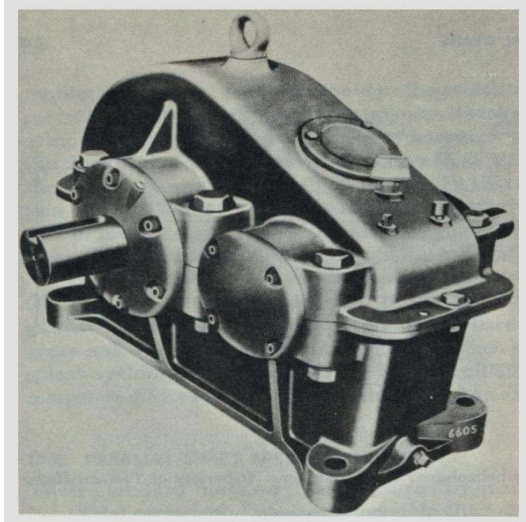


Lecture 17 – DESIGN OF GEARBOX

Contents

1. Commercial gearboxes
2. Gearbox design.



COMMERICAL GEARBOX DESIGN

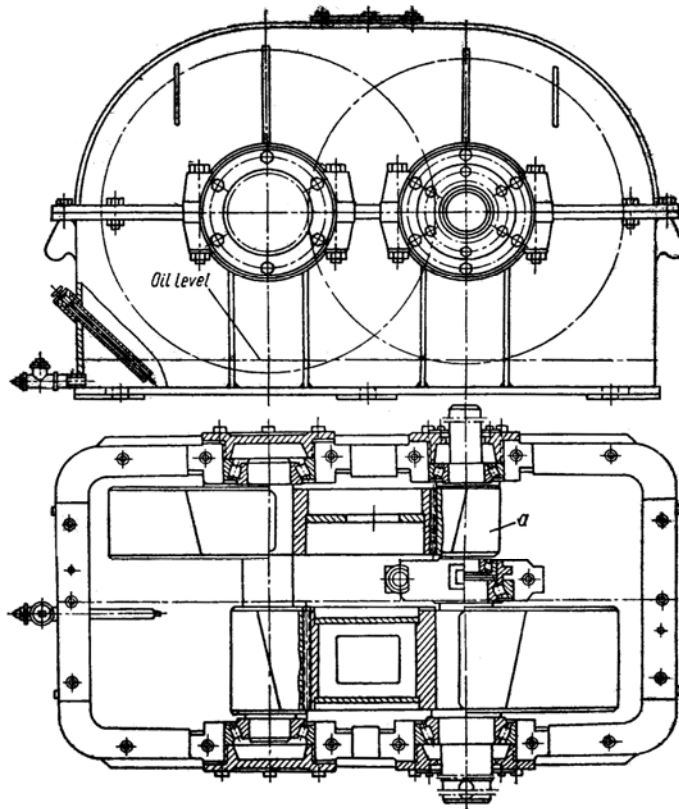


Fig. 1 Two stage helical gearbox

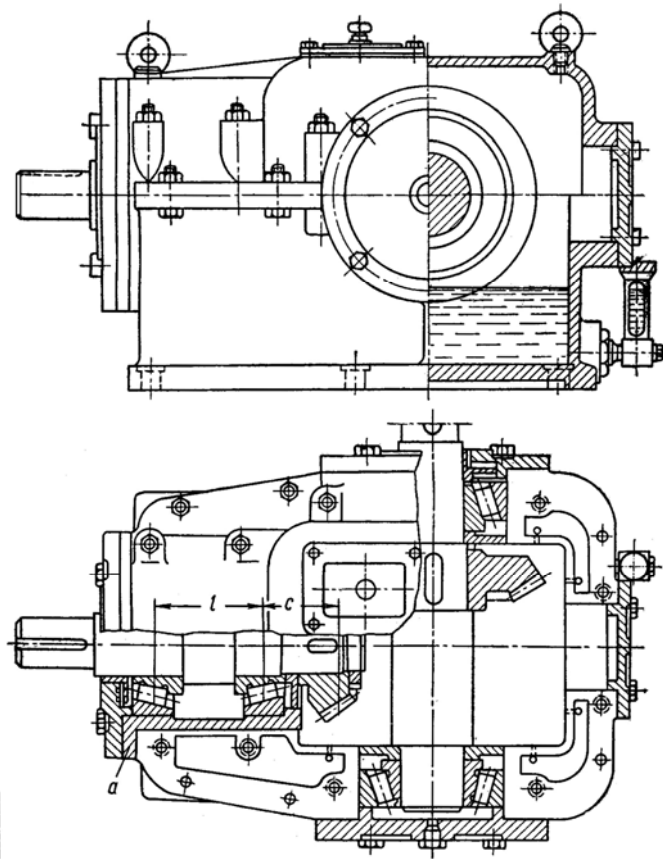


Fig. 2. A single stage bevel gearbox

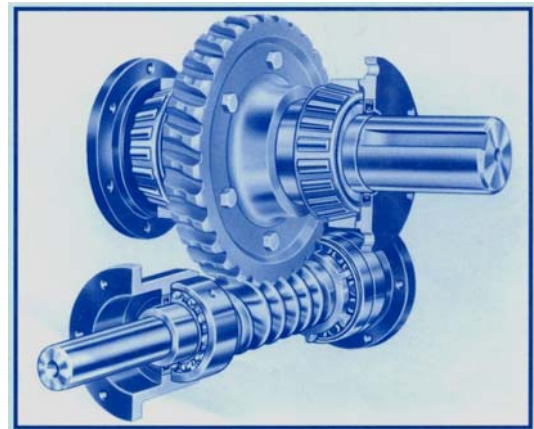
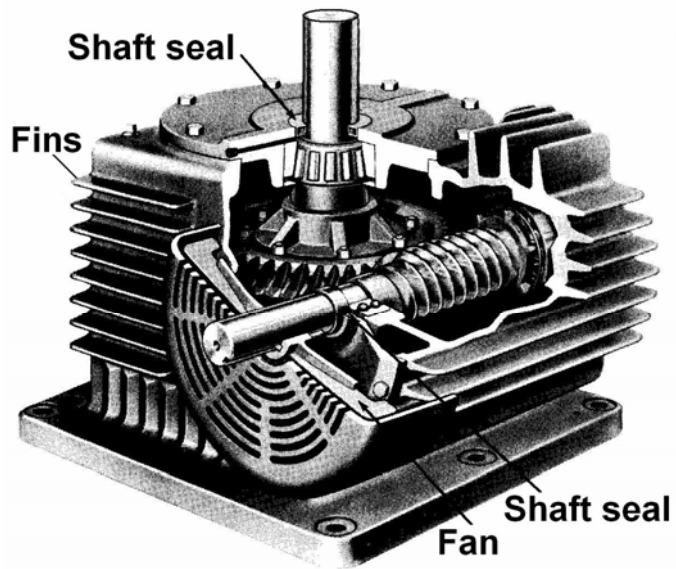


Fig.3 Worm gearbox.

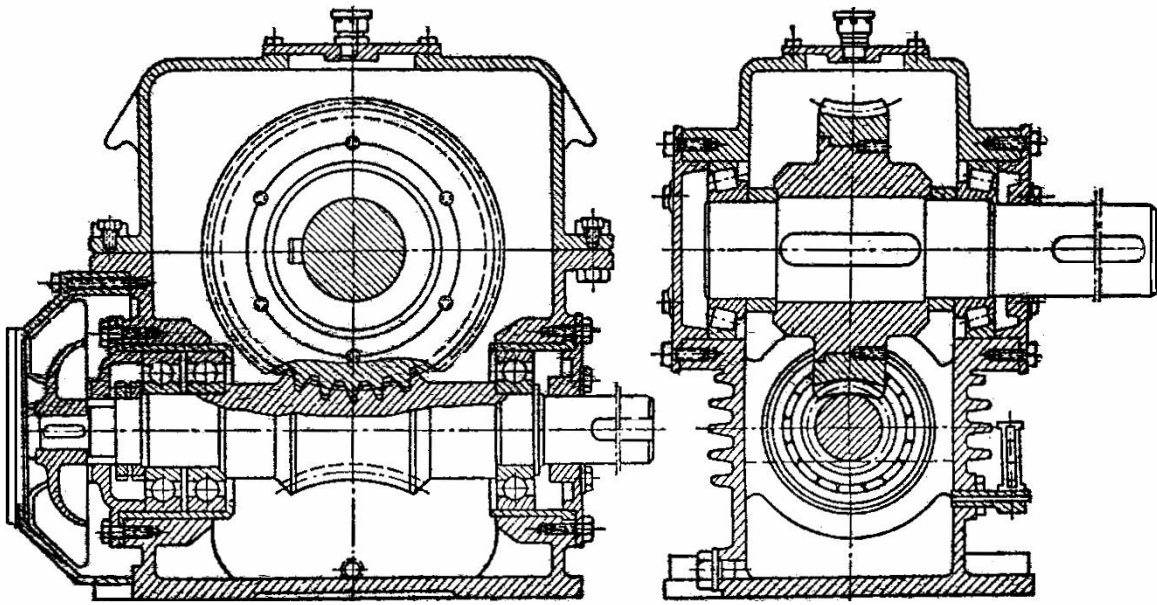


Fig. 4 Worm gearbox

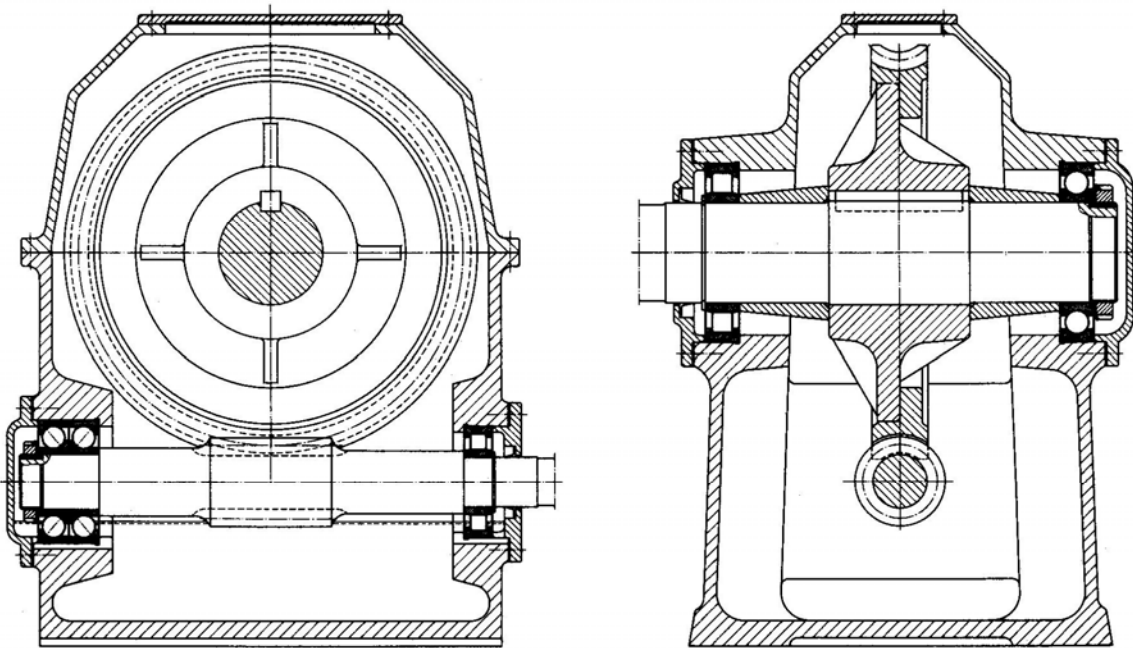


Fig.5. Worm gearbox.

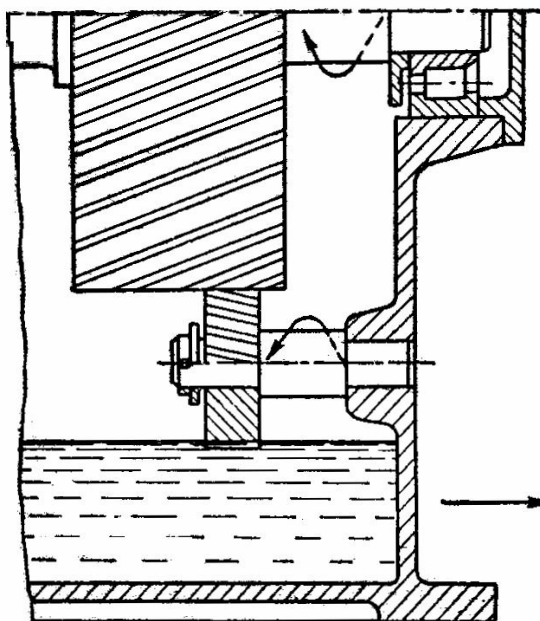


Fig.6. Helical gear lubrication with idler gear.

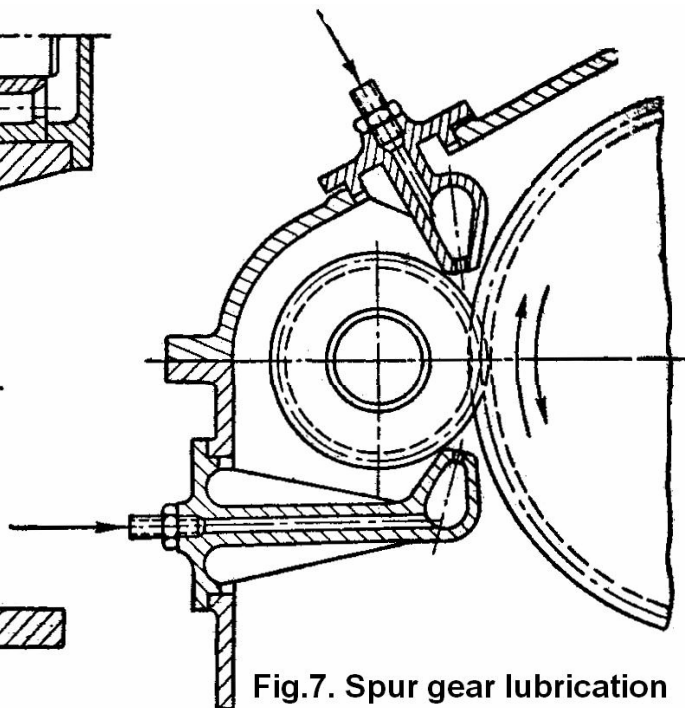


Fig.7. Spur gear lubrication with stream by nozzles.

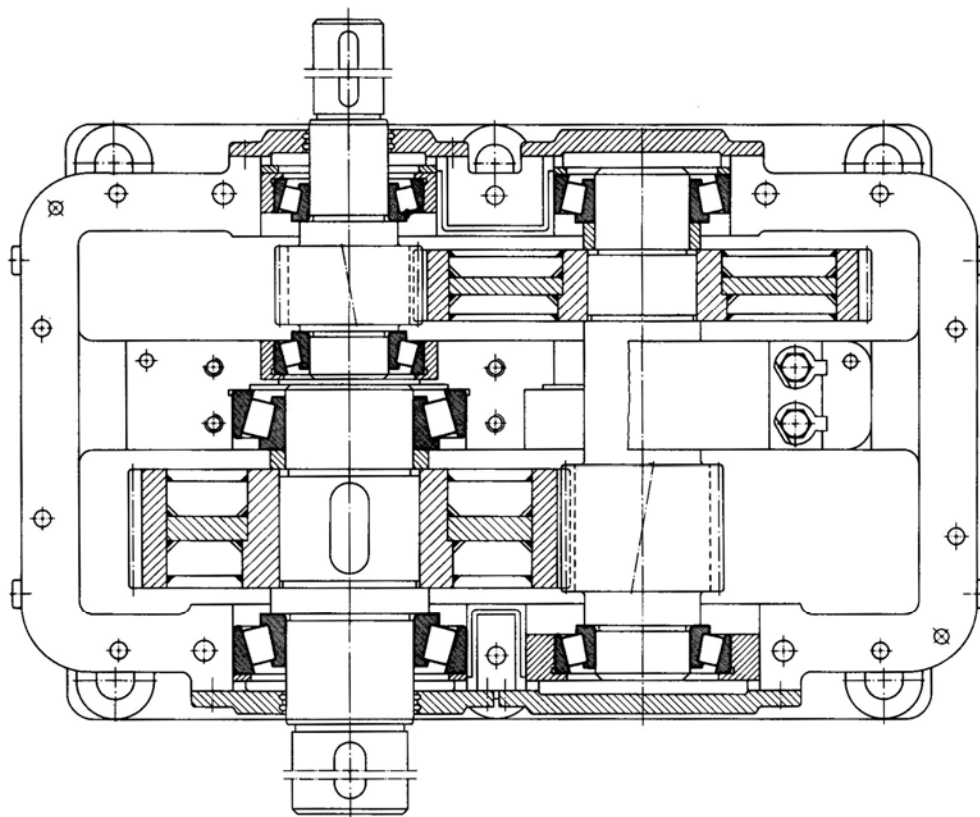


Fig. 8. A double reduction spur gearbox.

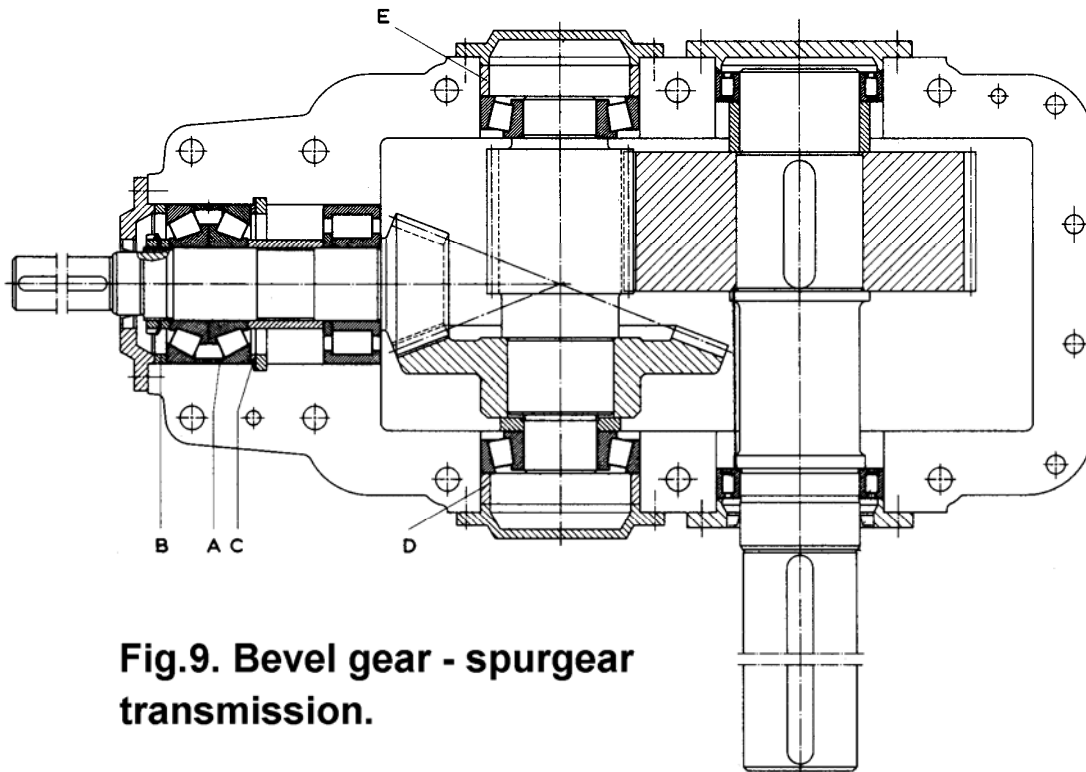


Fig.9. Bevel gear - spurgear transmission.

HELICAL GEARBOX DESIGN - PROBLEM 1

In a turbine drive 300 kW power is transmitted using a pair of double helical gear. The pinion speed is 2950 rpm and that of the gear is about 816.5 rpm. There are no space constraints on the gear drive. Selecting suitable materials, design the pinion and the gear to last for 10^8 cycles. Design the gearbox completely.

Data: $W = 300\text{kW}$; $n_1 = 2950\text{rpm}$; $n_2 \approx 816.5\text{ rpm}$;
Life 10^8 cycles.

Solution:

1. Angular speed of the input shaft

$$\omega_1 = \frac{2\pi n_1}{60} = \frac{2\pi \times 2950}{60} = 308.77 \text{ rad/s}$$

2. Torque: $T_1 = \frac{1000W}{\omega} = \frac{1000 \times 300}{308.77} = 971.6 \text{ Nm}$

3. The details of the gear design carried out are given in Table 1 and 2.

The final specifications of the pinion and gear are:
 20° pressure angle involute teeth with helix angle of 35° , $h_a = 1m_n$, $h_f = 1.25m_n$;
 $i = Z_2 / Z_1 = 105/29 = 3.62$

Table1. Gear dimensions

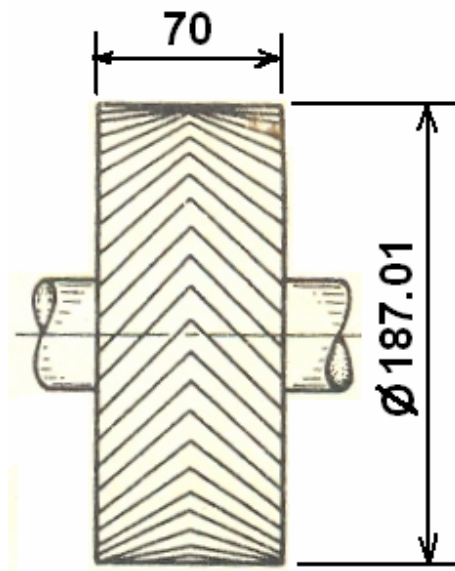
	Z	m_n mm	d mm	d_a mm	d_b mm	d_r mm	m_t mm
Pinion	29	5	177.01	187.01	161.76	164.51	6.104
Gear	105	5	640.92	650.92	585.69	628.42	6.104

Table 2a. Gear specifications

	Φ_n	ϕ_t	b mm	p_t mm	p_a mm
Pinion	20°	23.96°	70	19.165	27.37
Gear	20°	23.96°	70	19.165	27.37

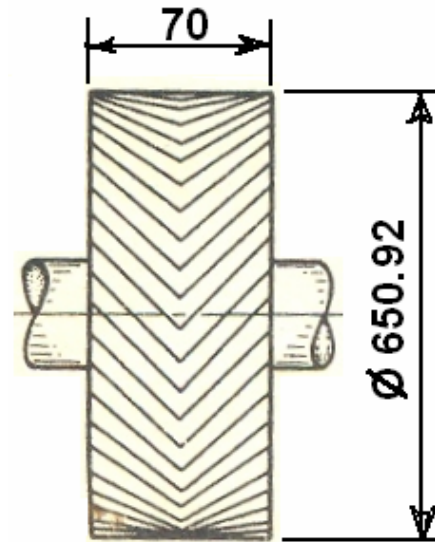
Table 2b. Gear specifications

	CR_t	CR_a	CR	FS s_b	FS s_H
Pinion	1.3044	1.2787	2.583	1.99	1.73
Gear	1.3044	1.2787	2.583	1.89	1.53



Pinion

Fig.10a. Pinion



Gear

Fig.10b. Gear

All dimensions are in mm and not to scale

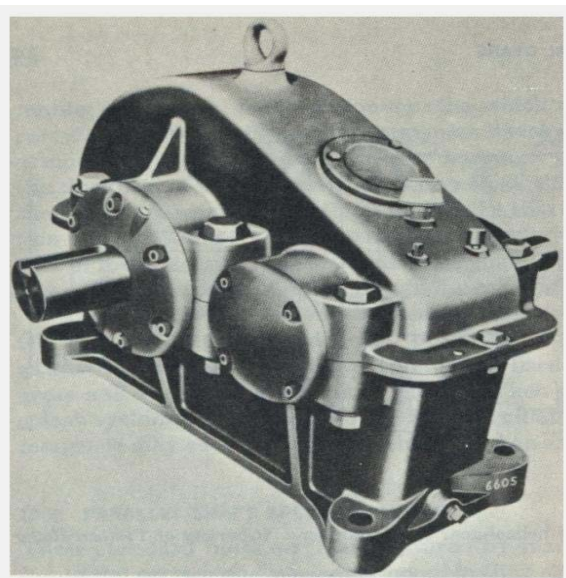
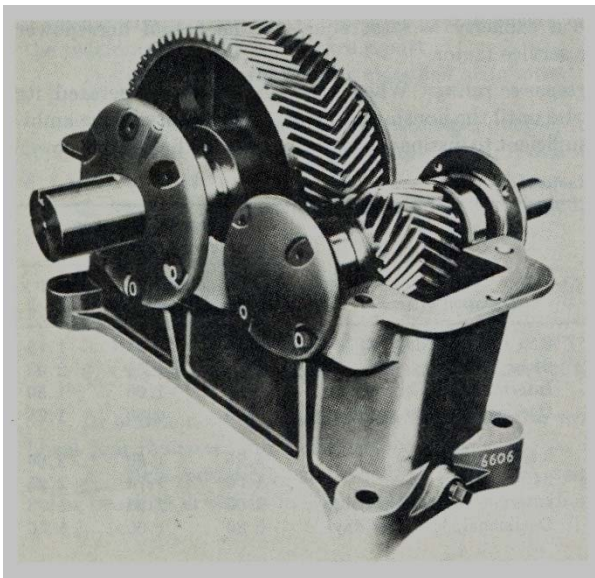


Fig.11 A commercial double helical gearbox

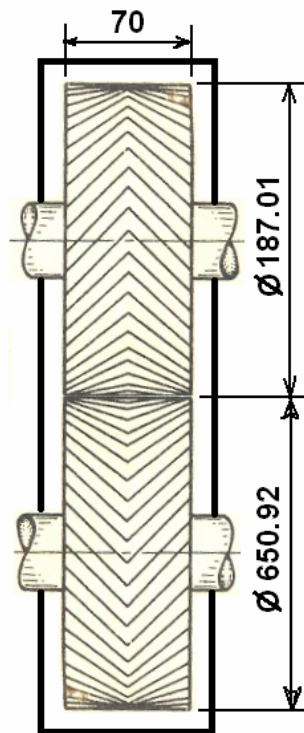


Fig. 12 Gearbox outer dimensions (tentative)

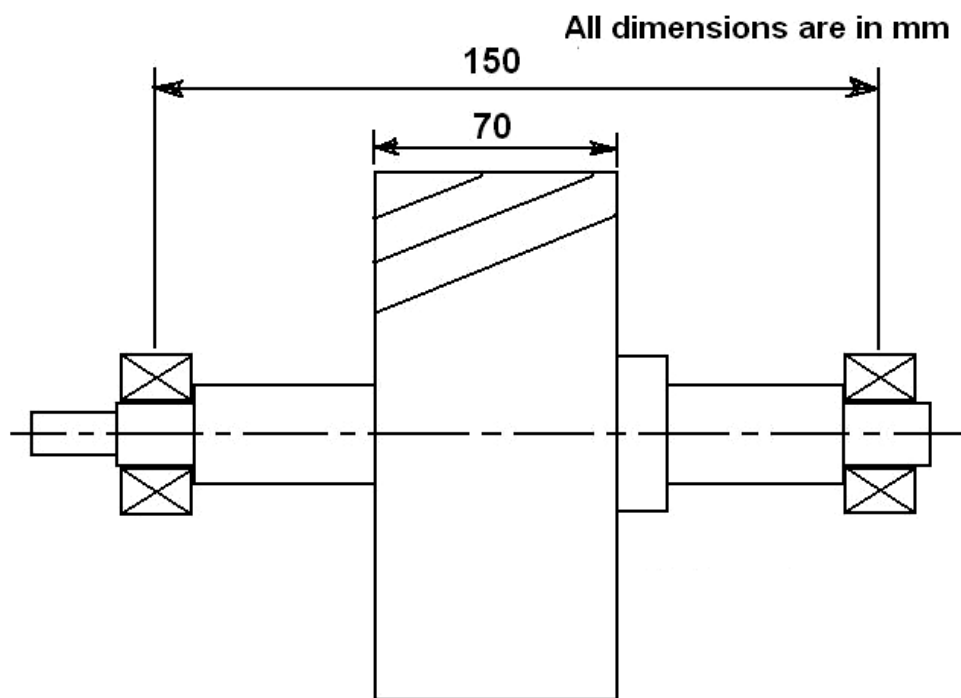


Fig.13. Pinion shaft layout diagram.

4. Shaft design is based on the ASME equation:

Tangential load on the shaft: $F_t = T/r = 971.6/0.088.5 = 11\text{kN}$

$F_r = F_t \tan \phi = 11 \tan 23.96^\circ = 4.89\text{kN}$

$$F = (F_t^2 + F_r^2)^{0.5} \\ = (11^2 + 4.89^2)^{0.5} = 15.42\text{kN}$$

Bending moment at C

$$M = Fl/4 = 15.42 \times 0.15/4 = 0.58\text{ kNm}$$

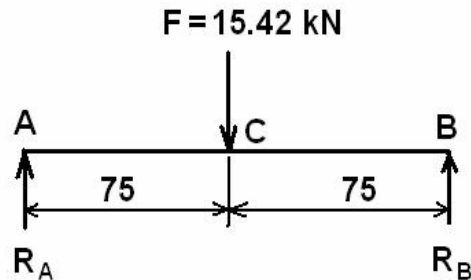


Fig.14. Shaft loading.

By ASME code equation for shaft design we have,

$$d = \frac{16}{\pi(1-k)[\tau]} \sqrt{(K_b M)^2 + (K_t T)^2}$$

$k = 0.2$ i.e, 20% reduction in strength due to keyway is assumed. From Table 3, for rotating shaft with minor shock loads, $K_b = 1.5$ and $K_t = 1.0$.

Taking C45 steel for the shaft, $\sigma_{yp} = 360\text{ MPa}$

$\tau_{yp} = 360/2 = 180\text{ MPa}$ and taking factor of safety of 2 , $[\tau_{yp}] = 180/2 = 90\text{ MPa}$

Table 3. Combined shock and fatigue factors for ASME code shaft design equation.

Type of loading	K_b	K_t
Stationary shaft		
Gradually applied load	1.0	1.0
Suddenly applied load	1.5 - 2.0	1.5 - 2.0
Rotating shaft		
Gradually applied load	1.5	1.0
Suddenly applied load		
With minor shocks	1.5 - 2.0	1.0 - 1.5
With heavy shocks	2.0 - 3.0	1.5 - 3.0

$$d = \frac{16}{\pi(1-k)[\tau]} \sqrt{(K_m M)^2 + (K_t T)^2}$$

$$d = \frac{16 \times 10^6}{\pi(1-0.2) \times 90} \sqrt{(1.5 \times 0.58)^2 + (1 \times 0.9716)^2} = 46 \text{ mm}$$

Take $d = 50 \text{ mm}$. Check for deflection at the pinion centre.

$$\text{Deflection at C: } \delta = \frac{F l^3}{48 E I} = \frac{15420 \times 150^3}{48 \times 2.1 \times 10^5 \left(\frac{\pi \times 50^4}{64} \right)} = 0.017 \text{ mm}$$

Since $\delta < 0.01 \text{ m} = 0.01 \times 5 = 0.05 \text{ mm}$, the design is OK.

Check for slope at the bearing at A.

$$\text{Slope: } \alpha = \frac{F l^2}{16 E I} = \frac{15420 \times 150^2}{16 \times 2.1 \times 10^5 \times \left(\frac{\pi \times 50^4}{64} \right)} = 0.00034 \text{ rad.}$$

$\alpha < 0.0008 \text{ rad}$. Hence the design is OK.

5. Check for the pinion size. The minimum pitch diameter of the pinion should be

$$\blacksquare \quad d_{1\min} \geq 2 \times \text{bore} + 0.25 \text{ m}$$

\blacksquare where d is the bore diameter and m is the module expressed in mm.

$$D_{1\min} \geq 2\text{bore} + 0.1m = 2 \times 50 + 0.1 \times 5 = 100.5 \text{ mm}$$

Since $d_1 = 177.01 \text{ mm} > D_{1\min}$. The design is satisfactory. Pinion drawing is shown in Fig.15 with full dimensions.

6. The outside diameter of the hubs in larger gears should be 1.8 times the bore for steel. The hub length should be at least 1.25 times the bore and never less than the width of the gear.

$$\text{Gear shaft diameter} = d \text{ (i)}^{1/3} = 50 (3.62)^{1/3} = 77 \text{ mm.}$$

Gear shaft diameter of 80 mm is taken.

The hub diameter: $d_H = 1.8 \times 80 = 144 \text{ mm}$, 150 mm is taken.

Hub length is taken as $L = 1.25d = 1.25 \times 80 \approx 100 \text{ mm}$

Other dimensions of the gear are given in Fig. 16.

In view of the dimensions of the pinion and the gear, the dimensions of the shaft layout is revised as shown in Fig.17. When the calculations are redone, there is no change in shaft diameters. The same diameters are adopted for further computations.

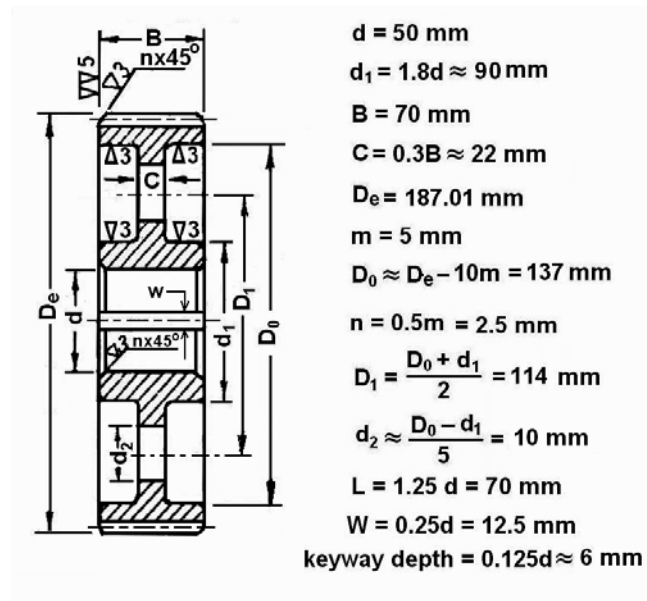


Fig.15. Pinion blank drawing showing all the dimensions.

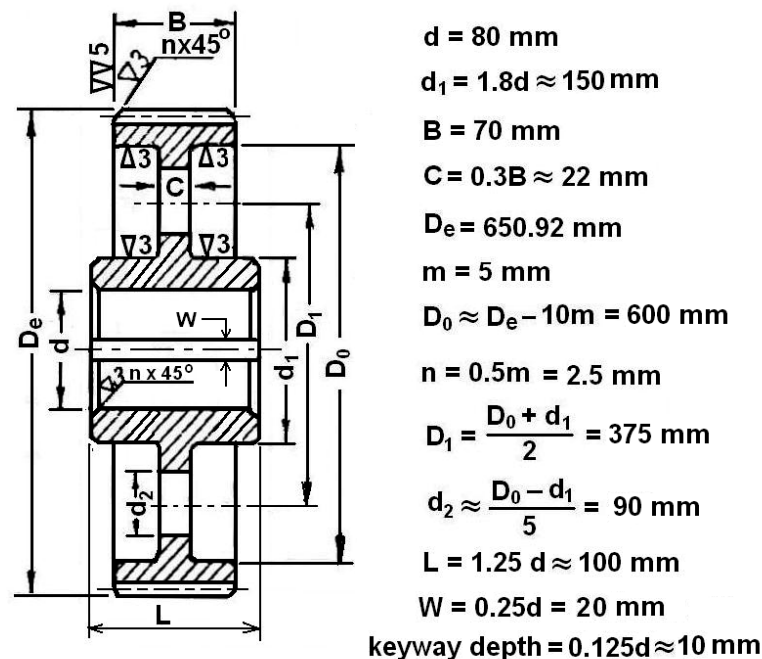


Fig.16. Gear blank drawing showing all the dimensions

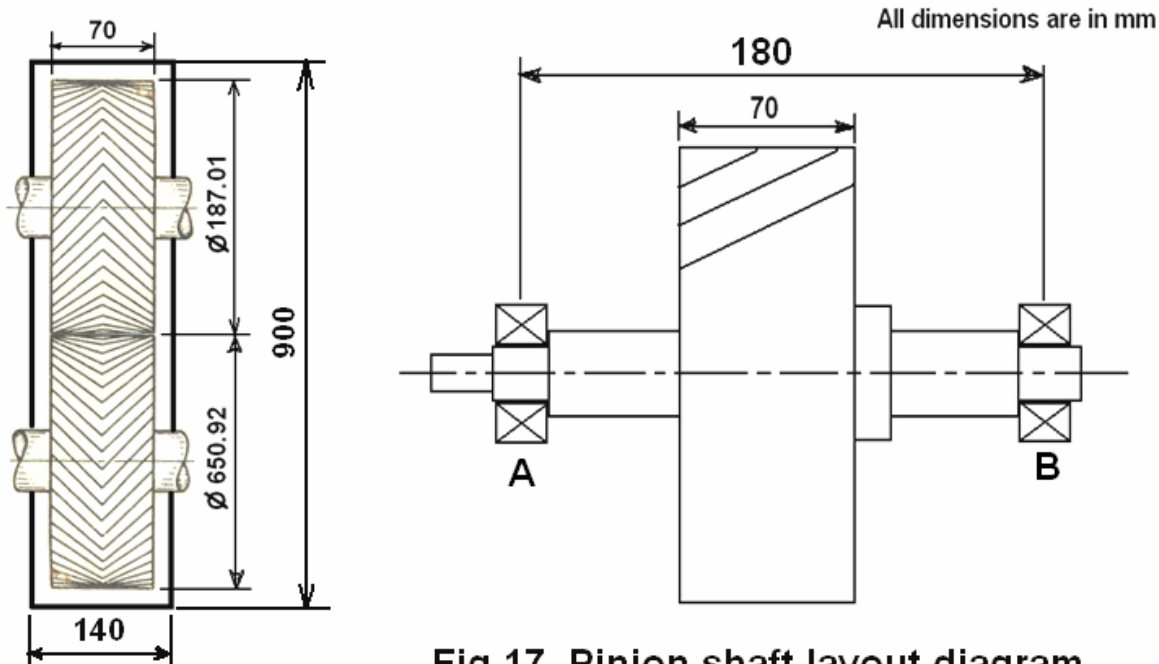


Fig.17. Pinion shaft layout diagram.

7. Bearings selection is based on 90% reliability for the following life:

8 hrs. operation per day life= 20,000- 30,000 hrs.

Consider the bearings at A & B with Life = 30,000 hrs,

$$P = 15420 / 2 = 7710 \text{ N,}$$

$$f_n = 0.224 \text{ for } n = 2950 \text{ rpm from FAG catalog.}$$

$$f_L = 3.91 \text{ for 30000 hrs life assuming 16 hrs/day working from FAG catalog.}$$

$$C = (f_L / f_n)P = (3.91/0.224) \times 7710 = 134581 \text{ N} = 134.6 \text{ kN}$$

Giving 2.5 mm abutment for the bearings, shaft diameter of the bearing should be 45 mm.

Roller bearing NJ 2309 satisfies this requirement

$$C = 137 \text{ kN, } C_o = 153 \text{ kN, } d_o = 100 \text{ mm, } d_i = 45 \text{ mm, } b = 36 \text{ mm.}$$

For the gear shaft of diameter 80mm, giving abutment of 2.5 mm, bearing bore diameter should be 75mm.

$$P = 7710 \text{ N, } f_n = 0.345 \text{ for shaft speed of 815 rpm.}$$

$$P = 7710 \text{ N,}$$

$$f_n = 0.345 \text{ for shaft speed of 815 rpm.}$$

$$f_L = 3.91 \text{ for Life of 30,000 hrs.}$$

$$C = (f_L / f_n) \times P = (3.91/0.345) \times 7710 = 87,380 = 87.38 \text{ kN}$$

Deep groove ball bearing 6315 with $C=114 \text{ kN}$, $C_o=67 \text{ kN}$; $d_o = 160 \text{ mm}$; $d_i = 75 \text{ mm}$; $b=37 \text{ mm}$.

8. Gearbox dimensions are fixed based on thumb rule given in Table 4.

Table 4. Wall thickness 's' in mm of the gearboxes.

	Non-case hardened gears	Case hardened gears
CI castings	$0.007L + 6 \text{ mm}$	$0.010 L + 6 \text{ mm}$
Steel castings	$0.005L + 4 \text{ mm}$	$0.007L + 4 \text{ mm}$
Welded construction	$0.004L + 4 \text{ mm}$	$0.005L + 4 \text{ mm}$

Where L is the largest dimension of the housing in mm.

$$s = 0.005L + 4 \text{ mm} = 0.005 \times 900 + 4 \approx 10 \text{ mm}$$

$$\text{Top cover thickness: } S_c = 0.8s = 8 \text{ mm.}$$

$$\text{Flange thickness : } s_f = 2s = 2 \times 10 = 20 \text{ mm}$$

$$\text{Flange cover bolt diameter: } d_{cb} = 1.5s \approx 16 \text{ mm M16 bolts.}$$

$$\text{Bolt spacing: } 6d = 6 \times 16 \approx 100 \text{ mm}$$

$$\begin{aligned} \text{Foundation bolt diameter: } d_{fb} &= (2T)^{1/3} \geq 12 \text{ mm} \\ d_{fb} &= (2 \times 3.62 \times 971.6)^{1/3} = 19.2, \text{ Take M20 bolts.} \end{aligned}$$

The thickness of the foundation flange should be:

$$s_{ff} \geq 1.5 d_{fb}.$$

$$S_{ff} = 1.5 d_{fb} = 1.5 \times 20 = 30 \text{ mm}$$

$$\text{The width of the flange at the base: } w_b = 2.5d = 2.5 \times 20 = 50 \text{ mm}$$

The width of the flanges at the two halves of the housing should be:

$$w_f = 2.5d = 2.5 \times 16 = 40$$

With welding bead of 5mm, $w_f = 45 \text{ mm}$ is taken.

Outside dimension of the bearing housing 1.2-1.5 times outside diameter of the bearing.

Bearing housing diameters are : $1.5 \times 100 = 150 \text{ mm}$

and : $1.3 \times 160 = 210 \text{ mm}$ taking 6 Nos. M10 bolts for the bearing covers.

The views of the bottom and top half of the gearbox are shown in Fig. 18 and Fig.19.

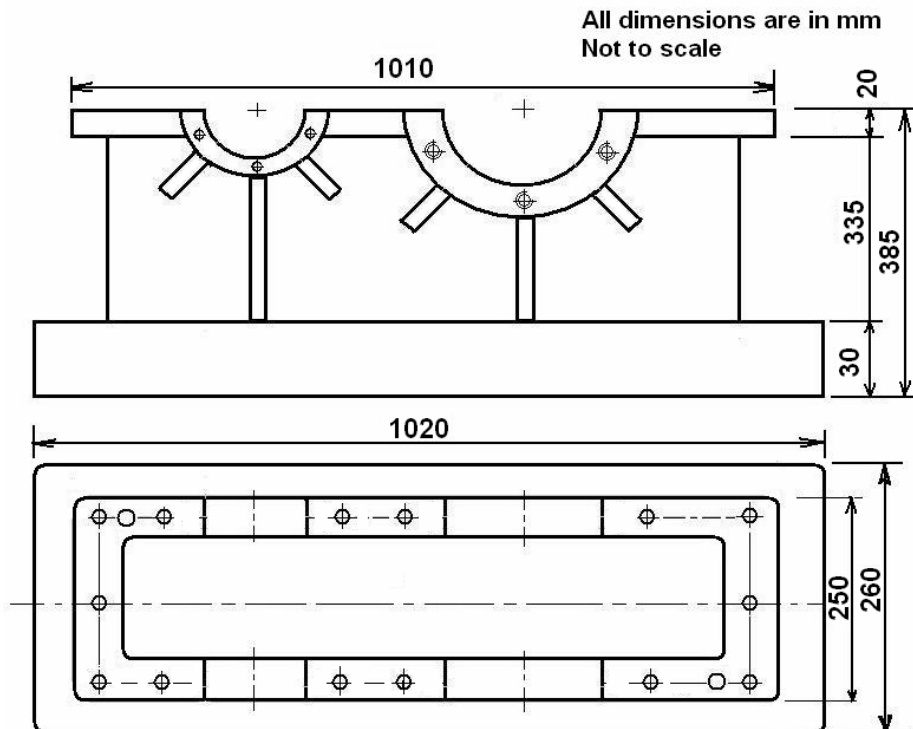


Fig.18. View of bottom half of the gearbox.

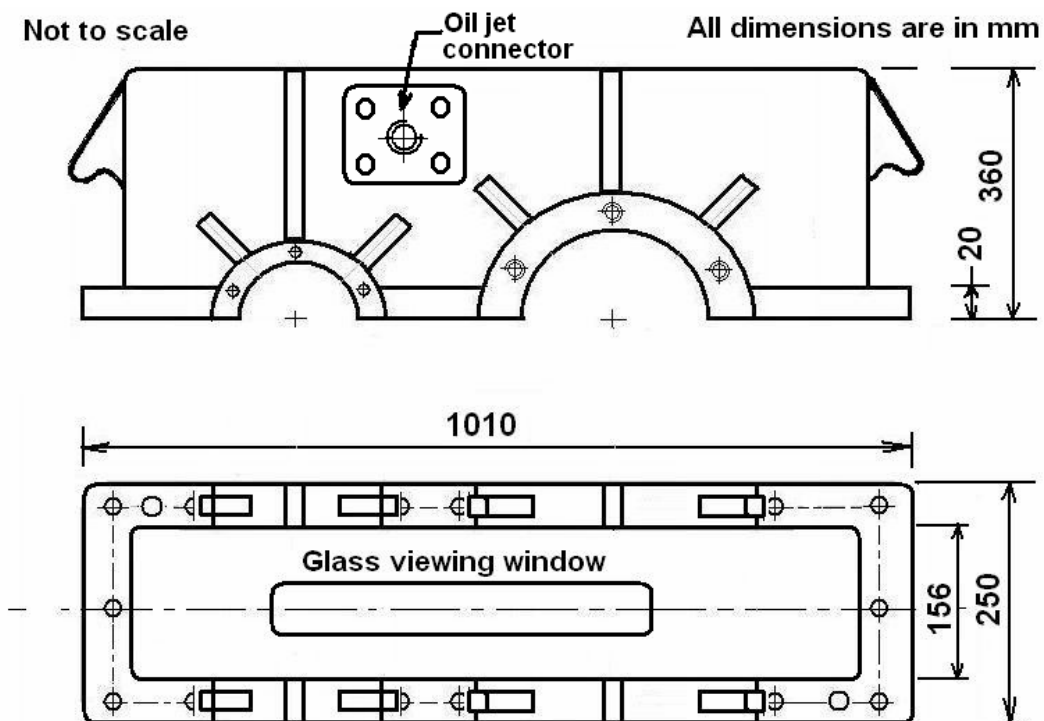


Fig.19. View of top cover of the gearbox.

RECOMMENDED OIL FOR VARIOUS SLIDING SPEEDS

Table 5. Recommended oil viscosity V₅₀ [cSt at 50 °C]
For different sliding speeds

v(m/s)	0.25	0.4	0.63	1.0	1.6	2.5
V ₅₀ min	175	145	120	100	83	69
V ₅₀ max	350	290	240	200	166	138

v(m/s)	4.0	6.3	10	16	25	40	63
V ₅₀ min	57	47	39	32	27	22	18
V ₅₀ max	114	94	78	64	54	44	36

The gears are operating at a sliding speed of $v = \omega r = 308.77 \times 0.0885 = 27.33$ m/s.

From the Table 5, the recommended oil viscosity at 50°C for this operation is V₅₀ between 25 to 51 cSt (interpolated values). ISO VG 100 satisfies this, see Fig. 20

The equivalent grade from chart in Fig. 21, SAE 30 oil comes under this range and is recommended for the operation.

ISO VG GRADE LUBRICANTS

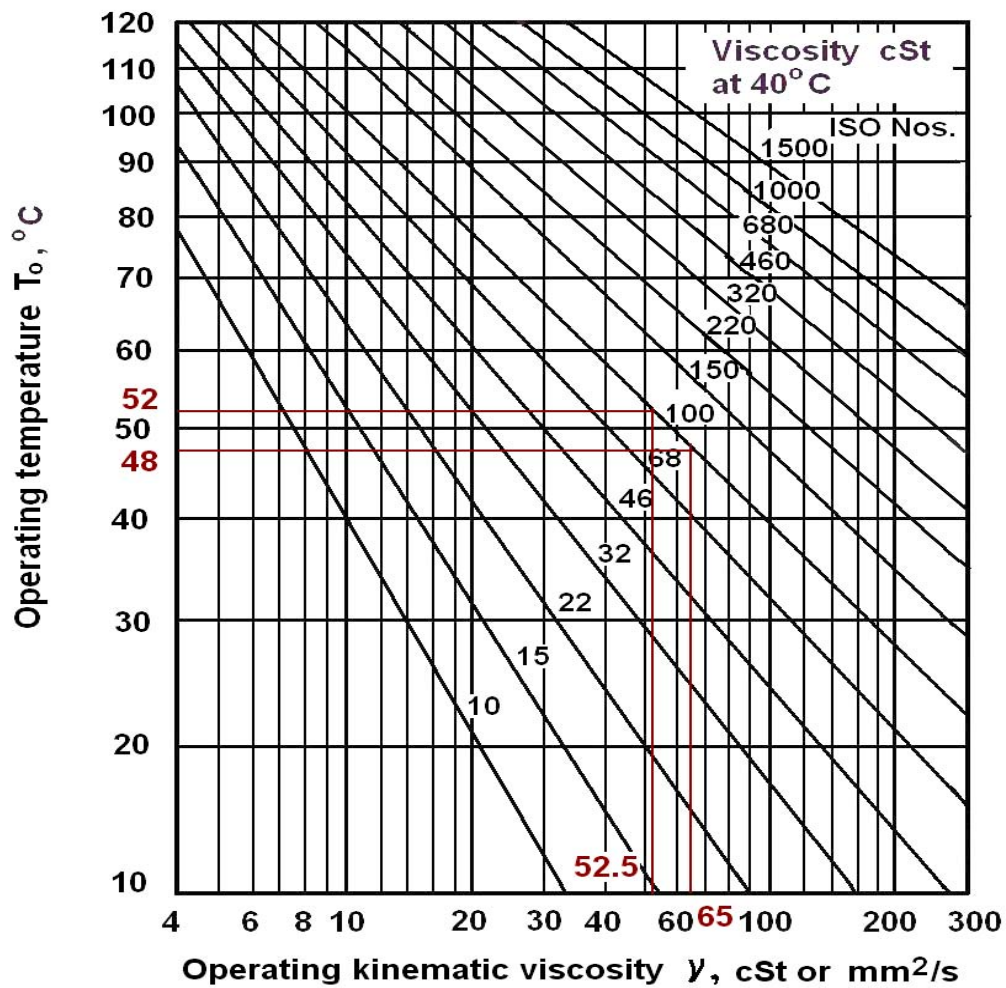


Fig.20. Viscosity -temperature diagram for ISO VG graded oils.

SAE OIL VISCOSITY CHART

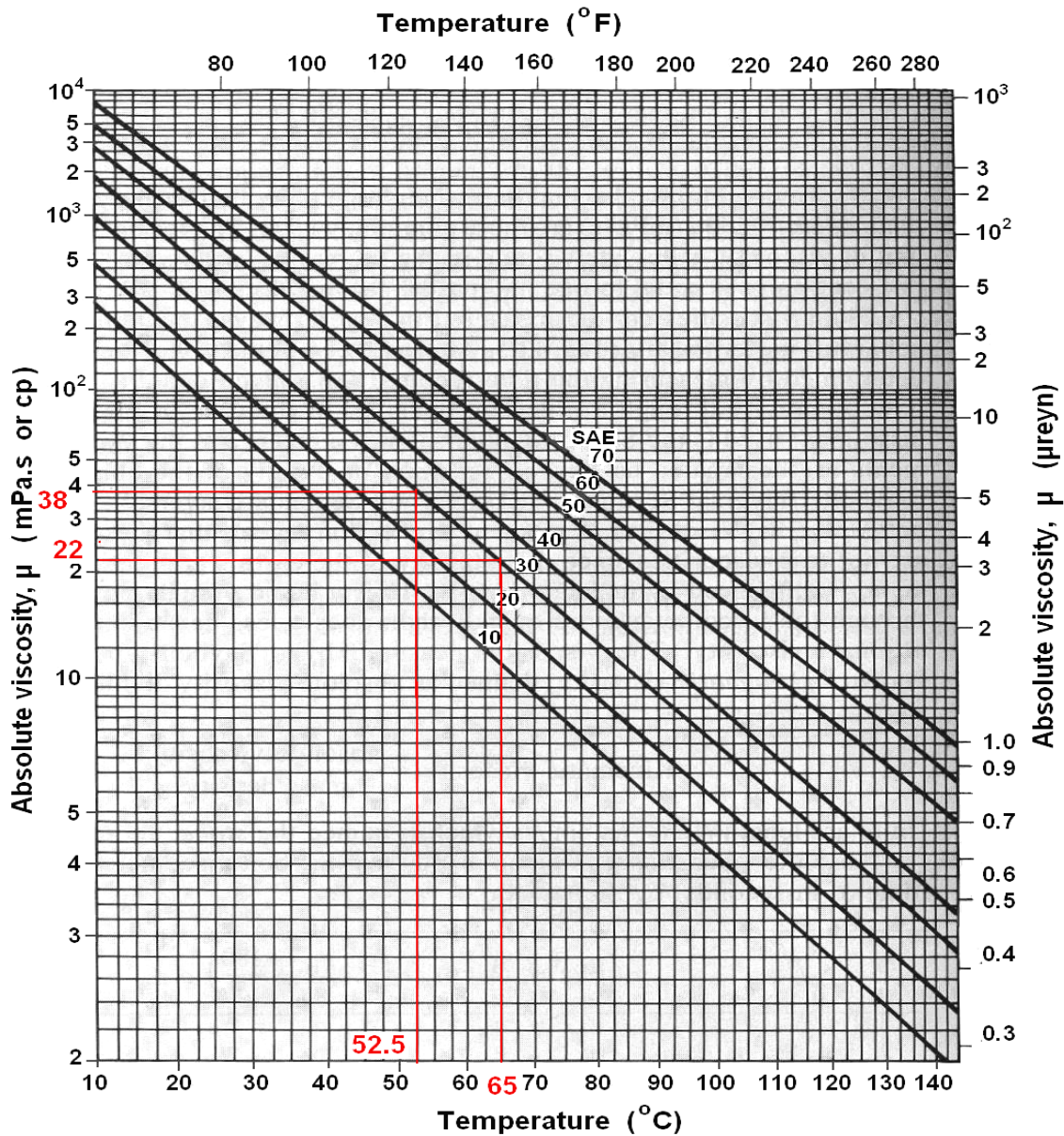


Fig. 21. Viscosity - temperature curves of SAE graded oils.

9. Losses in gear boxes :

$$\text{Total power loss } L = L_t + L_{ch} + L_b + L_s$$

L_t - power loss at tooth engagement.

L_{ch} - churning power losses

L_b - bearing power losses & L_s -seal frictional power loss.

$$L_t = W \left(\frac{0.1}{Z_1 \cos \psi} + \frac{0.3}{V + 2} \right) \text{ kW}$$

$$L_t = 300 \left(\frac{0.1}{29 \cos 35^\circ} + \frac{0.3}{27.3 + 2} \right) = 4.33 \text{ kW}$$

$$L_{ch} = c b V \left(\frac{200 V \mu}{Z_1 + Z_2} \right)^{0.5} \times 10^{-3} \text{ kW}$$

Where

V - peripheral speed (m/s)

b - face width of the gear (mm)

c - factor equal to 0.009 for splash lubrication, 0.006 for stream lubrication

μ - viscosity of oil at the operating temperature (cP)

$$L_{ch} = c b V \left(\frac{200 V \mu}{Z_1 + Z_2} \right)^{0.5} \times 10^{-3} \text{ kW}$$

$$L_{ch} = 0.006 \times 70 \times 27.3 \times \left(\frac{200 \times 27.3 \times 35}{29 + 105} \right)^{0.5} \times 10^{-3} = 0.433 \text{ kW}$$

$$L_b = 5.23 \times 10^{-8} F f_b d n \text{ kW}$$

where

F - radial load on the bearing (N)

f_b - coefficient of friction at the bearing reduced to the shaft diameter 0.005 - 0.01 for rough estimation or refer to catalog.

d - shaft diameter (mm)

n - shaft speed (rpm)

From the catalog $f_b = 0.002$ for roller bearings and 0.003 for ball bearings.

Bearings at A & B

$$\begin{aligned} L_b &= 5.23 \times 10^{-8} F f_b d n \\ &= 5.33 \times 10^{-8} \times 15420 \times 0.002 \times 45 \times 2950 = 0.218 \text{ kW} \end{aligned}$$

Bearings at D & E

$$\begin{aligned} L_b &= 5.23 \times 10^{-8} F f_b d n \\ &= 5.33 \times 10^{-8} \times 15420 \times 0.003 \times 75 \times 814.92 = 0.151 \text{ kW} \\ L_B &= 0.369 \text{ kW} \end{aligned}$$

Seal frictional power loss:

$$L_s = T_s \omega \times 10^{-3} \text{ kW}$$

Where T_s seal friction torque

ω – angular velocity of the shaft.

$$T_s = f P_r r$$

Where r = radius of the shaft [m]

f – seal friction

P_r – Radial lip load [N]

Coefficient of friction: f

$$f = \varphi (\mu v b / P_r)^{1/3}$$

φ = Characteristic Number

μ = Oil Viscosity [N.s/cm²]

v = Linear Speed [m/s]

b = Lip Contact Width [m]

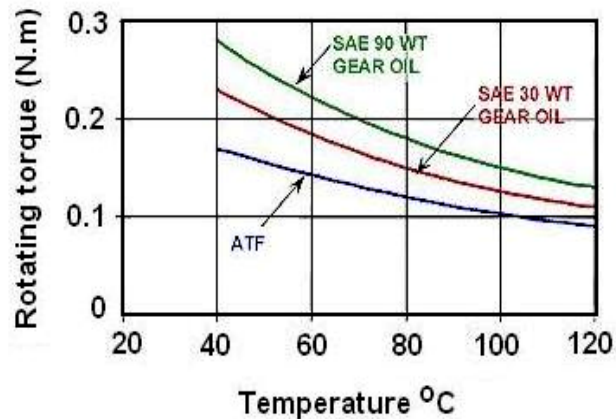


Fig.22. Friction torque at various temperature for nitrile rubber S type oil seal. Shaft diameter 50 mm, speed 1000 rpm.

Fig. 22 gives the torque vs temperature chart for seal. Let the outlet oil temperature be 65°C

At 65°C, $T_s = 0.17 \text{ Nm}$ from Fig.23.

$$V = \pi d n / 60000$$

$$= \pi \times 50 \times 1000 / 60000$$

$$= 2.36 \text{ m/s}$$

The operating Velocity

$$V = \pi \times 45 \times 2950 / 60000$$

$$= 6.95 \text{ m/s}$$

T_s at operating speed of pinion shaft

$$\text{speed} = 0.17 \times (6.95 / 2.36)^{1/3} = 0.244 \text{ Nm}$$

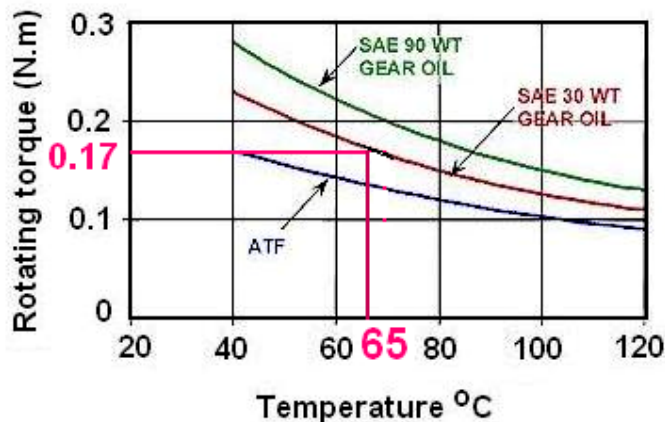


Fig.23. Friction torque at various temperature for nitrile rubber S type oil seal. Shaft diameter 50 mm, speed 1000 rpm.

Pinion shaft seals power loss

$$L_s = T_s \omega \times 10^{-3} = 0.244 \times 308.77 \times 2 \times 10^{-3} = 0.151 \text{ kW}$$

Gear shaft seal power loss

$$V = \pi \times 75 \times 814.92 / 60000 = 3.2 \text{ m/s}$$

$$T_s = 0.17 (3.2 / 2.36)^{1/3} = 0.188 \text{ Nm}$$

$$L_s = T_s \omega \times 10^{-3} = 0.188 (308.77 / 3.62) \times 2 \times 10^{-3} = 0.032 \text{ kW}$$

$$\text{Total seal friction} = 0.151 + 0.032 = 0.183 \text{ kW.}$$

Total power loss in the gearbox:

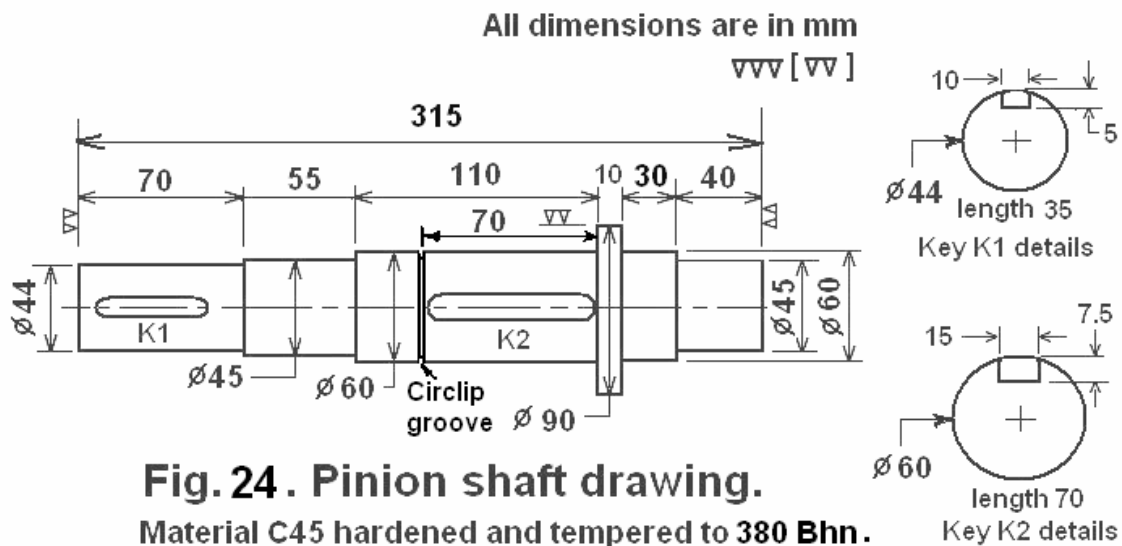
$$L = L_t + L_{ch} + L_b + L_s = 4.33 + 0.433 + 0.369 + 0.183 = 5.315 \text{ kW}$$

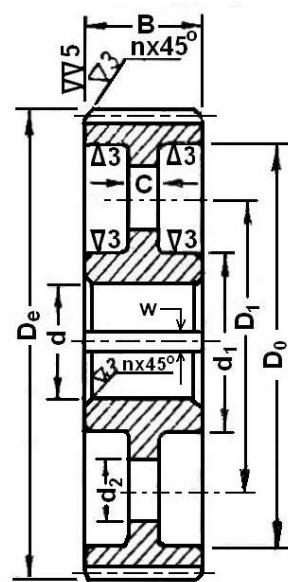
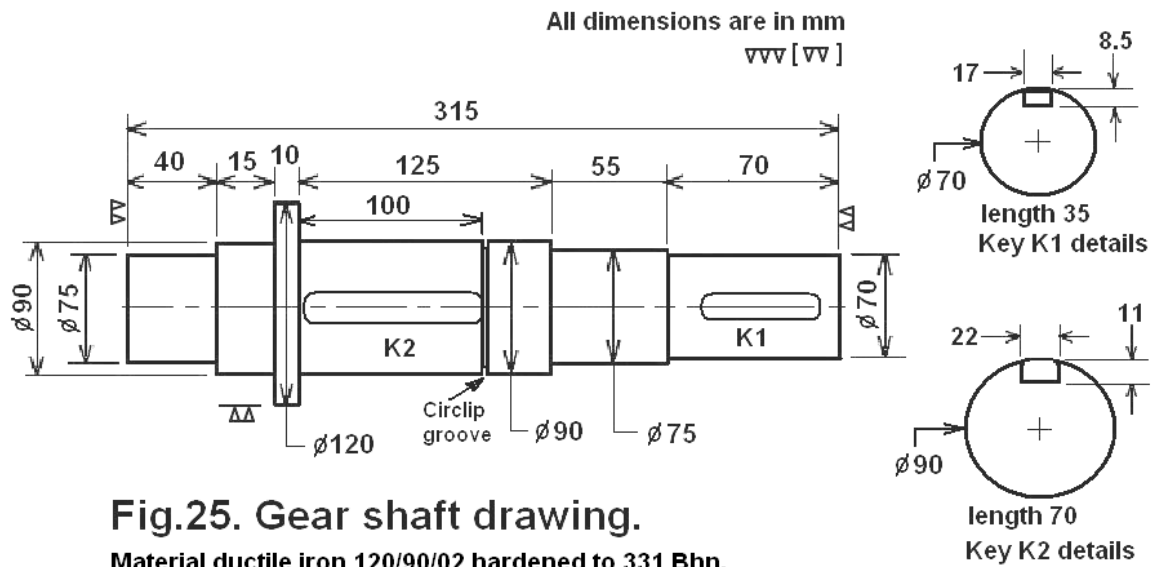
For the operating speed of the gear 27.33 m/s suggested type of lubrication is oil jet lubrication.

Assuming inlet oil temperature of 40°C and outlet oil temperature of 65°C, the oil supply rate has to be:

$$Q_e = L \times 10^3 / \rho c \Delta T = (5.315 \times 10^3 / 0.88 \times 1670 \times 25) = 0.1447 \text{ lps} = 0.01447 \times 60 = 8.68 \text{ lpm.}$$

Based the details of the gearbox the shaft details are worked out and detailed pinion shaft drawing is shown in Fig. 24 and that of the gear shaft in Fig.25. The corresponding revised dimensions of the gears are shown in Fig.26 and 27.





$$\begin{aligned}
 d &= 60 \text{ mm} \\
 d_1 &= 1.8d \approx 100 \text{ mm} \\
 B &= 70 \text{ mm} \\
 C &= 0.3B \approx 22 \text{ mm} \\
 D_e &= 187.01 \text{ mm} \\
 m &= 5 \text{ mm} \\
 D_0 &\approx D_e - 10m = 137 \text{ mm} \\
 n &= 0.5m = 2.5 \text{ mm} \\
 D_1 &= \frac{D_0 + d_1}{2} = 118 \text{ mm} \\
 d_2 &\approx \frac{D_0 - d_1}{5} \approx 10 \text{ mm} \\
 L &= 1.25 d \approx 70 \text{ mm} \\
 W &= 0.25d = 15 \text{ mm} \\
 \text{keyway depth} &= 0.125d \approx 7.5 \text{ mm}
 \end{aligned}$$

Fig. 26. Pinion blank revised drawing showing all the dimensions.

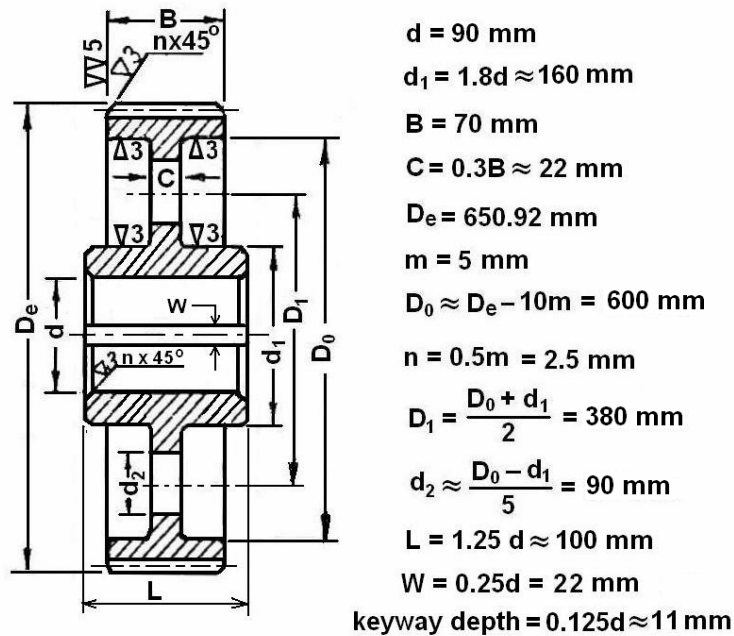


Fig. 27. Gear blank revised drawing showing all the dimensions.

Details of the gearbox Table 6.

Gearbox size & wt	745x260x1020 mm 330 kg	MS welded construction
Pinion C45 steel with hardness 380 Bhn Hobbed and ground	Gear ductile iron grade 120/90/02 of hardness 331 Bhn Hobbed and ground	Shafts C-45 hardened and tempered and ground
Lubricant SAE 30	Oil jet lubrication 10 lpm	$\eta = 98.2\%$

The gearbox is of split type with radial assembly. Gears and bearings are mounted on the shafts separately outside and assembled radially in the gearbox and the top cover is bolted in position. The oil jet and the outlet connections are made subsequently. 8 lpm oil is directed at the gear mesh and 2 lpm is directed at the bearings and seals.

The gearbox assembly views are shown in Fig. 28 to 30. Fig.28 shows the front view separately, Fig.29. The end view end view separately for clarity. and Fig.30 shows the complete view front and side together.

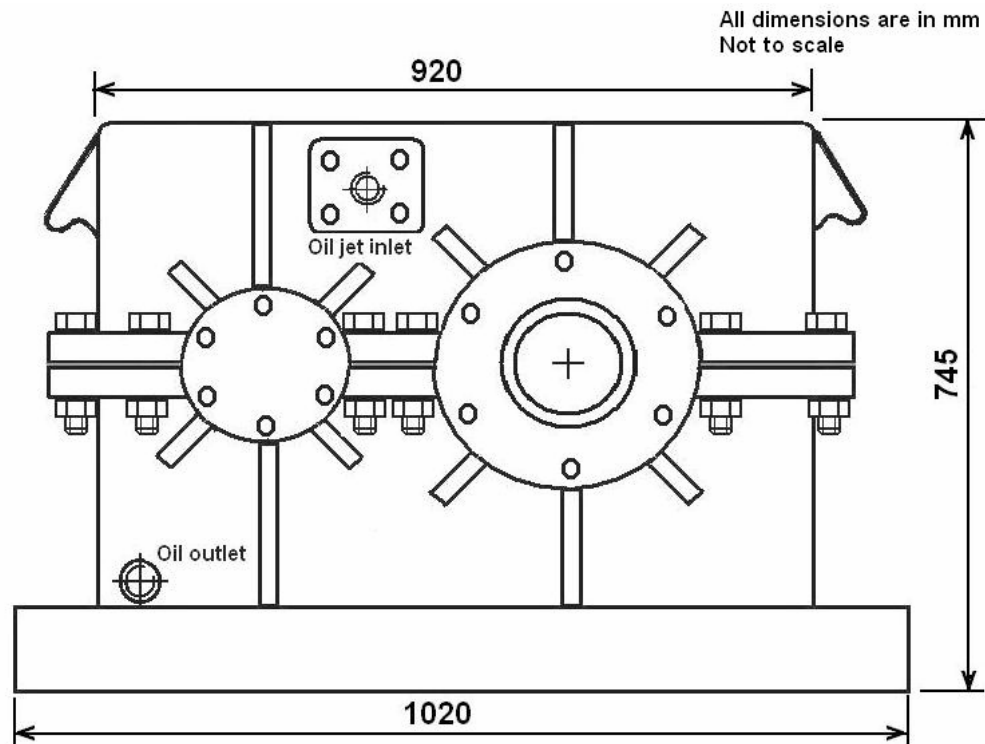


Fig. 28. Assembled view of the helical gearbox.

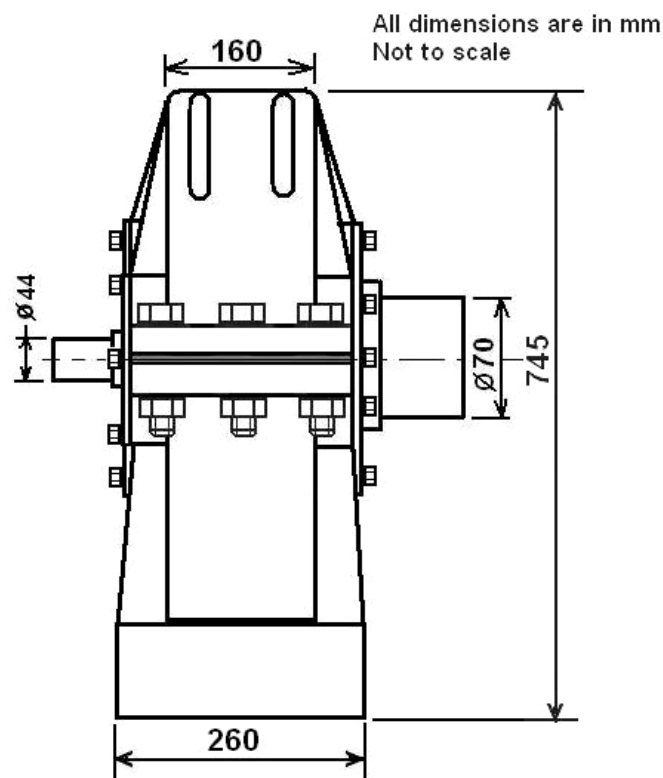


Fig.29. End view of the gearbox.

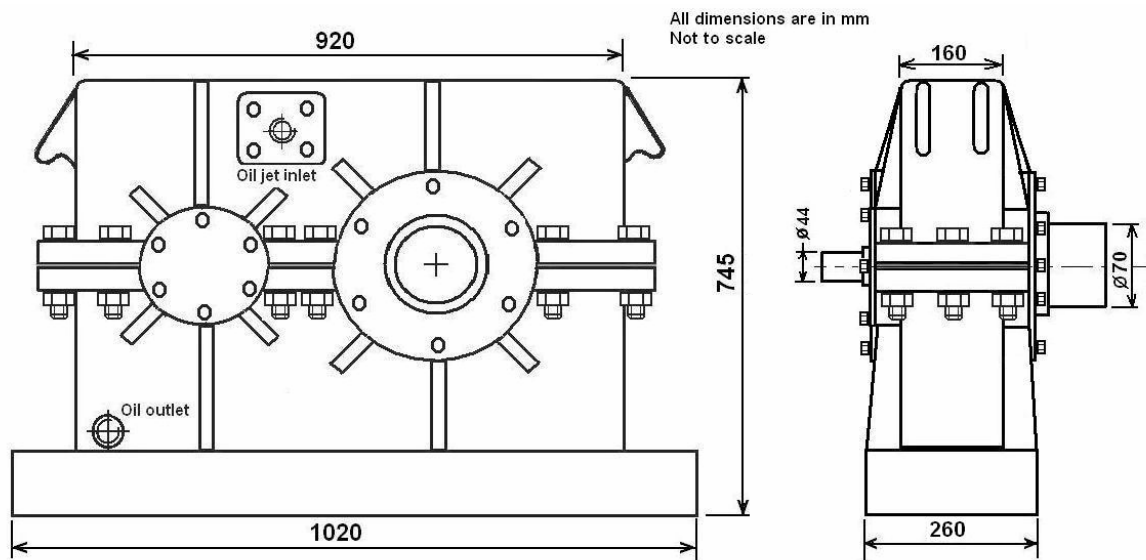


Fig.30. Assembled view of the helical gearbox.