DESIGN OF BACKGEAR MECHANISM IN LATHE

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Abstract

The objective of this project is to determine the design procedure involved in designing back gear mechanisms in lathe. It also gives detail about back gear mechanisms, its layout, its function and its design calculations. Design procedure include the design of hollow spindle, lay shaft, belt drive, gears and bearings. Calculations are made and compared with allowable stresses and allowable dimensions. This ensures a safe design for back gear mechanism's design data book is used for design calculation.

Keywords:

1. Introduction

1.1 Back gear mechanisms

As its name implies, "back gear" is a gear mounted at the back of the headstock that allows the chuck to rotate slowly with greatly-increased turning power. For a novice the ability to run a workpiece slowly might seem unnecessary, but a large-diameter casting, fastened to the faceplate and run at 200 r.p.m would have a linear speed at its outer edge beyond the turning capacity of a small lathe. By engaging back gear, and so reducing r.p.m. but increasing torque, even the largest faceplate-mounted jobs can be turned successfully.

1.2. Layout diagram

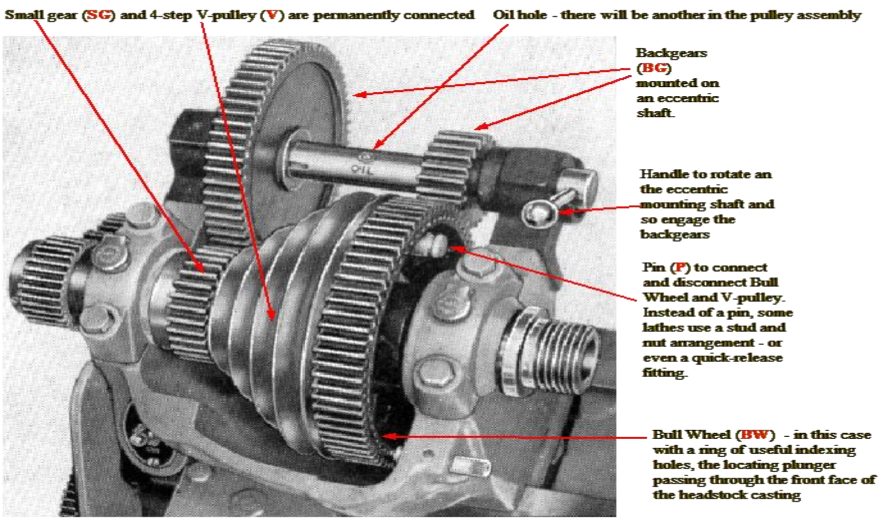

Fig1: This is layout diagram of 1934 Atlas lathe which has the 4-step V-pulley (V) with a small gear (SG) permanently attached to its smaller end. The entire length of V-pulley and gear are bushed - and able to rotate freely on the headstock spindle.

1.3. Working of back gear mechanism

In normal use the V-pulley is rotated by the drive belt and the spindle turned through the action of the pin driving the Bull Wheel and hence the spindle to which it is connected.

To engage back gear the lathe is stopped, the pin (**P**) withdrawn leaving the pulley and small gear free to rotate and the Back gear (**BG**) rotated on its eccentric shaft to bring it into mesh with the other gears. On starting the lathe action is now as follows: the pulley is rotated by the drive belt with the small gear (**SG**) on the V-pulley (**V**) driving the larger of the two rear-mounted back gears - which in turn causes the smaller gear at the other end of its shaft to rotate. This smaller gear drives the Bull Wheel (**B**) , and hence the spindle at a much lower speed (normally in the order of a 8 : 1 reduction) but with greatly increased torque.

2. Design calculation

2.1 Requirements.

Initial requirement for deign calculation of drive arrangement are taken as follows,

The lifting is provided by a 7 kW two speed electrical motor running at 720 rpm. The power is transmitted to the rope drum through a belt drive and a two stage spur gear reduction unit. The speed reduction in the belt drive is 2, while the overall speed reduction in the spur gear reducer is 16.

Design following

1. Design of belt drive
2. Design of gear
3. Design of main hollow spindle

2.2 Design of belt drive

Type of belt drive used in drive arrangement for a tiller crane is V Belt drive.

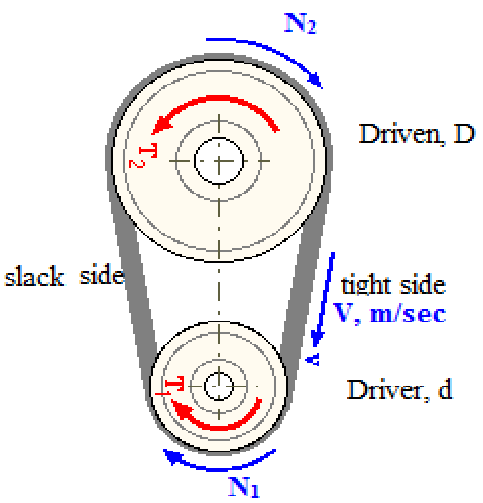

Fig:2.2.1 Pulley Arrangement

2.2.1Known Parameters

Center to center distance of driver and driven pulley C = 250 mm
Power P = 3.5 HP
= 3.5 X 736 KW
P= 2576 KW

Speed of input shaft (from motor) N_1= 550 rpm
Diameter of input shaft d_1= 300 mm
Diameter of driven shaft d2= 550 rpm
Tension on tight side of belt T_1= 200 N
$T_1/T_2 = 3$
Ratio of tension between tight sides to slag side [ɞ] = 40 MPa
Safe stress

2.2.2Selection of belt section

From design data book page no. 7.58 *B Section* is selected for the given power of 2.5 KW

- Nominal top width w = 17 mm
- Nominal thickness T = 11 mm

Speed of driven pulley (N_2)

$d_2/d_1 = N_1/N_2$

$550/300 = 550/N_2$

$N_2 = 300$ rpm

Therefore speed of driven pulley is 300rpm

Nominal pitch length of belt

From design data book 7.61

$L = 2C + \pi (d1 + d_2) / 2 + (d_2 - d_1)^2 / 4C$

$= (2 \times 250) + \pi (300 + 550) / 2 + (550 - 300)^2 / (4 \times 250)$

L = 1948 mm

Arc of contact angle

$\theta = 180^0 - 60^0 (d_2 - d_1) / 4C$

$\theta = 165^0$

Maximum Power Capacity Of Belt

Maximum power capacity = $(0.79 \times S^{-0.09} - 50.8/d_e - 1.32 \times 10^{-4} S^2) S$

S - Belt speed in m/s

De-Equivalent pitch diameter in mm

- Belt speed $S = \pi d_1 N_1 / 60$

$= \pi \times 300 \times 550 / 60$

S = 8.640 m/s

Equivalent pitch diameter $d_e = d_p \times F_b$

- pitch dia of smaller pulley (d_p) = 300 mm
- F_b for p/d (= 1.833) = 1.13

$d_e = 1.13 \times 300$

$d_e = 339$ mm

Maximum power capacity = $(0.79 \times S^{-0.09} - 50.8/d_e - 1.32 \times 10^{-4} S^2) S$

= 4.32 KW

Determination of number of belts

$n_b = (P \times F_a) / (KW \times F_c \times F_d)$

Service factor, F_a = 1.1 (up to 10 hrs light duty per day, from design data book page no 7.67)

Length correction factor, F_c = 0.97 (from design data book page no 7.68)

Correction factor for arc of contact, F_d = 0.79 (from design data book page no 7.68)

$n_b = (2.5 \times 1.1) / (4.32 \times 0.97 \times 0.79)$

$n_b = 0.83 = 1$ belt

Modified center distance or actual center distance

$C_{actual} = A - \sqrt{(A^2 - B)}$

$A = L/4 - \pi (D + d)/8$

$= 1948/4 - \pi (550 + 300) / 8$

$A = 153$ mm

$B = (D - d)^2 / 8 = (550 - 300)^2 / 8$

$B = 7813$

$C_{actual} = 153 + \sqrt{(153^2 - 7813)}$

$= 277.88$ mm

$C_{actual} = 280$ mm

2.2.3Calculation of belt tension

Power transmitted per belt = $(T1 - T_2)$ v

$2500 = (T_1 - T_2) \times 8.64$

$T_1 - T_2 = 290$ N

$T_1 / T_2 = 3$

T2 = 290 N

T1 = 435 N

Calculation of stress induced

Stress induced $[\sigma] = \dfrac{\text{maximum tension}}{\text{Cross sectional Area}}$

Cross sectional Area of V Belt = 140 mm^2

$[\sigma]_{induced} = 435 / 140$

$= 3.1$ N/mm^2 (> safe stress)

Therefore the design of safe

2.3 Design of gear

In speed reduction unit spur gear is used. In spur gear the teeth are straight and parallel to the shaft axis, Transmits power and motion between rotating two parallel shafts. The features of spur gear are easy to manufacturing, there are no axile force, relatively easy to produce high-quality gears, most common type of gear.

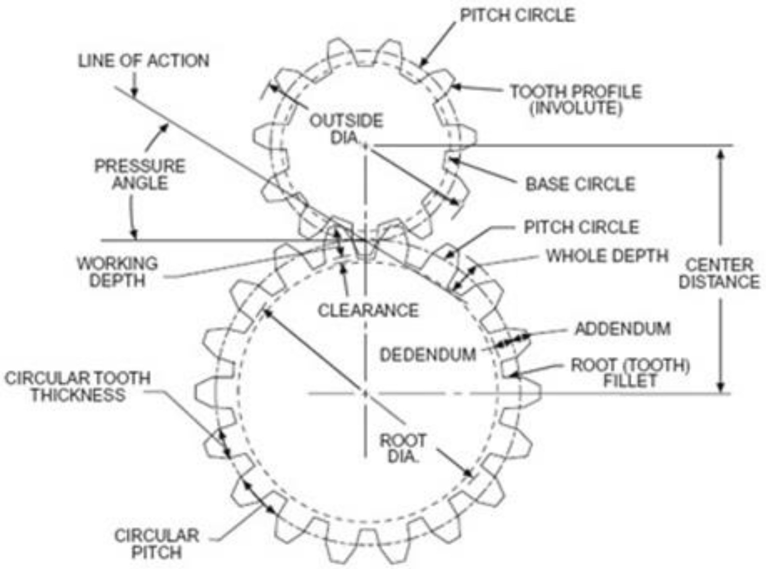

Fig.2.3.1. Spur gear drawing

Gear used in this mechanism is 20^0 involute spur gear

Speed of gears in input shafts

- $N_1 = N_3 = 300$ rpm
- $N_2 = N_4 = 550/8 \approx 70$ rpm(1:8 speed reduction)

Transmission ratio (i)

$i = N_1 / N_2$

$i = 300 / 70 = 4.3$

Number of teeth

Assume number of teeth in Z_1 and $Z_3 = 18$

$Z_2 = i \text{ X } Z_1 = 4.3 \text{ X } 18$

$Z_2 = 78$

$Z_2 = Z_4 = 78$

2.3.1Material selection

From design data book page number 8.4 ,for i > 4, C45 is taken material for both pinion and wheel

Design twisting moment [Mt] $[M_t] = M_t \text{ X } k_d . K$

k_d = Dynamic load factor

k = load concentration factor

$k_d . k$ = 1.3 (Assume for symmetric scheme)

M_t = 97420 X KW / n

= 97420 X 2.5 / 300

M_t = 811.8 kgf.cm

$[M_t]$ = 811.8 X 1.3

= 1055 kgf.cm

Design surface stress $[\sigma_c]$

Design surface compressive stress $[\sigma_c] = C_R .HRC .K_{cl}$

Rockwell Hardness, HRC = 55 (for CI Grade 20, 25)

C_R = 205

Life factor for compressive stress, K_{cl} = 1

$[\sigma_c]$ = 205 * 1 * 55

= 14575 kgf/cm^2

Center distance

$$a >= (i + 1) \sqrt[3]{[(0.74/ [\sigma_c])^2 (E [M_t]/ i \text{ X } \varphi)]}$$

ψ = 0.3 (from DDB Page no. 8.14)

i = 4.3

E = 2.15 X 10^6 kgf/cm^2

Substituting the above values we get center distance a= 20 cm

Module (m)

$$m = 1.26 \text{ X } \sqrt[3]{([M_t] / (y \text{ X } [\sigma_b] \psi_m \text{ X } Z_1))}$$

y = 0.377 (from DDB Page no. 8.18, for Z = 18)

ψ_m = 6 (from DDB Page no. 8.14)

Substituting the above values we get module, m = 0.3043 cm=0.4cm(standard module)

Checking for surface stress σ_c

$$\sigma_c = 0.74 (i+1)/a \sqrt{((i+1)/ib) E [M_t]}$$

ψ_m = b/m = 6

b = 6 X .03 = 1.8 cm

Substituting the above values we get σ_c = 225 kgf/cm^2 <[σ_c]=14575 kgf/cm^2

Therefore design is safe.

Checking for bending stress σ_b

Bending stress σ_b = [(i+1)/(a_m b y)] [M_t]

= (4.3+1)/(20 X 0.3 X 1.8 X 0.377)

σ_b = 1373 kgf/cm^2 < [σ_b]=1840 kgf/cm^2

Therefore design is safe

Dimensions Of Pinion And Gear

- Module m = 0.4cm = 4 mm
- Center distance a = 20cm = 200 mm
- Bottom clearance c = 0.25 * m = 0.75 mm
- Tooth depth h = 2.25 * m = 6.75 mm
- Pitch diameter d_1 = m * z_1 = 72 mm , d_2=m * z_2 = 312 mm

2.4. Design of main hollow spindle

Pulley D

Power at pulley D is 2576 watts and its diameter is 550mm

Torque Transmitted

Power P = (2πnT_D)/60

T_D = (60*2576)/(2π*300)

Therefore torque T_D = 82 Nm

Total Load On Pulley D

Tension on tight side T_1 = 447 N

Tension on slack side T_2 = 149 N

Total load on pulley D is W = T_1+T_2 = 447+149

W = 596 N

Gear C

Diameter of gear C is 72mm and it has 20^0 involute angles. Since shaft is rotating at 300rpm as same as for pulley speed ,its torque is 82N.m which is to be transmitted.

Normal Load In Gear C

Tangential force F_t*R =T_D, Radius R=36 mm

Tangential force F_t = 2278 N

Normal load = F_t/cos20^0 =2278/cos20^0

=2424 N

Vertical Component Force

Vertical component force = F_t*cos20^0

= 2424*cos20^0

= 2278 N

Horizontal Component Force

Horizontal component force = F_t*sin20^0

= 2424*sin20^0

= 829 N

2.4.1. Gear E

Diameter of gear E is 312mm and it has 20^0 involute angle. Since shaft is rotating at 300rpm as same as for pulley speed, its torque is 82N.m which is to be transmitted.

NORMAL IN GEAR E

Tangential force F_t*R = T_D, Radius R=156 mm

Tangential force F_t = 526 N

Normal load = F_t/cos20^0 = 559/cos20^0

= 559 N

Vertical Component Force

Vertical component force = F_t*cos20^0

= 559*cos20^0

= 526 N

Horizontal Component Force

Horizontal component force =F_t*sin20^0

= 559*sin20^0

= 191 N

Bending Moment on Shaft

Vertical Loading

Reaction Forces For Vertical Loading

Taking moment about RBV

RAV*200-2278*180-596*120-559*100=0

RAV=2687 N

RAV+RBV=2278+596+559

RBV=746 N

Bending Moment For Vertical Loading

Bending moment at C

M_{CV}=RAV*0.02=2687*0.02

M_{CV}=53.7 Nm

Bending moment at D

M_{DV}=RAV*0.08-C*0.06=2687*0.08-2278*0.06

M_{DV}=78.3 Nm

Bending moment at E

M_{EV}=RAV*0.1-C*0.08-D*0.02

M_{EV}=74.5 Nm

Horizontal loading

Reaction Force

Taking moment about RBH

RAH*200-829*180-191*100 =0

RAH =842 N

RAH+RBH=829+191=1020

RBH=178 N

2.4.2 Bending moment for horizontal loading

Bending moment at C

M_{CH}=RAH*0.02=842*0.02

M_{CH}=16.8 Nm

Bending moment at D

M_{DH}=RAH*0.08-C*0.06=842*0.08-829*0.06

M_{DH}=17.6 Nm

Bending moment at E

M_{EH}=RAH*0.1-C*0.08-D*0.02

M_{EH}=17.8 Nm

Maximum Bending Moment

Resultant bending moment at D

$M_D = [(M_{DV})^2 +(M_{DH})^2]^{0.5}$

$= [(78.3)^2 +(17.6)^2]^{0.5}$

M_D = 80.25 Nm

Resultant bending moment at E

$M_E = [(M_{EV})^2 +(M_{EH})^2]^{0.5}$

$= [(74.5)^2 +(17.8)^2]^{0.5}$ =76.5 Nm

From above it is clear that max bending moment occur at D

Therefore maximum bending moment M_{max} = 80.25 Nm

2.4.3Torque Equivalent

Torque equivalent $T_{eq} =[(M_{DES})^2 +(T_{DES})^2]^{0.5}$

Design bending moment

$M_{DES} = K_b * M_{max}$

Combined fatigue and shock factor for bending K_b is assumed to be 2

M_{DES} = 2* 80.25

M_{DES} = 160.5 Nm

Design torque

$T_{DES} = K_t * T_{max}$

Combined fatigue and shock factor for twisting K_t is assumed to be 1.5

T_{DES} = 1.5* 82

T_{DES} = 123 Nm

Torque equivalent

$T_{eq} = [(M_{DES})^2 +(T_{DES})^2]^{0.5}$

$= [(160.5)^2 +(123)^2]^{0.5}$

T_{eq} = 202 Nm

2.4.4 Diameter of hollow spindle

$T_{eq} = [(\pi/16)*\tau][(D_O^4 - D_I^4)/ D_O]$

$$202*10^3 = [(\pi/16)*\tau][(D_O^4 - D_I^4)/ D_O]$$

Assume that the ratio of outside diameter to inside diameter is to be 1.35,i.e

$$D_O / D_I = 1.35$$

$$1.72D_I^4 = (16*202*10^3)/(\pi*40)$$

Inside diameter D_I = 24.66 mm and

Outside diameter D_O = 33.21 mm

From DDB page no:7.20,standard diameter of shaft is taken as

Inside diameter D_I =25 mm

Outside diameter D_O =35 mm

3. Design of bearing

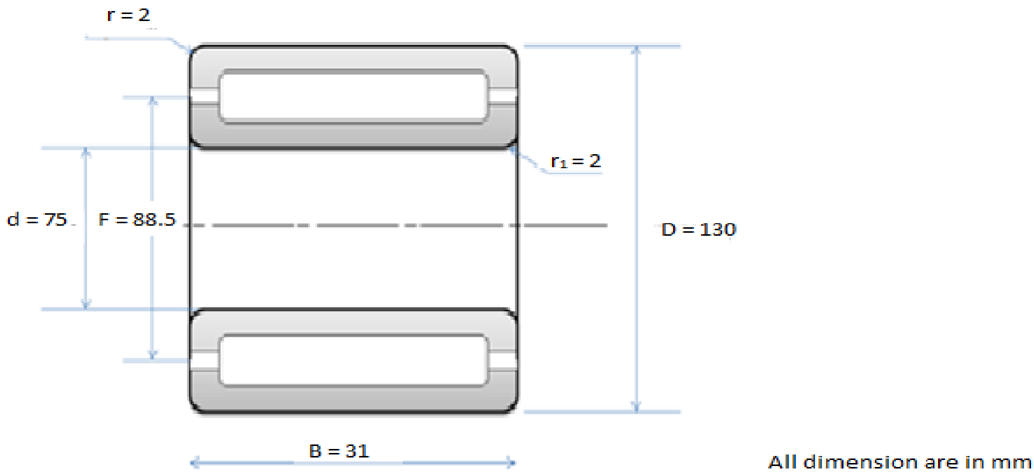

Fig3.1 Designed Bearing

Type of bearing used in back gear mechanism is ***roller bearing***

From Nu 2.2 series for shaft diameter 60 mm

Diameter of bearing = 60 mm

Outer dia , D = 110 mm

Width , B = 28 mm

Outer corner radius, r_2 = 2.5 mm

Inner corner radius, r_1 = 2 mm

Static capacity, C_o = 6200 kgf

Dynamic capacity, C = 7100 kgf

f = 73.5 mm

Max. permissible feed = 6000 rpm

Equivalent bearing load (P)

$C / P = 4.5$

$P = C / 4.5$

$= 7100 / 4.5$

$P = 1578$ kgf

Life of bearing

$L = (C / P)^{10/3}$

$= (7100 / 1578)^{10/3}$

$L = 150.14$ mm

$= (150.14 \text{ X } 10^5) / 60 \text{ X rpm}$

$= (150.14 \text{ X } 10^5) / 60 \text{ X } 300$

$L = 8358$ hrs.

Comparing in graph from DDB Page no. 4.7

For 300 rpm and C/P = 4.5,

L= 8500 hrs. , which is approximately equal to calculated life time

4. RESULTS AND DISCUSSIONS

4.1. DESIGN OF BELT DRIVE

B Section is selected for the given power of 7 kW,

a. Nominal top width w = 17 mm

b. Nominal thickness T = 11 mm

c. Nominal pitch length L = 2888 mm

d. Arc of contact angle θ = 155^0

e. Maximum power capacity kW = 5.4 KW

f. Number of belt n_b = 3

g. Actual Center Distance C_{actual} =755mm

4.2. DESIGN OF GEAR

The gear used are 20^0 full depth involute spur gear

a. Number of teeth in $Z_1 = 20$, $Z_2 = 80$

b. Module m = 0.4cm = 4 mm

c. Center distance a = 20cm = 200 mm

d. Bottom clearance c = 0.25 * m = 0.75 mm

e. Tooth depth h = 2.25 * m = 6.75 mm

f. Pitch diameter d_1 = m x Z_1 = 80 mm

d_2 = m x Z_2 = 320 mm

4.3. DESIGN OF INTERMEDIATSHAFT

Standard diameter of hollow spindle is

a. Diameter d = 75mm

4.4. DESIGN OF BEARING

a. Diameter of bearing d = 75 mm

b. Outer diameter D = 130 mm

c. Width B = 31 mm

d. Outer corner radius, r_2 = 2.5 mm

e. Inner corner radius, r_1 = 2.5 mm

f.	Static capacity,	C_o	= 8650 kgf
g.	Dynamic capacity,	C	= 9800 kgf
h.	Distance between rollers	F	= 88.5 mm

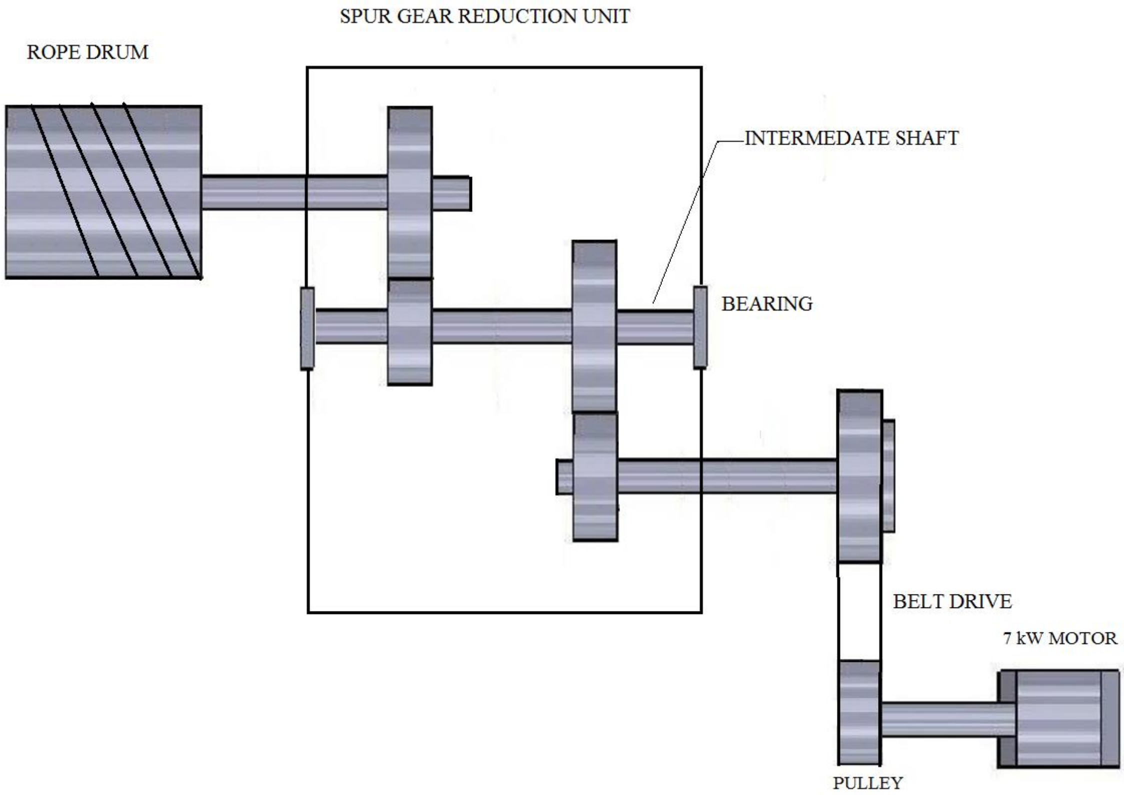

Fig 4.1 Layout Of Drive Arrangement

5. CONCLUSION

The detailed study of back gear mechanism about its working and its layout is made. Also discussed about the design procedure involved in design of back gear mechanism in lathe. The design procedure include the following design calculation

1. DESIGN OF BELT DRIVE
2. DESIGN OF GEAR
3. DESIGN OF MAIN HOLLOW SPINDLE
4. DESIGN OF LAY SHAFT
5. DESIGN OF BEARING

The design calculations made are compared with its allowable stress values which ensure the safe design for back gear mechanism.

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