Module 2 - GEARS

Lecture 17 – DESIGN OF GEARBOX

Contents

17.1 COMMERCIAL GEARBOXES

Various commercial gearbox designs are depicted in Fig. 17.1 to 17.10. These include single to multistage ranging from spur, helical, bevel to worm gears.

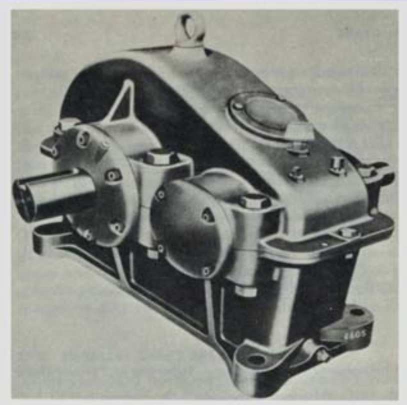


Fig.17.1 Commercial Gearbox Design

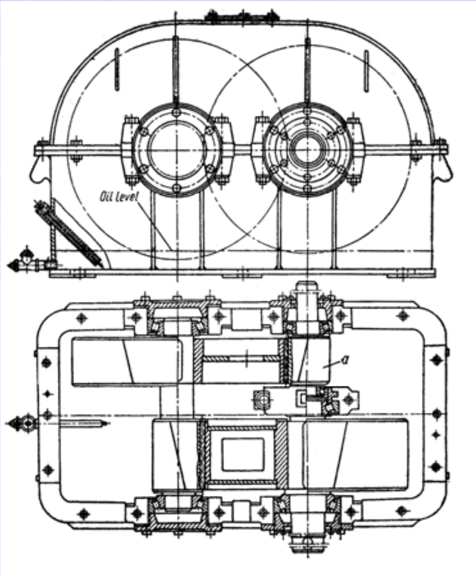


Fig. 17.2 Two stage helical gearbox

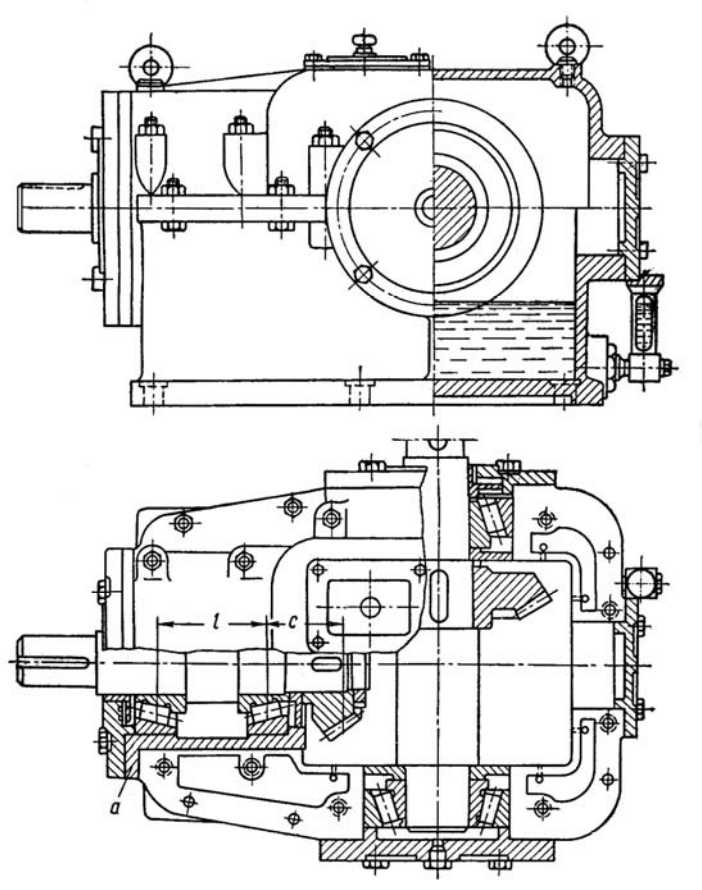
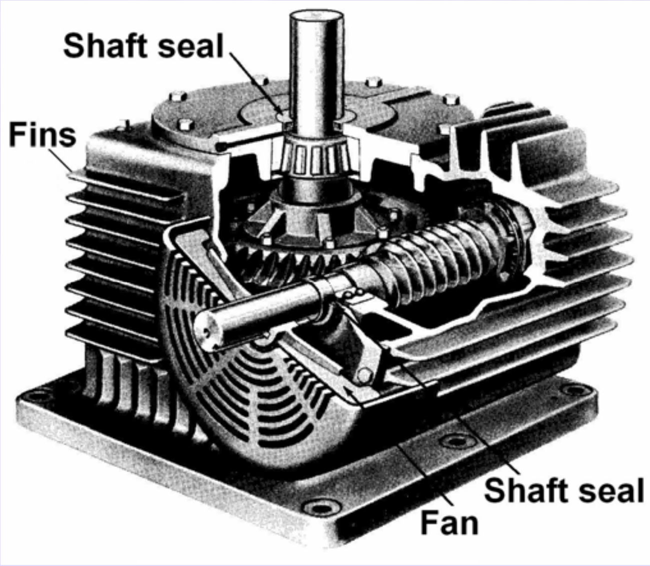


Fig.17.3 A single stage bevel gearbox



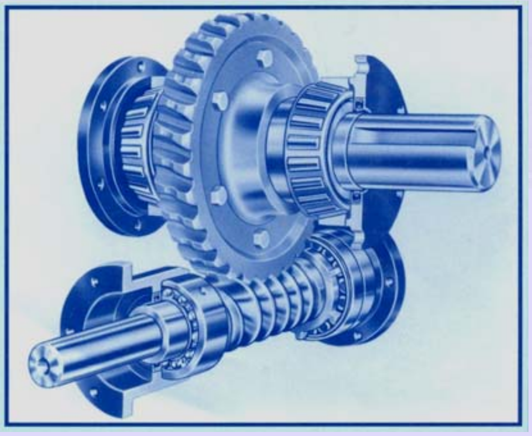


Fig.17.4 Worm gearbox

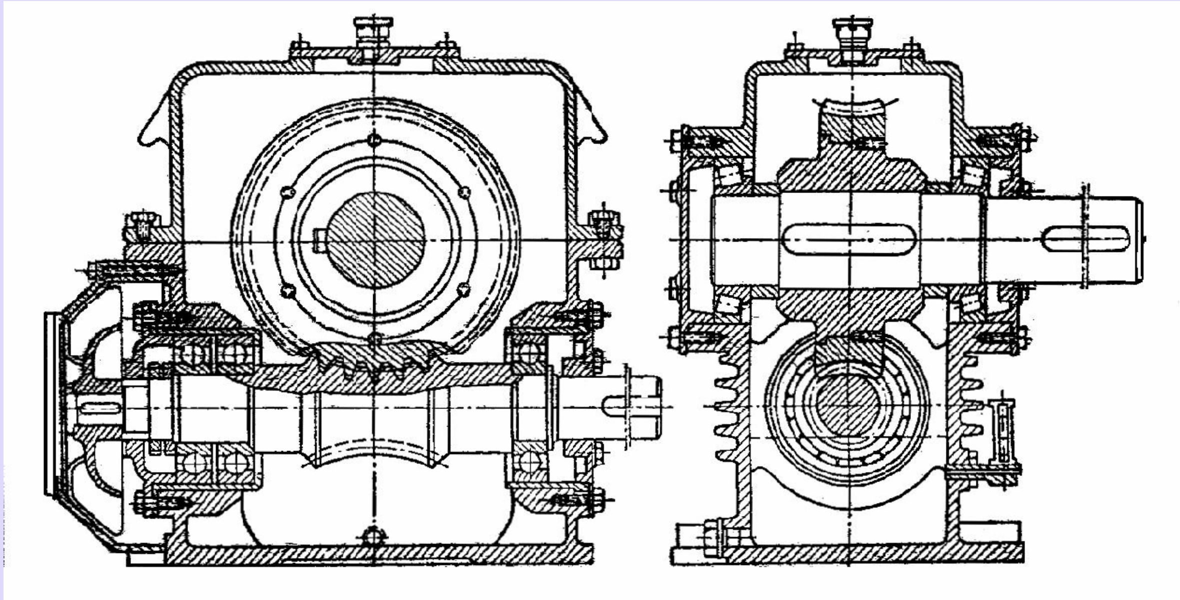


Fig.17.5 Worm gearbox, sectional front and side views

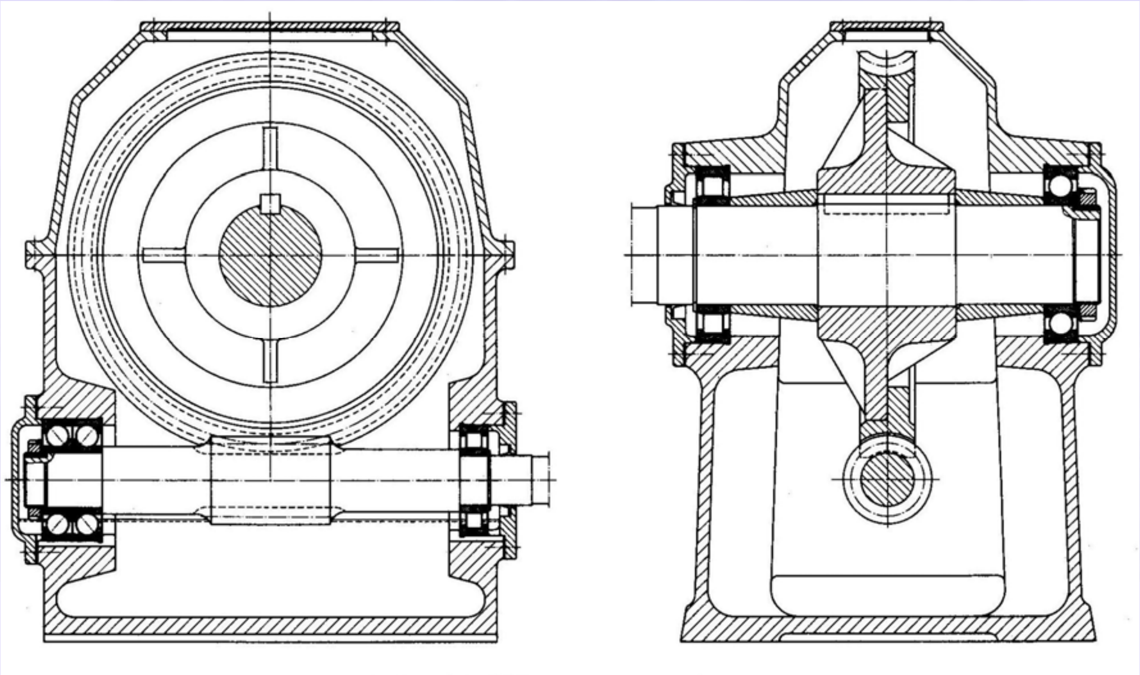


Fig.17.6 Worm gearbox, without cooling fins, sectional front and side views

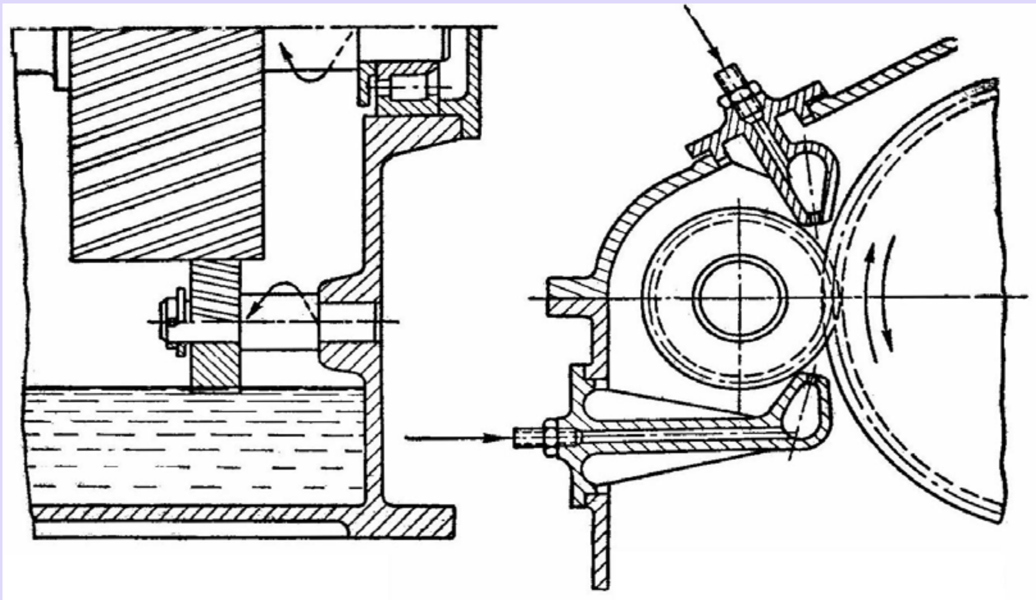


Fig.17.7 Helical gear lubrication with idler gear

Fig.17.8 Spur gear lubrication with stream by nozzles

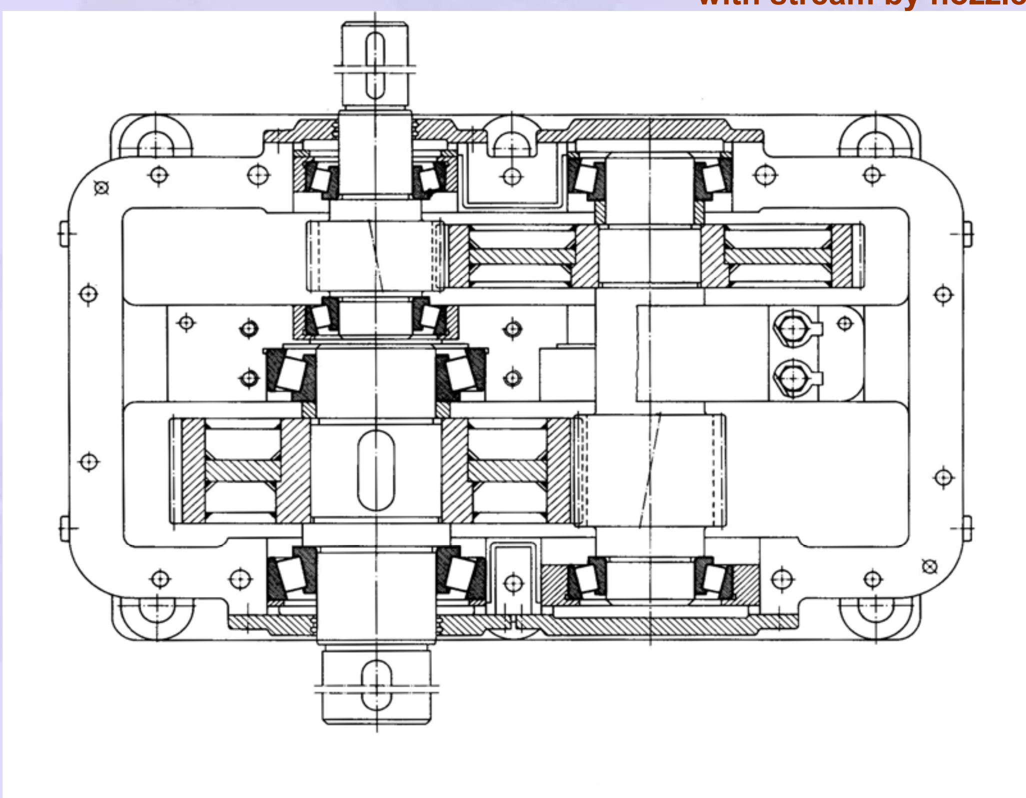


Fig.17.9 A double reduction spur gear box

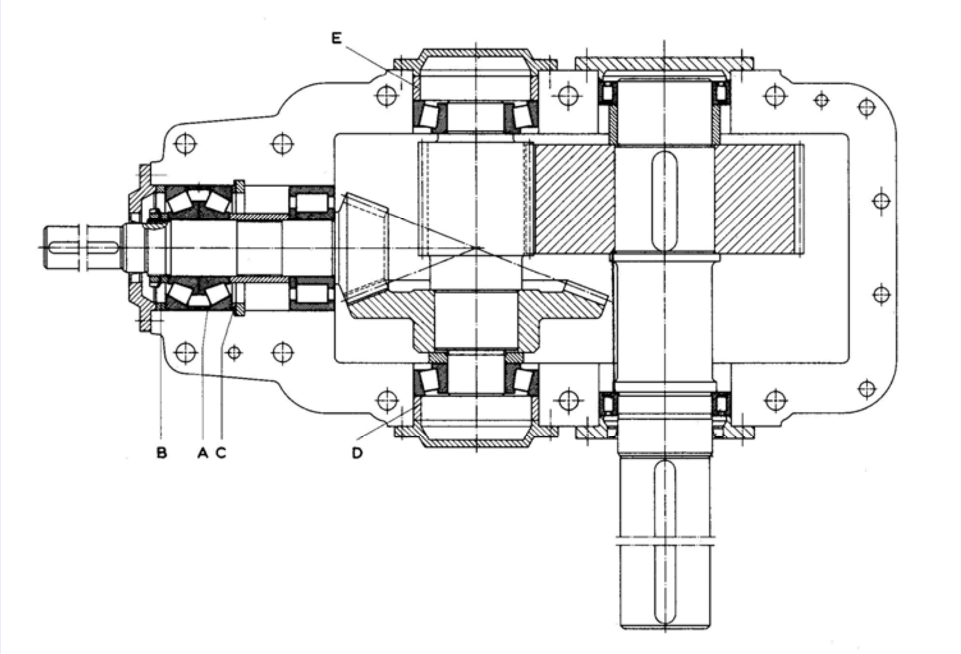


Fig.17.10 Bevel gear – Spur gear transmission

17.2 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 = 300kW; n_1 = 2950rpm; $n_2 \approx$ 816.5 rpm; Desired 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^o pressure angle involute teeth with helix angle of 35^o, h_a =1m_n, h_f =1.25m_n.

i= Z_2 / Z_1 =105/29= 3.62

Table 17.1 Gear dimensions

Element	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 17.2a Gear specifications

Element	Φ_n	φ_t	B mm	p_t mm	p_a mm
Pinion	20^o	23.96^o	70	19.165	27.37
Gear	20^o	23.96^o	70	19.165	27.37

Table 17.2b Gear specifications

Element	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

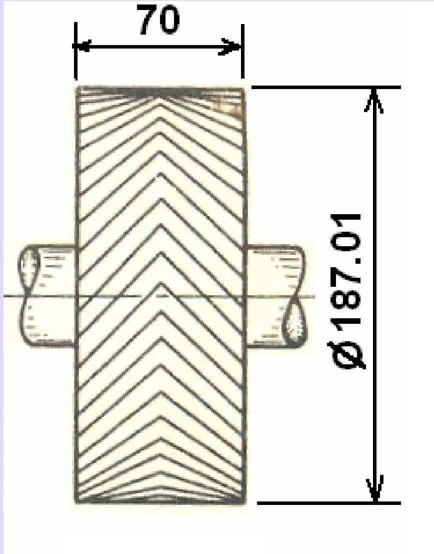


Fig.17.11 (a) Pinion

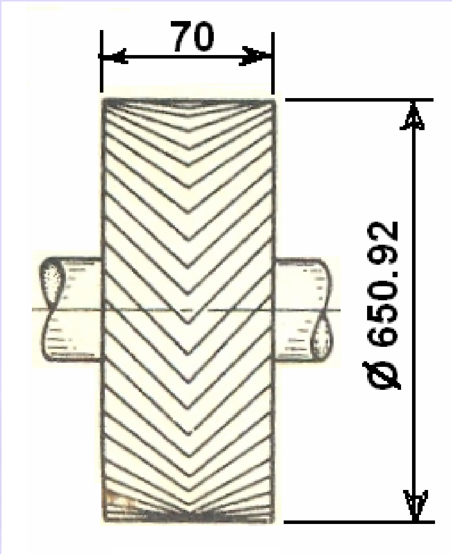
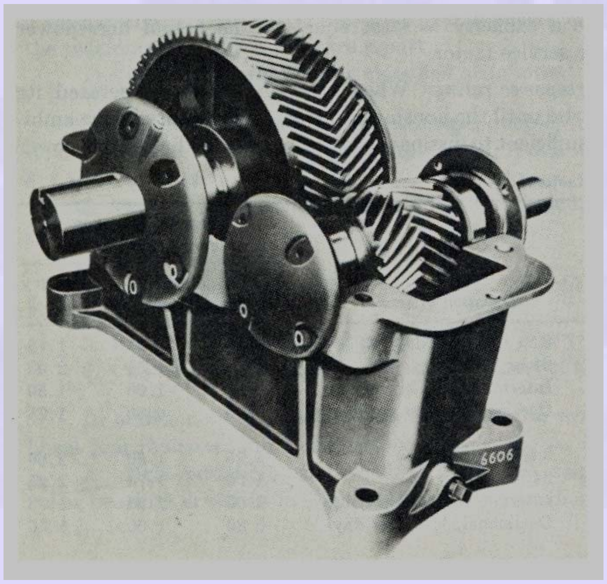


Fig.17.11 (b) Gear

All dimensions are in mm and not to scale



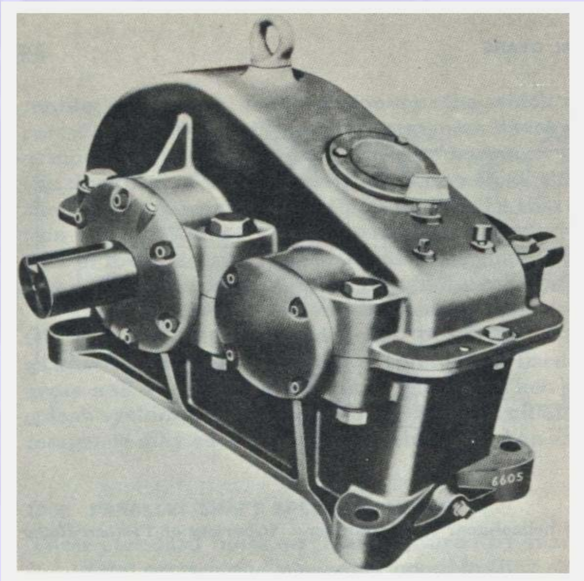


Fig.17.12 A commercial double helical gearbox

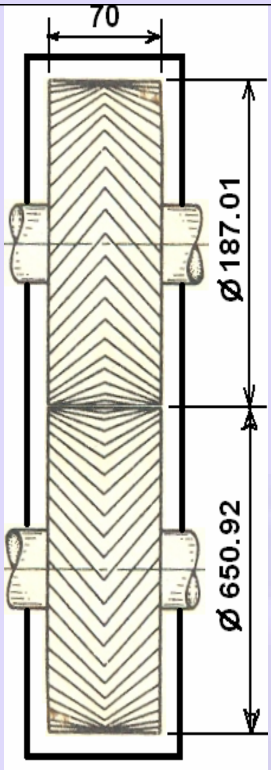


Fig.17.13 Gear box outer dimensions (tentative)

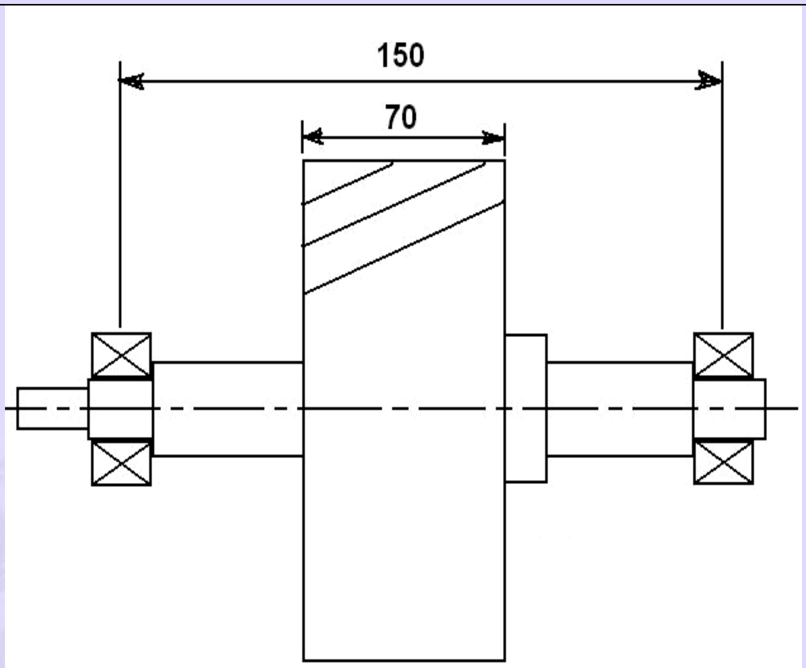


Fig.17.14 Pinion shaft layout diagram

All dimensions are in mm

4. Shaft design is based on the ASME equation:

Tangential load on the shaft: F_t = T/r = 971.6/0.088.5 = 11kN

$F_r = F_t \tan Ø = 11 \tan 23.96^o = 4.89$kN

$F = (F_t^2 + F_r^2)^{0.5}$

$= (11^2 + 4.89^2)^{0.5} = 15.42$kN

Bending moment at C

M = Fl /4 =15.42x0.15/4=0.58 kNm

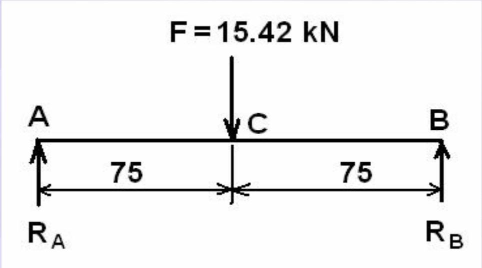


Fig.17.15 Shaft Loading

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 17.3, for rotating shaft with minor shock loads, K_b = 1.5 and K_t = 1.0.

Taking C45 steel for the shaft, σ_{yp} = 360 MPa

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

Table17. 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

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

$$d = \frac{16 \times 10^6}{\pi(1-0.2)x90}\sqrt{(1.5x0.58)^2 + (1x0.9716)^2} = 46\,mm$$

Take d = 50 mm. Check for deflection at the pinion centre.

Deflection at C: $\delta = \frac{Fl^3}{48EI} = \frac{15420 \times 150^3}{48 X 2.1x10^5 (\frac{\pi x 50^4}{64})} = 0.017\,mm$

Since δ < 0.01m = 0.01x5 = 0.05 mm, the design is OK.

Check for slope at the bearing at A.

Slope: $\alpha = \frac{FL^2}{16EI} = \frac{15420x150^2}{16x2.1x10^5 x(\frac{\pi x 50^4}{64})} = \mathbf{0.00034\ rad.}$

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

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

- $d_{1min} \geq 2$ x bore + 0.25 m

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

$D_{1min} \geq$ 2bore +0.1m= 2x50 + 0.1x5 = 100.5 mm

Since d_1 = 177.01 mm > D_{1min}. The design is satisfactory. Pinion drawing is shown in Fig.17.16 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.

Gear shaft diameter = d $(i)^{1/3}$ = 50 $(3.62)^{1/3}$ = 77 mm.
Gear shaft diameter of 80 mm is taken.
The hub diameter: d_H = 1.8 x 80 = 144 mm, 150 mm is taken.

Hub length is taken as L =1.25d =1.25 x80 ≈100 mm

Other dimensions of the gear are given in Fig. 17.17.

In view of the dimensions of the pinion and the gear, the dimensions of the shaft layout is revised as shown in Fig.17.18 When the calculations are redone, there is no change in shaft diameters. The same diameters are adopted for further computations.

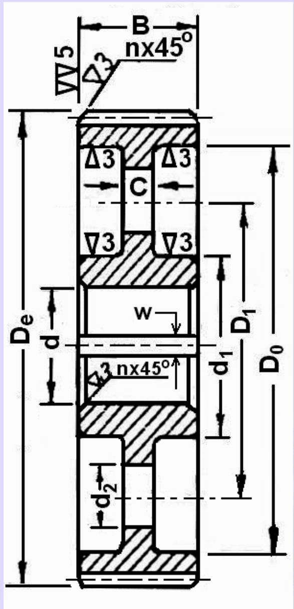


Fig.17.16. Pinion blank drawing

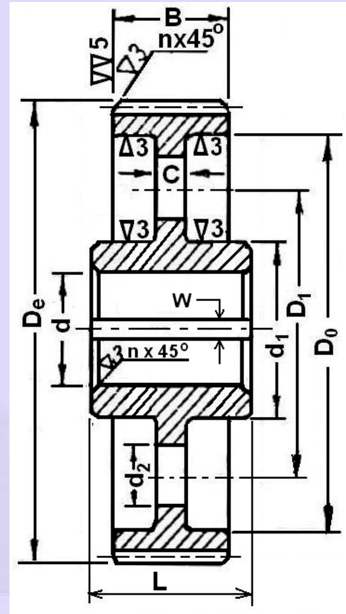


Fig.17.17. Gear blank drawing

Table 17.4 Dimensions of pinion and gear blank

Parameters	Pinion	Gear
d	50 mm	80 mm
d_1	1.8d ≈ 90 mm	1.8d ≈ 150 mm
B	70 mm	70 mm
C	0.3B ≈ 22 mm	0.3 B ≈ 22 mm
D_e	187.01 mm	650.92 mm
m	5 mm	5 mm
D_o	$\approx D_e - 10m = 137$ mm	$\approx D_e - 10m = 600$ mm
n	0.5 m = 2.5 mm	0.5 m = 2.5 mm
D_1	$(D_o + d_1) / 2 = 114$ mm	$(D_o + d_1) / 2 = 375$ mm
d_2	$\approx(D_o - d_1) / 5 = 10$ mm	$\approx(D_o - d_1) / 5 = 90$ mm
L	1.25 d = 70 mm	1.25d ≈ 100 mm
W	0.25 d = 12.5 mm	0.25d = 20 mm
Keyway depth	0.125d ≈ 6 mm	0.125d ≈ 10 mm

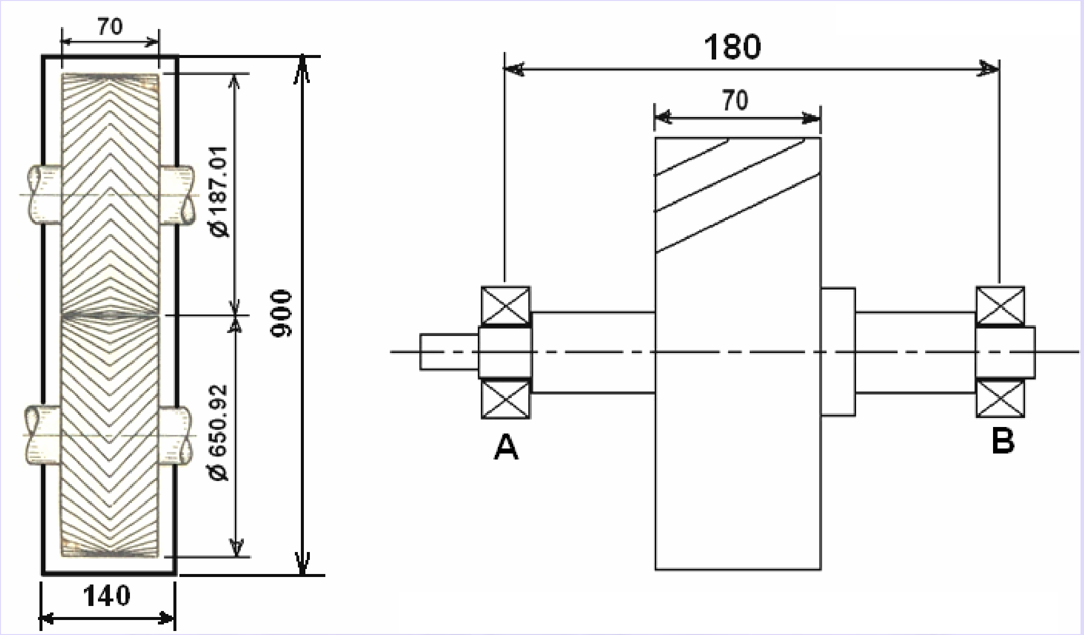


Fig 17.18 Pinion shaft layout diagram

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 N,

f_n = 0.224 for n = 2950 rpm from FAG catalog.

f_L = 3.91 for 30000 hrs life assuming 16 hrs/day working from FAG catalog.

C = (f_L/ f_n) P = (3.91/0.224)x7710 =134581N =134.6kN

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 kN, C_o = 153 kN, d_o =100 mm, d_i = 45mm, b = 36mm.

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

P = 7710 N,

f_n = 0.345 for shaft speed of 815 rpm.

f_L = 3.91 for Life of 30,000 hrs.

C= (f_L / f_n) x P = (3.91 / 0.345)x 7710 = 87,380 = 87.38 kN

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

8. Gearbox dimensions are fixed based on thumb rule given in Table 17.5.

Table 17.5 Wall thickness 's' in mm of the gearboxes

Material	Non-case hardened gears	Case hardened gears
CI castings	0.007L + 6 mm	0.010 L + 6 mm
Steel castings	0.005L + 4 mm	0.007L + 4 mm
Welded construction	0.004L + 4 mm	0.005L + 4 mm

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

s = 0.005L + 4 mm =0.005x 900 + 4 ≈ 10 mm

Top cover thickness: S_c = 0.8s =8 mm.

Flange thickness: sf = 2s = 2x10 = 20 mm

Flange cover bolt diameter: d_{cb} = 1.5s ≈16 mm M16 bolts.

Bolt spacing: 6d = 6x16 ≈ 100 mm

Foundation bolt diameter: $d_{fb}= (2T)^{1/3} \geq 12$ mm

$d_{fb} = (2x3.62x971.6)^{1/3}$ = 19.2, Take M20 bolts.

The thickness of the foundation flange should be:

$S_{ff} \geq 1.5\, d_{fb}$ = 1.5x20 = 30 mm

The width of the flange at the base: $w_b = 2.5d \quad = 2.5 \times 20 = 50$ 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, wf = 45 mm is taken.

Outside dimension of the bearing housing 1.2-1.5 times outside diameter of the bearing.
Bearing housing diameters are: 1.5x100 = 150mm and: 1.3x160 = 210 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. 17.19 and Fig.17.20.

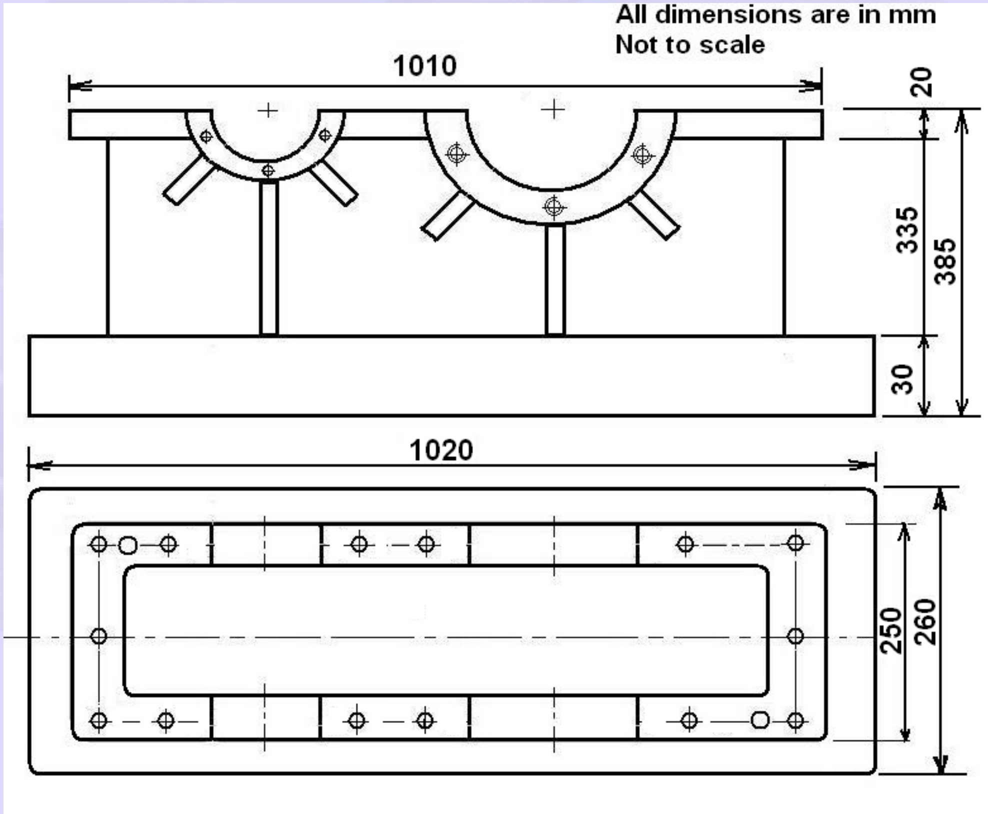


Fig.17.19 View of bottom half of the gearbox

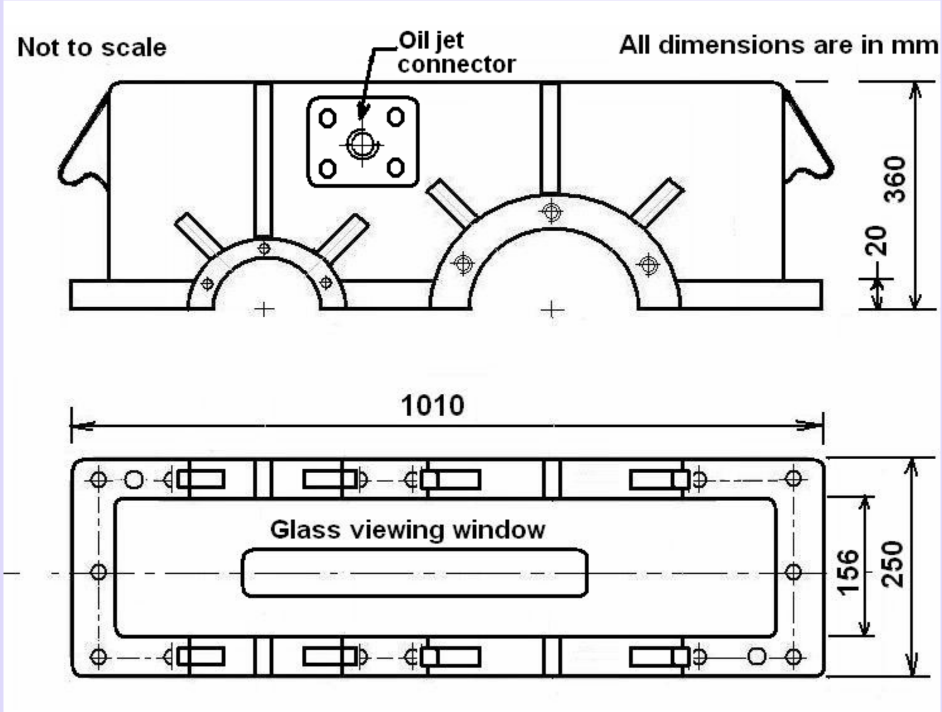


Fig.17.20 View of top cover of the gearbox

RECOMMENDED OIL FOR VARIOUS SLIDING SPEEDS

Table 17.6 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 = ωr = 308.77x0.0885 = 27.33 m/s.

From the Table 17.6, the recommended oil viscosity at 50^{o}C for this operation is V_{50} between 25 to 51 cSt (interpolated values). ISO VG 100 satisfies this, see Fig. 17.21

The equivalent grade from chart in Fig. 17.22, SAE 30 oil comes under this range and is recommended for the operation.

ISO VG GRADE LUBRICANTS

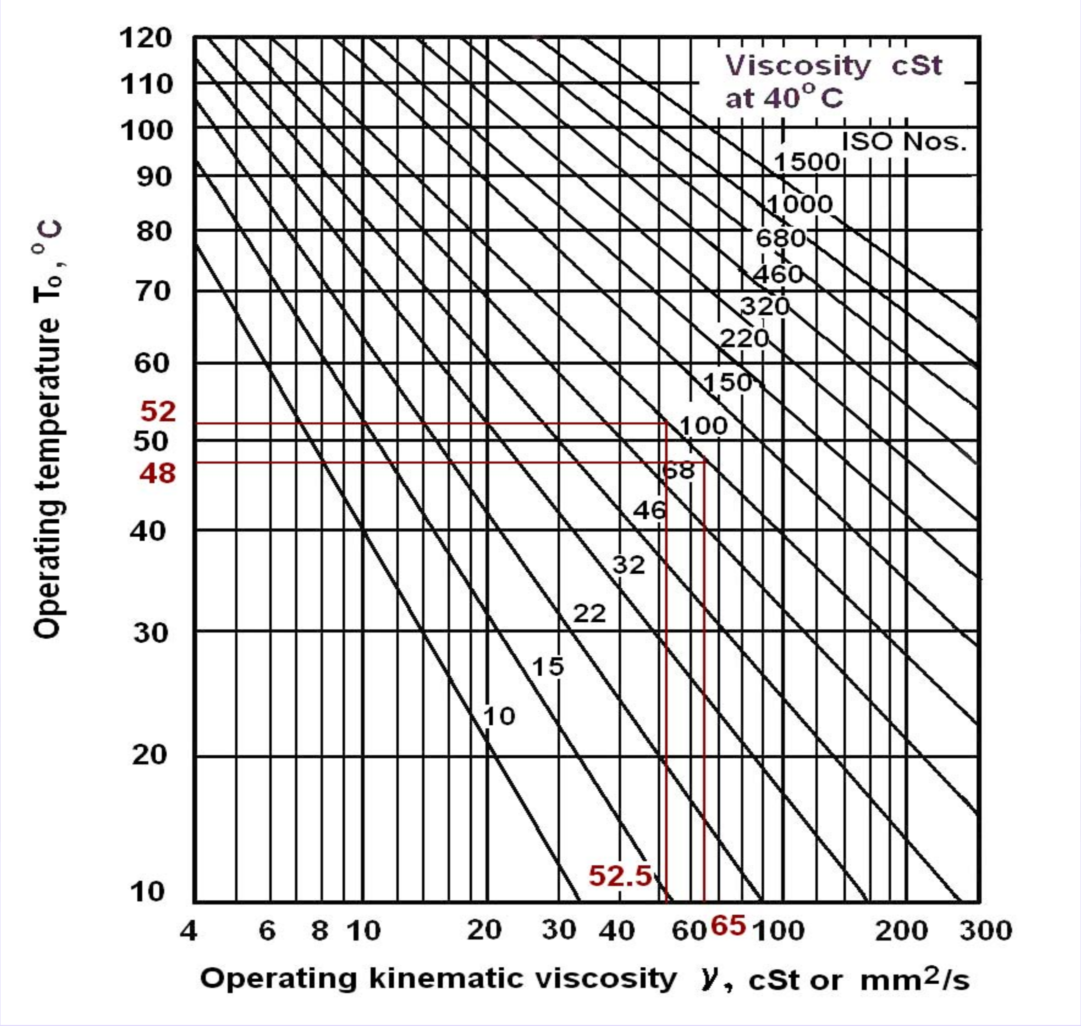


Fig.17.21 Viscosity – Temperature curve for ISO VG graded oils

SAE OIL VISCOSITY CHART

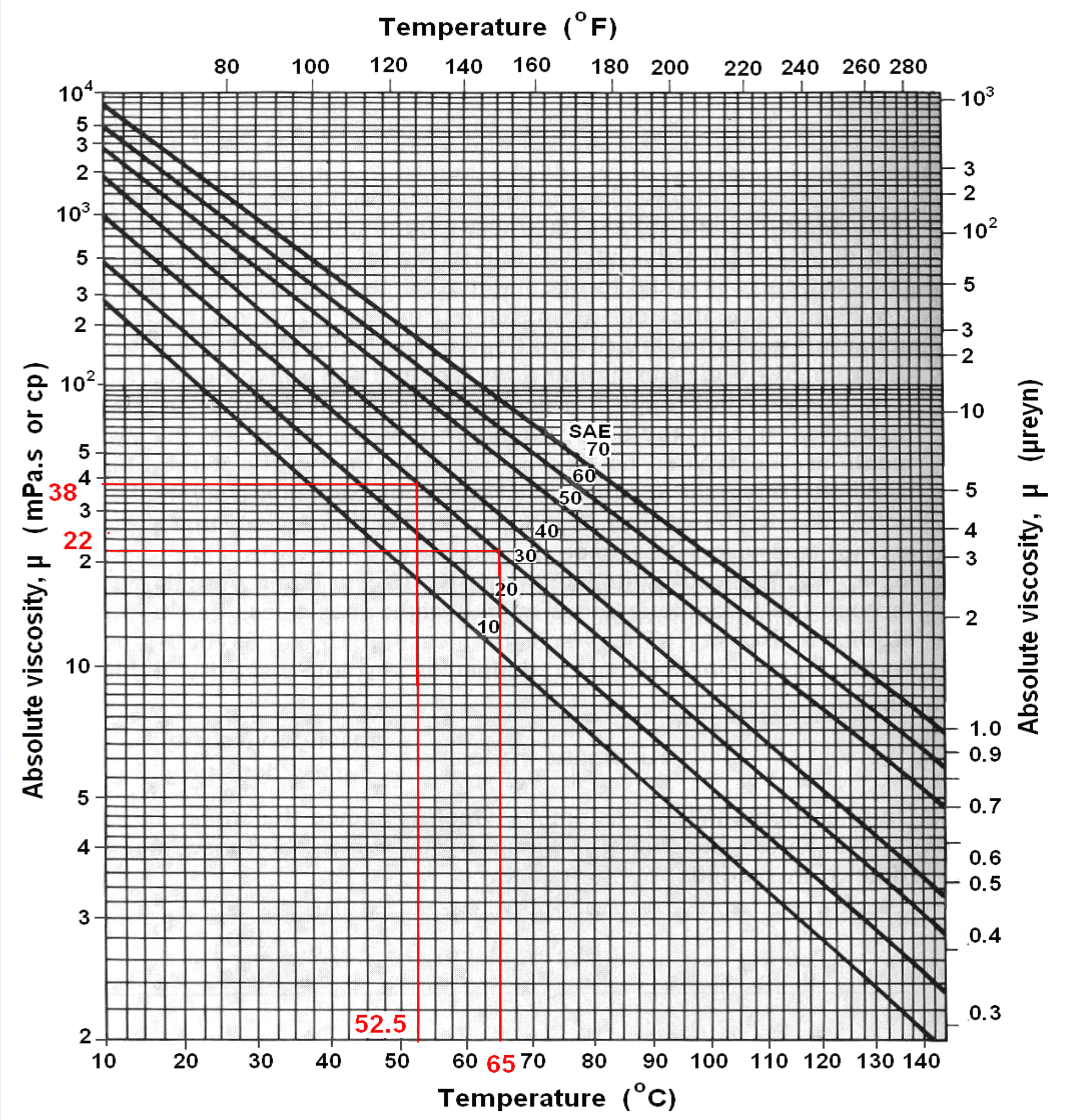


Fig.17.22 Viscosity – Temperature curves of SAE graded oils

9. Losses in gear boxes :

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 & Ls-seal frictional power loss.

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

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

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

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)

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

$$L_{ch} = 0.006x70x27.3x\left(\frac{200x27.3x35}{29+105}\right)^{0.5} x10^{-3} = 0.433\ kW$$

L_b = 5.23 x 10^{-8} F f_b d n 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

L_b = 5.23 x 10-8 F f_b d n

= 5.33x10-8x 15420x0.002 x 45x2950 =0.218 kW

Bearings at D & E

L_b = 5.23×10^{-8} F f_b d n

= $5.33 \times 10^{-8} \times 15420 \times 0.003 \times 75 \times 814.92 = 0.151$kW

$L_B = 0.369$ kW

Seal frictional power loss:

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

Where T_s seal friction torque

ω – angular velocity of the shaft.

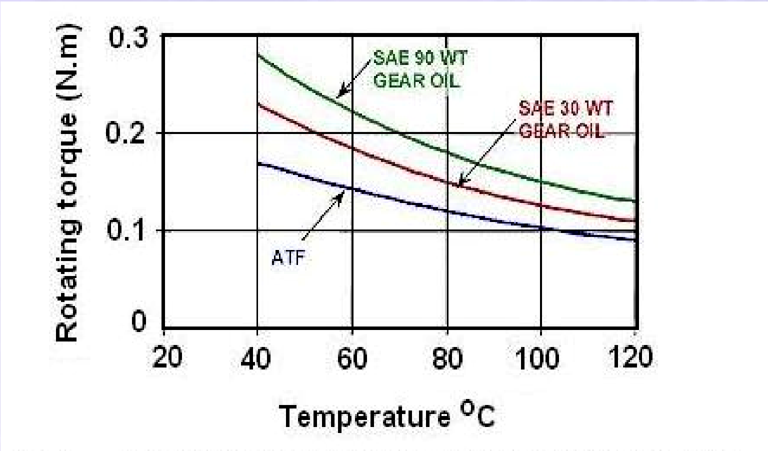


Fig. 17.23 Friction torque at various temperature for nitrile rubber S type oil seal, Shaft diameter 50mm, speed 1000 rpm

$T_s = f\, P_r\, r$

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/cm^2]

v = Linear Speed [m/s]

b = Lip Contact Width [m]

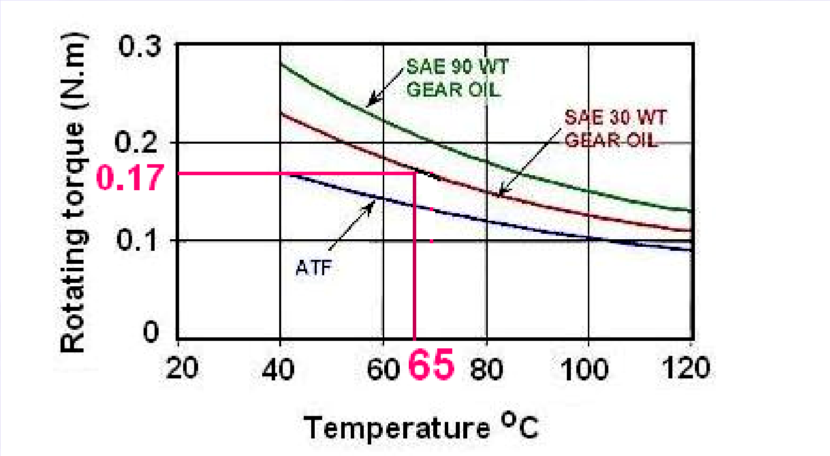
Fig. 17.23 gives the torque vs temperature chart for seal. Let the outlet oil temperature be 65^{o}C

At 65^{o}C, T_s =0.17Nm from Fig.17.23a

V = πdn/60000 = πx50x1000/60000 = 2.36 m/s

The operating Velocity V = πx45x2950/60000 = 6.95 m/s

T_s at operating speed of pinion shaft speed = 0.17 x $(6.95/2.36)^{1/3}$ =0.244 Nm



Graph 17.23a Friction torque at various temperatures for nitrile rubber S type oil seal, Shaft diameter 50 mm, speed 1000 rpm

Pinion shaft seals power loss

$L_s = T_s\ \omega\ x10^{-3}$ = 0.244x308.77 x 2 x10^{-3} = 0.151 kW

Gear shaft seal power loss

V = π x 75x814.92/60000 = 3.2 m/s

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

$L_s = T_s\ \omega\ x10^{-3}$ = 0.188 (308.77/3.62)x2x10^{-3} = 0.032 kW

Total seal friction = 0.151 + 0.032 = 0.183 kW.

Total power loss in the gearbox:

$L = L_t + L_{ch} + L_b + L_s$

$= 4.33 + 0.433 + 0.369 + 0.183 = 5.315$ kW

For the operating speed of the gear 27.33 m/s, the 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:

$Q_e = L \times 10^3 / \rho\, c\, \Delta T = (5.315\times10^3 / 0.88\times1670\times25) = 0.1447$ lps $= 0.01447\times60 = 8.68$ lpm.

Based on the details of the gearbox, the shaft details are worked out. The detailed pinion shaft drawing is shown in Fig.17.24 and that of the gear shaft in Fig.17.25. The corresponding revised dimensions of the pinion and gears are shown in Fig.17.26 and 17.27.

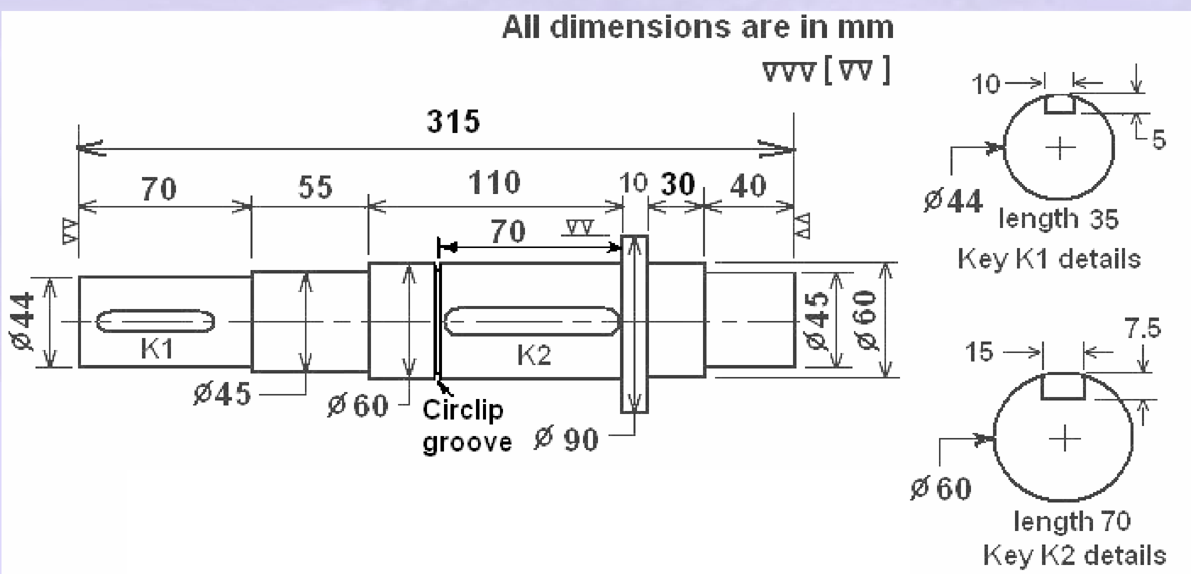


Fig.17.24 Pinion shaft drawing material C 45 hardened and tempered to 380 Bhn

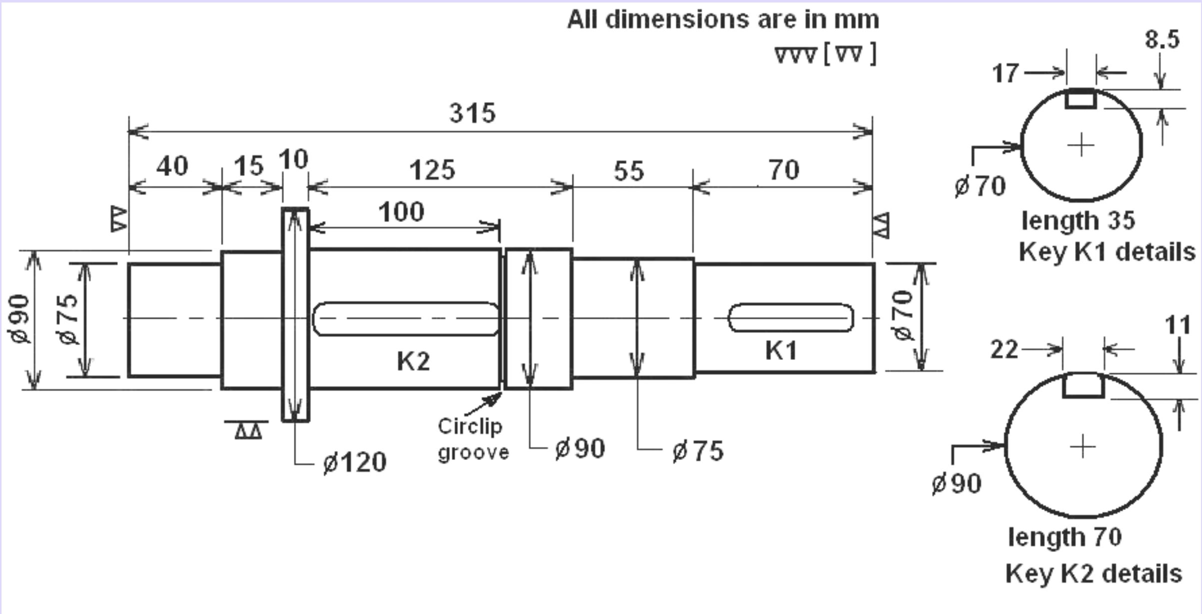


Fig.17.25 Gear shaft drawing material ductile iron 120/90/02 hardened to 331 Bhn

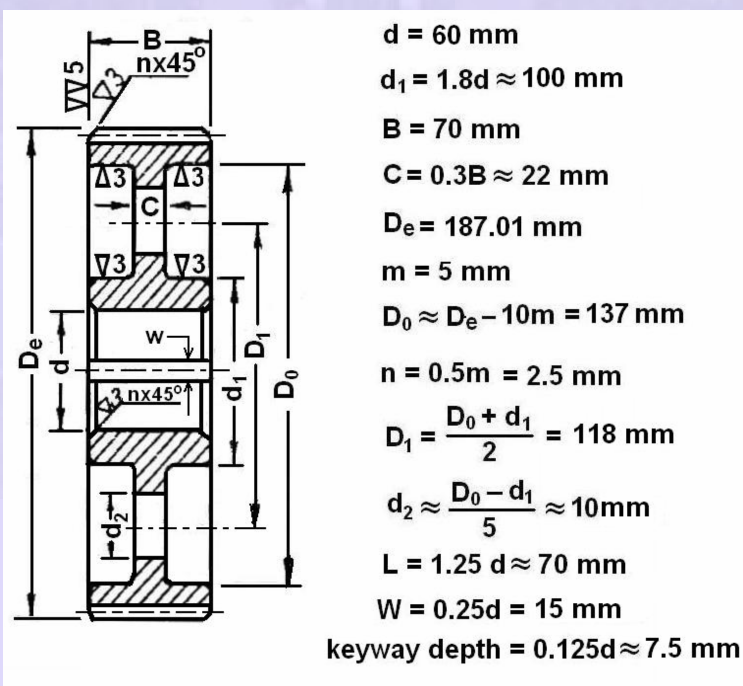


Fig.17.26 Pinion blank revised drawing showing all the dimensions

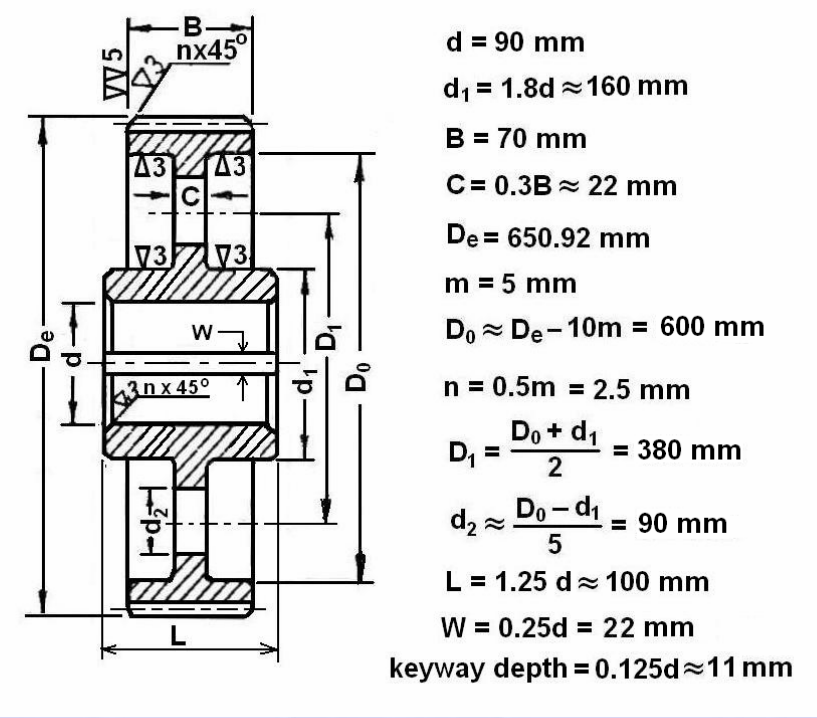


Fig.17.27 Gear blank revised drawing showing all the dimensions.

Table 17.6 Details of the gearbox

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	η = 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. 17.28 to 17.30. The front view separately and end view are shown separately in Fig 17.28 and Fig. 17.29 for clarity. The assembly view front and side together is shown in Fig.17.30.

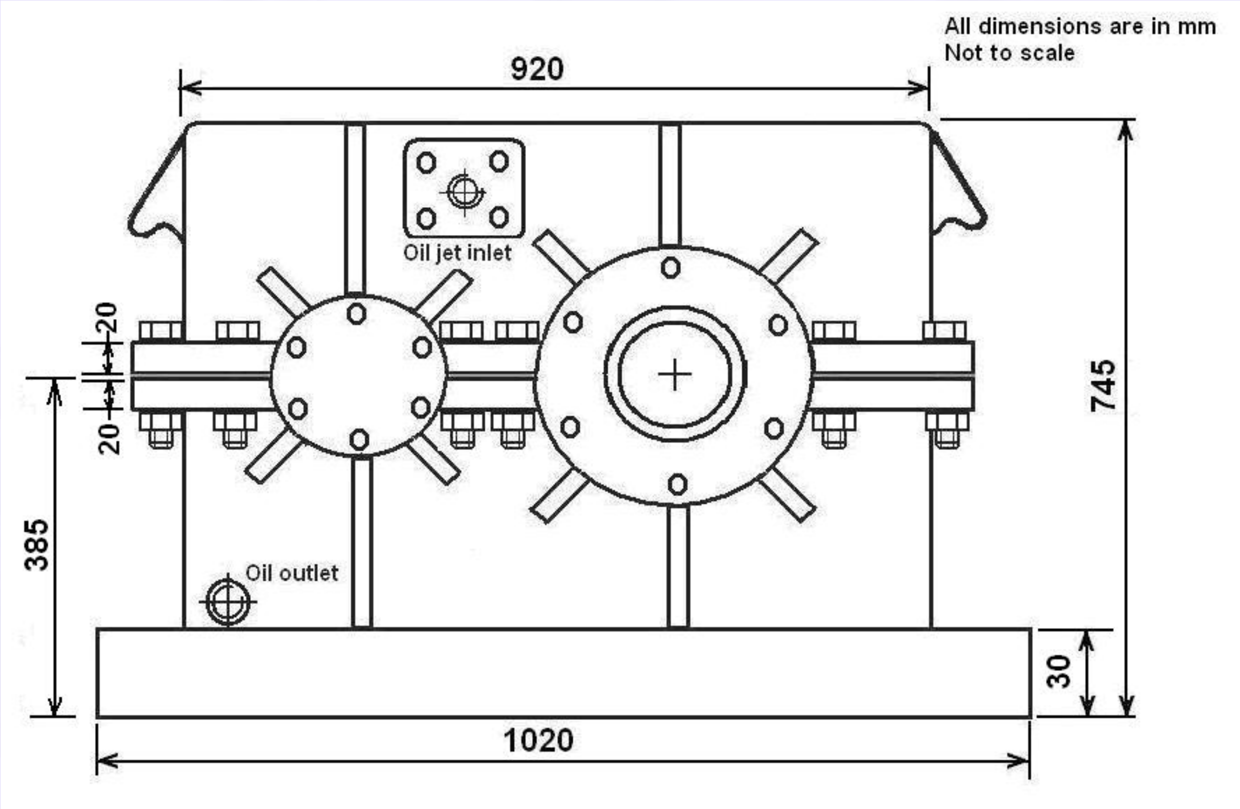


Fig.17.28 Front view of the assembly of the gearbox

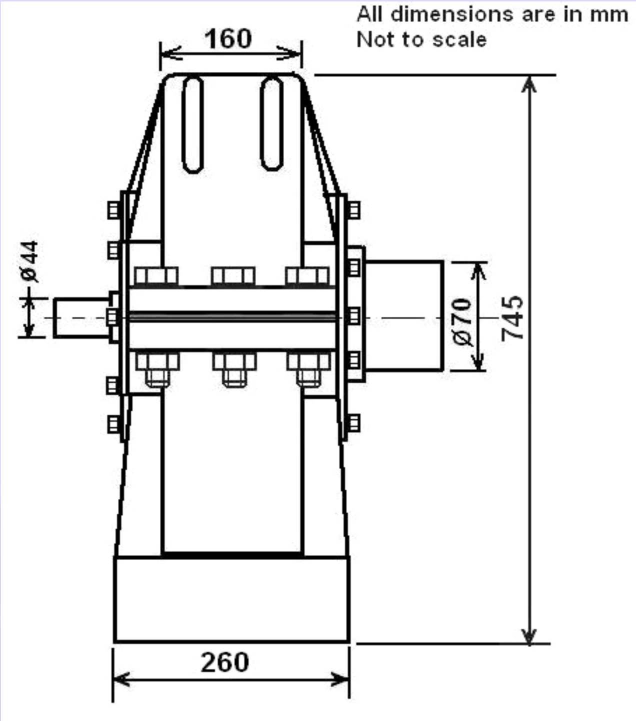


Fig.17.29 End view of the gearbox

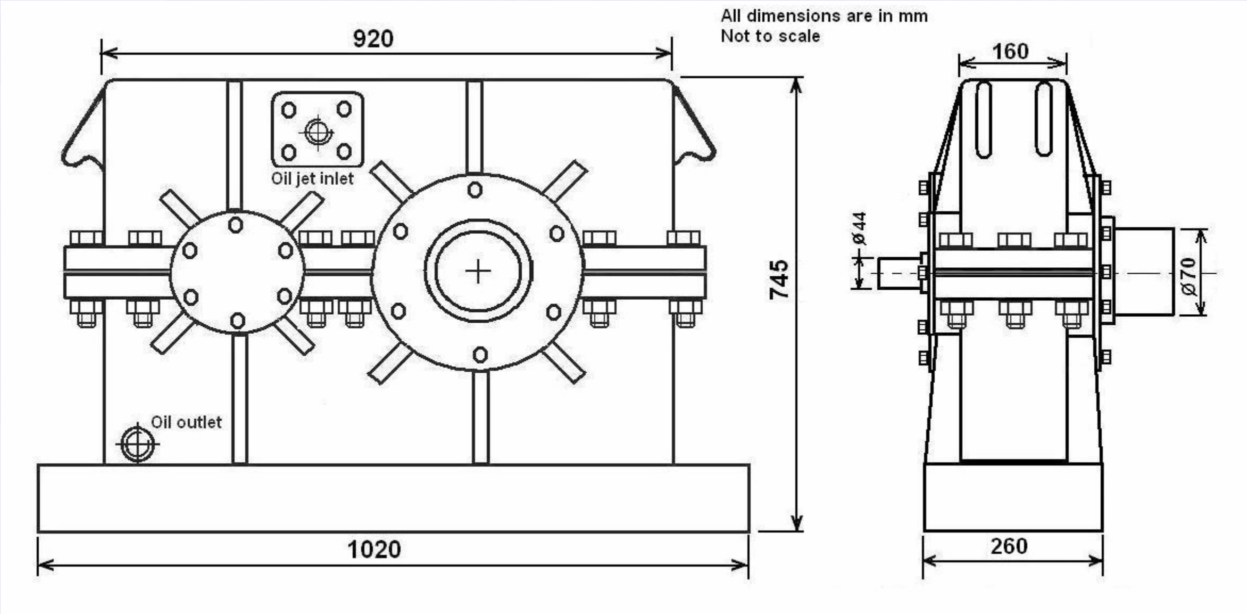


Fig.17.30 Assembled view of the helical gearbox
