SNS COLLEGE OF TECHNOLOGY

Coimbatore-35
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DEPARTMENT OF MECHANICAL ENGINEERING

DESIGN OF Transmission System

III YEAR VISEM

UNIT 4– Design of Gear Box

Contents

INTRODUCTION

Machine Tools - the **cutting speed varies** with

- **Material** to be machined,
- Type of **cutting tool** used,
- Nature of **operation**, etc.

- For **optimum results**, the cutting speed should be maintained within limits.
- Different working conditions - need for changing the spindle speeds.
- Achieved by **employing a single motor and a gear box between the motor and work spindle.**

In automobile,

- need of economical speed range irrespective of speed of the vehicle.
- Achieved by providing a **multi-speed gear box** between the engine and road wheels

Vehicle transmission system include a gearbox which will:

help the engine drive the car.

enable the vehicle to be reversed

connect the engine to the transmission system

- **Required wheel torque** is **not constant**
- **High torque** to accelerate the car
- Cruising torque does not need to be as high

So, gearbox needs to be capable of **different speed ratios**

Need for easily **connecting/disconnecting engine** from the transmission system - done by including a **clutch** between the engine and the gearbox.

OBJECTIVES OF GEAR BOX DESIGN

provide the **maximum possible efficiency** in all gears

be the **minimum possible weight** while being capable of handling with the **requisite torque** throughput

have an overall **simplicity in design** and, more importantly, **in assembly**, as ratio changing and **maintenance** often have to be carried out under fairly primitive conditions

require the **minimum amount of time and effort for maintenance** - this point is effected by the simplicity in design and assembly

reduce the number of components to be moved, when **changing gears**

GEAR BOX

Gearboxes serve the purpose of changing the **engine torque** and **speed**, as per the requirement at the wheels of the vehicle on the road.

Gearboxes operate on the principle of **reducing the engine speed** before the drive from the engine crank shaft is given to the driving wheels through the **use of a pair of meshing gears**.

Depending upon the mechanism of meshing the gear pairs, gearboxes are classified as:

(a) Sliding Mesh Gearbox

(b) Constant Mesh Gearbox

(c) Synchromesh Gearbox

(d) Epicyclic Gearbox

SLIDING MESH GEAR BOX

Contd.

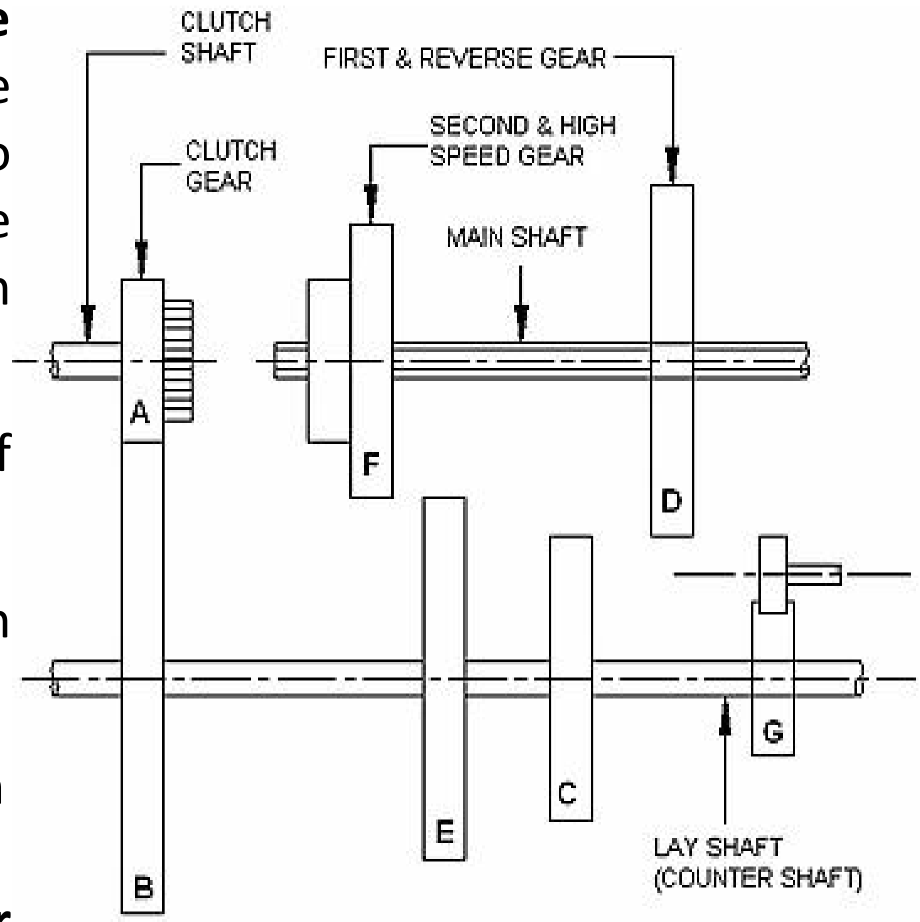
When gears are **meshed by sliding the gears (over a splined shaft)** with the help of a mechanism, in order to obtain different speed ratios, the gearbox is known as sliding mesh gearbox.

It is the **simplest and oldest type** of gearbox.

Driving shaft of the gearbox is known as **primary or clutch shaft**.

A clutch gear rigidly fixed to the clutch shaft always remains in the mesh with the bigger gear fixed on the **counter shaft or lay shaft**.

Other gears also remain fixed on the lay shaft, which are required for speed change.

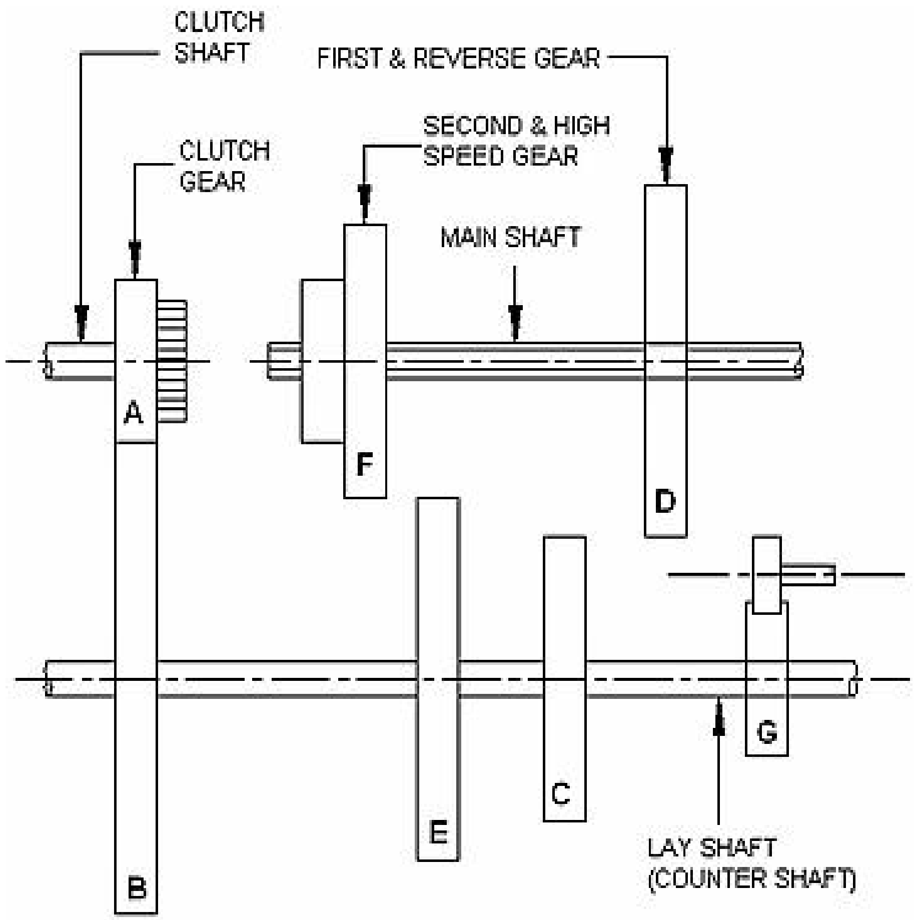


SLIDING MESH GEAR BOX

The **power comes** from the engine **to the clutch shaft** and hence to the **clutch gear** which is always in mesh with a gear on the **lay shaft**.

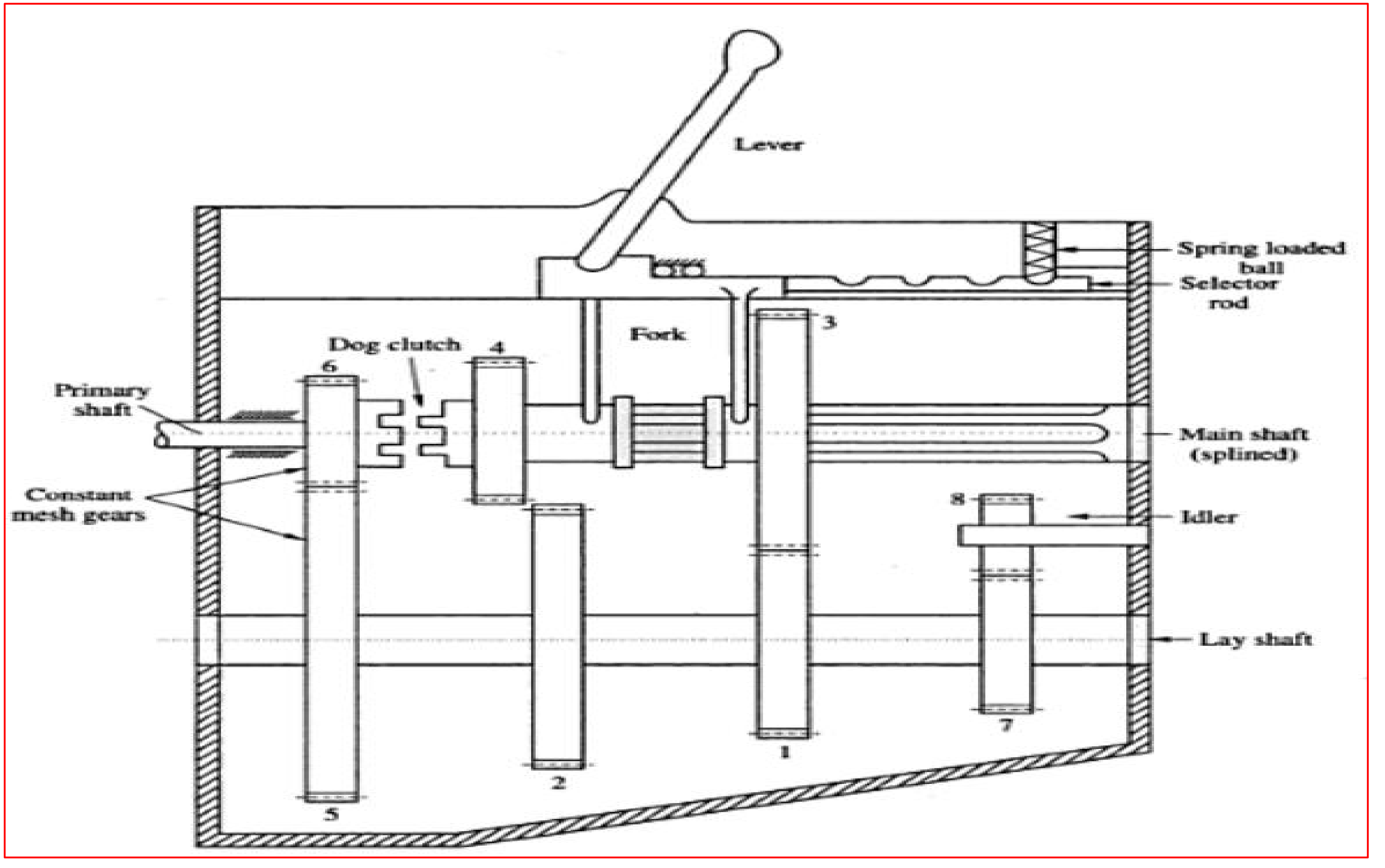
All the gears on **the lay shaft** are fixed to it and as such they are **all the time rotating** when the engine is running and the **clutch is engaged**.

Three direct and one reverse speeds are attained on suitably moving the gear on the main shaft by means of selector mechanism.



SLIDING MESH GEAR BOX

Contd.



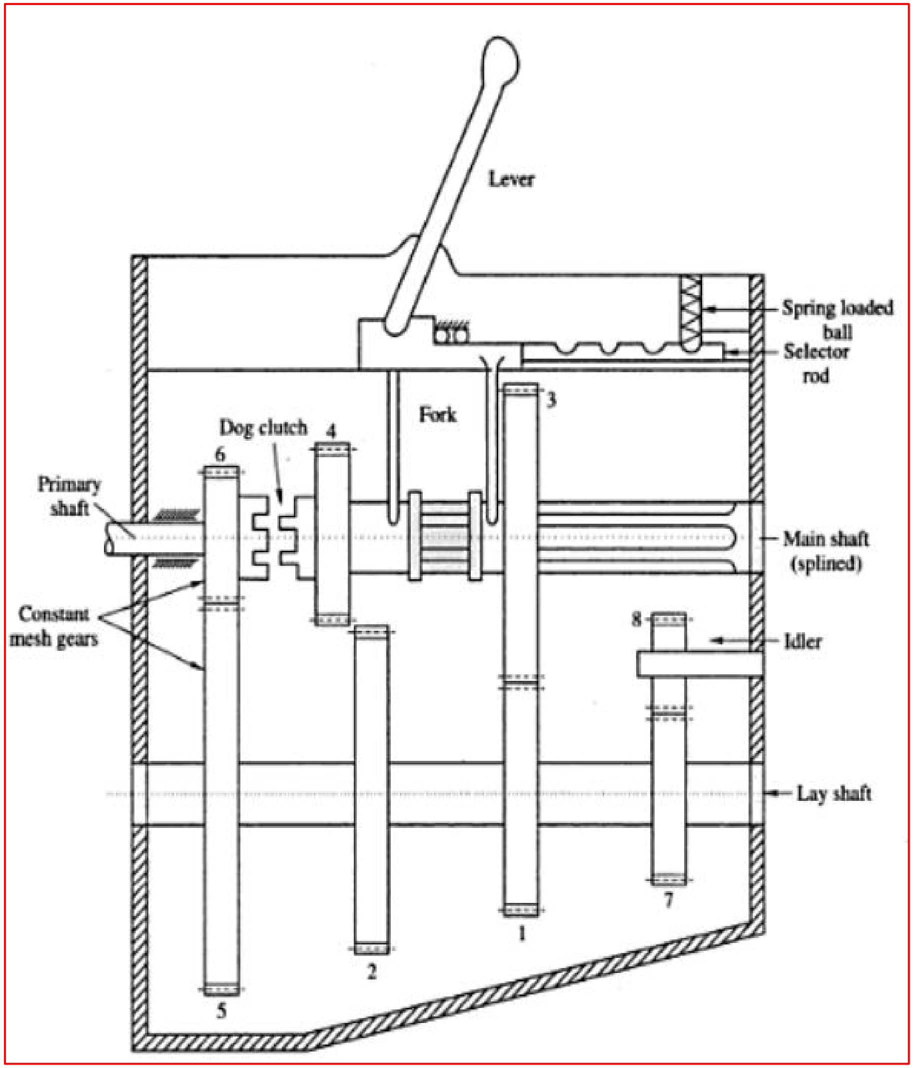
SLIDING MESH GEAR BOX

Contd.

The lay shaft carries three gears (two for direct, and one for reverse) or four gears (three for direct and one for reverse)

One gear of lay shaft is in constant mesh with the clutch gear.

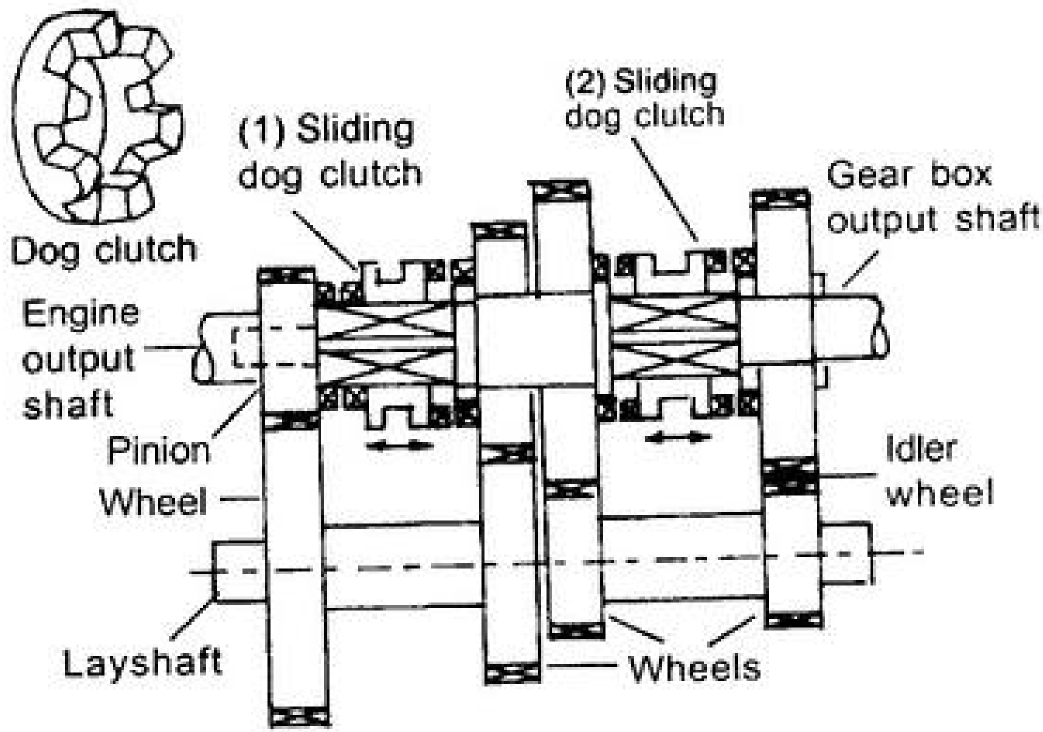
When the main shaft is driven from the lay shaft, speed reduction is obtained by the first gear pair which is always in mesh and is thus called constant mesh gear.



CONSTANT MESH GEAR BOX

In the constant – mesh gearbox, **all pairs of gear wheels are always in mesh**.

A dog clutch couples a particular gear pair to the main shaft, while other gear pairs are not coupled to the main shaft. This provides a **noiseless operation**.



Pairs of **helical gears** (in place of spur gears) can be used, which provide **noiseless operation** with **high efficiency**.

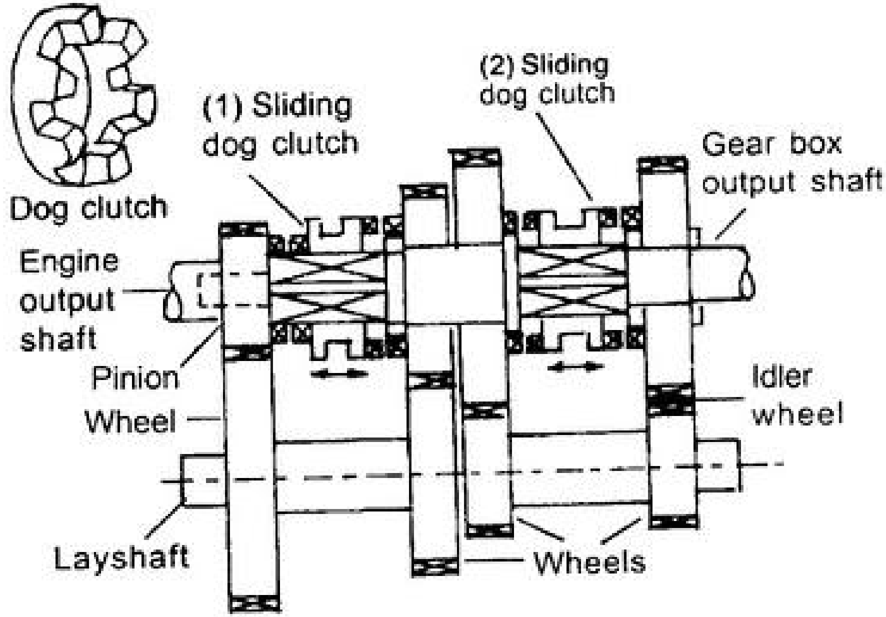
The **primary shaft** that carries the clutch is **splined**. Moreover, the **primary shaft carries a gear** that meshes with the **largest gear of the lay shaft**.

CONSTANT MESH GEAR BOX

CONSTANT MESH GEAR BOX

The **main shaft has a number of gears** which meshes with the gears of the lay shaft.

These gears on **bushes or bearings** are **free to rotate** on the main shaft **without transmitting any power.**



All the gears on the lay shaft are rigidly fitted to it.

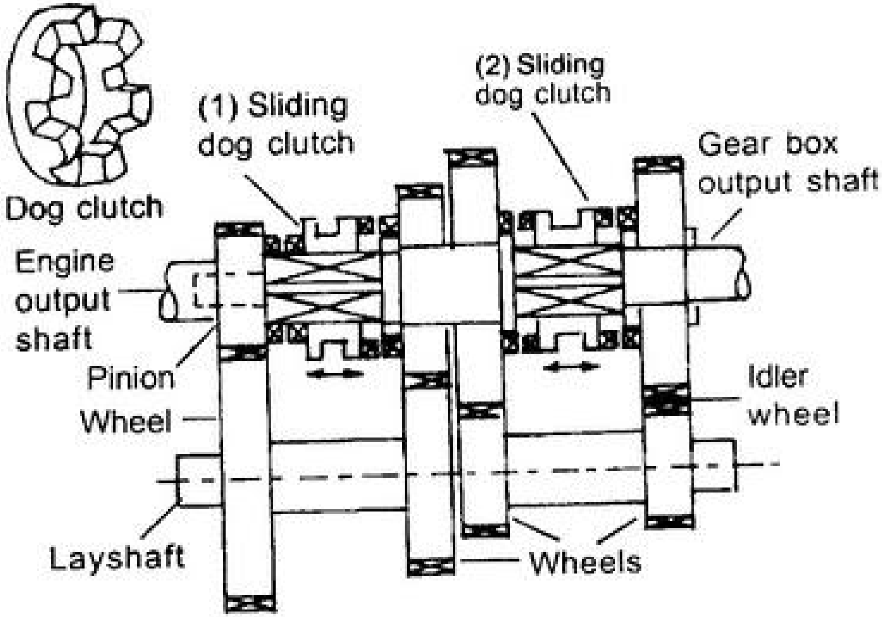
The **three main shaft gears** are, therefore **constantly** driven by the engine shaft, but **at different speeds**.

The first gear and the second gear rotate in the **same direction** as the engine shaft while the **reverse gear** rotates in **the opposite direction** to the engine shaft.

CONSTANT MESH GEAR BOX

Contd.

If anyone of the **gears on/the main shaft is coupled up to the main shaft**, then there **will be a driving connection** between the main shaft and the engine shaft. The **coupling** is affected **by the dog clutch units**.



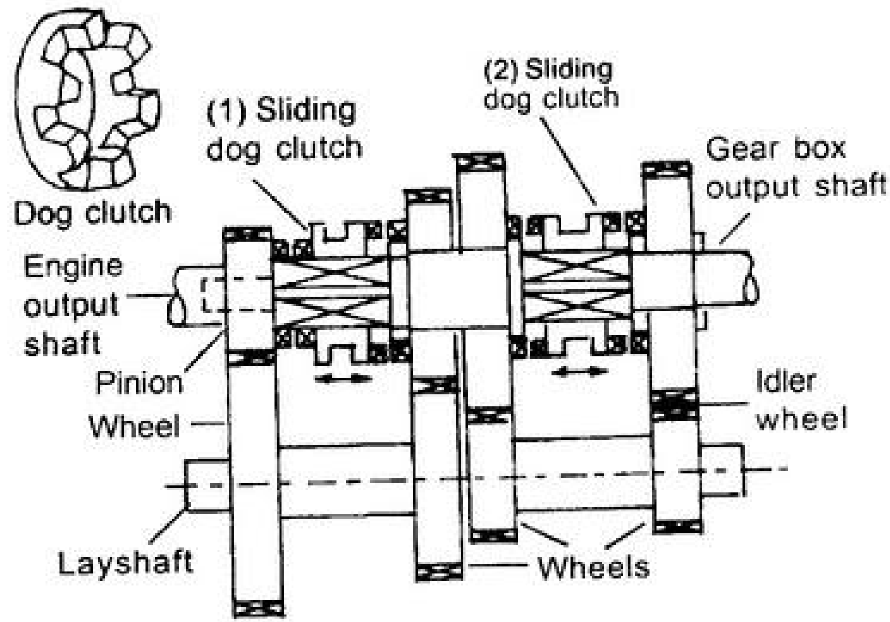
The d**og clutch** members are **carried on splined (or squared)** portions of the main shaft. They are **free to slide** on those squared portions, but **have to revolve** with the shaft.

CONSTANT MESH GEAR BOX

If one of the dog clutch members (I) is slid to the left it will couple the wheel (first gear) to the main shaft giving the first gear.

The drive is then through the wheels and this dog clutch member. The other dog clutch is meanwhile in its neutral position.

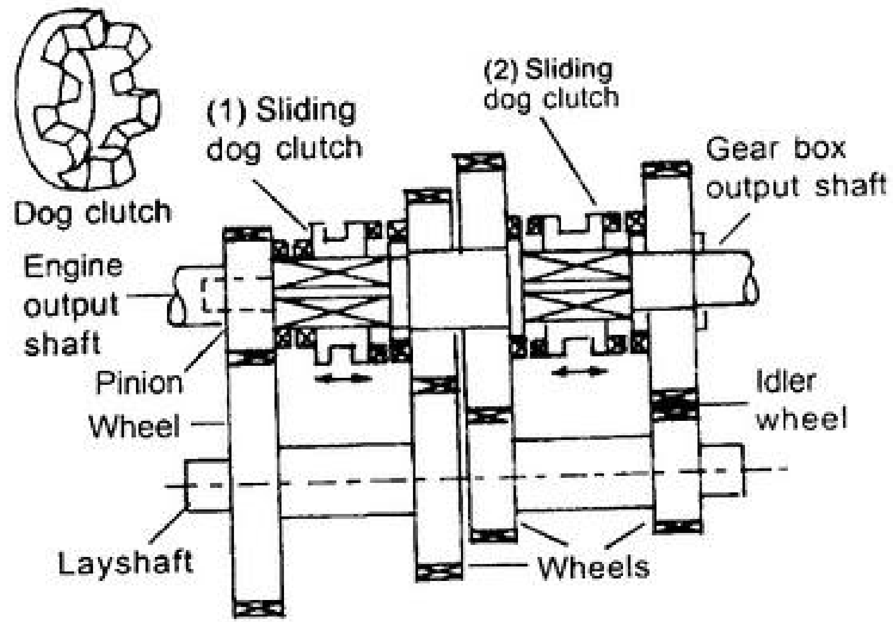
If, with the above dog clutch member in its neutral position, the other dog clutch member (2) is slid to the right, it will couple the wheel (second gear) to the main shaft and give second gear.



CONSTANT MESH GEAR BOX

If this dog clutch member is slid to the left, it will couple the main shaft directly to the pinion fixed to the engine shaft.

This will give a direct drive, as in the sliding mesh gear box.



The reverse gear is engaged by sliding the dog clutch member (which gives the first gear) to the right. Then it will couple the wheel (reverse gear) to the main shaft. The drive is then through the wheels, the idler and the dog clutch member.

In the constant mesh gear box, the gears on the main shaft must be free to revolve. For this, they are either be bushed or be carried on ball or roller or needle bearings.

CONSTANT MESH GEAR BOX

Contd.

The main advantages of the constant mesh gear box over the sliding mesh type are as follows:

Helical or double helical gear teeth can be used for the gears instead of spur gears. Then gearing is quieter.

Synchronizing devices can be used for smooth engagement.

Any damage that results from faulty manipulation occurs to the dog clutch teeth and not to the teeth of the gear wheels.

Once the dog clutches are engaged, there is no motion between their teeth. But when gear teeth are engaged, the power is transmitted through the sliding action of the teeth of one wheel on those of the other. The teeth have to be suitably shaped to transmit the motion properly.

If the teeth on the wheel are damaged, the motion will be imperfect and noise will result.

Damage is less likely to occur to the teeth of the dog clutches, since all the teeth engage at once, whereas in sliding a pair of gears into mesh the engagement is between two or three teeth.

SYNCHROMESH GEAR BOX

In the synchromesh gearbox, provision of synchromesh device avoids the necessity of double declutching.

These synchromesh devices work on the principle that the two gears to be engaged are first brought in to frictional contact, which equalizes their speed, after which they are readily and smoothly engaged.

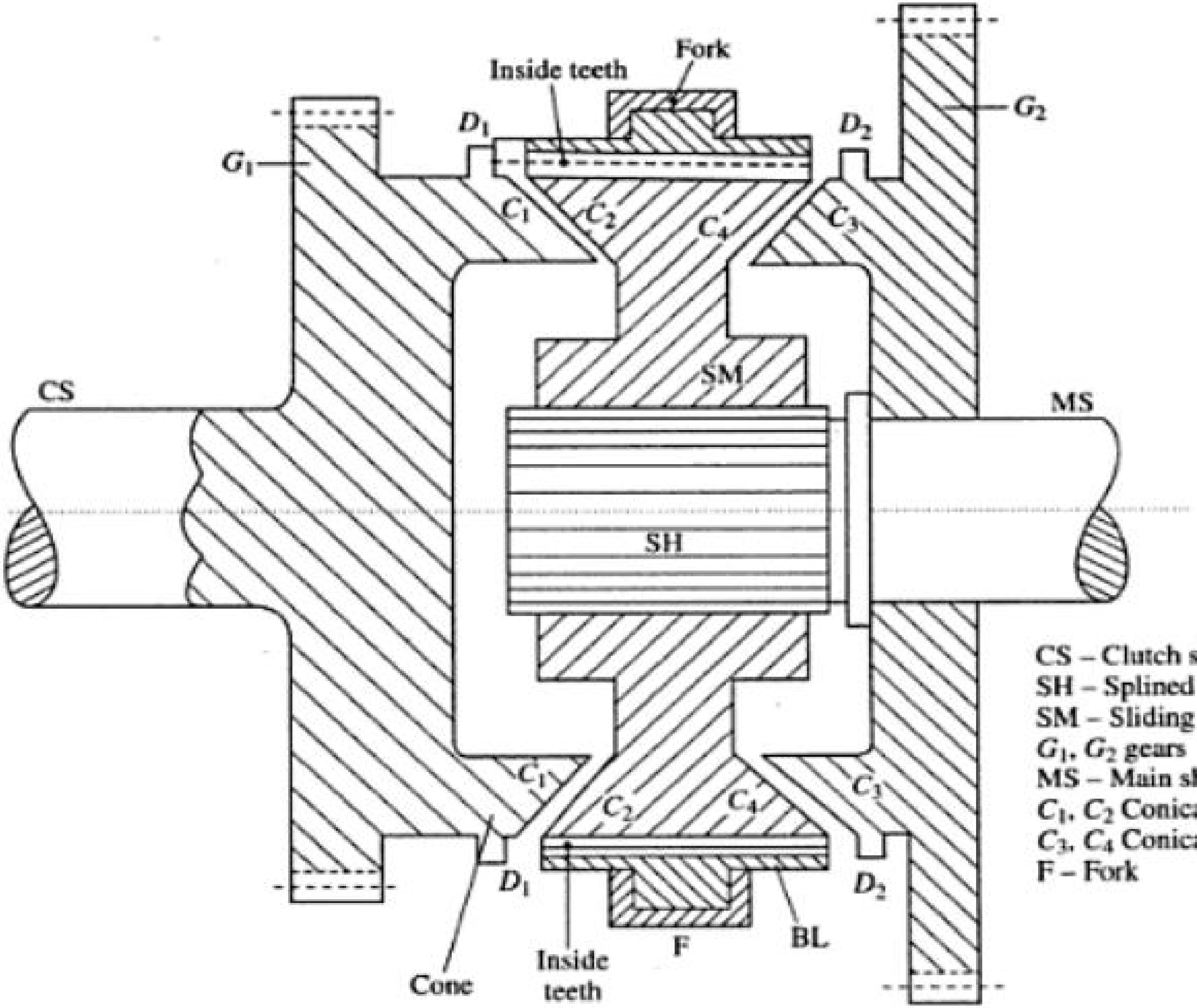
By using this device, even unskilled drivers can engage the gears without clash or damage to the gears.

Synchromesh gearbox is similar to the constant – mesh gearbox and its main features are:

1. Gears on main shaft are free to rotate on bushes. The end of the main shaft at the rear of transmission is called output shaft.
2. Gears on the main shaft are locked to the shaft by the dog clutch of the synchronising hub when their speeds have been equalized by their cones.

SYNCHROMESH GEAR BOX

Contd.



CS – Clutch shaft
SH – Splined hub
SM – Sliding member
G_1, G_2 gears
MS – Main shaft
C_1, C_2 Conical surfaces
C_3, C_4 Conical surfaces
F – Fork

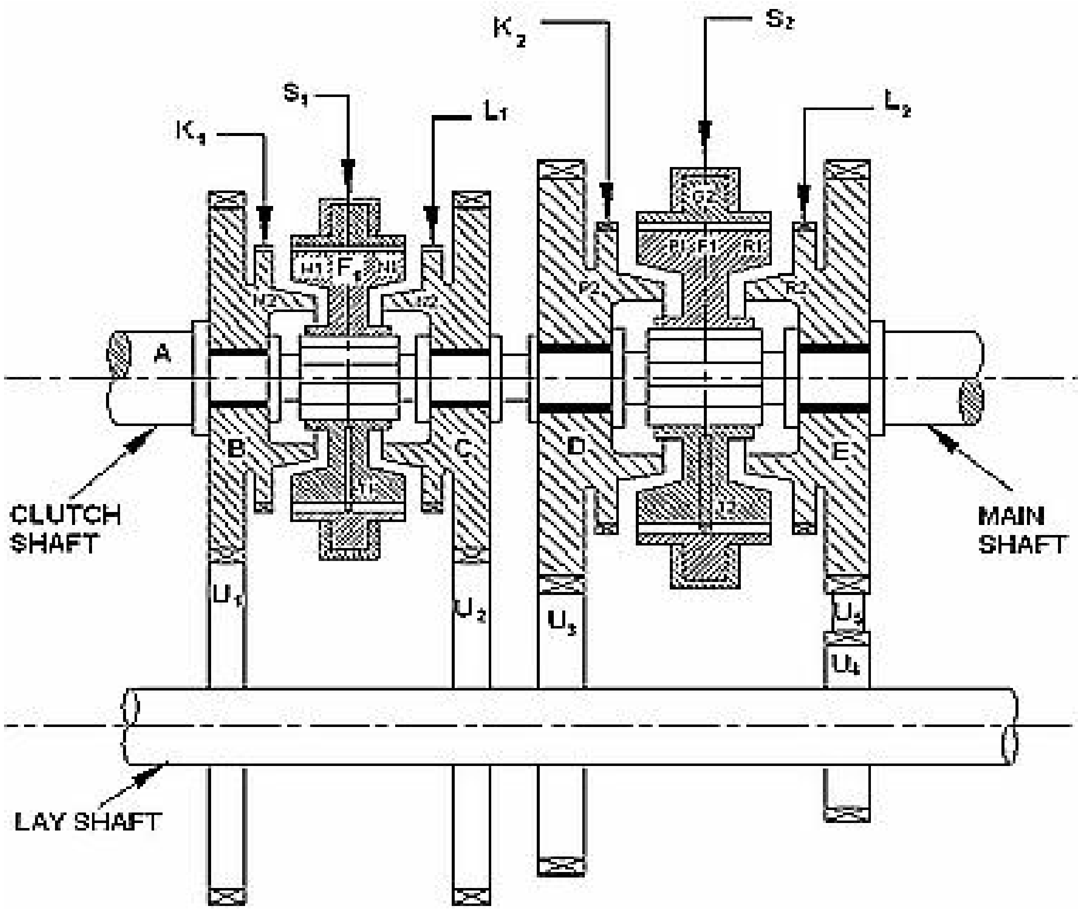
SYNCHROMESH GEAR BOX

Contd.

The engine shaft, Gears *B, C, D, E* are free on the main shaft and are always in mesh with corresponding gears in the lay shaft.

Thus all the gears on main shaft as well as on lay shaft continue to rotate so long as shaft A is rotating.

Menders F_1 and F_2 are free to slide on splines on the main shaft. *G1* and *G2* are ring shaped members having internal teeth fit onto the external teeth members F_1 and F_2 respectively.

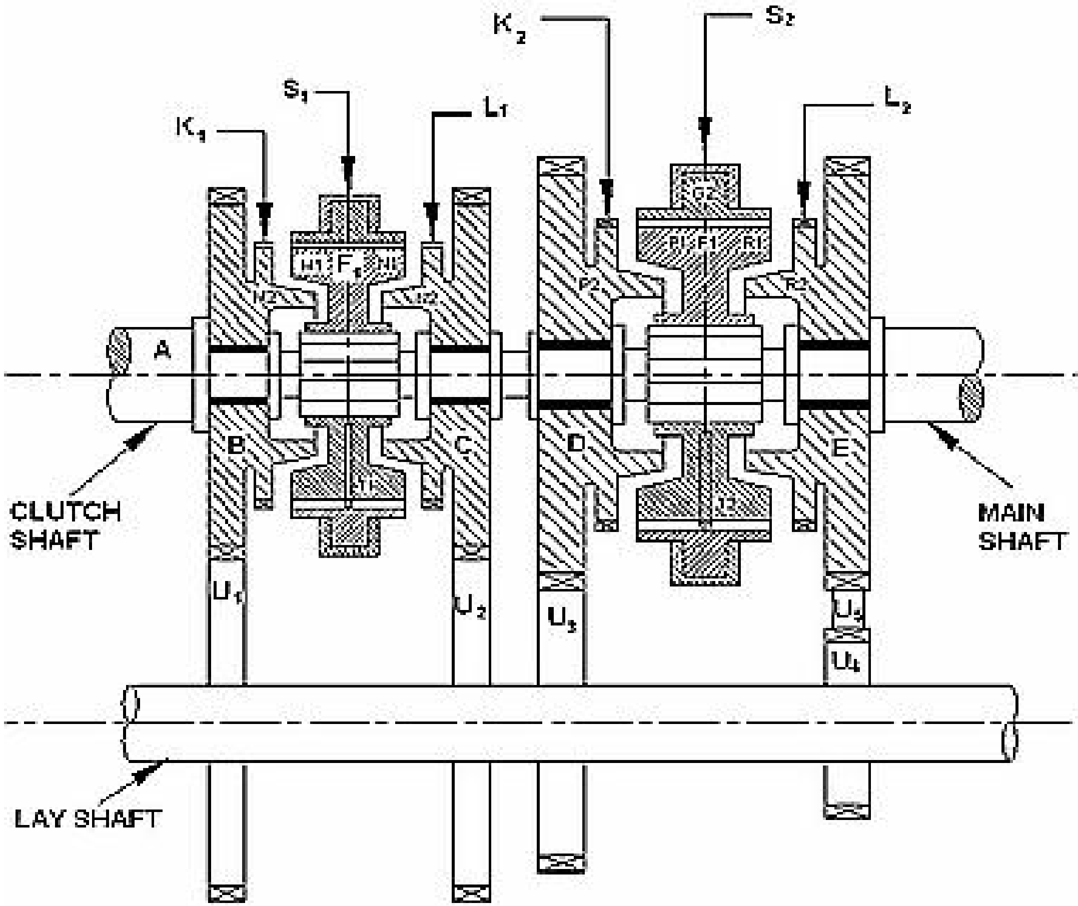


SYNCHROMESH GEAR BOX

Contd.

K_1 and K_2 are dog teeth on *B* and *D* respectively and these also fit onto the teeth of G_1 and G_2. $S_1$and S_2 are the forks.

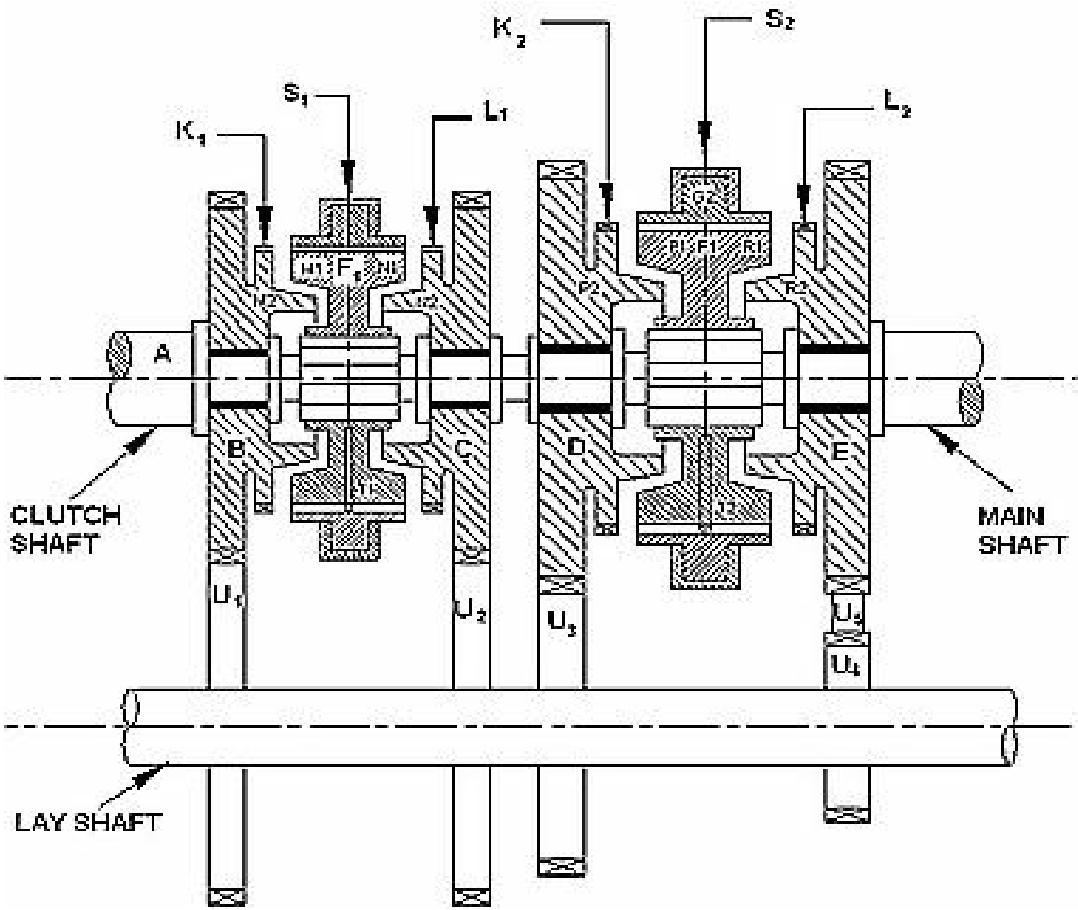
$T_1$and T_2 are the ball supported by springs. These tend to prevent the sliding of members G_1 (G_2) on F_1 (F_2). however, when the force applied in G_1 (G_2) through fork S_1 (S_2) exceeds a certain value, the balls are overcome and member G_1 (G_2) slides over F_1 (F_2).



SYNCHROMESH GEAR BOX

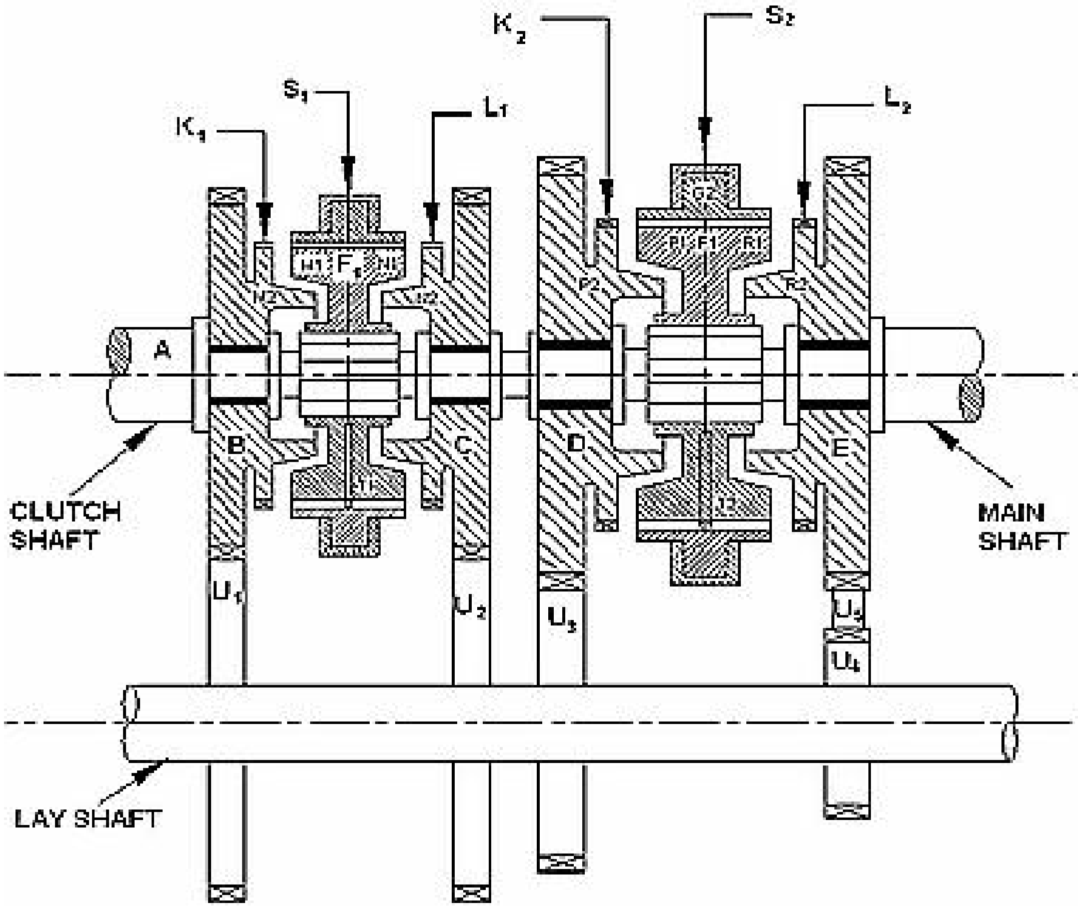
There are usually six of these balls symmetrically placed circumferentially in one synchromesh device. M_1, M_2, N_1, N_2, P_1, P_2, R_1, R_2 are the frictional surfaces.

The working of the gear box is as follows .for direct gear, member G_1 and hence member F_1 (through spring –loaded balls) is slid towards left till comes M_1 and M_2 rub and friction makes their speed equal.



Further pushing the member G_1 to left causes it to override the balls and get engaged with dogs K_1. Now the drive to the main shaft is direct from B via F_1 and the splines.

We have to give sufficient time for synchronization of speeds, otherwise clash may result.



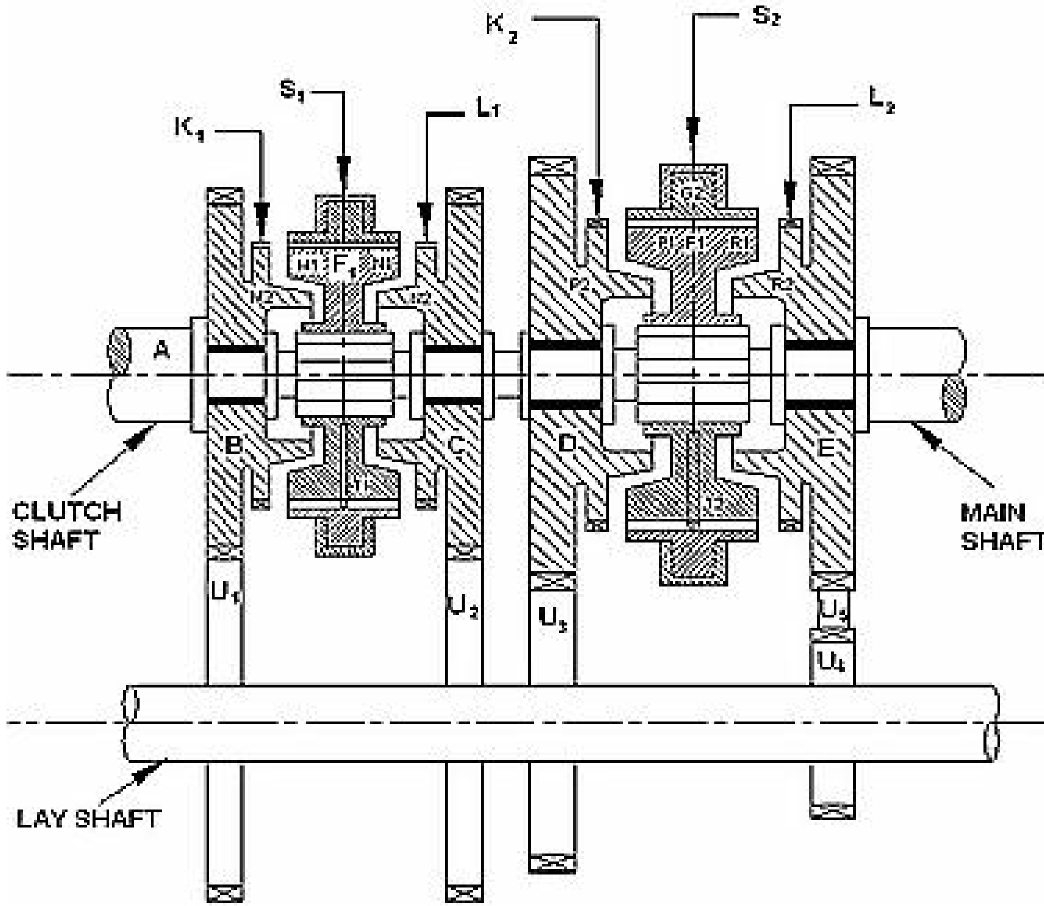
SYNCHROMESH GEAR BOX

Contd.

For the second gear the members F_1 and G_1 are slid to the right so that finally the internal teeth on G_1 are engaged with L_1.then the drive to main shaft will be from *B* via U_1, U_2, *C*, F_1 and splines.

For first gear, G_2 and F_2 are moved towards right. In this case the drive will be from *B* via U_1, U_3, *D*, F_2 and splines to the main shaft.

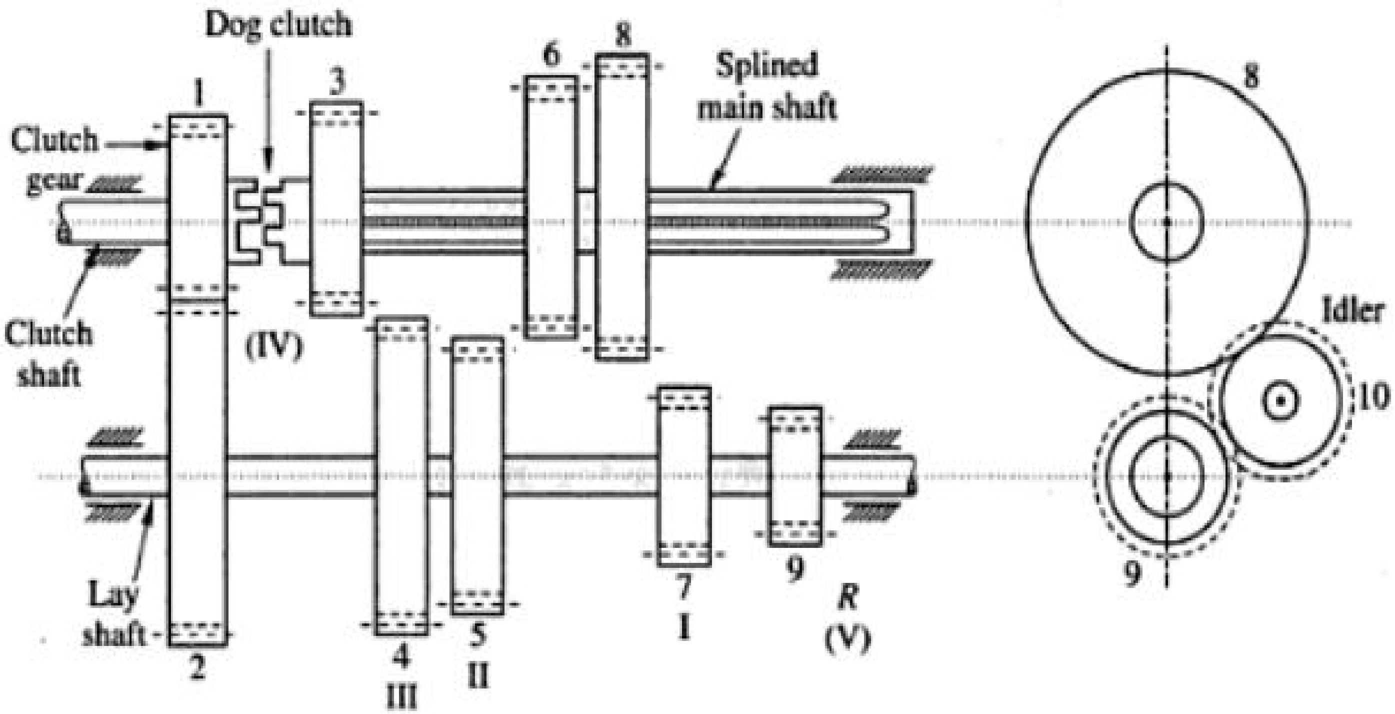
For reverse, G_2 and F_2 are slid towards right. In this case the drive will be from *B* via, U_1, U_4, U_5, *D*, F_2 are splines to the main shaft.



Gearboxes used in popular automobiles

Make of Vehicle	Type of Gearbox	Gear Ratios					
		I	II	III	IV	Top	Reverse
Maruti 800	Four forward speeds, all synchro-mesh and reverse	3.585	2.166	1.333	—	0.999	3.365
Maruti Gypsy	Four speeds, forward, all synchro-mesh, and reverse	3.136	1.946	1.422	—	1:1	3.463
Fiat Uno	Five speeds, all synchromesh with overdrive in five and reverse gear.	3.909	2.238	1.280	1.029	0.838	—
Tata Indica (Petrol)	Five speeds, all synchromesh, over-drive in five and reverse	3.42	1.95	1.36	0.95	0.770	3.58
Hyundai Santro	Five speeds, all synchromesh, over-drive on fourth and fifth gear, reverse	3.833	2.105	1.310	0.919	0.784	4.00
Maruti Zen	Five-speed, all synchromesh and reverse	3.416	1.894	1.280	0.914	0.757	—

Figure shows a five-speed sliding mesh gearbox, with 4 forward speeds and one reverse speed.



Five-speed sliding mesh gearbox

A clutch gear (1) on clutch shaft always remains in mesh with the gear (2) on the lay shaft. *Gears 2, 4, 5, 7 and 9 are rigidly fitted on the lay shaft.* With the help of fork and sliding mechanism, gears 3, 6 and 8 on main splined shaft are slid so as to get different speeds. Gears 3, 6 and 8 on main shaft rotate with main shaft, without any power transmission, when the engine is in neutral position.

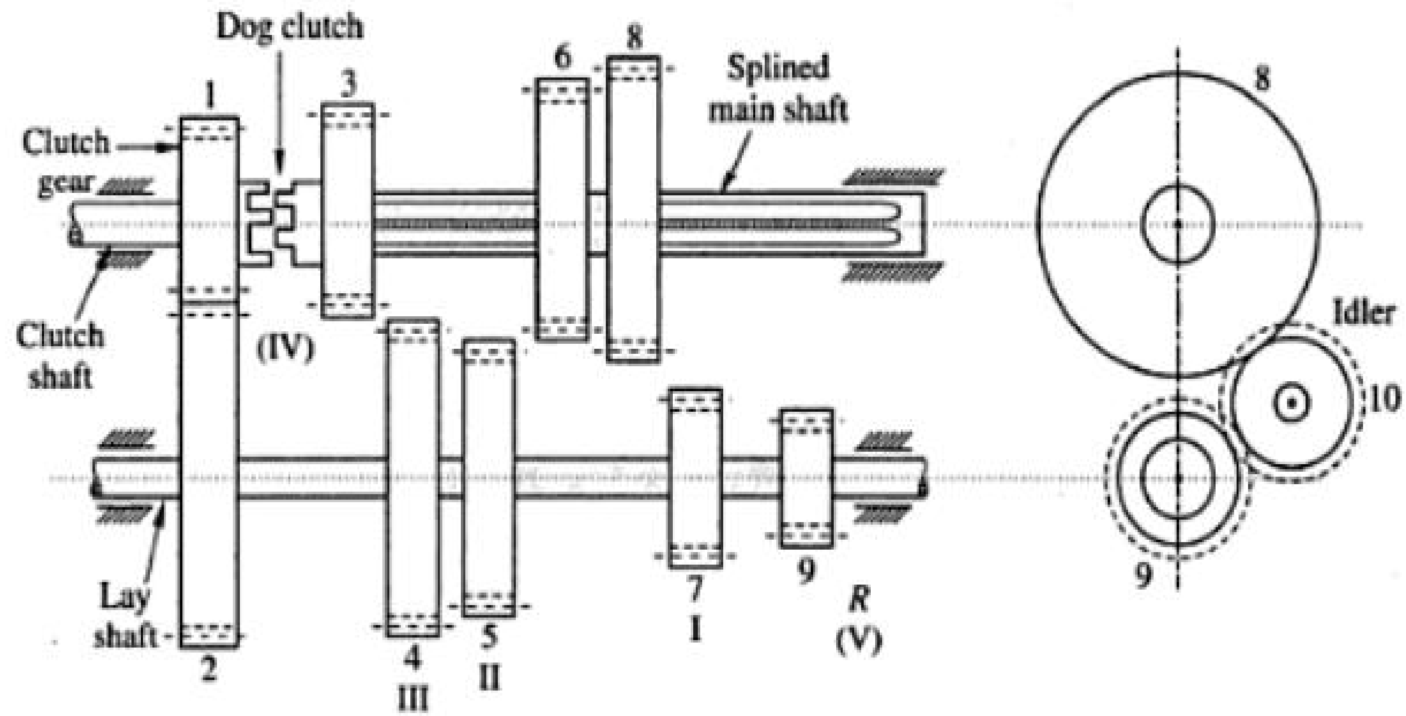


Figure 29-8 Five-speed sliding mesh gearbox

First gear: With the sliding mechanism, gear 8 on main shaft is brought in mesh, with gear 7 of lay shaft.

Second gear: Gear 6 of main shaft is brought in mesh with gear 5 of lay shaft.

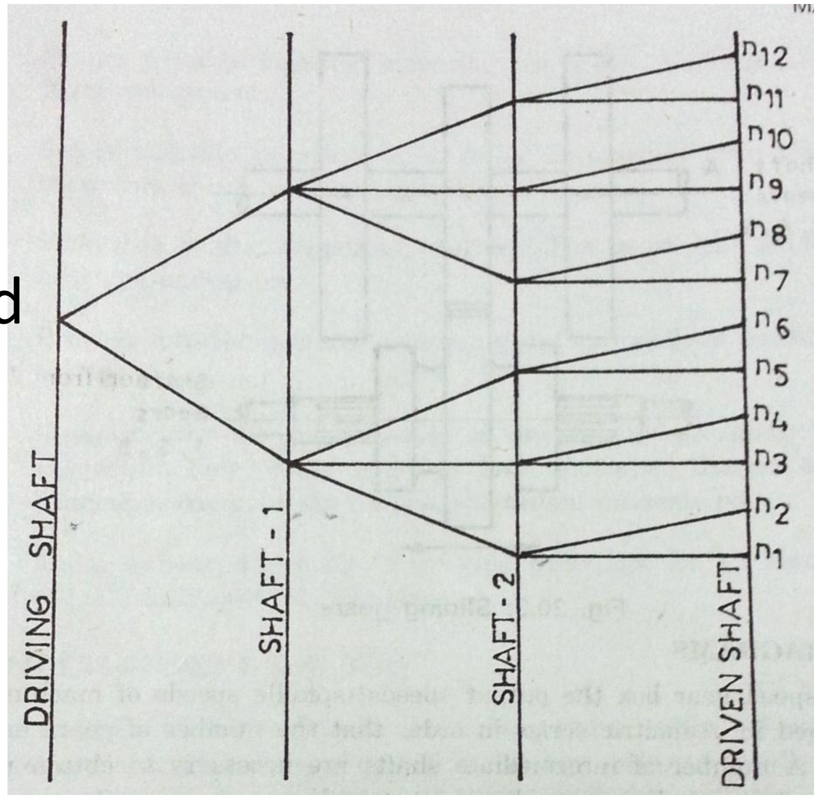
Third gear: Gear 3 of main shaft is brought in mesh with gear 4 of lay shaft.

Fourth gear (top gear): With the help of dog clutch, main shaft is coupled with the clutch shaft. The main shaft starts rotating at same speed as the engine shaft.

Reverse gear: Gear 8 of main shaft is brought in mesh with idler gear 10 on auxiliary shaft, which is already rotating with gear 9 of lay shaft. Speed of the vehicle is reversed.

RAY / SPEED / STRUCTURAL DIAGRAM

- Can be called as Ray Diagram or Structural Diagram
- If we required ‘n’ no of speed it is not possible to fix ‘n’ no of gears in a pair of shaft so.
  - For 2 shaft max 3 number of speed can be attained
  - Intermittent shaft should be used for more than 3 speed.
  - Required number of speeds can be obtained in multiples of 2 & 3

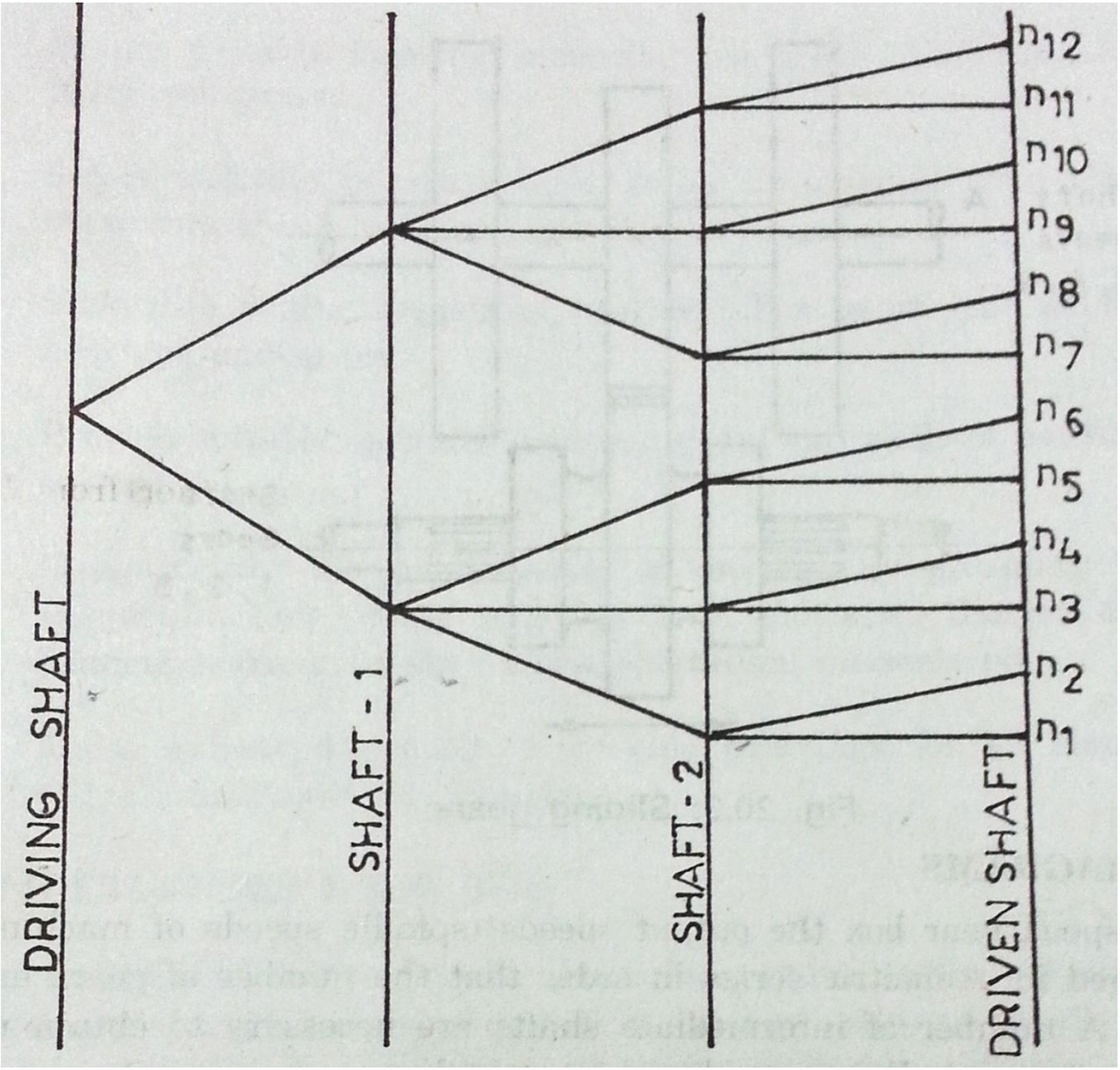


SPEED DIAGRAM

Contd.

2 x 3 x 2

(2) (3) (2)



SPEED DIAGRAM

Contd.

In a multi-speed gear box the output speeds (spindle speeds of machine tools) should be arranged in **geometric series** in order that the number of gears employed **to be minimum**.

A number of intermediate shafts are necessary to obtain required number of steps of speed. For example, if **12 spindle speeds are required** ranging from n_1 to n_2 from a constant speed motor two intermediate shafts are necessary as shown in earlier figure

The number of speeds from motor shaft (driving shaft) to intermediate shaft 1 is (2). That from shaft 1 to shaft 2 is (3) and that from shaft 2 to the driven shaft is (2).

Then the number of spindle speeds is equal to 2 x 3 x 2 = 12.

The speed of driving shaft, driven shaft and intermediate shafts 1 and 2 can be conveniently represented by **speed diagram or structural diagram**.

SPEED DIAGRAM

Contd.

It is seen that the driving shaft is running at one constant speed, the intermediate shaft 1 can run at two speeds, intermediate shaft 2 can run at six speeds and the driven spindle shaft can run at twelve speeds.

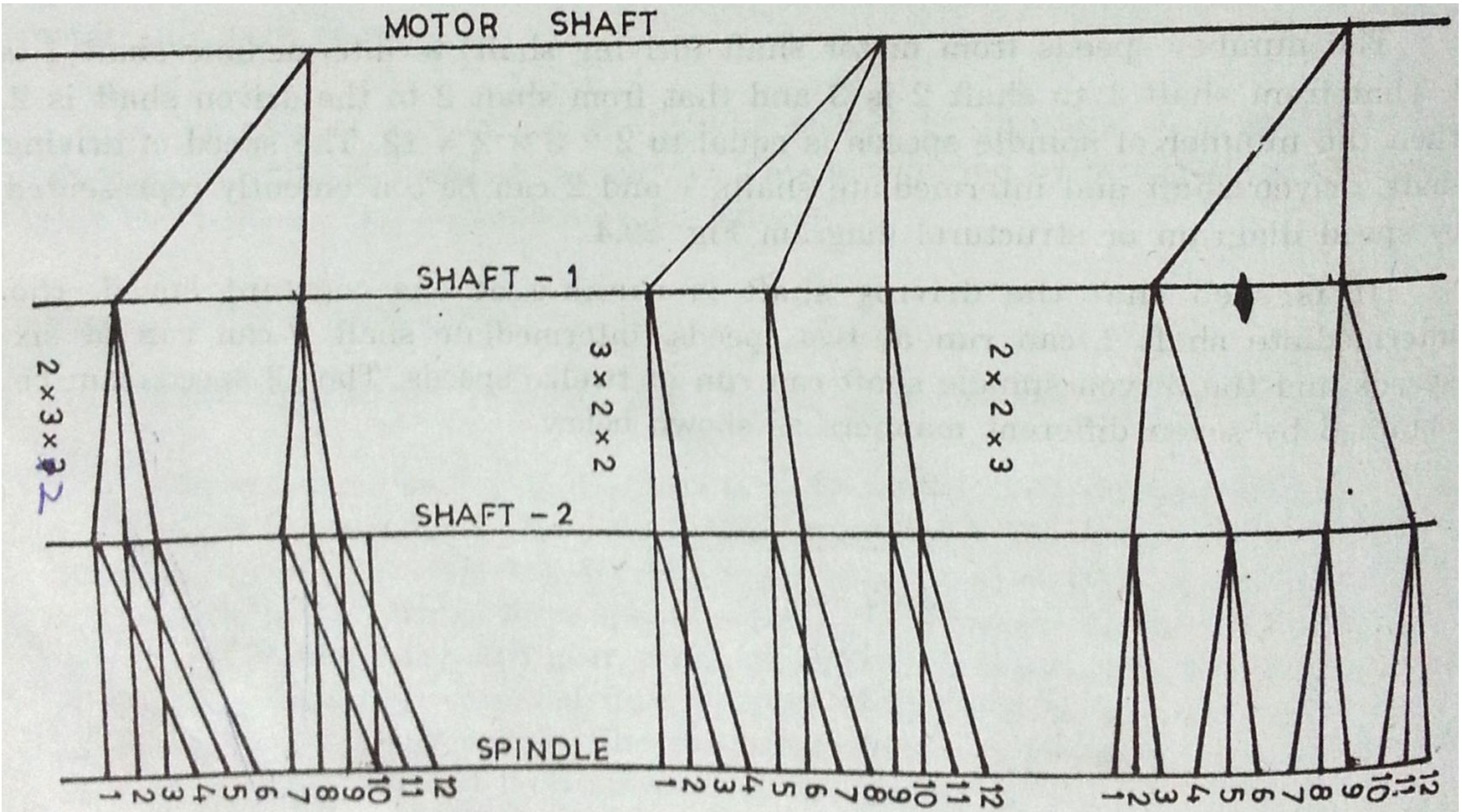
The 12 speeds can be obtained by seven different manners as shown below.

$$\left.\begin{array}{l} \mathbf{2 \times 3 \times 2} \\ \mathbf{2 \times 2 \times 3} \\ \mathbf{3 \times 2 \times 2} \\ \mathbf{4 \times 3} \\ \mathbf{3 \times 4} \\ \mathbf{6 \times 2} \\ \mathbf{2 \times 6} \end{array}\right\} = 12 \text{ speeds}$$

SPEED DIAGRAM

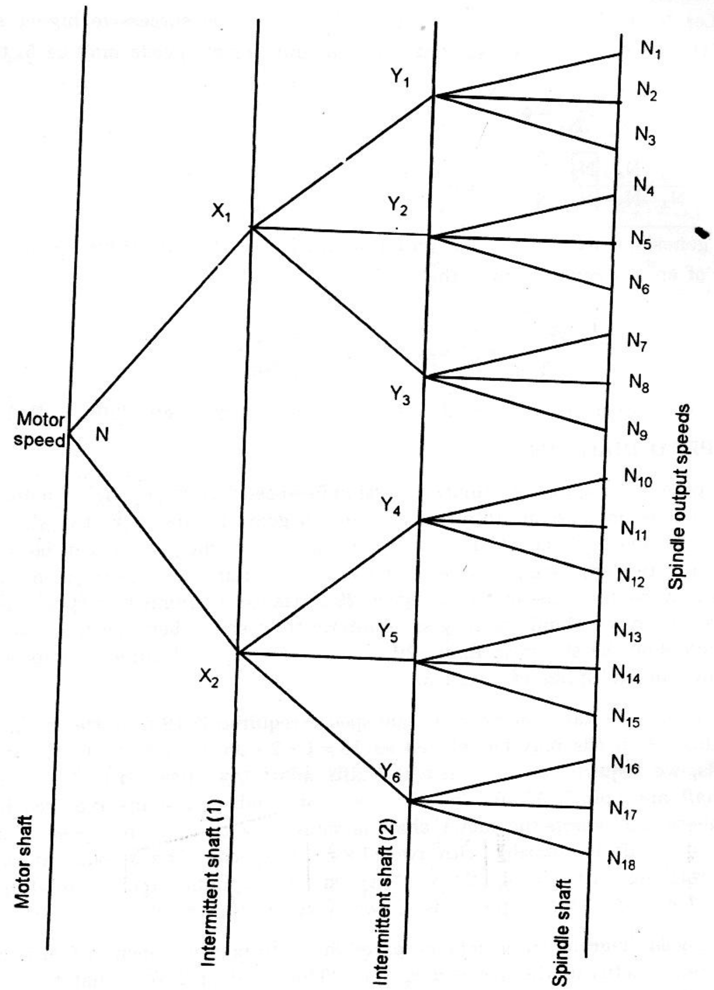
Contd.

Some possible structures for 12 Speed

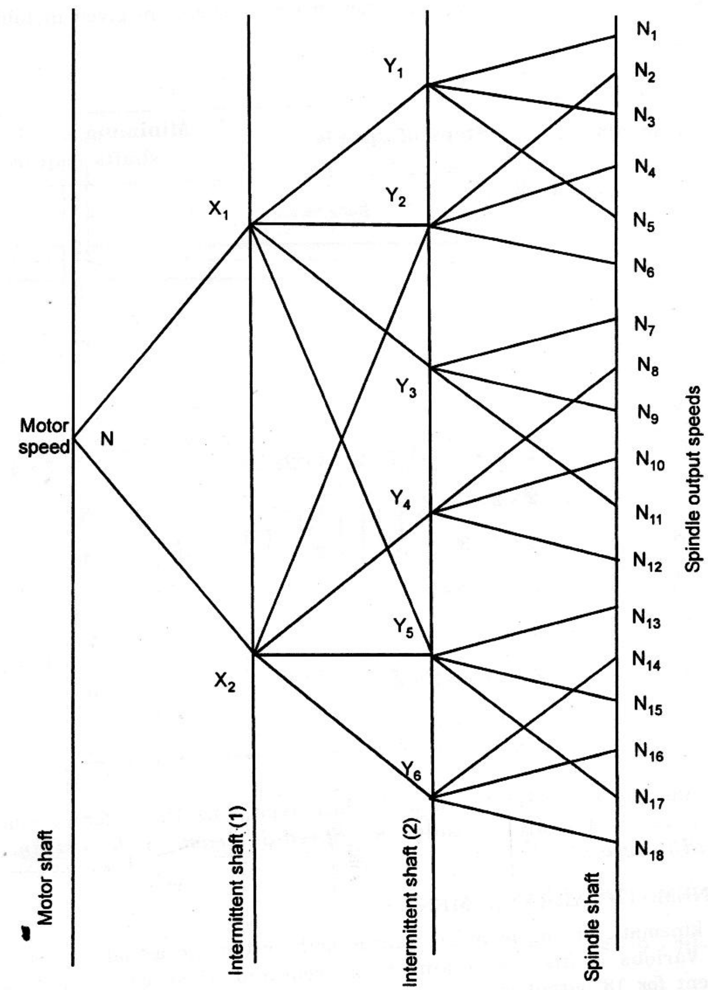


SPEED DIAGRAM

Contd.



Speed diagram for 18 output speeds



Speed diagram for 18 output speeds

Total Speeds	Steps of speeds	Minimum number of shafts required
2	1×2	2
3	1×3	2
4	$1 \times 2 \times 2$	3
6	$1 \times 2 \times 3$ (or) $1 \times 3 \times 2$	3
8	$1 \times 2 \times 2 \times 2$	4
9	$1 \times 3 \times 3$	3
12	$1 \times 2 \times 2 \times 3$ (or) $1 \times 2 \times 3 \times 2$	4
16	$1 \times 2 \times 2 \times 2 \times 2$	5
18	$1 \times 2 \times 3 \times 3$	4
24	$1 \times 2 \times 2 \times 2 \times 3$	5
27	$1 \times 3 \times 3 \times 3$	4
32	$1 \times 2 \times 2 \times 2 \times 2 \times 2$	6
36	$1 \times 2 \times 2 \times 3 \times 3$	5

The diagram shown in figure 29.2 which represents the different number of speeds in different shafts is called as ***'Speed-diagram*** or ***Ray diagram*** or ***Structural diagram'***.

PROGRESSION RATIO

Since the spindle speeds are arranged in geometric progression, the ratio of adjacent speeds will be a constant and it is denoted by φ.

Let N_1 be the minimum speed, N_2, N_3, N_4 be the successive higher speeds, and N_5 the maximum speed, so that the total number of speeds may be 5, then

$$\frac{N_5}{N_4} = \frac{N_4}{N_3} = \frac{N_3}{N_2} = \frac{N_2}{N_1} = \phi$$

or $$\frac{N_5}{N_1} = \frac{N_5}{N_4} \cdot \frac{N_4}{N_3} \cdot \frac{N_3}{N_2} \cdot \frac{N_2}{N_1} = \phi^4 = \phi^{(5-1)}$$

In general, if N_r, be the maximum speed, N_1 be the minimum speed and the number of speeds required is r, then

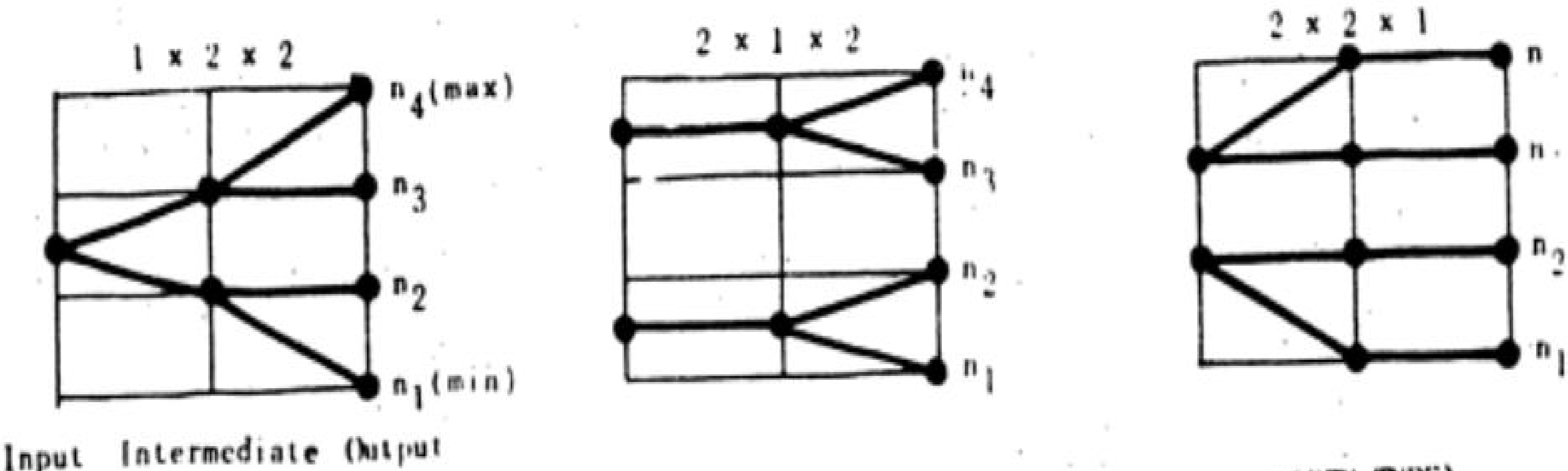
$$\frac{N_{max}}{N_{min}} = \frac{N_r}{N_1} = \phi^{(r-1)} \text{ (or) } \phi = \left[\frac{N_{max}}{N_{min}}\right]^{\frac{1}{(r-1)}}$$

The progression ratio φ should be a standard one and is taken either from R20 or R40 series from standard data.

Ray Diagrams

Ray diagram displays exact location of speed and then a ray diagram helps in calculating gear ratios. Various ray diagrams can result from a single structure diagram and these diagrams can be classified as unilateral, bilateral and skewed.

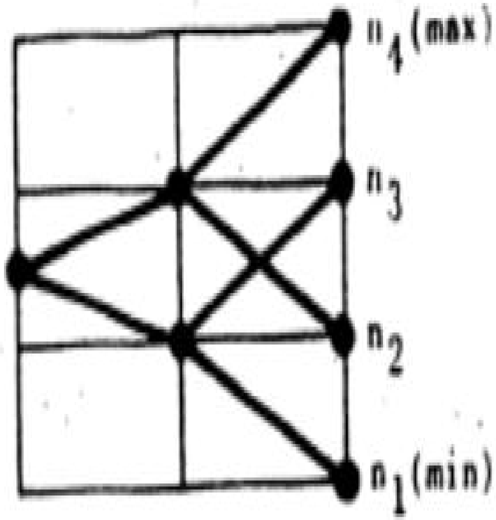
Example Let us consider 1×2×2 open type structure diagram of Fig.



(OPEN TYPE) (i)

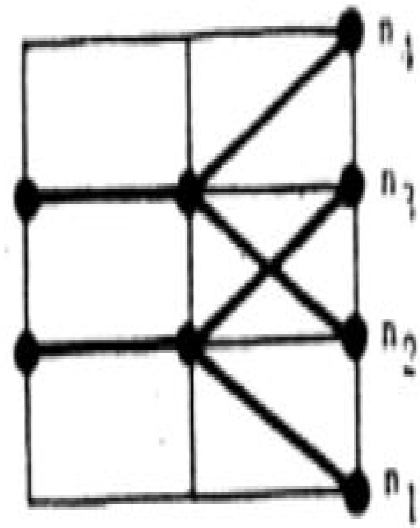
(OPEN TYPE) (ii)

(OPEN TYPE) (iii)



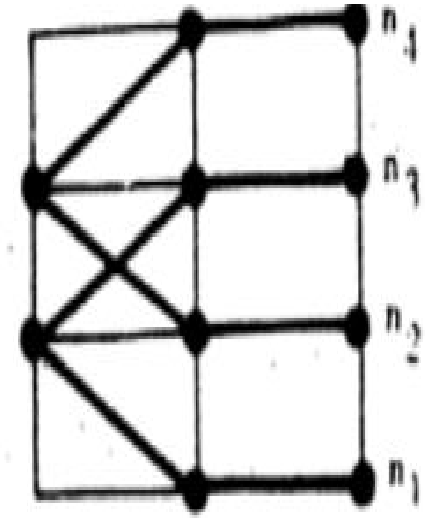
(CROSSED TYPE)

(iv)



(CROSSED TYPE)

(v)



(CROSSED TYPE)

(vi)

In this manner, many more ray-diagrams (both open and crossed type) can be drawn from a parent structure diagram. In fixing the input point it is, however, important to bear in mind that the input point should be preferably located towards higher speed to avoid large transformation ratio between the motcr shaft (which runs normally at a high speed) and the input shaft. With this consideration, ray diagrams (i), (v) are preferred ray diagrams.

Selection of Optimum Ray Diagram

An optimum ray diagram is one which will result in a compact gear box with less number of gears, shafts, bearings and shifting levers with a consequent minimisation of manufacturing cost. Size is minimised by minimising the shaft sizes subject to the constraints of ray and stage restriction. For minimising shaft sizes, summation of diameters of different gear box shafts have to be calculated from torque for alternative ray diagrams and the best layout is chosen. For a quick evaluation, node method is used which is illustrated later.

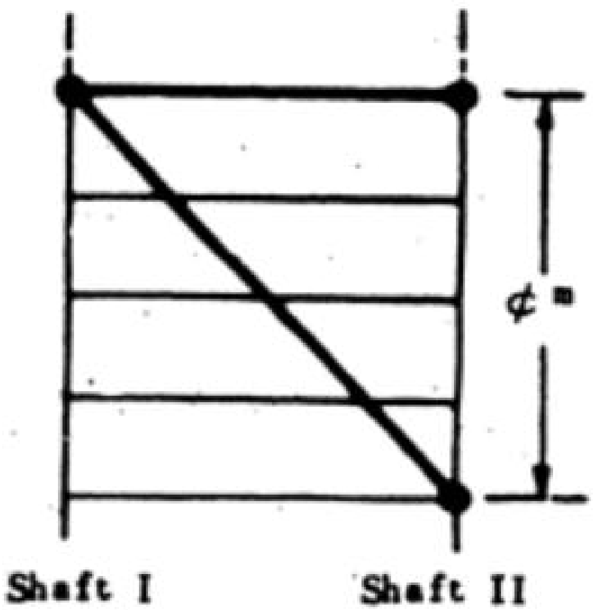
Constraints

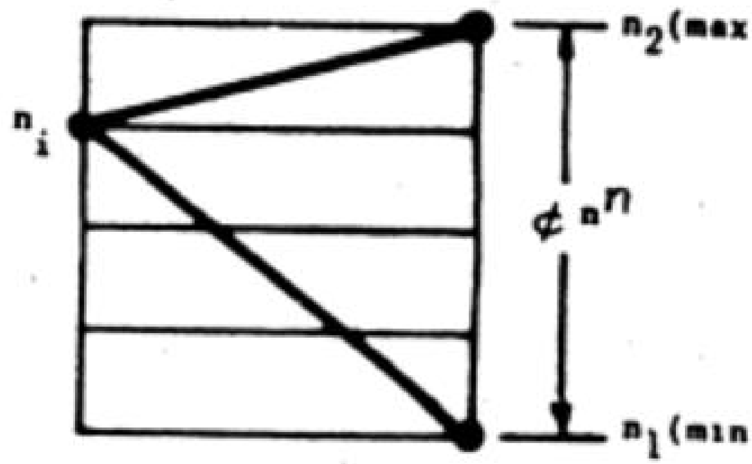
(i) *Ray restriction.* We know that in order to avoid interference, the minimum number of teeth in a set of gears should be greater than if (conveniently chosen as 20) and to avoid very large size gears, it should not be more than 120.

Therefore $|\phi^m|_{max} = \frac{120}{20} = 6$ where $m =$ no. of intervening spaces.

(ii) *Stage restriction*. We further know that due to limitations or space and pitch line velocity, the transformation ratio in a gear box is constrained in the limits

$$\frac{1}{4} \leqslant \text{transformation ratio} \leqslant \frac{2}{1}$$





Looking into the following ray diagram.

$$\therefore \qquad \frac{n_2}{n_1} \leqslant 2; \text{ and } \frac{n_1}{n_1} \leqslant 4 \;\; \frac{n_2}{n_1} \leqslant 2 \times 4 \leqslant 8$$

In a stage $\phi^n \leqslant 8$ where n is the number of intervening spaces between n_1 and n_2.

Node Method of Optimisation

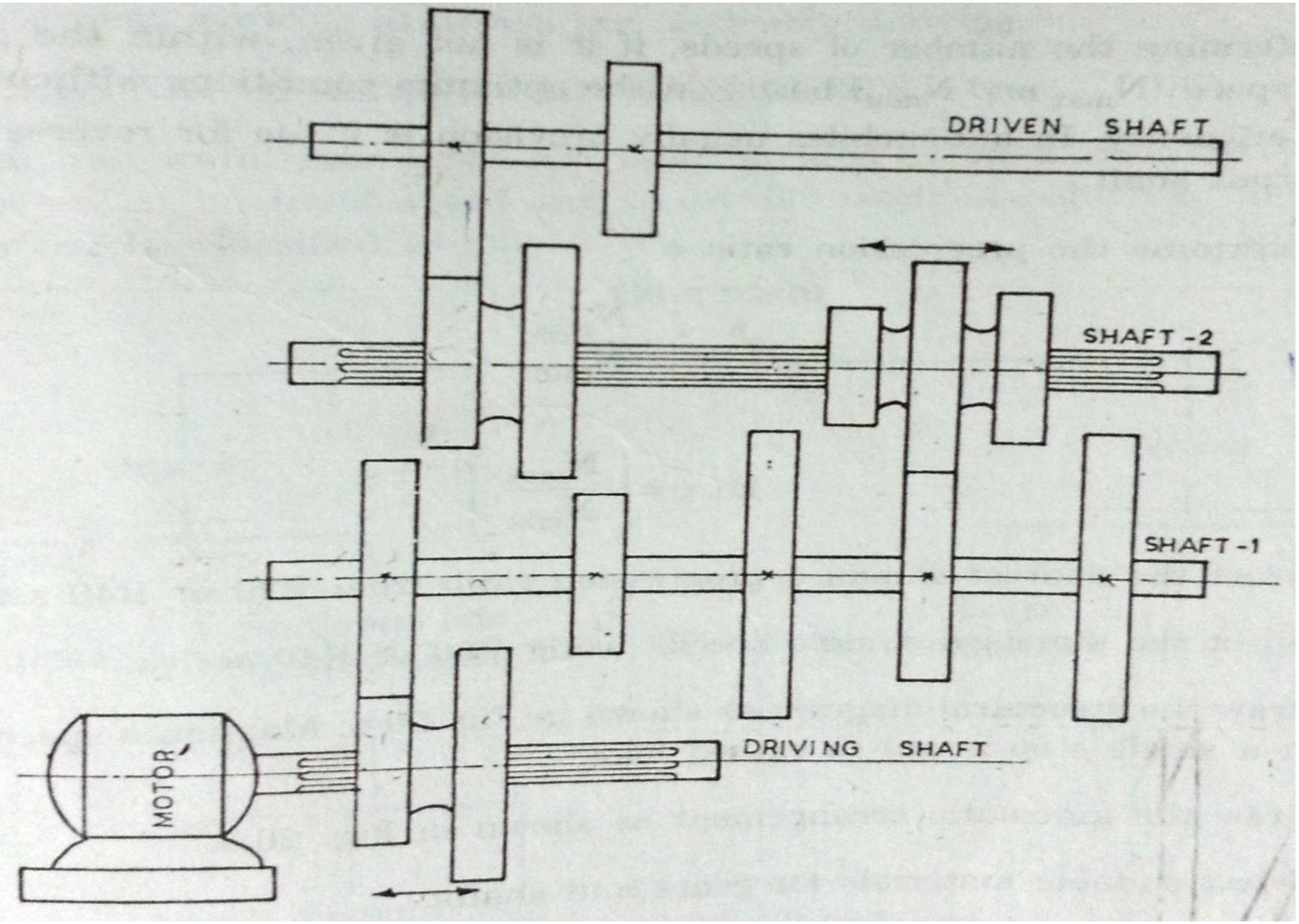
Node is a point from which a ray initiates or at which a ray terminates. Nodes are numbered from maximum speed and (corresponding to minimum torque) at each shaft and comparison is carried out with nodal sum (which, in fact, represents the sum of shaft diameters) as the criterion.

Ray Diagram	Shafts I	Shafts II	Shafts III	Nodal sum $\simeq \Sigma$ dia	Remarks
(i)	1	4	6	11	Best
(ii)	1	6	6	13	
(iii)	1	5	6	12	

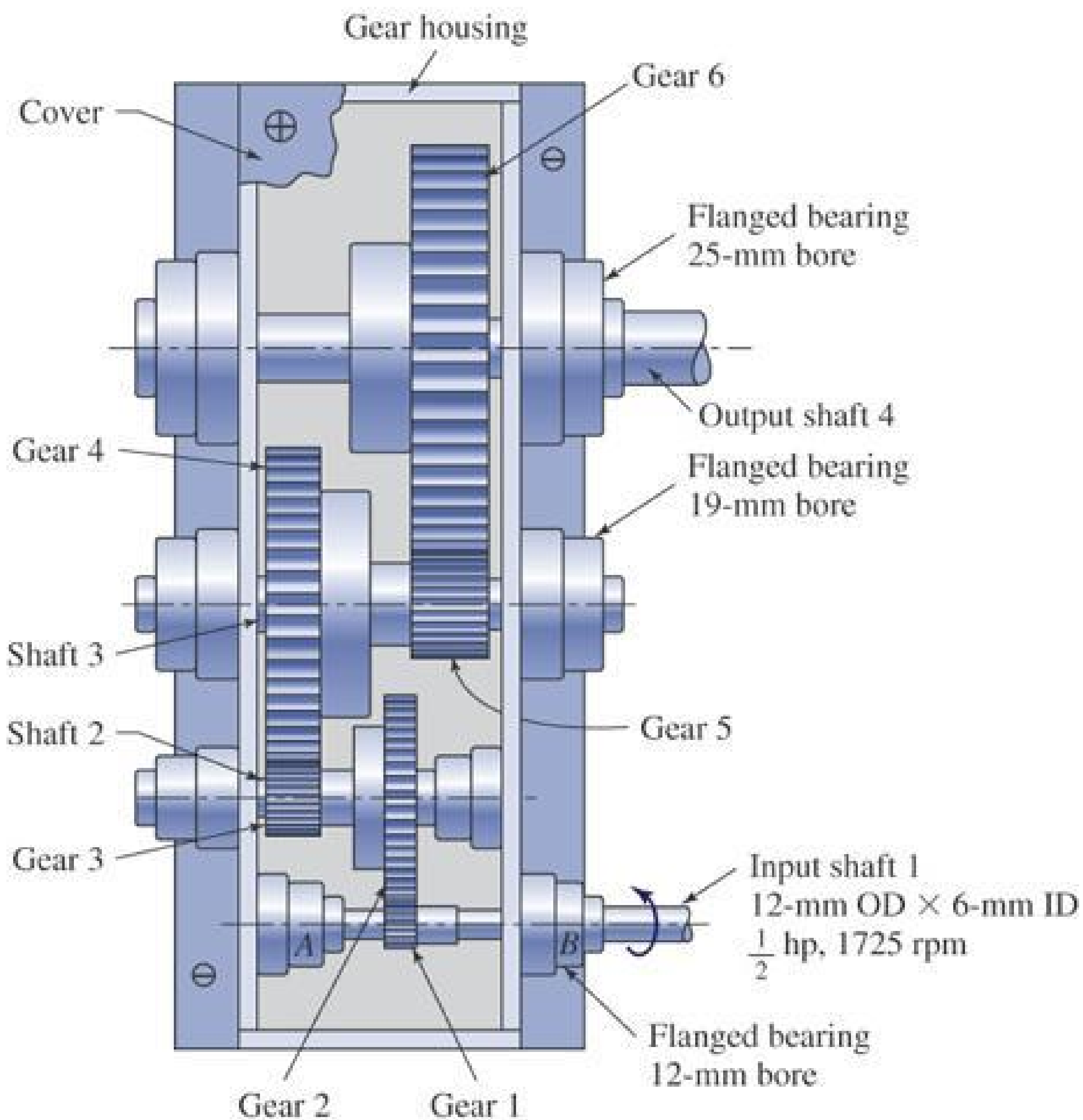
Therefore, type (i) is the best layout. Depending on the value of ϕ, it has to be, however, checked whether ray restrictions and stage restrictions are obeyed or not.

Kinematic Arrangement

In order to design gears in different shafts, the kinematic arrangement showing speeds of different shafts is drawn.

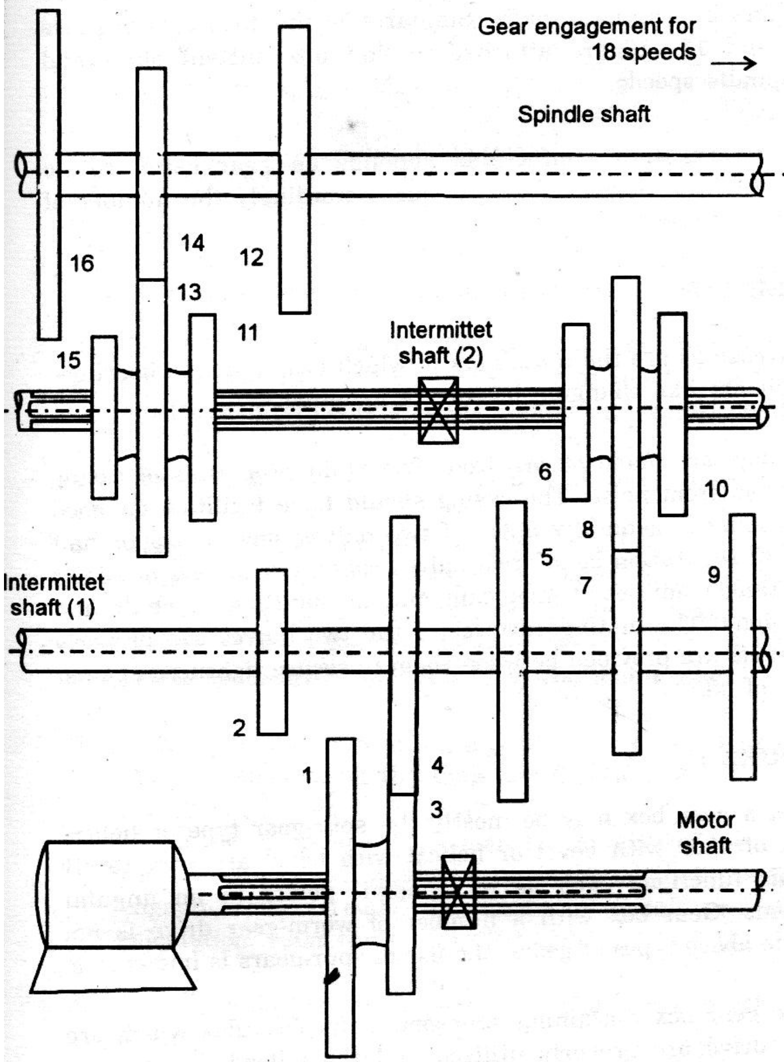


Gear Box Design



Kinematic Arrangement

Contd.



1. 1 – 2 – 5 – 6 – 11 – 12
2. 1 – 2 – 5 – 6 – 13 – 14
3. 1 – 2 – 5 – 6 – 15 – 16
4. 1 – 2 – 7 – 8 – 11 – 12
5. 1 – 2 – 7 – 8 – 13 – 14
6. 1 – 2 – 7 – 8 – 15 – 16
7. 1 – 2 – 9 – 10 – 11 – 12
8. 1 – 2 – 9 – 10 – 13 – 14
9. 1 – 2 – 9 – 10 – 15 – 16
10. 3 – 4 – 5 – 6 – 11 – 12
11. 3 – 4 – 5 – 6 – 13 – 14
12. 3 – 4 – 5 – 6 – 15 – 16
13. 3 – 4 – 7 – 8 – 11 – 12
14. 3 – 4 – 7 – 8 – 13 – 14
15. 3 – 4 – 7 – 8 – 15 – 16
16. 3 – 4 – 9 – 10 – 11 – 12
17. 3 – 4 – 9 – 10 – 13 – 14
18. 3 – 4 – 9 – 10 – 15 – 16

Fig. 29.3: Kinematic arrangement for 18 speeds

Design Procedure

1. From the given problem, decide the amount of power to be transmitted, motor speed, number of speeds required, maximum and minimum speeds, available space etc. For compact drive stronger materials such as alloy steels may be selected, usually all the gears may be made of same material.

2. Findout the progression ratio (or) step ratio 'ϕ' as

$$\phi = \left[\frac{N_{max}}{N_{min}}\right]^{\frac{1}{r-1}}$$

where N_{max} = Maximum speed.

N_{min} = Minimum speed.

r = Number of speeds required

The spindle speeds are standardised from R40 series.

3. Draw the speed diagram for which the number of shafts are selected based on the gear ratio which should not be more than four in a single step. The speeds of intermediate shafts should be marked on them.

Design Procedure

Contd.

4. Draw the kinematic arrangement according to speed diagram.

5. Compute the minimum centre distance between the shafts based on surface compressive stress considering the worst condition (i.e., maximum power and lowest speed condition). Usually, the determination of centre distance should be started from the spindle shaft, and then the design is proceeded to other shafts successively and finally to the motor shaft.

6. Calculate the minimum module based on design bending stress and standardise it using the table 25.16 (PSG 8.2).

7. Using the same module, findout the number of teeth of all the gears for that centre distance. It should be remembered that the total number of teeth of engaging pair is equal for the same module and same distance. For easy calculation, refer the tables 29.2 (PSG 8.6 to 8.12) (JDB 25.22).

8. Using similar procedure, find out the teeth of other gears fitted in other shafts.

9. Calculate the actual spindle speeds based on the designed number of teeth of all gears and their variation from the required speeds are tabulated for comparison.

10. Design the other elements of gear-box such as shafts, keys, bearings and gear changing levers etc and then draw arrangement of gear box neatly.

NUMBER OF TEETH IN A DRIVE FOR STANDARD RATIOS

Z, Total number of teeth $= Z_1 + Z_2$

Z_1, Number of teeth in driving gear

Z_2, Number of teeth in driven gear

i, Standard ratio, $\frac{Z_1}{Z_2}$

S_1, % deviation $= \left[\frac{\text{Actual ratio} - \text{Standard ratio}}{\text{Standard ratio}}\right] \times 100$

Values tabulated are :

$$Z_1 : \overset{\pm s}{Z_2}$$

Design Procedure

Contd.

Table 29.2(a)

i/Z	50	51	52	53	54	55	56	57	58	59
1	25:25		26:26		27:27		28:28		29:29	
1.06							$27{:}29^{0.4}$		$28{:}30^{-11}$	
1.12		$24{:}27^{0.3}$		$25{:}28^{0.2}$		$26{:}29^{0.5}$		$27{:}30^{-11}$		$28{:}31^{1.3}$
1.26	$22{:}28^{-11}$		$23{:}29^{0.2}$		$24{:}30^{0.7}$		$25{:}31^{1.5}$			$26{:}33^{0.5}$
1.41		$21{:}30^{-11}$		$22{:}31^{0.2}$		$23{:}32^{1.5}$			$24 : 34^{0.3}$	
1.58		$19{:}32\ F^{0.3}$	$20{:}32^{0.5}$		$21{:}33^{0.9}$			$22{:}35^{0.4}$		$23{:}36^{1.3}$
1.78	18:32			$19{:}34^{0.5}$			$20{:}36^{1.2}$		$21{:}37^{0.9}$	
2		$17{:}34^{-0.2}$			$18{:}36^{-0.2}$			$19{:}38^{-0.2}$		
2.24			$16{:}36^{-0.5}$			$17{:}38^{0.2}$				
2.51				$15{:}38^{-0.8}$			$16{:}40^{0.5}$			
2.82	$13{:}37^{-1.0}$			$14{:}39^{1.2}$	$14{:}40^{-1.4}$			$15{:}42^{0.7}$		
3.16	$12{:}38^{-0.1}$				$13{:}41^{0.2}$				$14{:}44^{0.5}$	
3.55					$12{:}42^{1.4}$	$12{:}43^{-1.1}$				13:469.3
3.98										$12.471.^{16}$

Design Problem

Design the layout of a 12 speed gear box for a lathe. The minimum and maximum speeds are 100 and 1200 rpm. Power is 5 kW from 1440 rpm. induction motor. Construct the speed diagram using a standard speed ratio. Calculate the number of teeth in each gear wheel and sketch the arrangement of the gear box.

(Anna University)

Solution :

Power	P	=	5 k.W.
Motor Speed	N	=	1440 rpm
Maximum output speed	N_{max}	=	1200 rpm
Minimum output speed	N_{min}	=	100 rpm
No. of speeds	r	=	12

$$\text{Progression ratio } \phi = \left(\frac{N_{max}}{N_{min}}\right)^{\frac{1}{r-1}} = \left(\frac{1200}{100}\right)^{\frac{1}{12-1}} = 1.2535$$

Nearest standard progression ratio $= 1.25$

Now the required twelve speeds are 100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250 rpm.

Let the arrangement of 12 speeds be $1 \times 2 \times 2 \times 3$

The structural or speed diagram and kinematic arrangement is then drawn which are shown in figure 29.4

Let the material for making all the gears is 40 Ni2Cr1 M_0 28 steel.

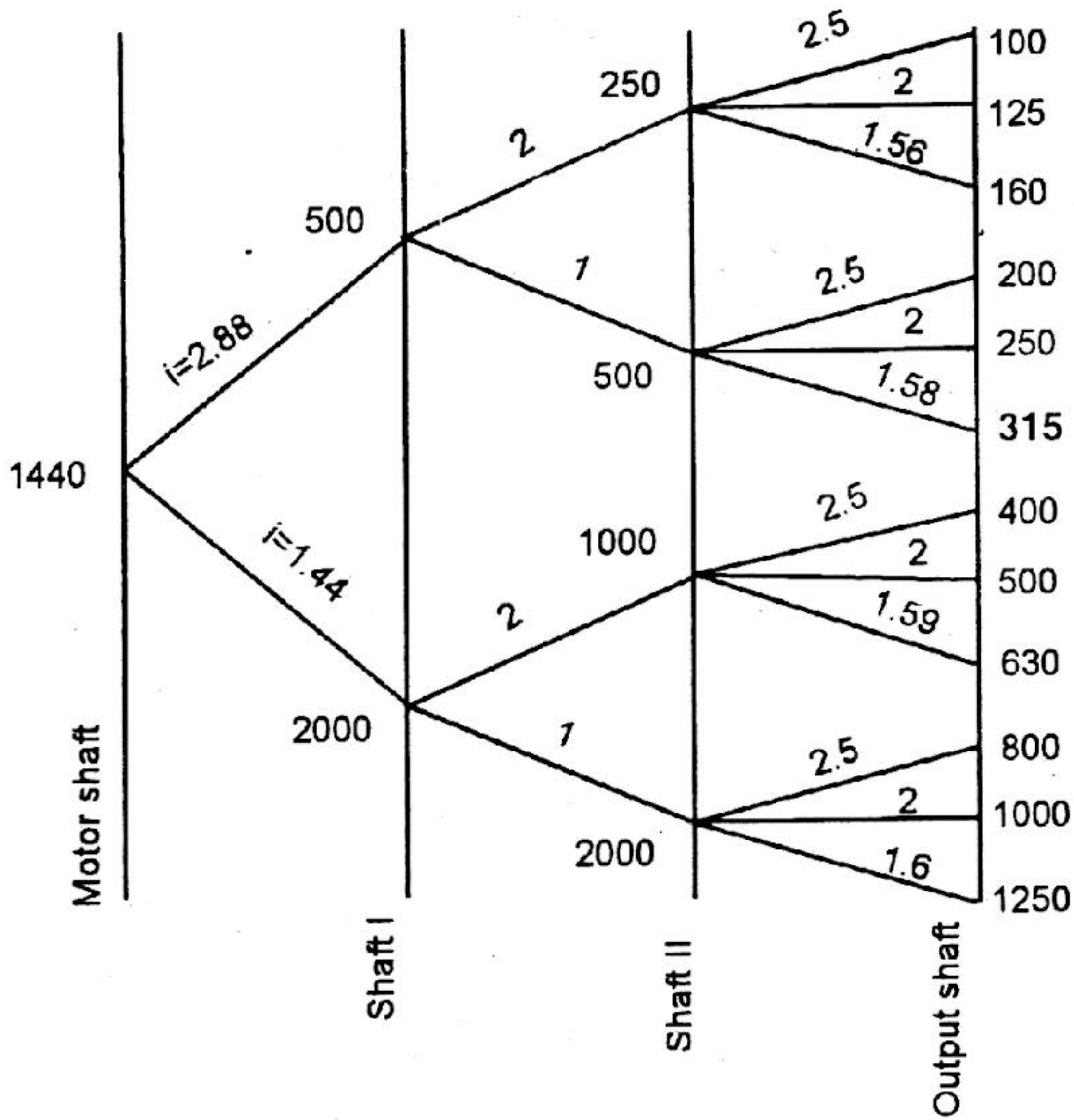
Hence $[\sigma_c] = 11000 \text{ kgf/cm}^2$

$[\sigma_b] = 4000 \text{ kgf/cm}^2$ (Table 25.7) (PSG 8.5) (JDB 25.9)

Design Problem

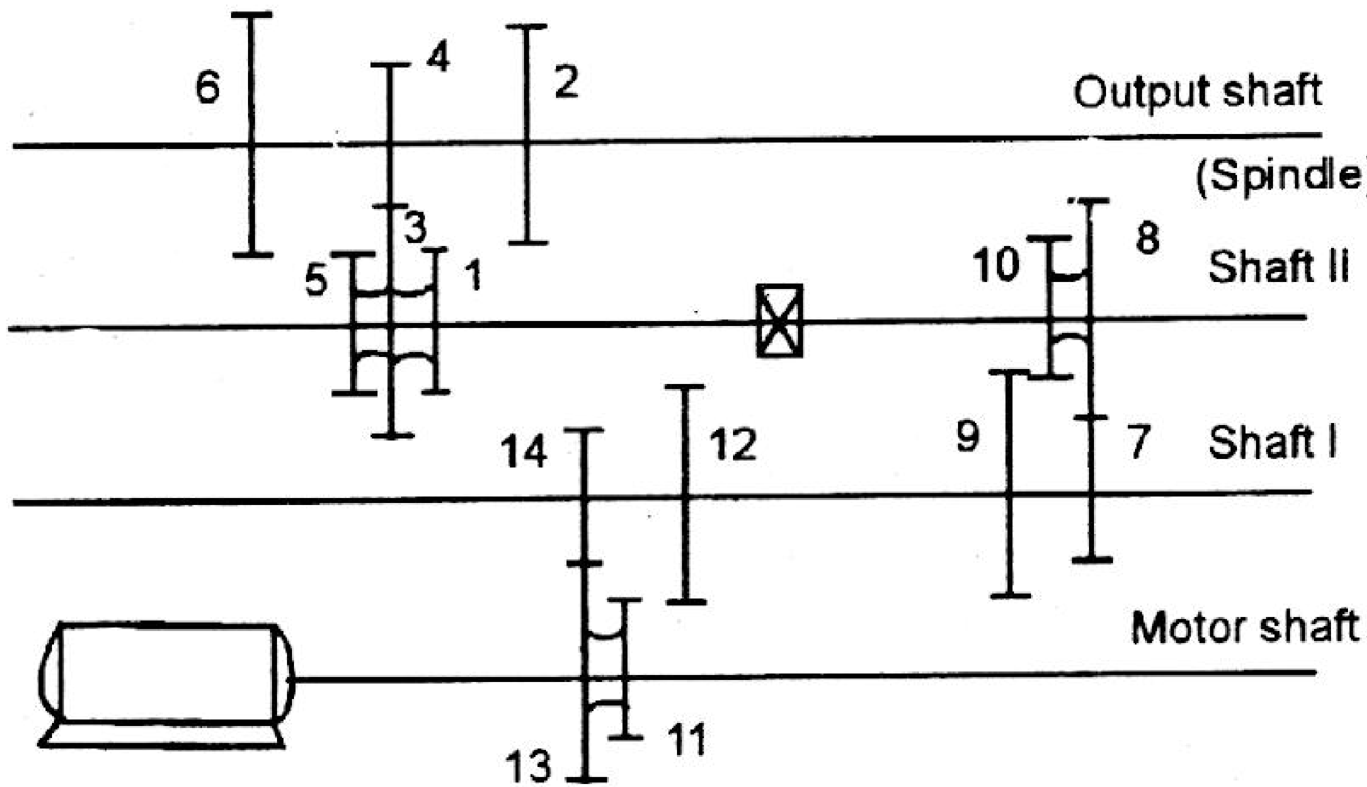
Contd.

Speed diagram (1X2X2X3)



Design Problem

Contd.



Kinematic arrangement

Speed diagram and kinematic arrangement

Design Problem

Contd.

Minimum centre disance between spindle shaft and shaft II

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{[\sigma_c]}\right)^2 \frac{E\,[M_t]}{i\,\psi}}$$

$[\sigma_c] = 11000 \text{ kgf/cm}^2$

$E = 2.15 \times 10^6 \text{ kgf/cm}^2$

$i = 2.5$

$\psi = \frac{b}{a} = 0.3$ (Assumed)

$[M_t] =$ Torque transmitted by pinion

$= 97420 \times \frac{k.W}{n} \times k.\,kd$ (Speed of gear (low) $= 100$ rpm

$= 97420 \times \frac{5}{250} \times 1.3$ Corresponding pinion speed $= 250$ rpm)

$= 2533 \text{ kgf} - \text{cm}$

$$\therefore\ a \geq (2.5+1)\sqrt[3]{\left(\frac{0.74}{11000}\right)^2 \times \frac{2.15 \times 10^6 \times 2533}{2.5 \times 0.3}}$$

≥ 11.2 cm

Design Problem

Contd.

Minimum module based on beam strength

$$m \geq 1.26 \times \sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m\,Z_1}}$$

$$[M_t] = 2533 \text{ kgf} - \text{cm}$$

$$[\sigma_b] = 4000 \text{ kgf/cm}^2$$

$$\psi_m = 10 \text{ (Assumed)}$$

$$Z_1 = 20 \text{ (Assumed)}$$

$$y = 0.389 \text{ (for } Z_1 = 20 \text{ teeth)}$$

Design Problem Contd.

$$\therefore m \geq 1.26 \times \sqrt[3]{\frac{2533}{0.389 \times 4000 \times 10 \times 20}}$$

$$\geq 0.253 \text{ cm} \geq 2.53 \text{ mm}$$

Next standard module $= 3 \text{ mm} = 0.3 \text{ cm}$.

Now, the number of teeth of pinion,

$$Z_1 = \frac{2a}{m(i+1)} = \frac{2 \times 11.2}{0.3\,(2.5+1)} = 21.3$$

Take $Z_1 = 22$ and $Z_2 = Z_1 \times i = 22 \times 2.5 = 55$

Final centre distance, $a = \frac{m}{2}(Z_1 + Z_2)$

$$= \frac{0.3}{2}(22 + 55) = 11.6 \text{ cm } \textbf{(O.K.)}$$

Also $Z_1 + Z_2 = 22 + 55 = 77$ teeth

Since the centre distance is same, the total number of teeth of engaging pair is equal.

ie., $(Z_3 + Z_4) = (Z_5 + Z_6) = (Z_1 + Z_2) = 77$

Since the gear ratio between Z_3 and Z_4 is 2,

let $Z_4 = 51$ and $Z_3 = 26$ so that $\frac{Z_4}{Z_3} = 1.96 \approx 2$

Similarly for the gear ratio between Z_5 and Z_6 as 1.56, let $Z_5 = 30$ and $Z_6 = 47$

Centre distance

Design Problem

Contd.

Centre distance between shaft II and shaft I :

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{[\sigma_c]}\right)^2 \frac{E \times [M_t]}{i\,\psi}}$$

Now $[M_t] = 97420 \times \dfrac{k.W}{n} \times k.kd$

$$= 97420 \times \frac{5}{500} \times 1.3 = 1267 \text{ kgf} - \text{cm}$$

$i = 2$

$$\therefore\ a \geq (2+1)\sqrt[3]{\left(\frac{0.74}{11000}\right)^2 \times \frac{2.15 \times 10^6 \times 1267}{2 \times 0.3}} \geq 8.2 \text{ cm}$$

Design Problem

Contd.

$$\text{Module (m)} \geq 1.26 \sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m\,Z_1}}$$

$[y = 0.389,\ [\sigma_b] = 4000\ \text{kgf/cm}^2$

$\psi_m = 10,\ Z_1 = 20$

$\therefore\ y\,[\sigma_b]\,\psi_m\,Z_1 = 311200]$

$$m \geq 1.26 \sqrt[3]{\frac{1267}{311200}} \geq 0.201$$

$$= 0.3\ \text{cm (or) 3 mm (Standard)}$$

$$\text{Now}\ Z_7 = \frac{2\,a}{m\,(i+1)} = \frac{2 \times 8.2}{0.3\,(2+1)} = 18.2 = 20\ \text{(say)}$$

$$Z_8 = i\,Z_7 = 2 \times 20 = 40$$

$$\therefore\ a = \frac{m}{2}(Z_7 + Z_8) = \frac{0.3}{2}(20 + 40) = 9\ \text{cm (O.K.)}$$

$$\because\ Z_9 + Z_{10} = Z_7 + Z_8 = 20 + 40 = 60;\ \text{and also}$$

$$\frac{Z_9}{Z_{10}} = 1\ ;\ \text{we have}\ Z_9 = Z_{10} = 30\ \text{teeth}$$

Centre distance between shaft I and Motor shaft :

$$a \geq (i+1)\sqrt[3]{\left\{\frac{0.74}{[\sigma_c]}\right\}^2 \frac{E\,[M_t]}{i\,\psi}}$$

$$i = 2.88$$

$$[M_t] = 97420 \times \frac{5}{1440} \times 1.3 = 440 \text{ kgf} - \text{cm}$$

$$\therefore\ a \geq (2.88+1)\sqrt[3]{\left\{\frac{0.74}{11000}\right\}^2 \times \frac{2.15 \times 10^6 \times 440}{2\ 88 \times 0.3}} \geq 6.61 \text{ cm}$$

$$m \geq 1.26\ \sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m\,Z_1}}$$

$$\geq 1.26\ \sqrt[3]{\frac{440}{311200}} \geq 0.14 \text{ cm} \geq 1.4 \text{ mm}$$

$$m = 1.5 \text{ mm} = 0.15 \text{ cm}$$

$[[\sigma_b] = 4000 \text{ kgf/cm}^2$

$\psi_m = 10.$

$Z_1 = 20,\ \ y = 0.389$

$\therefore\ y\,[\sigma_b]\,\psi_m\,Z_1 = 311200]$

Design Problem

Contd.

Now $Z_{11} = \dfrac{2 \times 6.61}{0.15\,(2.88 + 1)} = 22.7$

Let $Z_{11} = 24$ and $Z_{12} = 70$ so that $\dfrac{Z_{12}}{Z_{11}} = \dfrac{70}{24} = 2.91$

Now corrected centre distance,

$$a = \frac{m\,(Z_{11} + Z_{12})}{2} = \frac{0.15\,(24 + 70)}{2} = 7.0 \text{ cm}$$

Also $(Z_{13} + Z_{14}) = Z_{11} + Z_{12} = 24 + 70 = 94,$

We get $Z_{13} = 55;\ Z_{14} = 39$ so that

$$\frac{Z_{13}}{Z_{14}} = \frac{55}{39} = 1.41 \simeq \quad \frac{n_{14}}{n_{13}} = \frac{2000}{1440} = 1.39$$

Design Problem

Contd.

Face width (b)

$$b = \psi \cdot a \text{ or } \psi_m \cdot m$$

Taking minimum value as,

$$b = 0.3 \times 7 = 2.1 \text{ cm}$$

$$\text{(or) } b = 10 \times 0.15 = 1.5 \text{ cm}$$

For worst conditions assume,

$$b = 2.1 \text{ cm and } a = 7 \text{ cm}$$

$$[M_t] = 440 \text{ kgf} - \text{cm} \; ; \; i = 2.88$$

For the above parameters, the induced stresses are estimated as follows.

$$\text{i.e., } \sigma_c = 0.74 \left(\frac{i+1}{a}\right) \sqrt{\frac{(i+1)}{i\,b} E\,[M_t]}$$

$$= 0.74 \left(\frac{2.88+1}{7}\right) \sqrt{\frac{(2.88+1)}{2.88 \times 2.1} \times 2.15 \times 10^6 \times 440}$$

$$= 10105 \text{ kgf/cm}^2 < [\sigma_c] = 11000 \text{ kgf/cm}^2$$

$$\sigma_b = \frac{(i+1)\,[M_t]}{ab\,my} = \frac{(2.88+1)\,440}{7 \times 2.1 \times 0.15 \times 0.389}$$

$$= 1990 \text{ kgf/cm}^2 < [\sigma_b] = 4000 \text{ kgf/cm}^2$$

Design Problem

Since the induced stresses for even the lowest parameters are less than their allowable values, our design is correct.

Therefore,

Centre distance between spindle shaft and shaft II = 11.6 cm

Centre distance between shaft II and shaft I = 9.0 cm

Centre distance between shaft I and motor shaft = 7.0 cm

The number of teeth of various gears are as follows

No. of teeth of gears (Z)	Gear ratio (i)
$Z_1 = 22; Z_2 = 55$	2.5
$Z_3 = 26; Z_4 = 51$	1.96
$Z_5 = 30; Z_6 = 47$	1.57
$Z_7 = 20; Z_8 = 40$	2.0
$Z_9 = 30; Z_{10} = 30$	1.0
$Z_{11} = 24; Z_{12} = 70$	2.92
$Z_{13} = 55; Z_{14} = 39$	1.41

Calculation of actual speeds for different output shafts

Motor shaft speed, $N = 1440$ rpm

Speeds for Shaft I :

$$Y_1 = \frac{1440}{(70/24)} = 493 \text{ rpm and } Y_2 = \frac{1440}{(39/55)} = 2030 \text{ rpm}$$

Speeds for shaft II :

$$X_1 = \frac{493}{(40/20)} = 247 \text{ rpm}; X_2 = \frac{493}{(30/30)} = 493 \text{ rpm}$$

$$X_3 = \frac{2030}{(40/20)} = 1015 \text{ rpm}; X_4 = \frac{2030}{(30/30)} = 2030 \text{ rpm}$$

Speeds for spindle shaft :

$$N_1 = \frac{247}{(55/22)} = 99 \text{ rpm}; \; N_2 = \frac{247}{(51/26)} = 126 \text{ rpm}$$

$$N_3 = \frac{247}{(47/30)} = 157 \text{ rpm}; \; N_4 = \frac{493}{(55/22)} = 197 \text{ rpm}$$

$$N_5 = \frac{493}{(51/26)} = 252 \text{ rpm}; \; N_6 = \frac{493}{(47/30)} = 314 \text{ rpm}$$

$$N_7 = \frac{1015}{(55/22)} = 406 \text{ rpm}; \; N_8 = \frac{1015}{(51/26)} = 518 \text{ rpm}$$

$$N_9 = \frac{1015}{(47/30)} = 646 \text{ rpm}; \; N_{10} = \frac{2030}{(55/22)} = 812 \text{ rpm}$$

$$N_{11} = \frac{2030}{(51/26)} = 1035 \text{ rpm}; \; N_{12} = \frac{2030}{47/30} = 1292 \text{ rpm}$$

The actual speeds for various shafts and their variations are tabulated.

Tabulation of actual speeds and their variations

Design Problem

Contd.

Tabulation of actual speeds and their variations

Name of shaft.	Number of teeth in gears	Gear ratio	Actual speeds in rpm.	Preferred speed in rpm	Percentage of variations
	Z	i	A	B	$\frac{A-B}{B} \times 100$
Motor shaft	24 55		1440		
		2.92 1.41			
Shaft I	70 39 20 30		493 2030		
		2.0 1.0			
Shaft II	40 30 22 26 30		247 493 1015 2030		
		2.5 1.96 1.57			

Design Problem Contd.

Name of shaft.	Number of teeth in gears	Gear ratio	Actual speeds in rpm.	Preferred speed in rpm	Percentage of variations
Spindle shaft	55		99	100	– 1
	51		126	125	0.8
	47		157	160	– 1.9
			197	200	– 1.5
			252	250	0.8
			314	315	– 0.3
			406	400	1.5
			518	500	3.6
			646	630	2.5
			812	800	1.5
			1035	1000	3.5
			1292	1250	3.4

Design the shafts.

Select suitable bearings for the shafts.

Design the casing based mainly on manufacturing process and ease of assembly and disassembly of shafts and gears.

Design the gear changing levers.

Work out other components like keys, spacers, oil holes, oil level indicator, bearing seals, cooling fins etc.

Make neat working drawings and assembly drawing.

SELECTION OF STANDARD GEAR BOXES

Helical and worm gear boxes are manufactured as standard units to transmit required power at an input speed and to provide required reduction.

Helical gear boxes are employed where reduction per stage is less than 5 and where efficiency should be high. Besides, where self locking is not allowed, helical gear box is a must.

Worm gear boxes are used where high reduction is required with minimum space, though efficiency of these gear boxes are low as compared with helical gearbox. Maximum reduction per stage is restricted to 70, with two stage, it is possible to get reduction 70 x 70 = 900 with very compact space.

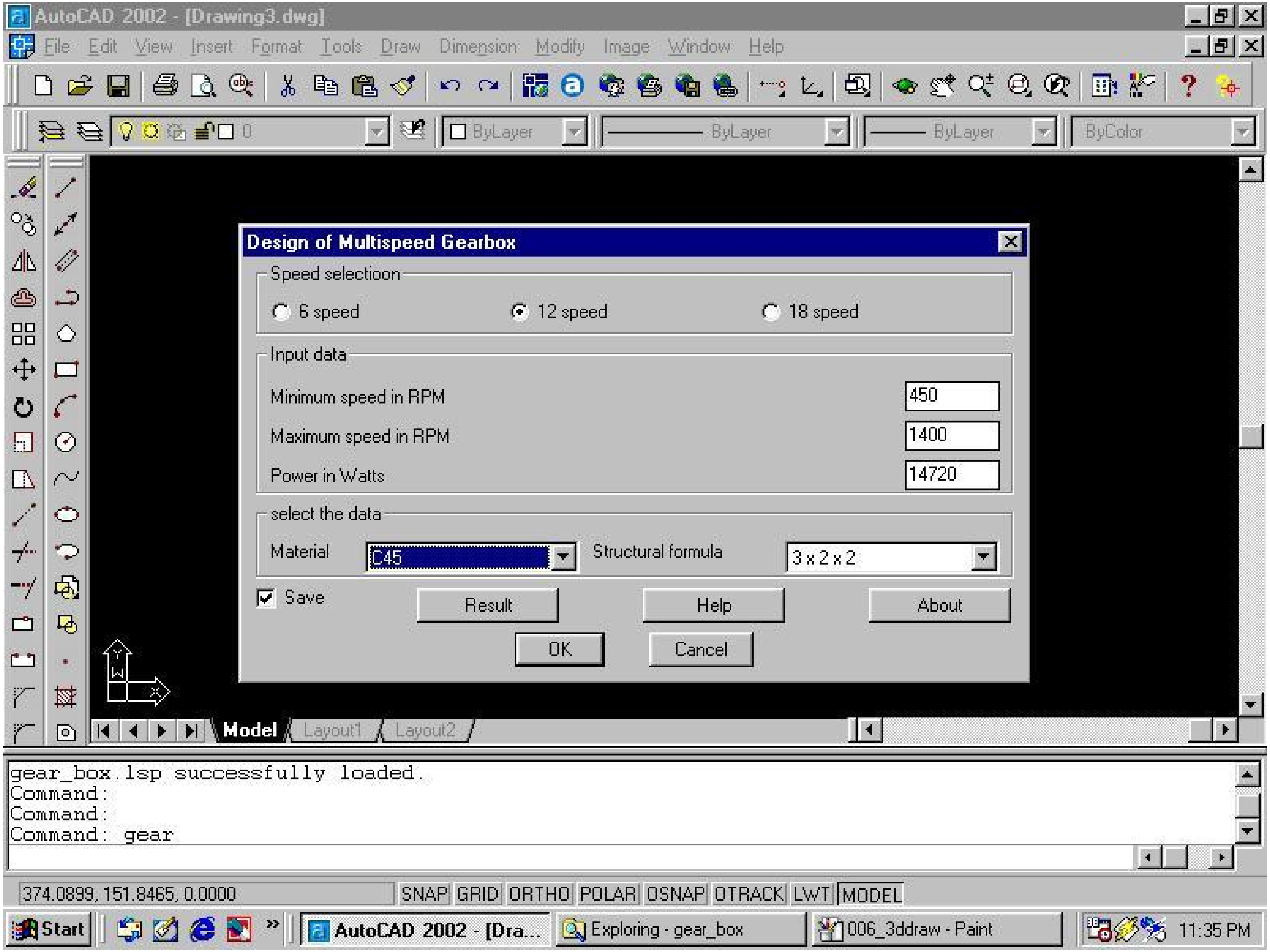
Worm gear boxes are used where high reduction is required with minimum space, though efficiency of these gear boxes are low as compared with helical gearbox. Maximum reduction per stage is restricted to 70, with two stage, it is possible to get reduction 70 x 70 = 900 with very compact space.

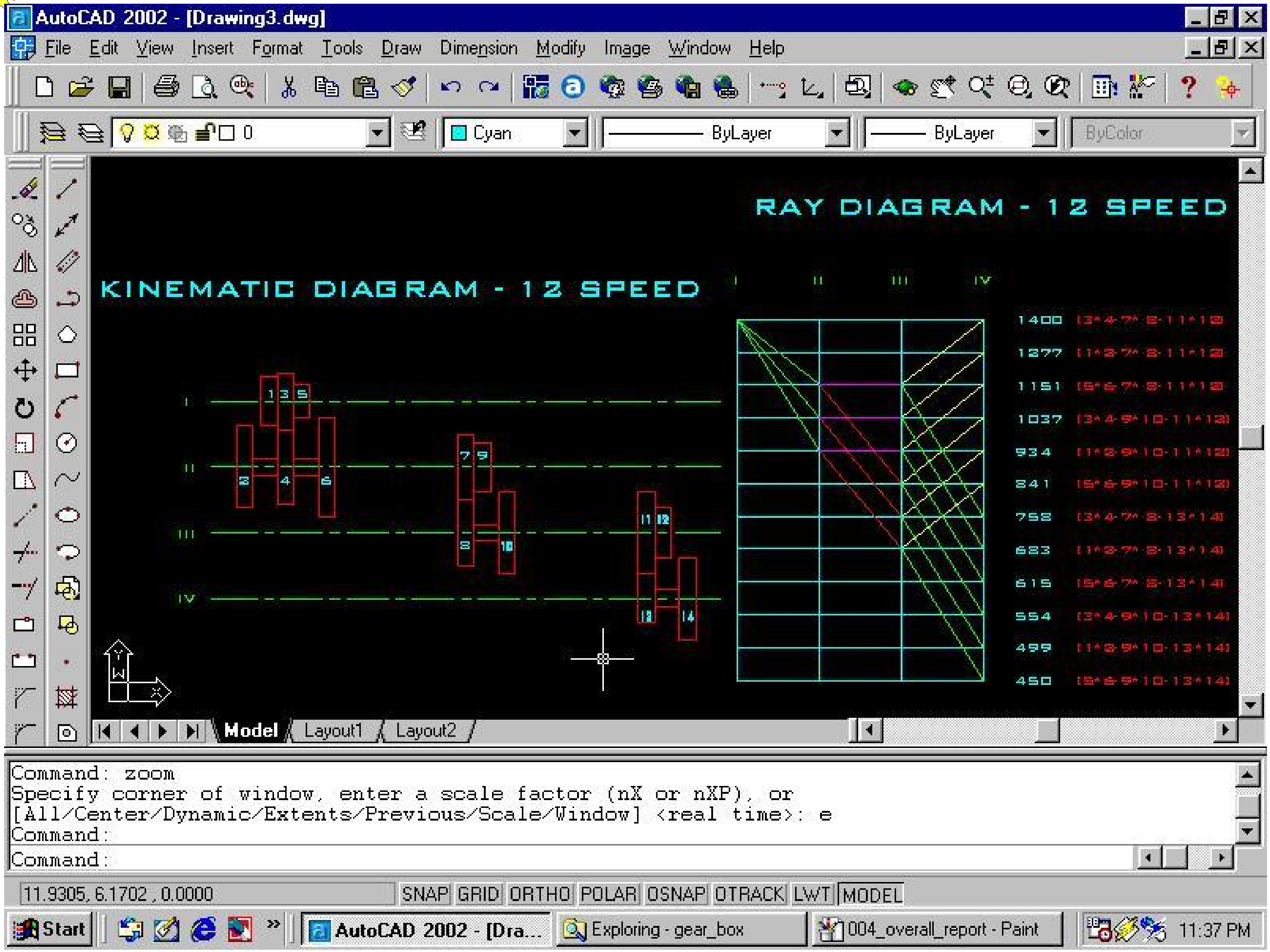
These gear boxes are used where self locking is desirable. This will avoid reversal of load under power failure.

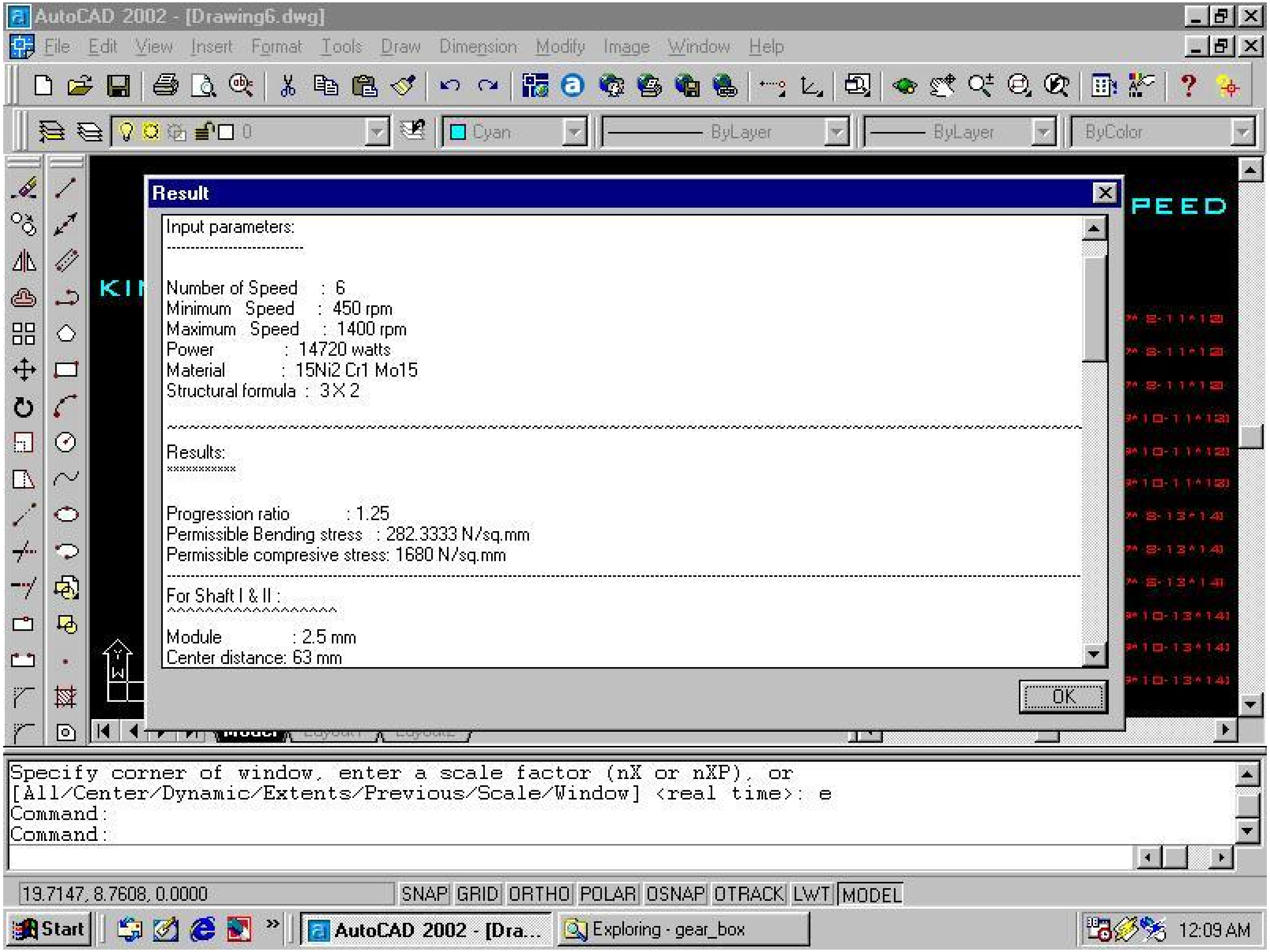
In such locations worm gears are provided, the inertia load of crane will act on worm wheel during breaking and damage may. occur.

Normally, for all higher power transmission, say above 25kW, invariably helical gear boxes are used.

COMPUTER AIDED DESIGN OF MULTI-SPEED GEAR BOX




AutoCAD 2002 - [Drawing3.dwg]
File Edit View Insert Format Tools Draw Dimension Modify Image Window Help
0
Cyan
ByLayer
ByLayer
ByColor
RAY DIAGRAM - 12 SPEED
KINEMATIC DIAGRAM - 12 SPEED
I II III IV
1 3 5
2 4 6
7 9
8 10
11 12
13 14
1400 (3^4-7^8-11^12)
1277 (1^2-7^8-11^12)
1151 (5^6-7^8-11^12)
1037 (3^4-9^10-11^12)
934 (1^2-9^10-11^12)
841 (5^6-9^10-11^12)
758 (3^4-7^8-13^14)
683 (1^2-7^8-13^14)
615 (5^6-7^8-13^14)
554 (3^4-9^10-13^14)
499 (1^2-9^10-13^14)
450 (5^6-9^10-13^14)
Model Layout1 Layout2
Command: zoom
Specify corner of window, enter a scale factor (nX or nXP), or
[All/Center/Dynamic/Extents/Previous/Scale/Window] <real time>: e
Command:
Command:
11.9305, 6.1702 , 0.0000
SNAP GRID ORTHO POLAR OSNAP OTRACK LWT MODEL
Start
AutoCAD 2002 - [Dra...
Exploring - gear_box
004_overall_report - Paint
11:37 PM


AutoCAD 2002 - [Drawing6.dwg]
File Edit View Insert Format Tools Draw Dimension Modify Image Window Help
Result
Input parameters:
Number of Speed : 6
Minimum Speed : 450 rpm
Maximum Speed : 1400 rpm
Power : 14720 watts
Material : 15Ni2 Cr1 Mo15
Structural formula : 3 X 2
Results:
Progression ratio : 1.25
Permissible Bending stress : 282.3333 N/sq.mm
Permissible compresive stress: 1680 N/sq.mm
For Shaft I & II :
Module : 2.5 mm
Center distance: 63 mm
OK
Specify corner of window, enter a scale factor (nX or nXP), or
[All/Center/Dynamic/Extents/Previous/Scale/Window] <real time>: e
Command:
Command:
19.7147, 8.7608, 0.0000
SNAP GRID ORTHO POLAR OSNAP OTRACK LWT MODEL
Start
AutoCAD 2002 - [Dra...
Exploring - gear_box
12:09 AM

Thank You !

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