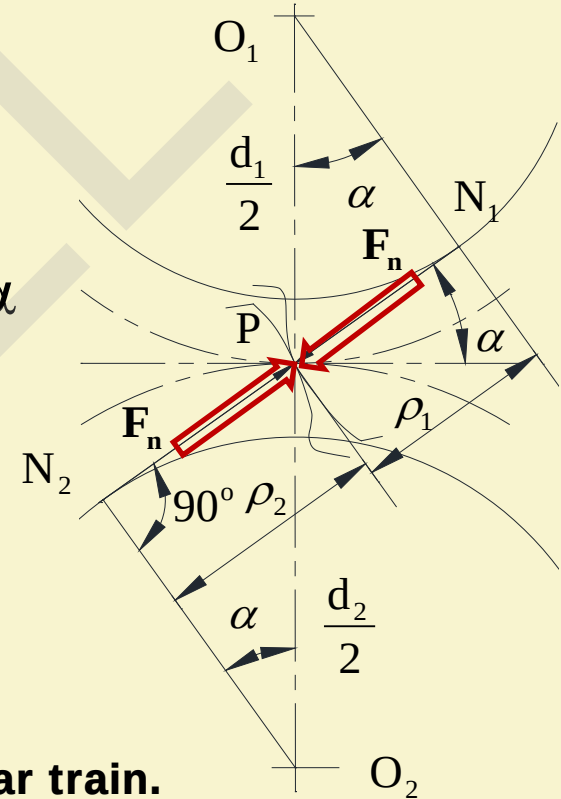
$$R_1 = \rho_1 = \frac{d_1}{2} \sin \alpha \quad \text{and} \quad R_2 = \rho_2 = \frac{d_2}{2} \sin \alpha$$

Therefore,

$$R = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2} = \frac{d_1 \sin \alpha}{2(1+i)} \times i$$

Where, $i = d_2/d_1 = Z_2/Z_1$ or, $= Z_g/Z_p$,

the 'Transmission Ratio', Gear ratio in simple gear train.



Wear Load Capacity of Teeth Pair

Substituting in

$$\sigma_c = \sqrt{\frac{F_n \left(\frac{1}{R_1} + \frac{1}{R_2} \right)}{\pi(1-\nu^2)b \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}}$$

and rearranging,

$$F_n = \left[\frac{\sigma_c^2 \sin \alpha}{1.4} \left(\frac{1}{E_1} + \frac{1}{E_2} \right) \right] \left[\frac{2i}{1+i} \right] d_1 b$$

Wear Load Capacity of Teeth Pair

Expressing in terms of 'Wear load carrying capacity (F_w)' in relation to allowable compressive endurance limit (S_{es}) of the gear and pinion materials-

$$F_n = \left[\frac{\sigma_c^2 \sin \alpha}{1.4} \left(\frac{1}{E_1} + \frac{1}{E_2} \right) \right] \left[\frac{2i}{1+i} \right] d_1 b$$

is expressed as: $F_w = K Q_g d_{pp} b$

Where $K = \left[\frac{S_{es}^2 \sin \alpha}{1.4} \left(\frac{1}{E_1} + \frac{1}{E_2} \right) \right]$ and $Q_g = \left[\frac{2i}{1+i} \right]$, function of relative size of gears.

$d_{pp} = d_1$ being the pitch circle diameter of pinion.

K Is available in tabular form (against surface hardness) in machine design and gear data books.

Wear Load Capacity of Teeth Pair

In $F_w = K Q_g d_{pp} b$, Q_g can also be expressed as: $Q_g = \left[\frac{2 Z_g}{Z_p + Z_g} \right]$

Now as $d_{pp} = Z_p m_n$ and $d_{pg} = Z_g m_n$ then F_w can also be expressed as:

$$F_w = K Q_p d_{pg} b \quad \text{Where, } Q_p = \left[\frac{2 Z_p}{Z_p + Z_g} \right]$$

$d_{pg} = d_2$ being the pitch circle diameter of gear.

In case of helical gear,
$$F_w = \frac{K Q_p d_{pg} b}{\cos \beta} = \frac{K Q_g d_{pp} b}{\cos \beta}$$

Where, d_{pg} and d_{pp} are pcd of helical gear and pinion respectively.

Probable Dynamic Load at Tooth Contact

Probable Dynamic Load $F_d = F_n + F_i$

Where, F_i is increase in load over the normal load.

Buckingham proposed a detail expression for F_i considering the accuracy and manufacturing errors in gear.

However, $F_d = \frac{F_n}{C_v}$ and $1.15F_d \leq F_w$ are in common practice for designing ordinary purpose industrial gear unit considering infinite life of gears.

Thank You



Module-03

Design of a General Purpose Industrial Helical Gear Reduction Unit- Part I

Lecture-14

Gear Unit Design -

1st. Stage Pinion & Gear Design-II

(Probable Dynamic Load and Wear Load Capacity
and Finalizing 1st. Stage Pinion & Gear set)

Outline of the Lecture

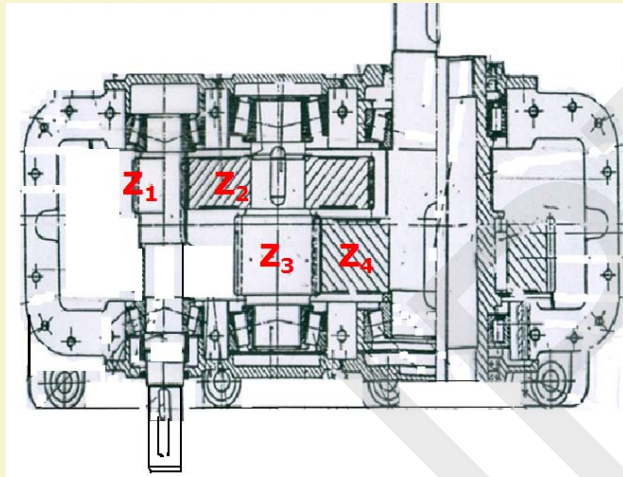
- Design of 1st. Stage pinion and gear- Estimation of Wear Load Capacity
 - Estimation of Probable Dynamic Load at Tooth Contact.
 - Comparison of Wear Load Capacity and Probable Dynamic Load
 - Finalizing 1st. Stage Module- (Considering Bearing ID)
- 2nd. Stage Module Calculation
 - Finalizing Gear Dimensions/Data

Design Steps:

1st. Step (Recapitulation).

➤ Selection of number of stages for a Total Transmission Ratio $i_t = 37$ to 40.

Considering two stage reduction, with 1500 rpm input speed, the numbers of teeth of pinions and gears were selected as follows:



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Which is accepted.

Gear Design (1st. Stage):

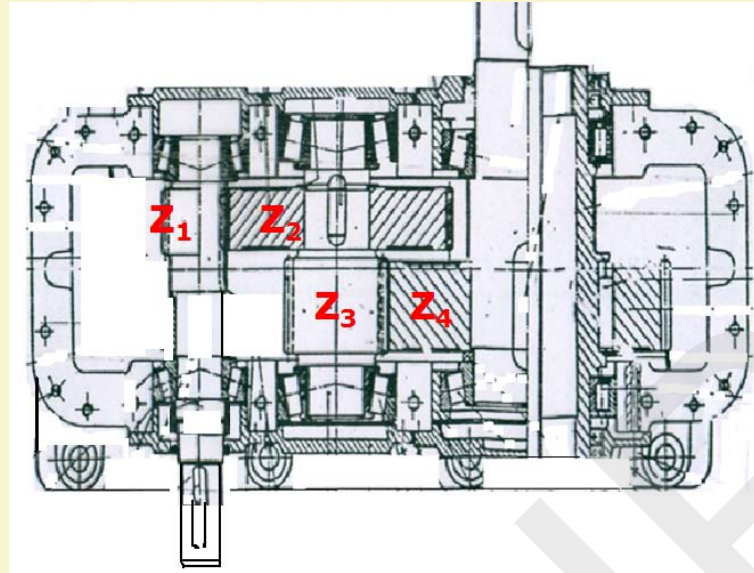
Step-2

Suggested Materials for Pinion & Gear (Recapitulation)

	<u>Pinion</u>	<u>Gear</u>	<u>Shaft</u>
	EN 19A	EN 18A	EN 8
Ultimate Strength :	$S_u = 940 \text{ MPa}$	$S_u = 860 \text{ MPa}$	$S_u = 570 \text{ MPa}$
Yield Strength :	$S_y = 600 \text{ MPa}$	$S_y = 550 \text{ MPa}$	$S_u = 280 \text{ MPa}$
BHN (Hardened and Tempered) :	300-340	250-300	

Gear Design (1st. Stage):

Step-2 Design of Gears (Contd.): 1st. Stage (Cont.) Verification for wear load:



Assembled plan view is of 2-stage gear box.

Selected normal module (on beam strength basis) for 1st. Stage is 2.5 mm.

$$F_w = K Q_g d_{pp} b$$

$$Q_g = \left[\frac{2 Z_g}{Z_p + Z_g} \right] = \frac{2 \times 81}{81 + 17} = 1.653$$

$$b = 2.5 \times 20 \times 10^{-3} = 5 \times 10^{-2} \text{ m}$$

Gear Design (1st. Stage):

Step-2

Wear load capacity at tooth contact- 1st. stage:

$$d_{pp} = \frac{17 \times 2.5}{0.978} \times 10^{-3} = 43.456 \times 10^{-3} \text{ m}$$

For 300 BHN in gear and 350 BHN in pinion $K=1.8 \times 10^6$ Pas for 20° Involute.

$$F_w = K Q_g d_{pp} b$$

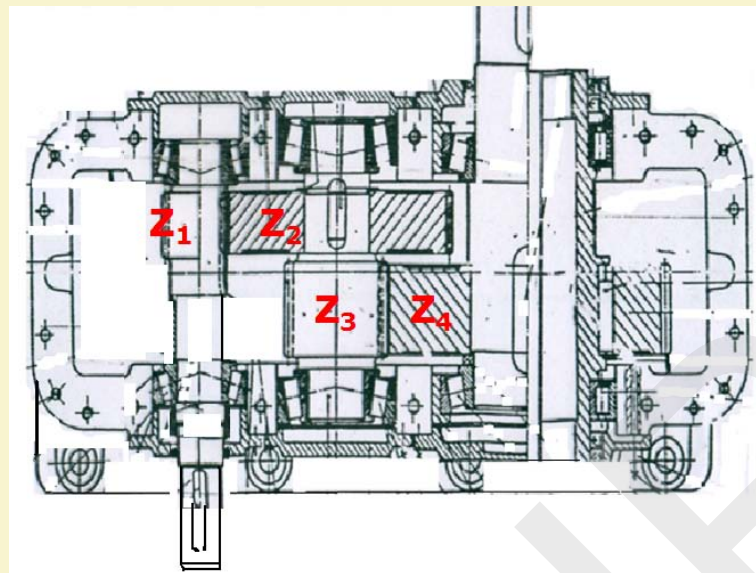
$$F_w = 1.8 \times 10^6 \times 1.653 \times 43.456 \times 10^{-3} \times 50 \times 10^{-3} = \mathbf{6465 \text{ N}}$$

Gear Design (1st. Stage):

Step-2

Design of Gears (Contd.):

1st. Stage (Cont.)



Assembled plan view is of 2-stage gear box.

Probable dynamic load:

$$F_d = F_n / c_v$$

$$F_t = \frac{2T}{d_{pp}} = \frac{2 \times 60}{43.456} \times 10^3 = 2761.4 \text{ N}$$

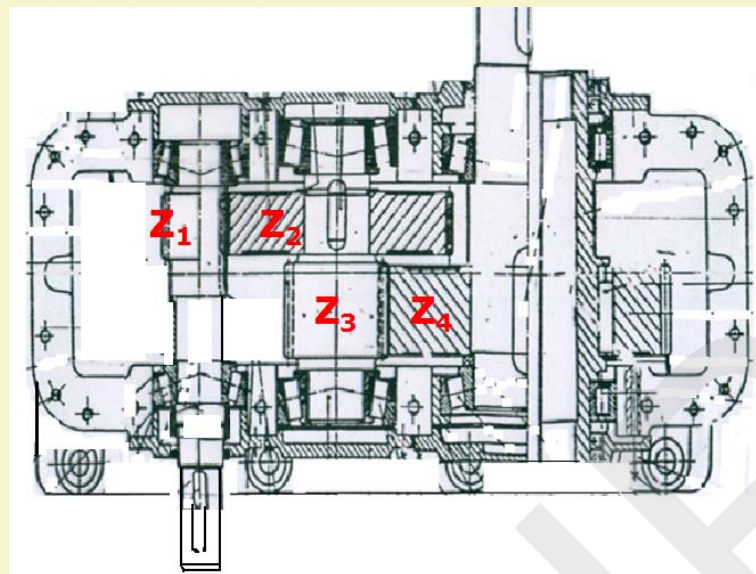
$$F_n = F_t \sec \beta \sec \alpha = 2761.4 \times \sec 12^\circ \times \sec 20^\circ \\ = 2761.4 \times 1.0223 \times 1.064 = 3004.15 \text{ N}$$

Gear Design (1st. Stage):

Step-2

Design of Gears (Contd.):

1st. Stage (Cont.)



Assembled plan view is of 2-stage gear box.

Probable dynamic load:

Substituting the values of F_n and $C_v (=0.57)$:

$$F_d = F_n / c_v = 3004.15 / 0.57 = 5270.4 \text{ N}$$

Wear load capacity should be higher than probable dynamic load. i.e., $1.15F_d \leq F_w$

$$1.15F_d = 1.15 \times 5270.4 = 6061 \text{ N}$$

It is less than $F_w (=6465 \text{ N})$

Therefore, 1st stage module 2.5 mm is safe.

Gear Nomenclature :

r_{bp}, r_{bg} Base Circle Radii

r_{dp}, r_{dg} Root or Dedendum Circle Radii

r_{pp}, r_{pg} Pitch Circle Radii

r_{ap}, r_{ag} Tip or Addendum Circle Radii

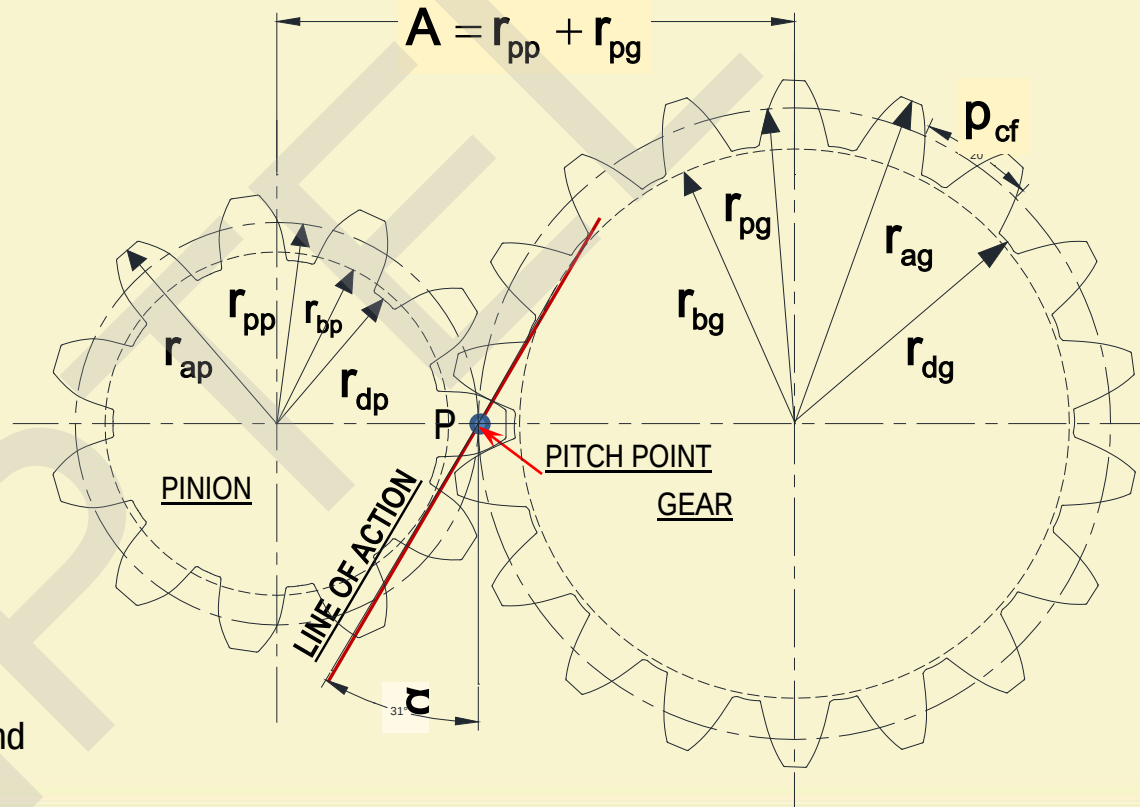
$A = r_{pp} + r_{pg}$ Centre Distance

α Pressure Angle

p_{cf} Circular Face Pitch

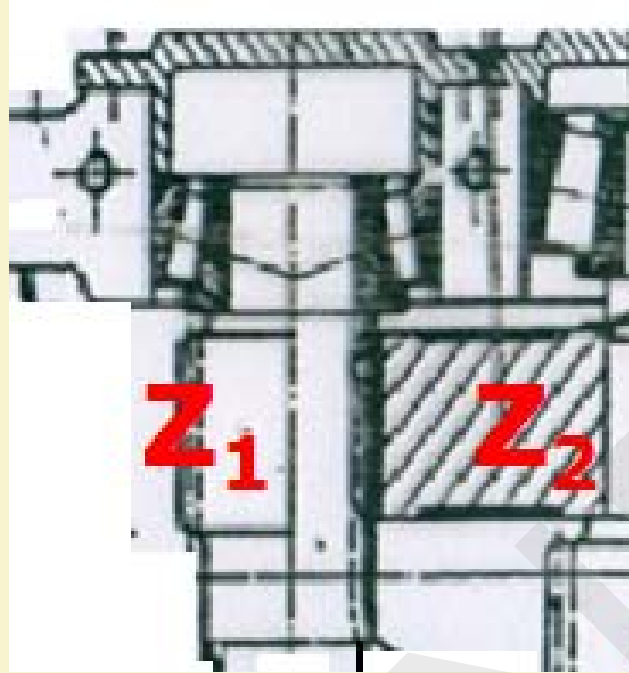
P Pitch Point

p & g As subscript at second place stand for pinion & gear respectively.



Gear Design (1st. Stage):

2nd. Step (Contd.):



Bearings on input shaft.

It is found that 1st stage module 2.5 mm is safe.

Pinion dimensions (nominal/not final):

$$\text{PCD of Pinion : } d_{pp} = \frac{Z_p \times m_n}{\cos \beta} = \frac{17 \times 2.5}{\cos 12^\circ} = 43.45 \text{ mm}$$

Root diameter of Pinion :

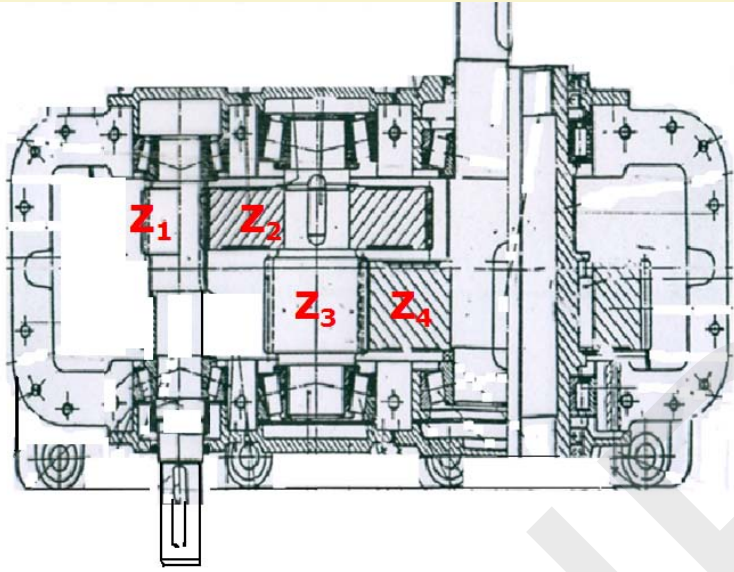
$$\begin{aligned} d_{dp} &= d_{pp} - 2 \times (m_n \times d_f) \\ &= (43.45 - 2 \times 2.5 \times 1.25) = 35.25 \text{ mm} \end{aligned}$$

Inner Diameter (ID) of Bearing close to the Pinion :
In normal course of design it should not be more than 25 mm.

As recommended bearing life is 10,000 hrs. (higher than normal), considering bearing ID at least 30 mm,
1st. Stage normal module is taken as 3 mm.

Gear Design (1st. Stage):

2nd. Step (Contd.):



Assembled plan view is of 2-stage gear box.

Following the same procedure as in 1st stage
the 2nd. Stage module is selected as:

Gears for 2nd. Stage and Centre Distances :

Estimated 1st. Stage module is $m_{n1} = 3 \text{ mm}$

Gear Design for 2nd. Stage

Torque= Design torque for first stage x
(Reduction) Ratio of 1st. Stage.

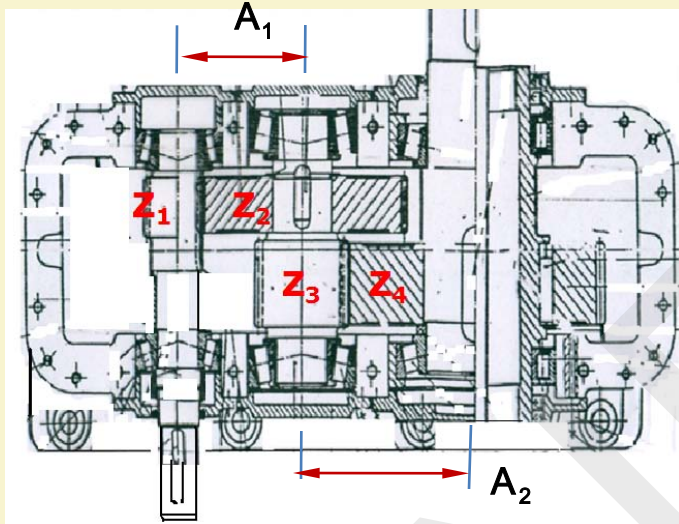
$$\begin{aligned} T_{2d} &= T_{id} \times (Z_{g1} / Z_{p1}) = T_{id} \times (Z_2 / Z_1) \\ &= 60 \times (81/17) = 60 \times 4.76 = 285.6 \text{ Nm} \end{aligned}$$

$$m_{n2} = 4 \text{ mm}$$

Dimensions of Gears and Gear data Table :

2nd. Step (Contd.):

With a suitable selection of helix angle, $\beta_1 = \beta_2 = 11^\circ 28' 42''$, for which $\cos \beta_1 = \cos \beta_2 = 0.98$ and centre distances become:



Assembled plan view is of 2-stage gear box.

$$A_1 = \frac{(Z_1 + Z_2) \times m_{n1}}{2 \times \cos \beta_1} = \frac{(17 + 81) \times 3}{2 \times 0.98} = \frac{98 \times 3}{2 \times 0.98} = 150 \text{ mm}^*$$

and

$$A_2 = \frac{(Z_3 + Z_4) \times m_{n2}}{2 \times \cos \beta_2} = \frac{(16 + 131) \times 4}{2 \times 0.98} = \frac{147 \times 4}{2 \times 0.98} = 300 \text{ mm}^*$$

Dimensions of Gears and Gear data Table :

2nd. Step (Contd.):

Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
1.	Number of Teeth (Z)	17	81	16	131
2.	Tooth Profile	20° Full Depth Involute, Uncorrected			
3.	Normal Module (m_n)	3 mm		4 mm	
4.	Helix Angle (β)	11°28'42"		11°28'42"	
		RH	LH	LH	RH
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	3 mm		4 mm	
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3.75 mm		5 mm	

Dimensions of Gears and Gear data Table :

2nd. Step (Contd.):

Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	534.694
8.	Centre Distance (mm)	150		300	
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	542.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	524.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm
12.	Material	EN 19A	EN 18A	EN 19A	EN 18A
13.	Hardness , Through Hardened (BHN)	350	300	350	300

End of Step 2. Now the 1st. Layout of pinions and gears in mesh is done (Step 3) and Rough shape to the shafts are given (Step 4).

Thank You



IIT KHARAGPUR



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CERTIFICATION COURSES

Prof. Rathindranath Maiti
Mechanical Engineering Department

Module-03

Design of a General Purpose Industrial Helical Gear Reduction Unit- Part I

Lecture-15

Gear Unit Design - 1st. Layout (After Gear Design)



Outline of the Lecture

- **Pinion and Gear Nomenclature**
- **Design Problem and Gear Data & Dimensions**
- **Lay out of Centre Lines – AutoCAD Input & Drawing**
- **Lay out of Gears – AutoCAD Input & Drawing**
- **Lay out of Intermediate Shaft (Draft) – AutoCAD Input & Drawing**

Gear Nomenclature :

r_{bp}, r_{bg} Base Circle Radii
 p & g As subscript at second place stand for pinion & gear respectively.

r_{dp}, r_{dg} Root or Dedendum Circle Radii

r_{pp}, r_{pg} Pitch Circle Radii

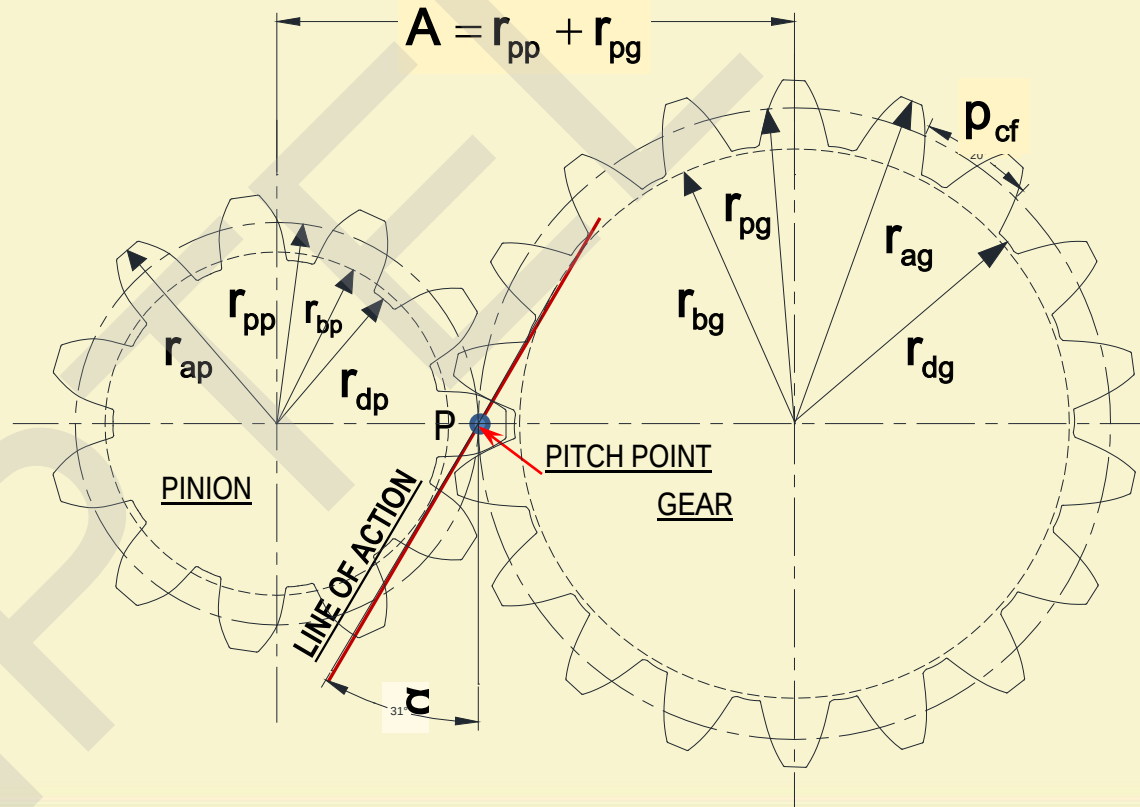
r_{ap}, r_{ag} Tip or Addendum Circle Radii

$A = r_{pp} + r_{pg}$ Centre Distance

α Pressure Angle

p_{cf} Circular Face Pitch

P Pitch Point



Dimensions of Gears and Gear data Table : 2nd. Step (Contd.):

The Gear Unit design problem is same as already considered. All data remain same except the Second stage reduction ratio is reduced to 5.125. Overall transmission ratio is now 24.42.

Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
1.	Number of Teeth (Z)	17	81	16	82
2.	Tooth Profile	20° Full Depth Involute, Uncorrected			
3.	Normal Module (m_n)	3 mm		4 mm	
4.	Helix Angle (β) and Direction of Helix	11°28'42"		11°28'42"	
		RH	LH	LH	RH
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	3 mm		4 mm	
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3.75 mm		5 mm	

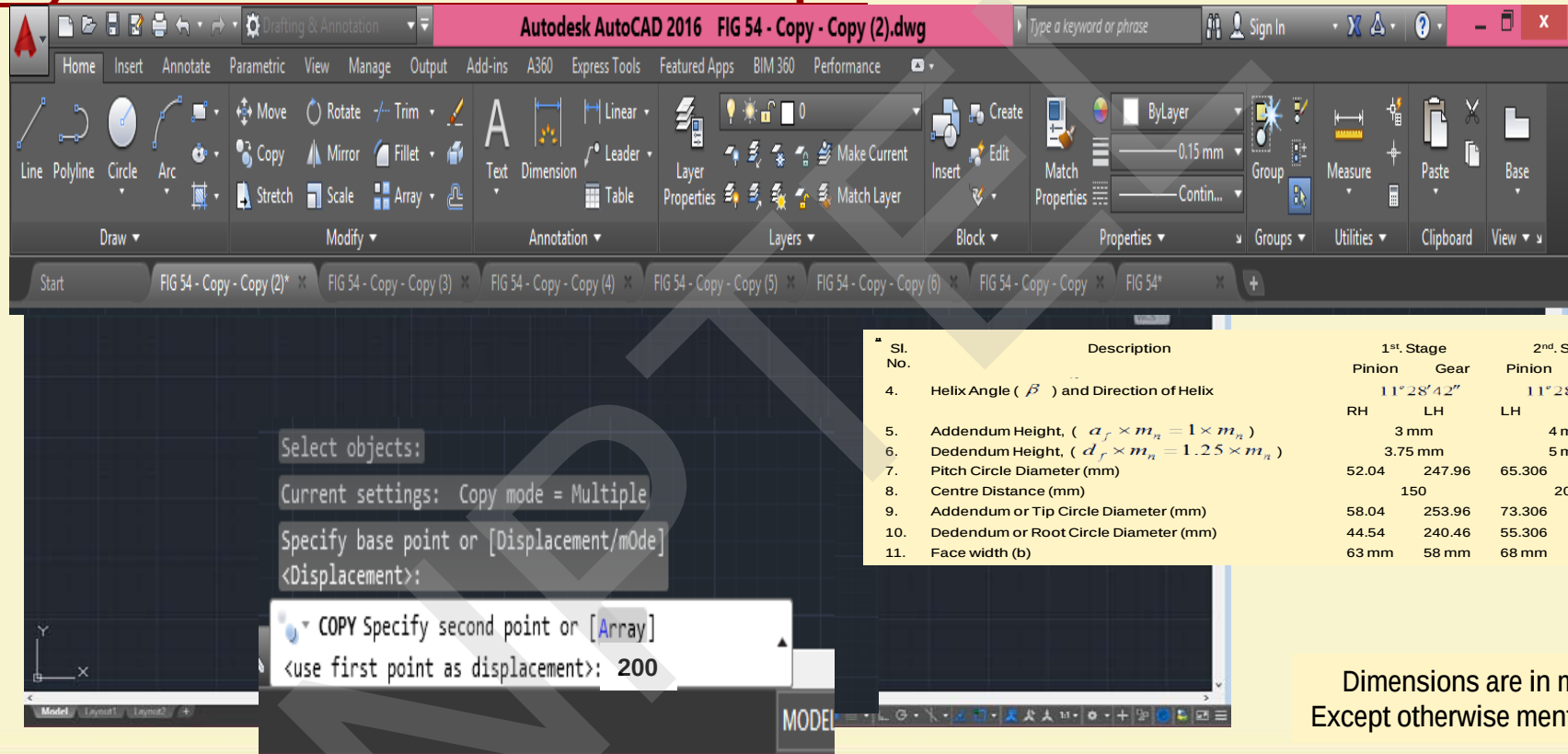
Dimensions of Gears and Gear data Table :

2nd. Step (Contd.):

Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	334.694
8.	Centre Distance (mm)	150		200	
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	342.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm
12.	Material	EN 19A	EN 18A	EN 19A	EN 18A
13.	Hardness , Through Hardened (BHN)	350	300	350	300

End of Step 2. Now the 1st. Layout of pinions and gears in mesh is done (Step 3) and Rough shape to the shafts are given (Step 4).

1st. Layout of Centre Lines - AutoCAD Input:



Select objects:

Current settings: Copy mode = Multiple

Specify base point or [Displacement/mOde]

<Displacement>:

COPY Specify second point or [Array]

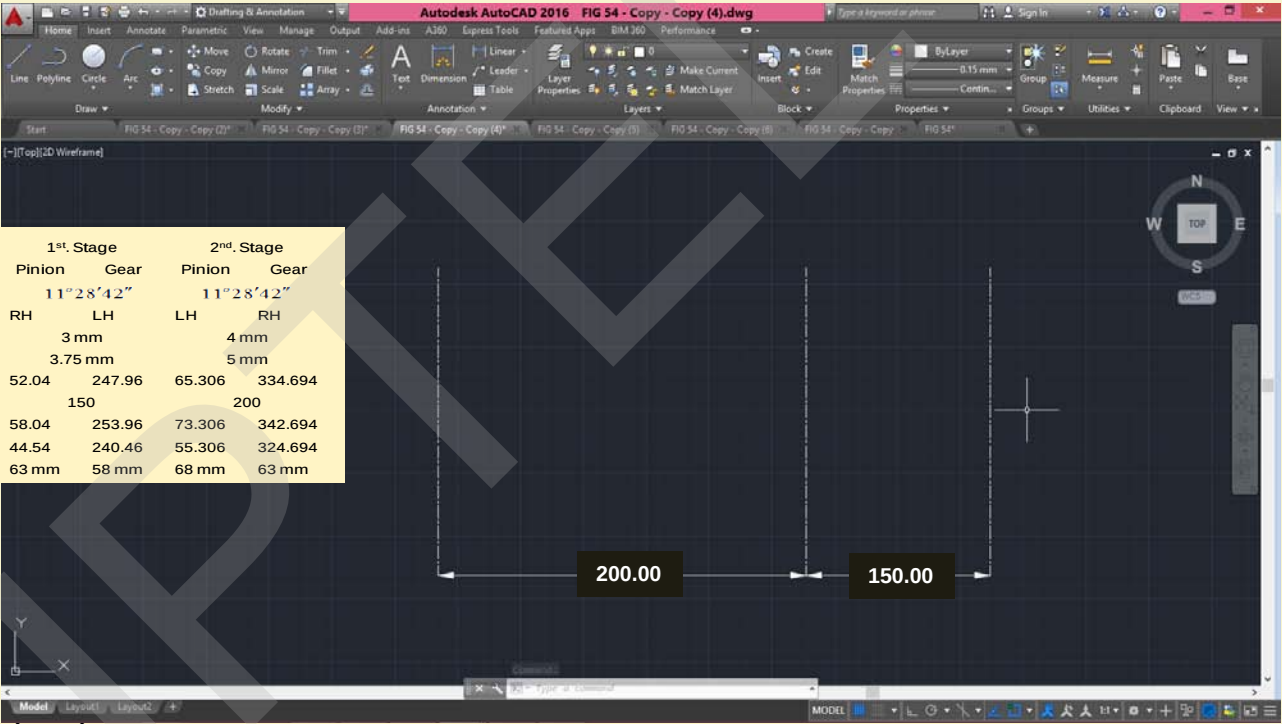
<use first point as displacement>: 200

Sl. No.	Description	1st. Stage		2nd. Stage	
		Pinion	Gear	Pinion	Gear
4.	Helix Angle (β) and Direction of Helix	11°28'42"		11°28'42"	
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	RH	LH	LH	RH
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3 mm	4 mm	5 mm	4 mm
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	334.694
8.	Centre Distance (mm)	150	200		
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	342.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm

Dimensions are in mm, Except otherwise mentioned

1st. Layout of Centre Lines - AutoCAD Input (Contd...) :

Sl. No.	Description
4.	Helix Angle (β) and Direction of Helix
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)
7.	Pitch Circle Diameter (mm)
8.	Centre Distance (mm)
9.	Addendum or Tip Circle Diameter (mm)
10.	Dedendum or Root Circle Diameter (mm)
11.	Face width (b)



Dimensions are in mm,
Except otherwise mentioned

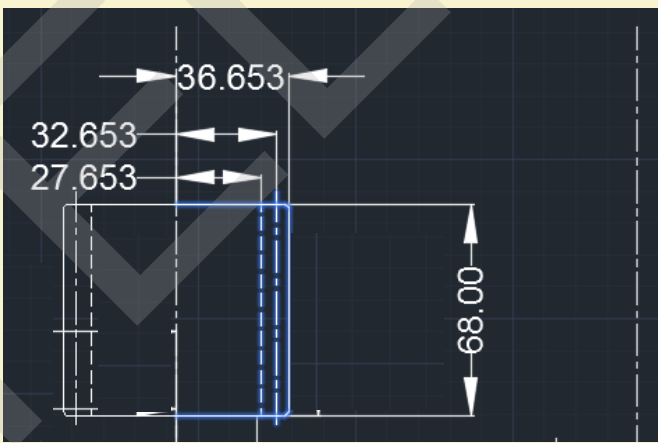
8.	Centre Distance (mm)	150	200
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Layout of 2nd. Stage Pinion - AutoCAD Input :

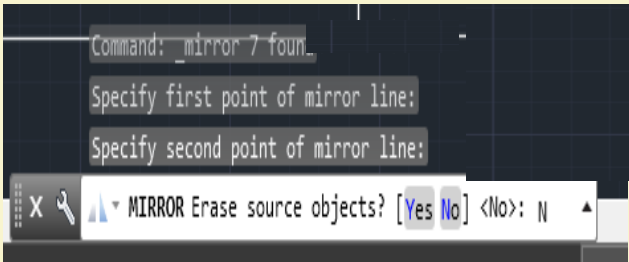
Dimensions are in mm,
Except otherwise mentioned

Intermediate Shaft Axis

Input Shaft Axis

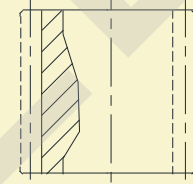


Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
4.	Helix Angle (β) and Direction of Helix	11°28'42"		11°28'42"	
		RH	LH	LH	RH
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	3 mm		4 mm	
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3.75 mm		5 mm	
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	334.694
8.	Centre Distance (mm)	150		200	
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	342.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm



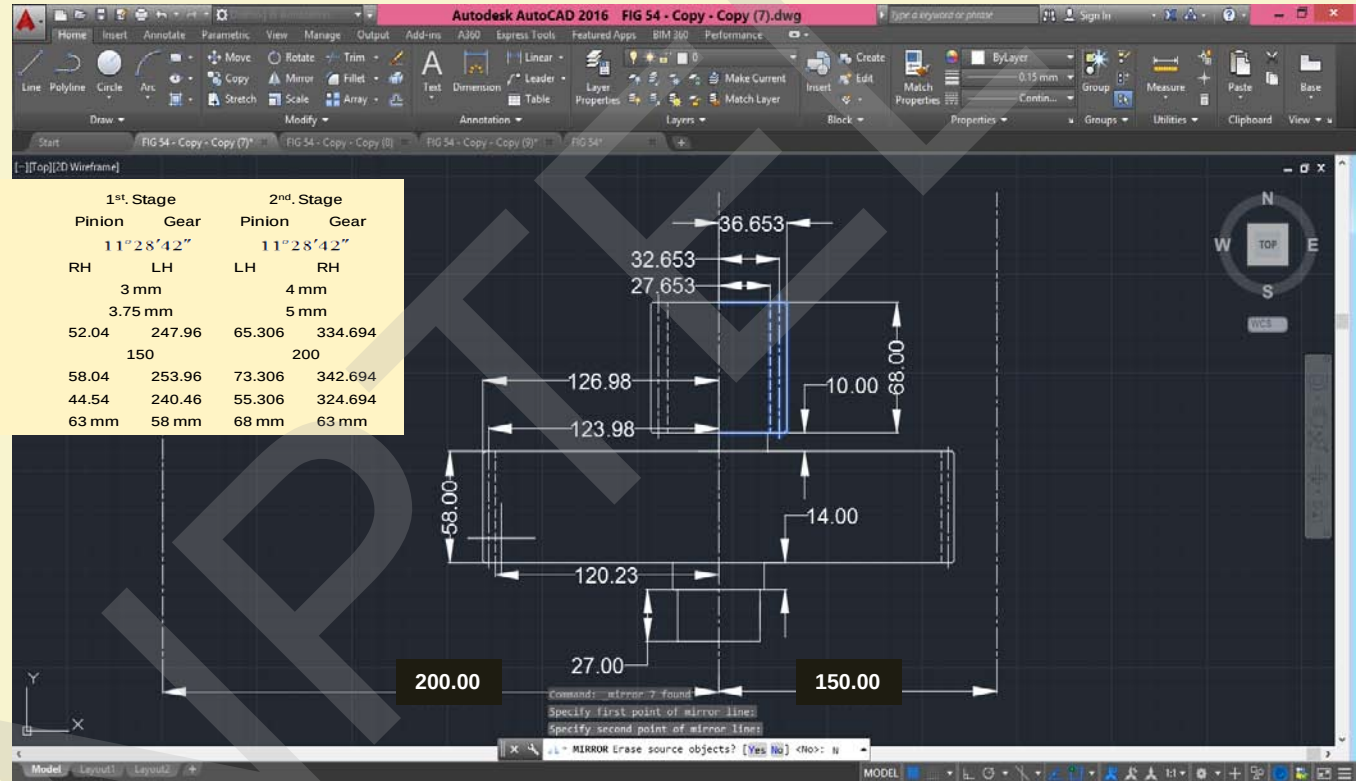
1st. Layout- Centre Lines and Gears:

- Centre Lines
Intermediate
Shaft i.e., 2nd.
Stage Pinion
- Pitch Lines
 - Boundary or Envelope
 - Root (Dedendum Lines)
 - Scrap view to show tooth mesh
 - Corner Finishing (Chamfer)



Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
4.	Helix Angle (β) and Direction of Helix	11°28'42"		11°28'42"	
		RH	LH	LH	RH
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	3 mm		4 mm	
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3.75 mm		5 mm	
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	334.694
8.	Centre Distance (mm)	150		200	
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	342.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm

Layout of Centre Lines & Gears - AutoCAD Input (Contd...) :



1st. Layout- Centre Lines and Gears (Contd...) :

Centre Lines &
2nd. Stage Pinion

Pitch Line of
1st. Stage Gear (on
Intermediate Shaft)

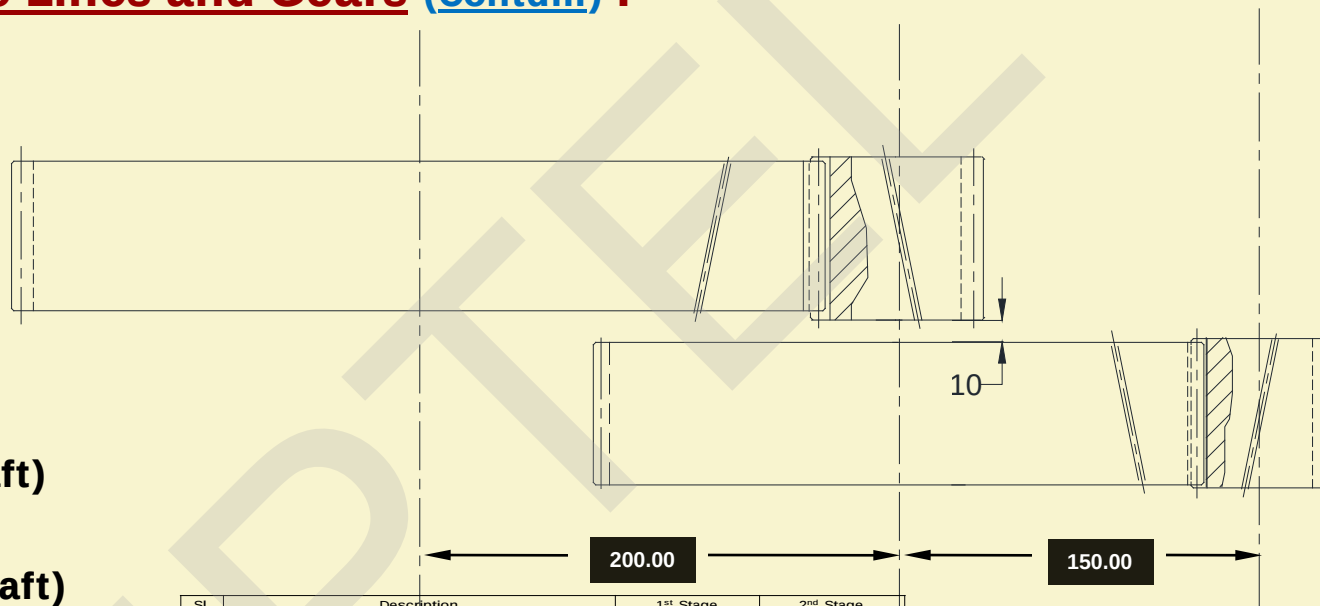
Envelope & Root of
1st. Stage Gear

2nd. Stage Gear
(Keyed to Output Shaft)

1st. Stage Pinion
(Integral on Input Shaft)

Scrap view on input
pinion to show
tooth mesh

Direction of Helix (& Angle)



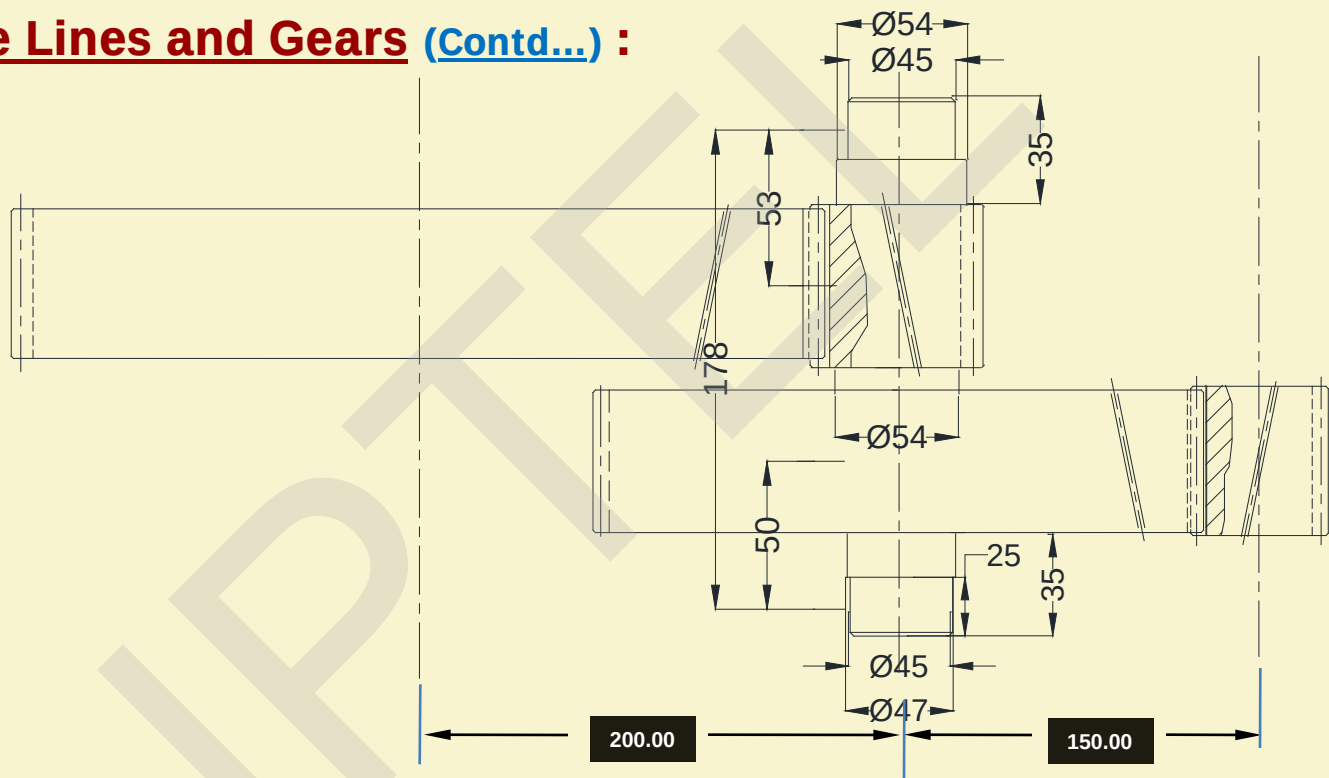
Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
4.	Helix Angle (β) and Direction of Helix	11° 28' 42"		11° 28' 42"	
		RH	LH	LH	RH
5.	Addendum Height, ($a_f \times m_n = 1 \times m_n$)	3 mm		4 mm	
6.	Dedendum Height, ($d_f \times m_n = 1.25 \times m_n$)	3.75 mm		5 mm	
7.	Pitch Circle Diameter (mm)	52.04	247.96	65.306	334.694
8.	Centre Distance (mm)	150		200	
9.	Addendum or Tip Circle Diameter (mm)	58.04	253.96	73.306	342.694
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694
11.	Face width (b)	63 mm	58 mm	68 mm	63 mm

1st. Layout- Centre Lines and Gears (Contd...) :

**Centre Lines &
All pinion & Gears
Intermediate shaft
Between Pinion
and Gear

1st. Stage Gear End

2nd. Stage Pinion End**



Sl. No.	Description	1 st . Stage		2 nd . Stage	
		Pinion	Gear	Pinion	Gear
10.	Dedendum or Root Circle Diameter (mm)	44.54	240.46	55.306	324.694

Layout & Preliminary Bearing Selection (Contd.....)

Step 3 to 5:

Roller Bearings:

15 to 25 mm
(Typical)

10 to 15 mm (Typical)

15 to 25 mm
(Typical)

(Not taper roller
in this case).

Considering
bearing widths,

finally distances
are marked.

53

50

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Layout of Intermediate Shaft (Referring to Design Problem)

Thank You



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