MACHINE SYSTEM DESIGN

A Project Report on

“DESIGN OF ESCALATOR”

Submitted by

Group 2D

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1. NAME OF THE MACHINE :

Escalator

2. PURPOSE OF THE MACHINE :

The purpose of the machine is to transport people to required height at a constant low speed.

3. DESIGN OBJECTIVE :

Underground metro stations are busy locations. The aim here is to design an escalator that will funnel heavy traffic at these stations and would function for around 20 hours a daily. The data of Delhi metro stations is taken as a reference for the magnitude of traffic flow to be dealt with. There are two peaks in the traffic flow generally and the escalator should be able to handle it.

4. BROAD DESIGN SPECIFICATION :

Location of escalator : Underground metro

Height to which transport is carried out : 7.5m.

Transport capacity : ~ 6000-7000 people /hour at peak hours

Number of people per step : 1.

Average weight of the people being transported : 75kg.

5. SELECTION OF PROCESS AND MECHANISM

Process Description:

The basic requirement of an escalator is that the surface on which people stand remain horizontal at every location on the step path. Essentially it should behave as a moving staircase. However, to ensure that the number of steps are not exhausted, the steps need to be moved in a loop, and this loop is then driven with a flexible drive such as a belt or a chain. The entire system is driven by a motor with a suitable transmission that reduces the rpm.

To ensure that the steps follow the intended path, ***guideways*** are provided over which the wheels attached to steps can roll. Guideways need to be supported by a truss network. The above process is a prerequisite for any escalator.

Selection of Mechanism:

To ensure the horizontality of step surface two mechanisms or structures were considered as follows :

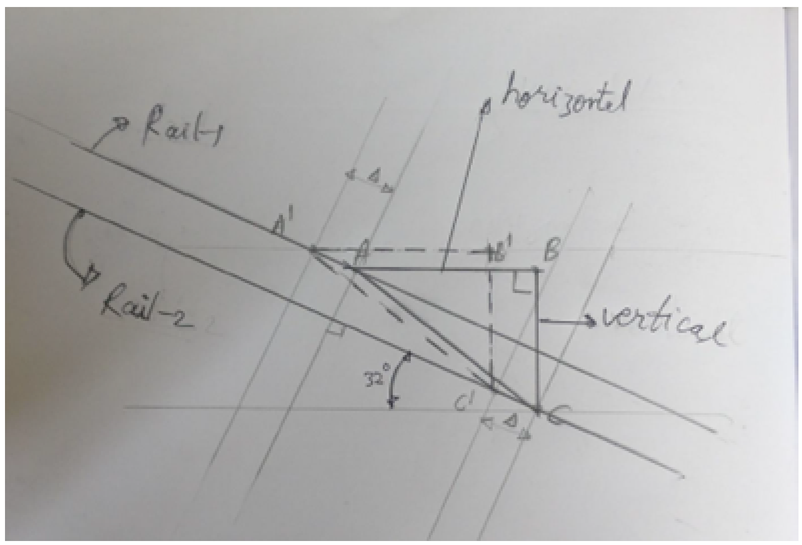


Figure 1

a) This structure for supporting the step is generally used in the escalators today. It utilizes 2 rails each side to keep the steps horizontal while it moves. The vertices A and C move along the rails (using rollers) at the pre-decided angle, but the surface AB (landing) remains horizontal as the step moves along the guideway.

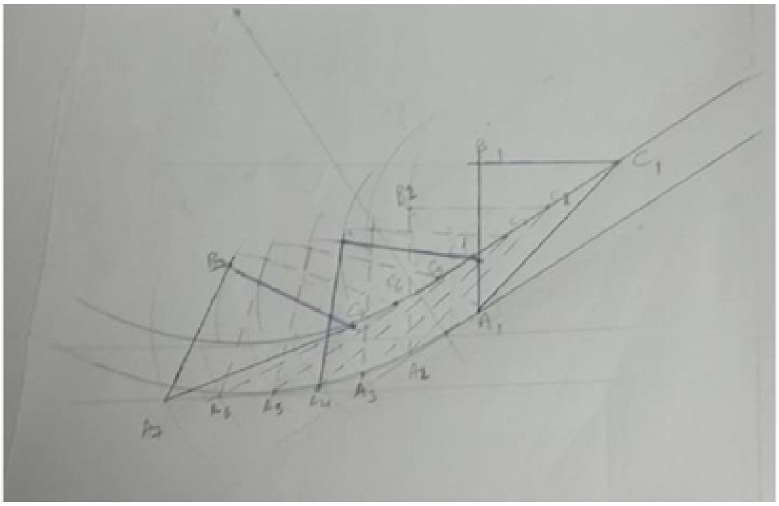


Figure 2

This construction is done to prove that we cannot have both the rails parallel in the transition zone because the step loses its horizontality. Thus, in the transition zone both rails cannot be parallel but need to diverge. It is expected that a similar situation exists at the upper transition zone.

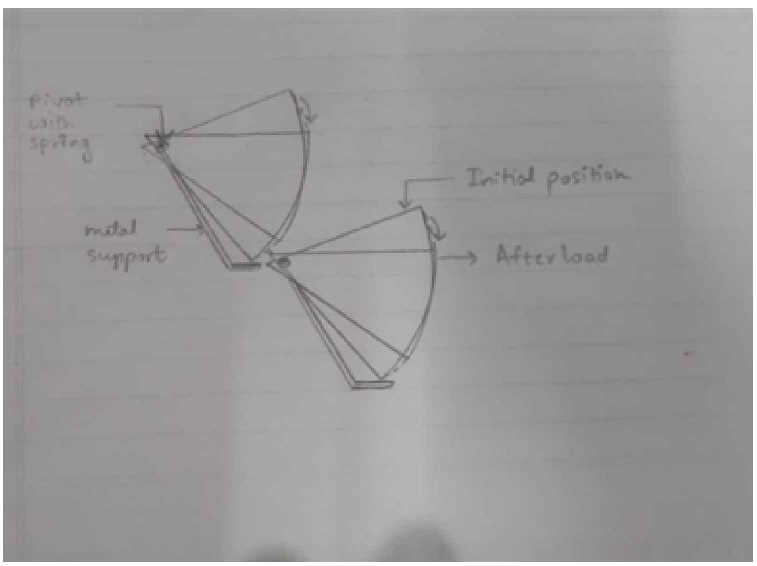


Figure 3

b) This structure is proposed as an alternative to the first scheme. It requires only one rail or guideway on each side. The entire step is pivoted by one point on each side. The unloaded step landing shall be at some angle governed by the torsional spring at the pivot. When a person steps on the landing, the landing shall become horizontal, and its travel shall be restricted by a positive locking. The plate at the bottom provides the locking and support.

Pros and cons of (a):

Pros – Here, the steps are always horizontal irrespective of a load being present. This makes it user friendly.

Cons – The manufacturing of the lower railing is difficult due to its unusual profile at transition.

Pros and cons of (b):

Pros - As is evident the complications that emerged in process 1 are absent here. Firstly, here we need only one rail each side and horizontality is retained even in transition without much hassle. Elimination of one guideway is a major advantage.

Cons - Requirement of torsion springs and plates at every step which makes manufacturing costly. However, the problems with the second scheme need a more detailed discussion.

Problems with the (b) structure:

Firstly, it is critical to understand that, for the plate to act as a support, it must be a cantilever beam on the axle. This is because it must have relative motion with respect to the axle.

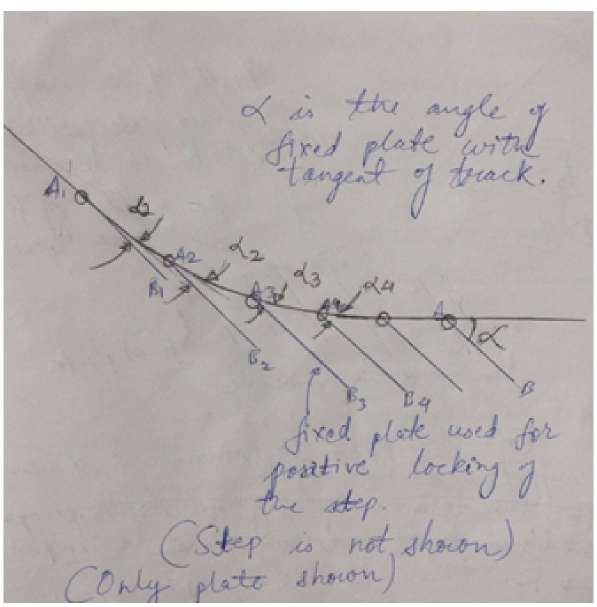


Figure 4

This construction shows only the base plate at different locations in transition. Say α is the angle it makes with the horizontal. To maintain the horizontal step, the angle with horizontal should not change anywhere in its travel, which means the angle with the track tangent changes continuously in the transition zone($\alpha_1<\alpha_2<\alpha_3<\alpha_4$).

If the base plate is rigidly fixed with the step frame, then its angle with horizontal will be altered due to the transition. Thus, the base plate needs to be attached via a **pivot** to the step frame. However, if we pivot it, the base plate cannot resist moments resulting from the loading of the step.

We need another support apart from point A (possibly at B) to ensure that the plate resists moments from the loaded step which can be done by **another track**. But, if we go for another track, then we are nullifying the main advantage that this structure is providing.

Another issue is that a rider of an escalator would not prefer the step surface to be moving. A moving surface can cause hesitation and anxiety in using an escalator.

Thus, a double guideway system (a) where the step is a rigid body is preferred to maintain the horizontality of the step.

<u>Drive:</u>

A **chain drive** is chosen to drive the steps on the guideway. The chain is to have a **special attachment** to house the step axle. Both the pivots would have rollers attached to them. Further, the chain drive would be applied to the upper pivot and the roller at the lower pivot roller would follow.

6. DESIGN OF MECHANISM

<u>Design of guideways:</u>

As noted earlier, in the 2-support system, both the guideways cannot be parallel throughout otherwise the step landing would lose its horizontality.

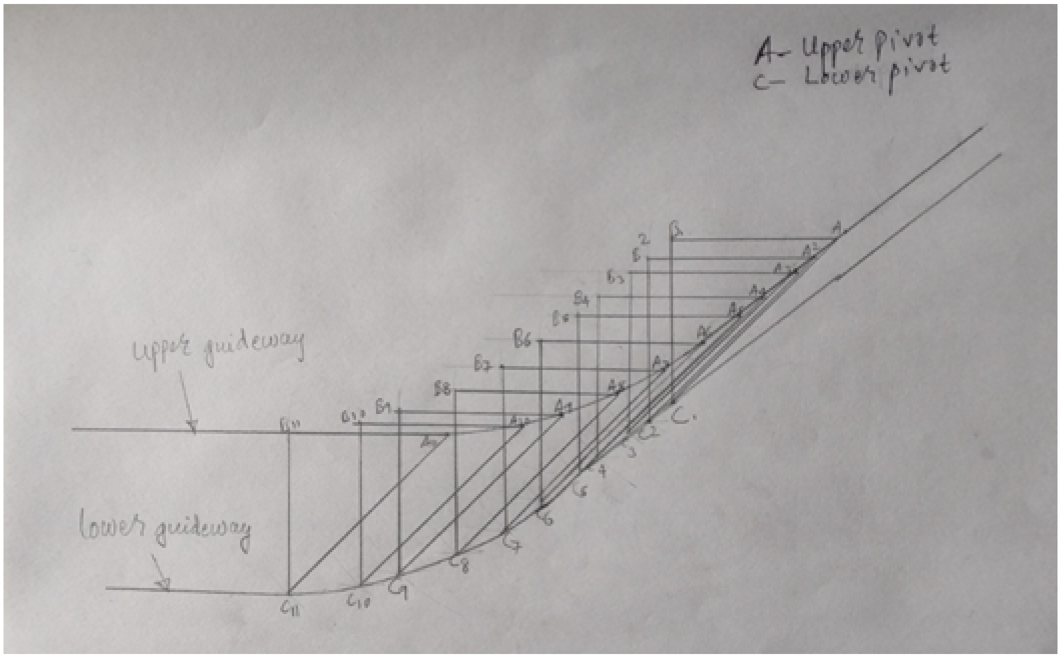


Figure 5

This construction is done by considering that A-B(Step surface/landing) is horizontal and then tracking the point C which is the lower pivot. The upper guideway is drawn with some radius of curvature**. It seems that the lower pivot C also follows a curved path, but its curvature begins sometime later to A depending on the step dimensions.** The result is that in the lower straight region, the two guideways are again parallel, but at a greater distance. A similar situation would exist in the upper transition zone of the escalator i.e., the distance between the two guideways would swell as the steps emerge at the top.

Step Dimensions:

Till now it was assumed that point A would be the pivot with the roller mounted on it. However, the upper pivot is driven by the chain and would require an axle. The axle mounting needs to be somewhere inside the step frame or the axle might protrude outside. So, the upper pivot is shifted slightly inwards.

It is observed from Figure 5 that as the step proceeds upwards along the track, the distance of point B with the track increases. Also, the angle of AB with track tangent continuously increases. So, from the frame of reference of the track, point B has a circular locus. Hence **a circular profile at BC instead of a straight line is chosen**.

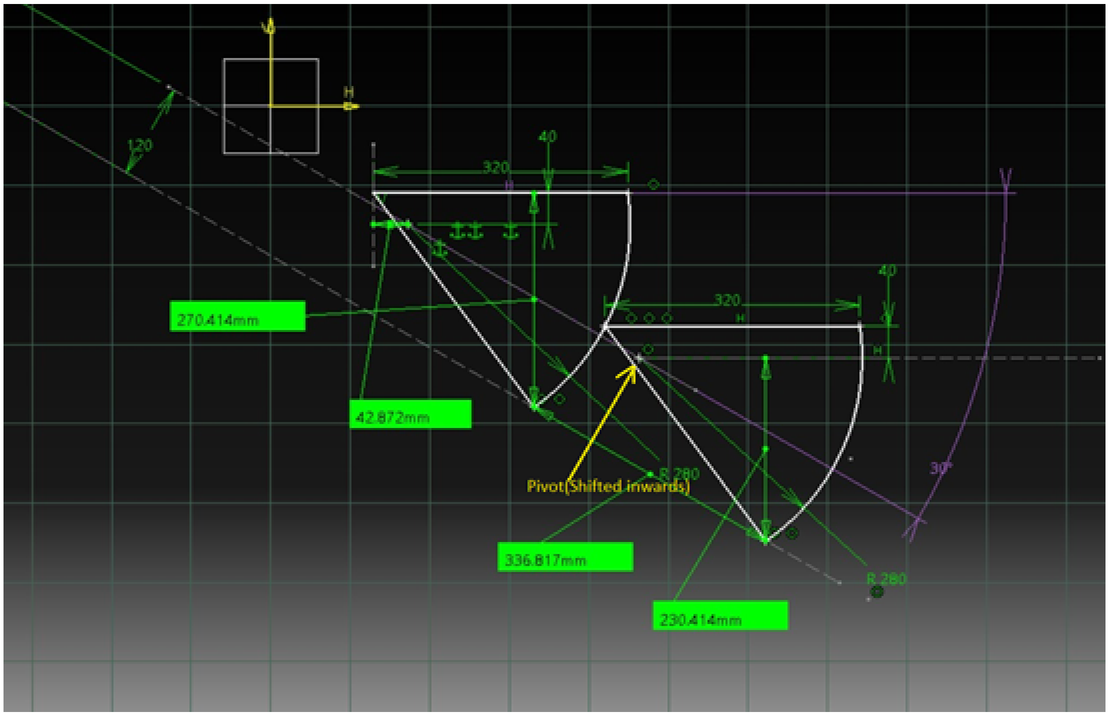


Figure 6

Considering a foot area space of 320 mm, assuming radius of curvature 280 mm and pivot 40mm below the landing, the other dimensions are as shown.

The step pitch was found to be ~337mm. After power calculations and chain selection, chain pitch was 63.5mm. However, 5*63.5 = 317mm and 6*63.5 = 381 mm, the latter being too big. Thus, the step dimensions were altered to suit the drive chain.

Figure 7

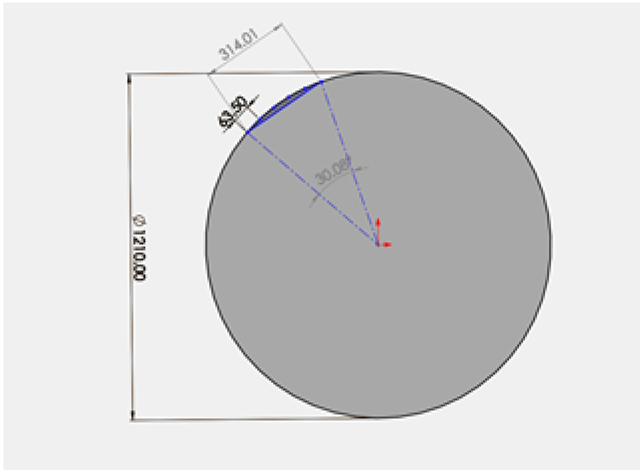


Figure 8

As can be seen, with chain pitch 63.5 mm (design of step chain discussed in Section11) the total chain length would be 63.5*5 = 317.5 mm between the steps. However, while on the sprocket or in transition, the chain would travel a longer path then the step pitch, hence the step pitch should be lesser than the total chain length. Managing a clearance of 3-4 mm, the above result is achieved (Figure 7). New step pitch = 314 mm. (Blue links are the chain links.)

Considering that at a time, around 5-6 steps are moving on the sprocket, we get sprocket diameter **Ds** = 1210mm.

Using this step pitch, the step dimensions are as follows.

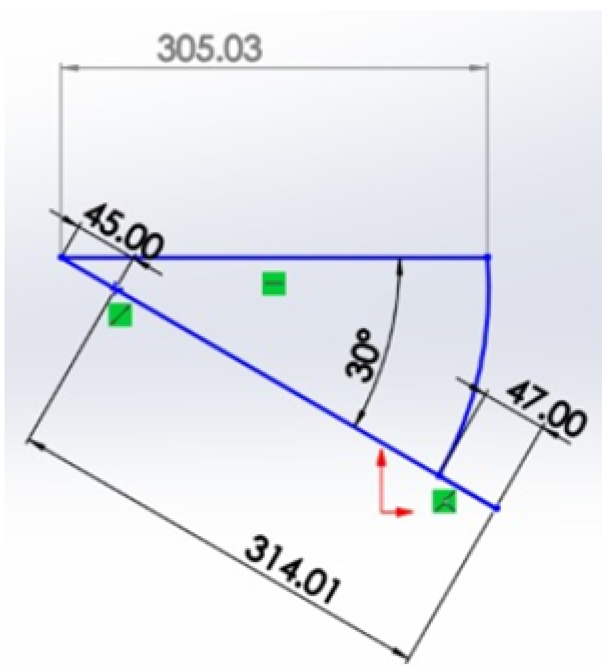


Figure 9

As can be seen in Figure 9, **step landing length is 305.03 mm**. Pivot is 45 mm from landing tip along the incline **(30º)**. A 2mm clearance is left between 2 steps as shown by the dimension 47 mm.

7. CALCULATION OF DESIGN LOAD

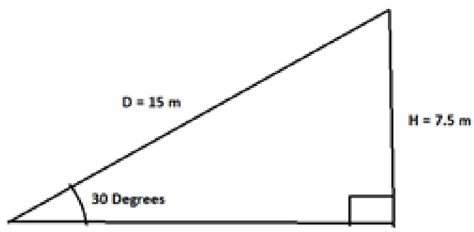


Figure 10

I = 314 mm. (Distance between two steps)

Height = **H =** 7.5 m.

Diagonal **D =** 7.5/sin (30) = 15 m.

Number of steps = 15/ 0.314 = 47.77 steps ~48 steps = **S**

S is the number of steps at a time on the diagonal.

One man per step implies 48 people at a time.

Considering v = **0.65 m/s** as speed,

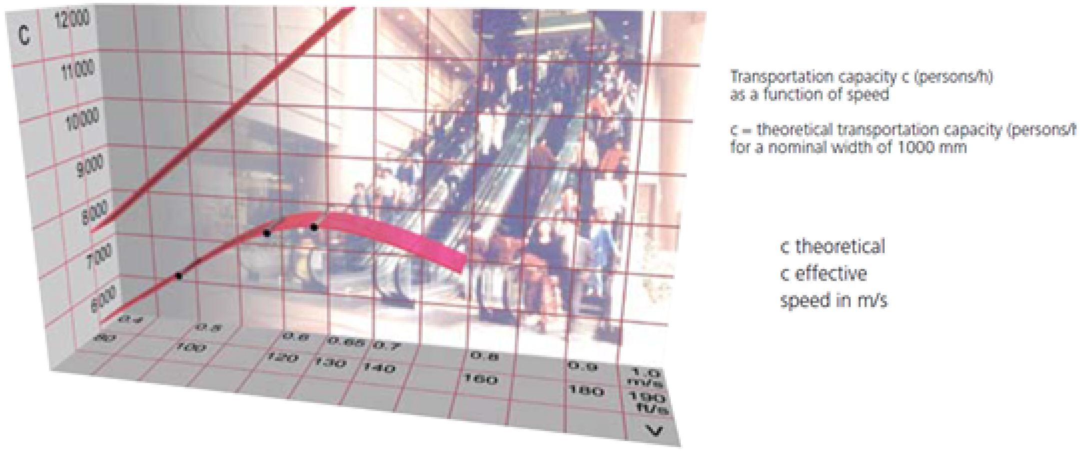


Figure 11

It can be seen in the above graph that while theoretically transport capacity increases with speed, practically it peaks at around *0.65 m/s.* [1 p.13]

Time required = D/v = 23.07 sec = **T.**

Total **Transport =** S*3600/ T = 7490 people/hour.

Thus, from the above calculations the maximum possible capacity of the escalator is 7490 people/hour and that is the **design load** of the escalator. All further calculations are done for this traffic flow.

8. CALCULATION OF POWER AND SPEED REQUIREMENT

Assuming average Indian weight = 65 kg and considering additional 10 kg, let

M = 75 kg.

Pr = Power requirement = (H*M*9.81*S)/T **= 11.48 kW = 15.4 hp.**

Pr is the power required or the rated power of the escalator to be designed.

Ds = 1210 mm. [Section 6. Step Dimensions]

N = Rpm of sprocket = **v*60/(*Ds)** = 10.259 rpm ~**10.3 rpm.**

Thus, **the speed requirement** is 10.3 rpm.

Motor Selection:

Types of Motor:

- **DC Motor**: This type of motor has a high starting torque which is very much required in an escalator but when there is no load and the escalator is started, it is injurious to the escalator.

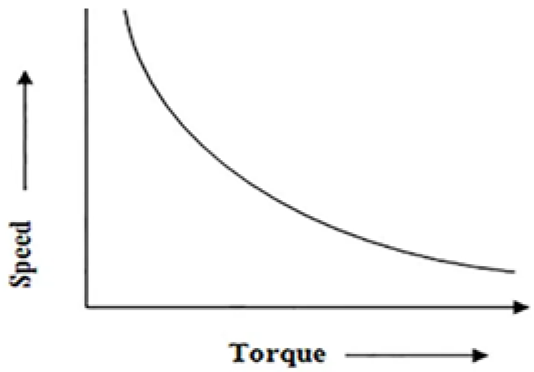


Figure 12

As for its Torque – Current Characteristics,

As the load increases, the armature current increases and the torque increases, Thus during peak working times the wires may burn.

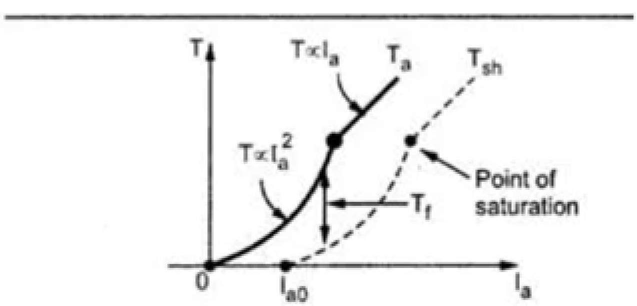


Figure 13

$$\phi \propto I_a$$
$$T_a \propto \phi I_a \propto I_a^2$$

· **AC motor:** Here, Though the starting torque is low at the same frequency, a Variable Frequency Device (VFD) can be used to increase it.

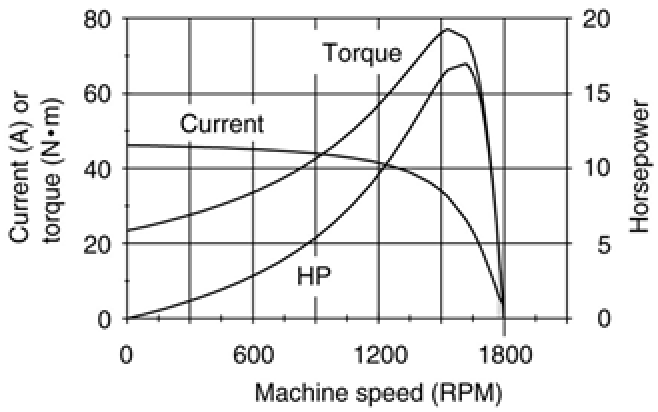


Figure 14

Moreover, in operating range, the slip is nearly 0.

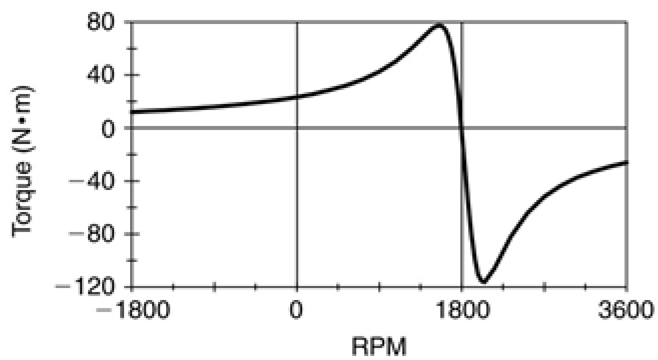


Figure 15

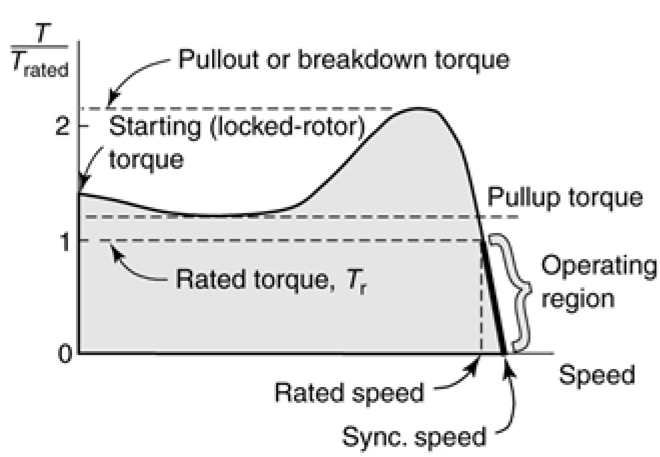
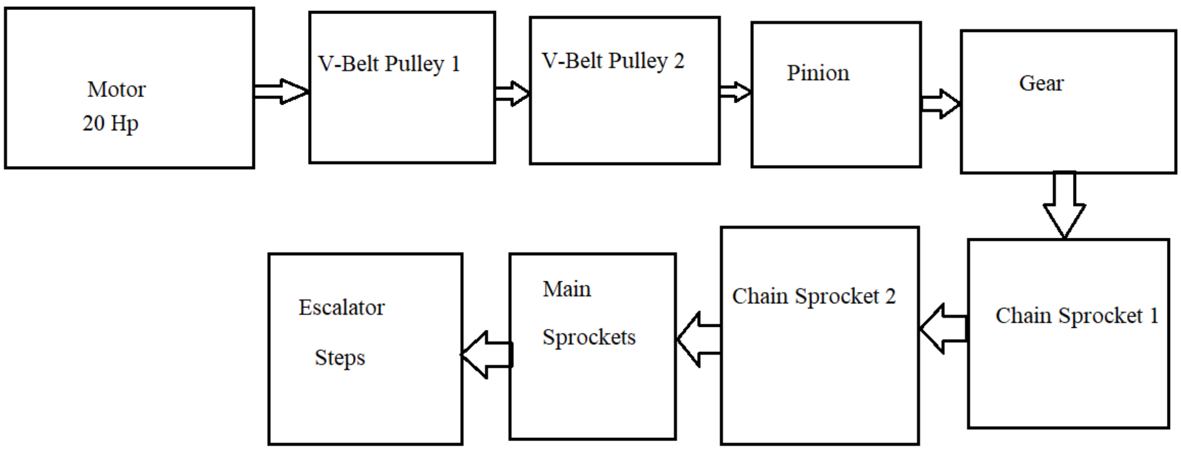


Figure 16

A double cage motor is chosen because of its high starting torque which is suitable for an escalator.

9. POWER TRANSMISSION CHAIN



10. FORCE ANALYSIS

Force in chain :

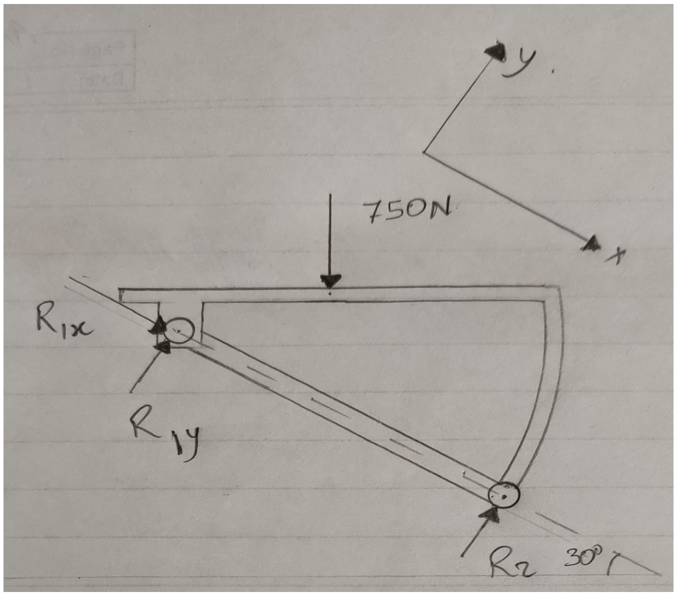
Maximum force in chains = Pr/v = 11.48kW/0.65m/s = 18.5 kN. However, there are 2 chains, therefore the force in one chain = 18.5/2 = 9.25 kN.

Ft = 9.25 kN.

Force in step frame:

Loading on the step frame: 750N

Case 1: When on inclined plane-



Taking X axis along the inclination of the guideway at 30° clockwise from the horizontal and the Y axis perpendicular to it upward.

$$[305/2 - 45\text{Cos}(30°)] * 750\text{Cos}(30°) = (314 - 47)R_2$$

$$R_2 = 276.43\text{N}$$

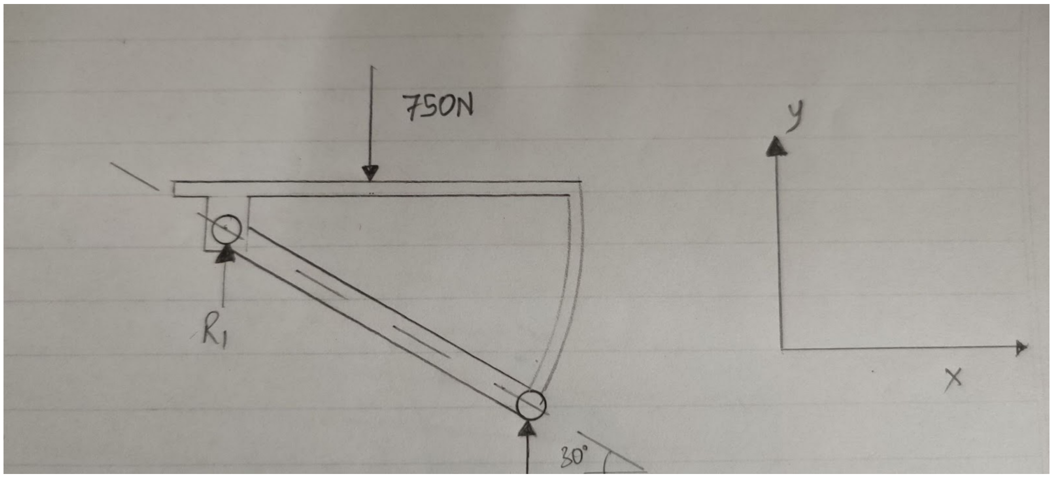
$$R_{1X} + R_2 = 750\text{Cos}(30°)$$

$$R_{1X} = 373.57\text{N}$$

$$R_{1Y} = 750\text{Sin}(30°) = 159.465\text{ N}$$

$$R_1 = (R_{1X}^2 + R_{1Y}^2)^{0.5} = 406.181\text{ N}$$

Case 2: When on horizontal plane-



Taking X axis along the horizontal and the Y axis perpendicular to it upward.

$[305/2 - 45\cos(30°)] * 750 = (314 - 47) R_2 \cos(30°)$

$R_2 = 368.23N$

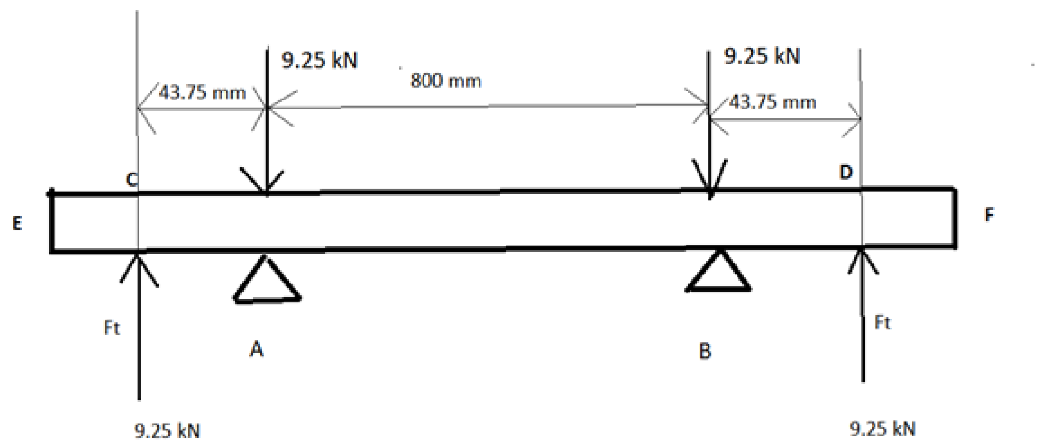
$R_1 + R_2 = 750N$

$R_1 = 381.76$ N

Forces in Axle :

The step length was taken as 800 mm and the axle is extended by 70 mm on both sides. Chain force Ft is acting effectively at 43.75mm of step supports A and B. Axle layout is discussed in section 11 in further detail.

Top View : (Looking in a direction perpendicular to the incline)



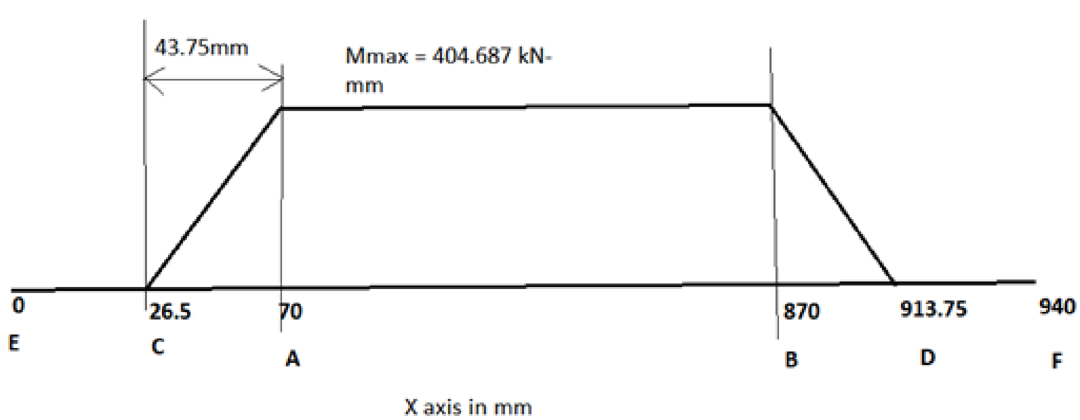
(Not to scale)

Chain force Ft acts in the plane of the incline.

The beam length is = 940 mm.

R1x = 375 N acts at A and B in the downward direction (down the incline). However, the support reaction at A and B have the same line. Hence R1x does not contribute to bending moment.

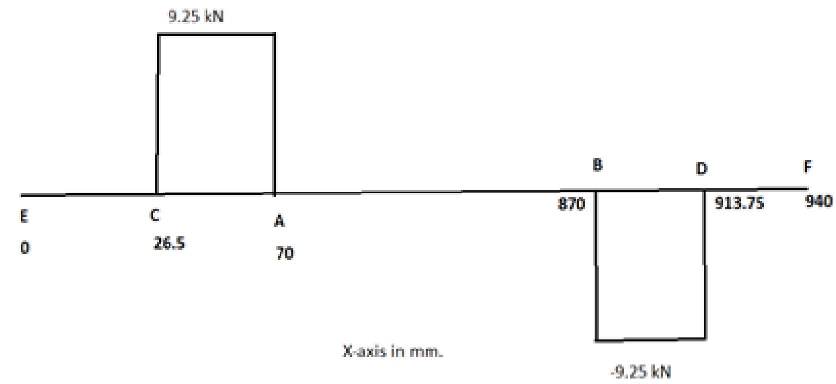
Bending moment diagram:



(Not to scale)

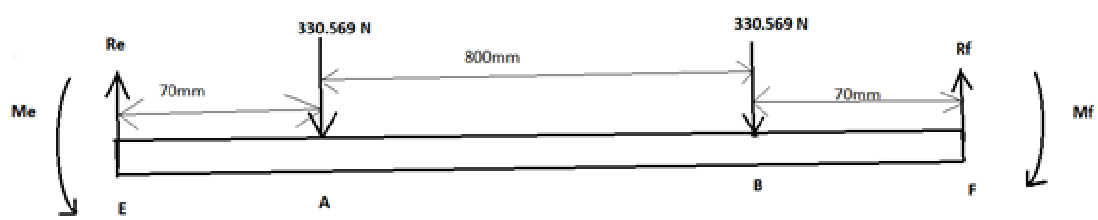
As can be seen the bending moment peaks at 70 mm from the roller end.

Mmax1 = $43.75*9.25*10^3 = 404.687*10^3$ N-mm.



Above shown is the shear force diagram of the beam in top view. Maximum shear = 9.25kN = **Smax.**

Front View : (Looking along the incline)



(Not to scale)

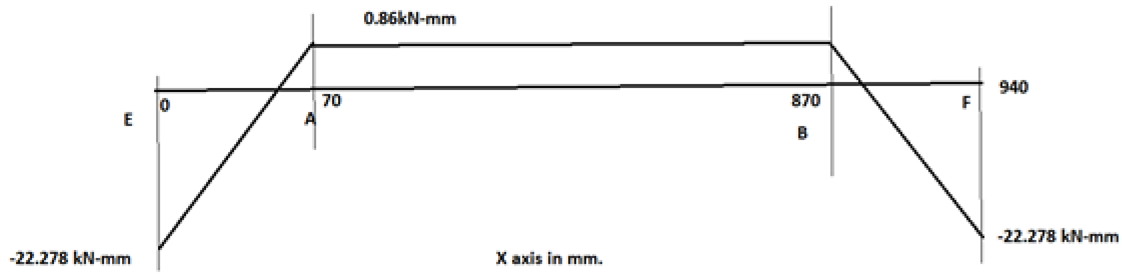
In front view, the rollers at its end act as fixed ends. Hence, a deflection analysis was performed to get the bending moment variation.

Re = Rf = 330.569 N.

Using the differential equation $M = E*I*d^2y/dx^2$ for deflection and integrating twice with boundary conditions as follows : y = 0 & dy/dy = 0 @ x = 0 ; y = 0 & dy/dy = 0 @ x = 940 mm.

We get Me = $22.278*10^3$ N-mm.

Since, Me = Mf, we get Mf = $22.278*10^3$ N-mm.



(Not to scale)

Hogging moment has been taken as negative in both the bending moment diagrams.

|Me| = **Mmax2** = 22.278 kN-mm.

11. DESIGN OF MACHINE ELEMENTS

<u>Step Chain:</u>

Considering that Pr is 15.4 hp and N is 10.3 hp, from Shiwalkar's data book, we get the chain pitch as 63.50 mm = **P**

(#200 chain from Fig 14.2 "Horsepower rating chart – low speeds").

Design power = Pd = Pr*K1, where K1 is the load factor.

From table XIV-2, K1 = 1.4 (moderate shock, 24 hrs-day).

Therefore, Pd = 16.072 kW.

Power Capacity of chain = Pc = (P)^2*[v/104 – (v^1.41/526)*(26 – 25*cos(180/T1))]*Kc,

Where P = pitch(mm), v = speed(m/s), Kc = strand factor = 1 for single strand, T1 = number of teeth on smaller sprocket.

T1 = (6steps)*(5 pitch per step) * 2 = 60 teeth.

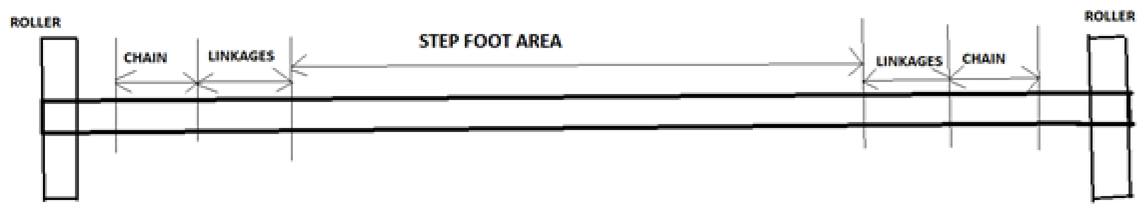
Therefore, Pc = 20.88 kW.

Pc > Pd. Thus, the chain can carry the required load. We will be attaching one of these chains on each side of the step. Thus, if one fails, the other can still function.

This chain requires a special attachment to house the step axle. The ANSI Double Pitch Standard Chain (pitch = 63.5mm) from Tsubaki's attachment chain catalogue [2 p.34] was chosen initially. However, the hole size on the attachment was too small for our application. Therefore, a special attachment for our purpose was designed (the company manufactures special attachments on request).

Step Axle:

Axle Layout :



(Not to scale)

The step length was taken as 800 mm. From Tsubaki's guide, the attachment chain width is 47.5 mm. [2 p.34]. Between the chain and the step, another 20 mm was left for mounting for mounting metal connections between the steps.

The beam length is taken as 800 + 2*(47.5 + 20) ~940 mm.

Material of axle :

C60 steel is chosen for the axle. C60 is for making spindles for machine tools, hardened screws and nuts, couplings, crank shafts, axles and pinions. [PSG p.1.10]

Yield Stress = **Syt** = 42 kgf/mm^2 = 412.02 N/mm^2 .

Diameter of axle :

As can be seen from the force calculation of axle, that Mmax1 is far greater than Mmax2. Also, they occur at different sections. Mmax1 at 70 mm and Mmax2 at 0 mm from the roller. So, the beam was designed for Mmax1.

Considering a fatigue factor of 1.5, [PSG p.7.21],

Mef = Mmax1*1.5 = 607.0305 kN-mm. Using beam theory for round shafts,

Syt = Mef*32/(π*d^3), where d is the diameter. This gives, d = 24.66 mm ~ 25 mm.

Diameter of the axle is 25 mm.

Checking for shear strength:

Shear strength of C60 = Ssyt = 430 Mpa.

Shear Strength = Sc = (π/4)*(d^2)*Ssyt = 211.075 kN . Smax = 9.25 kN. Sc >>Smax. The axle is safe in shear.

Chain Attachment:

As mentioned previously, ANSI Double Pitch Standard Chain (pitch = 63.5mm) from Tsubaki's attachment chain S type roller catalogue [2 p.34] was chosen initially.

However, the hole size (14.3 mm) is too small for the axle. Therefore, keeping the dimensions T, H, P, and N as given in the catalogue, the C1 and XS dimensions are altered to fit our application. (Tsubaki provides custom attachments on request.) The main chain body is also kept the same.

T = 4.8 mm. H = 28.6mm. P =63.5 mm. N = 47.60mm.

Material of chain :

Stainless Steel 304.

Yield strength = **Syt**= 215 Mpa.

Using Von Mises criterion, shear strength = **K** = Syt/√3 = 124.130 Mpa.

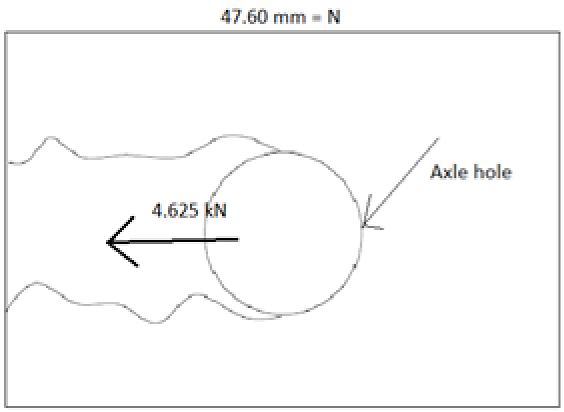
Compressive strength = **Sc** = 250 Mpa.

Dimensions :

Checking for shear due to the axle at the hole :

F = Ft/2 = 4.625 kN = Load. (Two plates will share the load of Ft)

Shear Strength = 2 * (N – d/2)*K*T = 13.451 kN > F. (Safe).

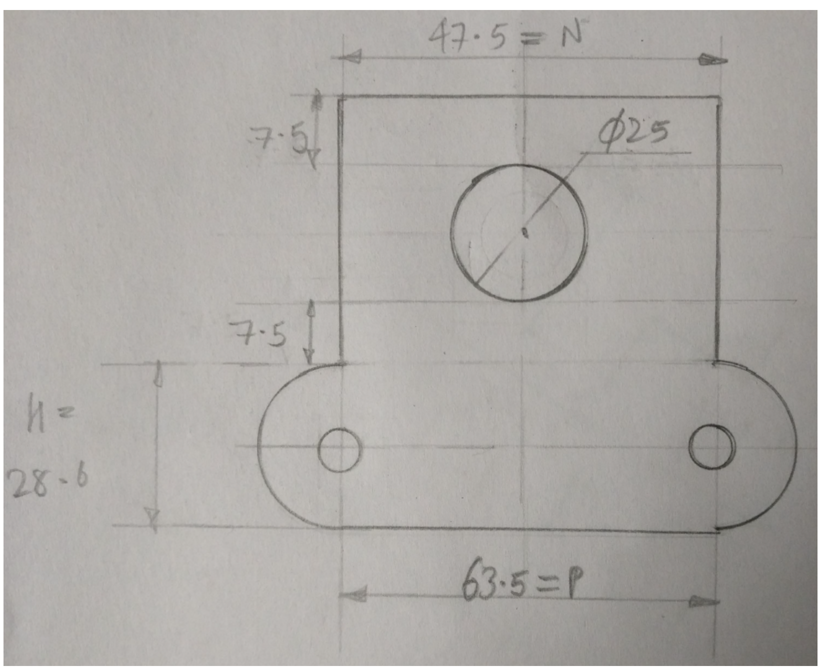


Checking for compressive strength at the hole:

Compressive Strength = d*T*Sc = 30kN > F. (Safe)

The plate is not subjected to tearing stress. Thus, with the above two checks, it is assured that the plate is safe.

Following are the final dimensions of the chain plate :



Step Frame:

Note: Assumption that a deflection less than 1 mm is considered safe.

Link 1: The dimensions of this beam 305mm * 15mm * 15mm.

Maximum force applied= 750N.

Bending Stress($\boldsymbol{\sigma}$)= $WL^2*y/2*I$

=750*305^2*7.5/2*(15^4)/12

[I=bd^3/12 for rectangular cross section]

=62 kN/mm^2

Yield Stress = 215 Mpa.

Deflection(δ)=$WL^3/48EI$

=$750*305^3/48*200*10^3*(15^4/12)$

=0.525mm

Link 2: The dimensions of this beam is assumed to be 746m * 35mm * 35mm.

Maximum force applied = 1500N.

Bending Stress = $WL^2*y/2*I$

= 67.172 kN/mm^2

Deflection = $5WL^3/384EI$

= 0.55mm

Link 3: The dimension of this link: cross section area is 8.5 mm each side from the arc of radius 260.72mm.

Maximum force applied = 1500N.

Bending Stress = M*y/I

$=330 \cos 30° * 167.25*12/17^4$

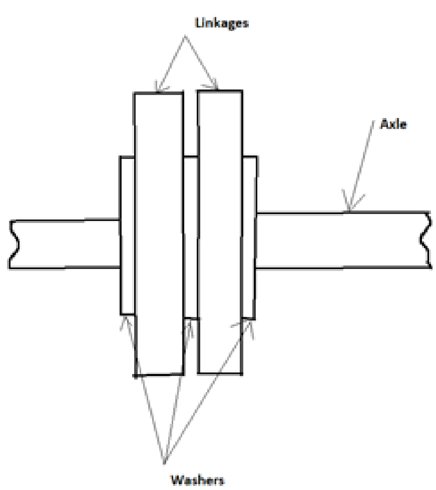
Deflection= $ML^2/(2EI)$

= 0.48mm

Step Linkages:

A metal link is provided between two steps to maintain the distance of 314 mm. This linkage is mounted on the axle. On each axle on each side, two metal linkages are mounted – one connecting with the forward step axle, other connecting with the behind step axle.

The lateral movement of the linkages on the axle are inhibited by the chain attachment plate on one side and the step frame on the other. (Both are not shown in the figure below). Metal washers are provided to mitigate friction at those locations. Bushings are provided in the eyes of the metal links.



(Linkage assembly)

Material of linkages:

The linkages do not sustain any appreciable forces; hence a thin cross section was adopted. **C25 Mn75 steel** was chosen for the linkages. It is general purpose steel for low stress components. [PSG p.1.10].

Dimensions of linkage:

Inside diameter = ID = 28mm.

Outside Diameter = OD = 45 mm.

Thickness of eye = 5mm.

Length between eye centers = 314 mm.

Cross section in straight region = 10mm*5mm.

Bushings:

Sintered bronze bushings are selected to allow relative motion between the linkages and the axle which is fixed. Sintered bronze eliminates the requirement of lubrication. A bushing would absorb any shocks in the system and is cheap compared to a bearing.

Bore diameter = 25mm.

Outside diameter = 28mm.

Thickness = 5 mm.

Washers: [PSG p.5.89 Machined Washer]

Machined washers are chosen over others due to their least thickness. In axle design, 20 mm space was left for the linkage assembly, exceeding which would impose higher bending stresses on the axle.

Nominal size = M24.

Inside diameter(d) = 26 mm.

Outside Diameter(D) = 45 mm.

Thickness(s) = 4mm.

Length of the assembly:

Length along the axle = 2*5+3*4 = 22 mm. This is slightly greater than 20mm, and its effect on axle diameter is insignificant. Performing all calculations again, gives axle diameter = 25.03mm ~25mm. (No change).

Main Rollers:

Roller 2C 75*24 with bearing **6005 2RS** was chosen. [Faigle p.19] the RS stands for rubber seal. The 2RS stands for **2 rubber seals**. One on each side of the bearing.

The bearing is a deep groove ball bearing. [Faigle p.15 see figure]

Fe = Equivalent load on the bearing ~ 330N. (See Force in Frame) Dynamic capacity (C) for 6005 bearing = 780 kgf ~7800N. [PSG p.4.12]

Therefore, life of the bearing = (1million cycles) *$(7800/330)^3$ = 13205.108 million cycles. (Practically infinite life. Safe.) (1 million cycles correspond to dynamic capacity C.)

Quantity of Roller 2C 75*24 per step = **4**.

Belt Drive:

Belt Specifications :

As discussed earlier, V belt drive has been chosen as the first stage.

Velocity ratio at the belt drive is 5 from 1500 rpm(**N1**) (motor speed) to 300 rpm(**N2**). **VR** = 5.

Motor power = 20 hp = 14.914 kW.

Therefore, Section C V-belt is chosen. [PSG p.7.58]. Nominal Thickness T = 14mm.

Minimum pulley diameter = D1 = 200mm from the table. **D1** = 225 mm taken.

Therefore, bigger diameter **D2** = VR * D1 = 1125 mm.

Minimum center distance = Cmin = 0.55(D1 + D2) + T = 756.5 mm. [PSG p. 7.61].

Maximum center distance = Cmax = 2*(D1 + D2) = 2700mm.

Taking center distance **C** = 1000mm,

A1 = contact angle at smaller pulley = π – (D2-D1)/C = 2.241^C = 128.4^o

Vp = Pitch line velocity = π*D1*N1/(60*1000) = 17.671 m/s.

Now, D2/D1 = 5. Therefore, small diameter factor **Fb** = 1.14. [PSG p.62]. Therefore, **De =** Effective diameter = Fb* D1 = 256.5 mm. Thus **Pr =** Rated power = 9.19kW. [PSG p.7.65 for belt speed 18m/s]

Arc of contact factor = **Ka =** 0.85. [PSG p.7.68]

Service factor = **Ks =** 1.3 [PSG p.7.69 for medium duty]

We have, Pitch length = Lp **=** $2*C + \pi*(D1+D2)/2 + (D2-D1)^2/4*C$ = 4323.075 mm.

Choosing standard **Lp** = 4450 mm [PSG p.7.60], we get inner length **Li** from table as 4394mm. Therefore, the belt chosen is the **C4394** V belt. **Kl =** Length correction factor = 1.04.

Actual power capacity of belt = **Pact =** Pr*Ka*Kl/Ks = **6.2491 kW**.

New centre distance **C =** 1070.09 mm.

Total number of belts = 14.914/Pact = 2.38 ~3.

C4394 belt. 3 in quantity. Smaller pulley diameter = 225mm, larger pulley diameter =1125mm.

Pulley Specifications :

C section belt with D1 = 225 mm has been chosen. Therefore,

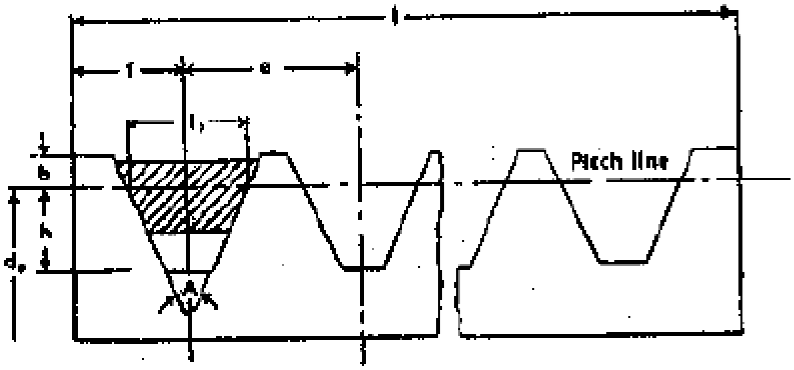
Pulley groove angle = A = 36º with 0.5º tolerance in both directions.

h = 14.3mm.

e = 25.5mm with 0.5mm tolerance in both directions.

f = 17mm. +2- and -1-mm tolerance.

b = 19 mm.



[PSG p.7.70 Dimensions of Standard V-grooved Pulleys]

Helical Gear Pair:

A helical gear of velocity ratio 8 is to be designed for a speed reduction of pinion speed 300 rpm to a gear speed of 37.5 rpm.

Assuming $\Psi = 30°$,$\Phi = 30°$

$N_g = 37.5$ rpm $\quad N_p = 300$ rpm $\quad P_r = 12.739$kW

$P_d = P_r * K_I * K_W$

Taking $K_{I = 1.8}$ [For Continuous duty medium shock]

K_W=1.15 [For Continuous lubrication oil bath]

$P_d = 26.371$ kW

$D_p = m_n * t/ \text{Cos}(\Psi) = 28.97m_n$

$V_p = \pi D_p N_p/60*10^3 = 0.44m_n$

$F_t = P_d / V_p = 53.33/m_n$

$F_b = S_o*y*b*m_n*C_v$

S_o for SAE 1030 = 140 MPa

$y = 0.485 - 2.87/t_f = 0.410$ [Assuming no of teeth in pinion as t=25 for good balance between strength and wear and $t_f = t/\cos^3(\Psi)$]

$b = 12m_n/\text{Sin}(\Psi)$

$C_v = 0.5$

$F_b = 688.8m_n^2$

For $F_b = F_t$

We get $m_n = 4.43 \approx 5$[next standard]

$(S_o y)_{gear} > (S_o y)_{pinion}$

Thus, The gear is designed with preference to the pinion.

$D_p = 144.35$ mm

$V_p = 2.2$ m/s

$C_v = 0.67$

$F_t = 11.986$ kN

$F_t = F_b$

$b = 62.33 \text{ mm} > b_{min} = 4.5m_n / \text{Sin}(\Psi)$

$F_D = F_t + F_i$

$F_i = 21V_p(Ceb + F_t)\text{ Cos}(\Psi) / (21V_p + (Ceb + F_t)^{0.5}) = 7.376\text{kN}$

$F_D = 19.362\text{kN}$

But, $F_D > F_b$(without C_v) $\therefore$ Unsafe.

So, increase b to $12m_n/\text{Sin}(\Psi) = 82$ mm

So, new F_b(without C_v) = 24.423 kN

$\therefore$ $F_D < F_b$(without C_v) $\therefore$ Safe

$F_w = kbDQ/ \text{Cos}^2(\Psi)$

$Q = 2t_g/(t_g + t_p) = 1.7777$

When we put $F_w = F_D$

We get,

k = 0.267 = 0.288 [next standard]

Thus, the hardness of both the pinion and the gear has a hardness of 150 BHN.

Specifications for pinion

m_n = 5 mm

D_p = 144.35mm

b = 120 mm

Specifications for gear

m_n = 5 mm

D_p = 721.75mm

b = 120 mm

Chain Drive:

Chain Drive is chosen as the 3rd stage of velocity reduction with a velocity ratio of 3.64.

2nd stage shaft angular velocity(N1)= 37.5 rpm

3rd stage shaft angular velocity(N2)= 10.3 rpm

Considering efficiency as 98%, Pr= 20*0.95*0.98*0.916 = 17 hp

For given N1 and Pr, we assume a tentative chain from Shiwalkar's Data Book (pg. 153)

For #160 chain, no. of teeth= 21, Pitch= 2 inch= 50.8 mm

From PSG data book (pg. 7.73) for R160 chain,

Sprocket pinion teeth (T1)= 29

T2= 1.875*29= 54.375~55

(from pg. 7.77 PSG) FoS=7 for N1=37.5 rpm

(from pg. 7.78 PSG) Total load= (Pt+Pc+Ps)= (Ft+Fc+Fs)*k1*k2*k3*k4*k5*k6

Pitch Dia D1(small sprocket)= P/sin(180/T1)= 50.8/sin(180/29)= 469.93 mm

Pitch Dia D2(sprocket wheel)= P/sin(180/T2)= 50.8/sin(180/55)= 890 mm

Vp= pi*D1*N1/60= 0.922 m/s

Ft= Pr/Vp= 13.047/0.922= 14150.75 N

Fc= w(Vp)^2/g= 9.9*9.81*(0.922)^2/9.81= 18.416 N

Fs= k*w*a

(here, a is the centre distance= 30p and k=1 from pg. 7.78 PSG)

Fs= 1*99*30*50.8/1000= 150 N

Taking from pg.7.76- 7.77 PSG db,

k1=1 (constant load)

k2=1.25 (fixed centre distance)

k3= 1 (c=30-50P)

k4= 1 (inclination of line joining centres is 45 deg<60 deg)

k5= 1 (drop lubrication)

k6= 1.25 (16 hrs/day)

Total load= (14150.75+8.416+150)(1*1.25*1*1*1*1.25)= 22358.072 N

FoS (calc)= Br. load/Total load= 226800/22358.072= 10.144> min FoS so the chain is **safe** from breaking failure.

Min FoS= 7 (from PSG db)

Bearing pressure

For R160 chain, Area= 6.42 cm^2

Load on pin= Ft*k1*k2*k3*k4*k5*k6= 22110.56 N

Bearing Pressure= 221100.5/642= 34.44 N/mm^2

Allowable bearing stress = 3.5 kgf/mm^2 for N1=37.5 rpm (pg. 7.77 PSG db)

Bearing Pressure< Allowable Bearing Stress so the chain is **safe** from bearing stress.

Length of Chain (Lp)= 2Cp + (T1+T2)/2 + (T2-T1)^2/Cp*30= 102.75~ 104 pitches

For finding actual Cp substitute the new Lp in above equation,

104= 2Cp+ 42+ 676/Cp*30

Actual Cp= 30.632*p

So, actual C= 1.556 m

Sprockets in Chain Drive:

From Shiwalkar's data book pg.154

For smaller sprocket,

Sprocket teeth width T0= 0.58P-0.15 mm= 29.314 mm

D(outside)= P[0.6+cot{180/T1}]= 497.82 mm

D(root)=Dp-0.625*p= 469.93-0.625*50.8= 438.25 mm

For larger sprocket,

T0= 0.58P-0.15= 20.314 mm

D(outside)= 50.8[0.6+cot{180/55}]= 918.9 mm

D(root)= 890-0.625*p= 858.25 mm

Shaft- Stage 3:

Length of the shaft between bearings ~ 950 mm.

Force in Stage 3 chain = 14.150 kN. Diameter of stage 3 sprocket = 890 mm.

Tc = Torque to stage 3 chain = (14.150kN)*(890/2)*K1 = 6611.5875 kN-mm.

K1 = load factor [PSG p.7.78 Position of Chain Drive]

Tm = Torque by main sprocket = 5596.25 kN-mm.

Tmax = 2*Tm = 11192.5 kN-mm.

M = Bending moment = 4688 kN-mm.

Teff = ((Kt*Tmax)^2 + (Kb*M)^2)^0.5 = 13218.210 kN-mm.

Kb =1.5, Kt =1.

KT = Stress concentration factor = 1.5.

Teff = 13218.210*KT = 19827.315 kN-mm

Material of shaft :

C50 [PSG p. 1.10]

Shear Strength = **S =** 400 Mpa.

Diameter of Shaft:

S = 16*Teff/(pi*D^3)

This gives a diameter of 63.201 mm. Taking standard diameter as 70 mm. [PSG p. 7.25]

Ds3 = 70 mm.

Shaft Stage 2:

Material of shaft :

C50 [PSG p. 1.10]

Shear Strength = **S =** 400 Mpa.

Diameter of Shaft:

Tmax = 6903.936 kN-mm.

Kt = Stress Concentration Factor = 1.5. **Teff =** 10355.904 kN-mm.

0.4 = Teff*16/(pi*Ds2^3)

Ds2 = 50.8978 mm. Taking standard shaft, **Ds2 =** 55 mm.

<u>Shaft Stage 1:</u>

Material of shaft :

C50 [PSG p. 1.10]

Shear Strength = **S =** 400 Mpa.

Diameter of Shaft:

Teff ~ 1000 kN-mm.

0.4 = Teff*16/(pi*Ds1^3)

Ds1 = 23.3508 mm. Taking standard shaft, **Ds1 =** 25 mm.

12. SUB-ASSEMBLY DRAWINGS

<u>Parts List:</u>

Sr. No.	Component Name	Component Number	Material	Quantity	Drawing Number/ Specification
1	Axle	ESC01001	C60	120	3
2	Guideway	ESC01002	-	4	-
3	Metal Linkages	ESC01003	C65	238	2
4	Washer	ESC01004	-	720	Machined Washer Nominal SizeM24
5	Bushings	ESC01005	Sintered Bronze	480	4
6	Circlip	ESC01006	-	240	LightA 25 IS : 3075-1965

7	Shaft Collar	ESC01007	Structural Steel St 37	240	Light Series Type A Bore25H8
8	Bolt	ESC01008	-	240	Hexagonal Bolt Precision P Grade M10
9	Nut	ESC01009	-	240	Hexagonal Nut Precision P Grade M10
10	Roller wheel	ESC01010	PAS-PU 54D-H blue	240	Faigle. 044.01473 Roller 2C 75x24
11	Roller Wheel	ESC01011	PAS-PU 90A-H blue	240	Faigle 044.02488 Roller 1C 38x13
12	Step Chain	ESC01012	Stainless Steel 307	2 (595 links each)	Tsubaki Chain No. RF2100
13	Special Attachment	ESC01013	Stainless Steel 307	240	1

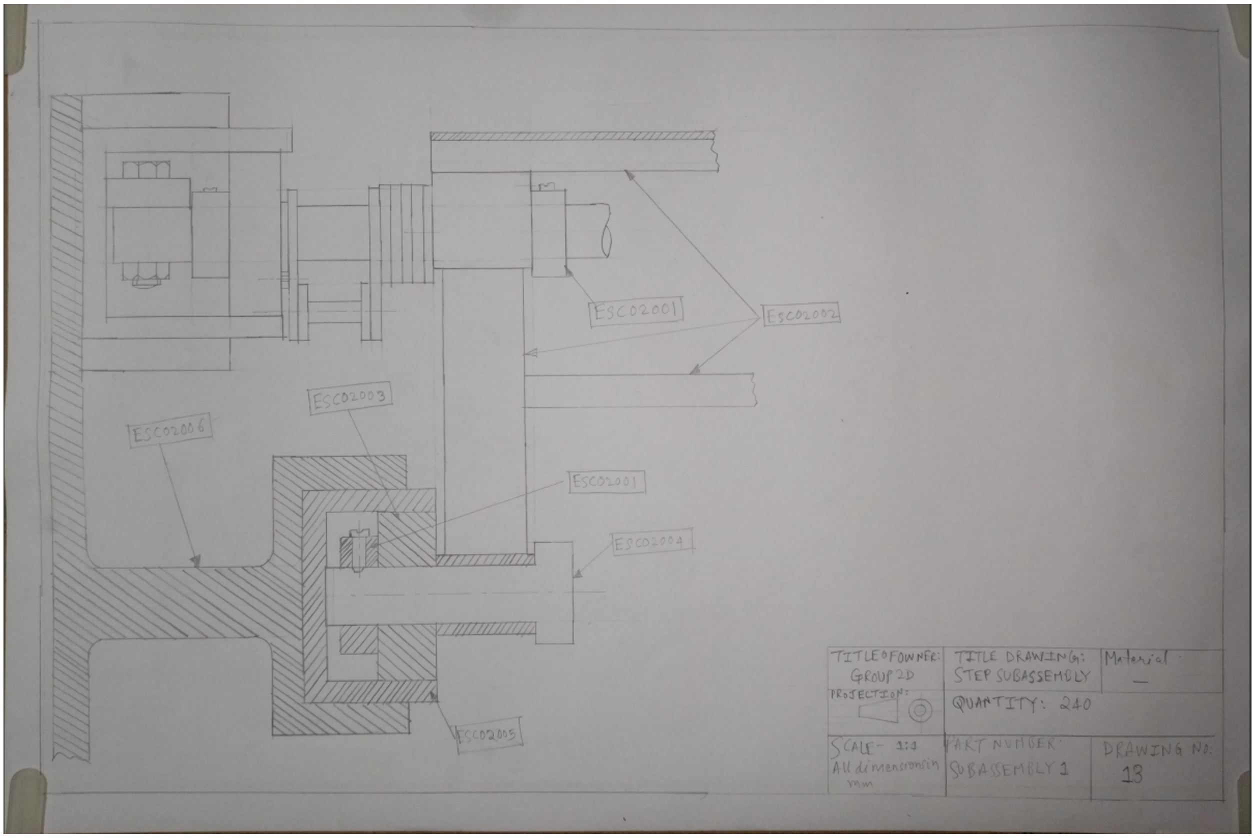
14	Shaft Collar	ESC02001	Structural Steel St 37	480	Light Series Type A Bore25H8
15	Step Frame	ESC02002	stainless steel ASTM A240 304H	120	5
16	Roller Wheel	ESC02003	PAS-PU 54D-H blue	240	Faigle. 044.01473 Roller 2C 75x24

17	Rod	ESC02004	C60	240	6
18	Lower Guideway	ESC02005	-	2	-
19	Guideway Support Structure	ESC02006	-	-	-
20	AC Induction Motor(Double Cage), 1500 Rpm, 20 Hp	ESC03001	-	1	-
21	V-Belt Drive Pulley1	ESC03002	CI	1	7
22	V-Belt Drive Pulley2	ESC03003	CI	1	-
23	V Belt	ESC03004	-	3	C4394
24	Helical Gear (Pinion)	ESC03005	SAE1030, Heat Treated	1	8
25	Helical Gear	ESC03006	SAE1030, Heat Treated	1	-
26	Sprocket 1	ESC03007	-	1	9
27	Sprocket 2	ESC03008	-	1	Bore dia. 70mm Pitch dia. 890mm Pitch 50.8mm
28	Sprocket 3	ESC03009	-	2	Bore dia. 70mm Pitch dia. 1210mm Pitch 63.5mm
29	Shaft 1	ESC03010	C50	1	10
30	Shaft 2	ESC03011	C50	1	11

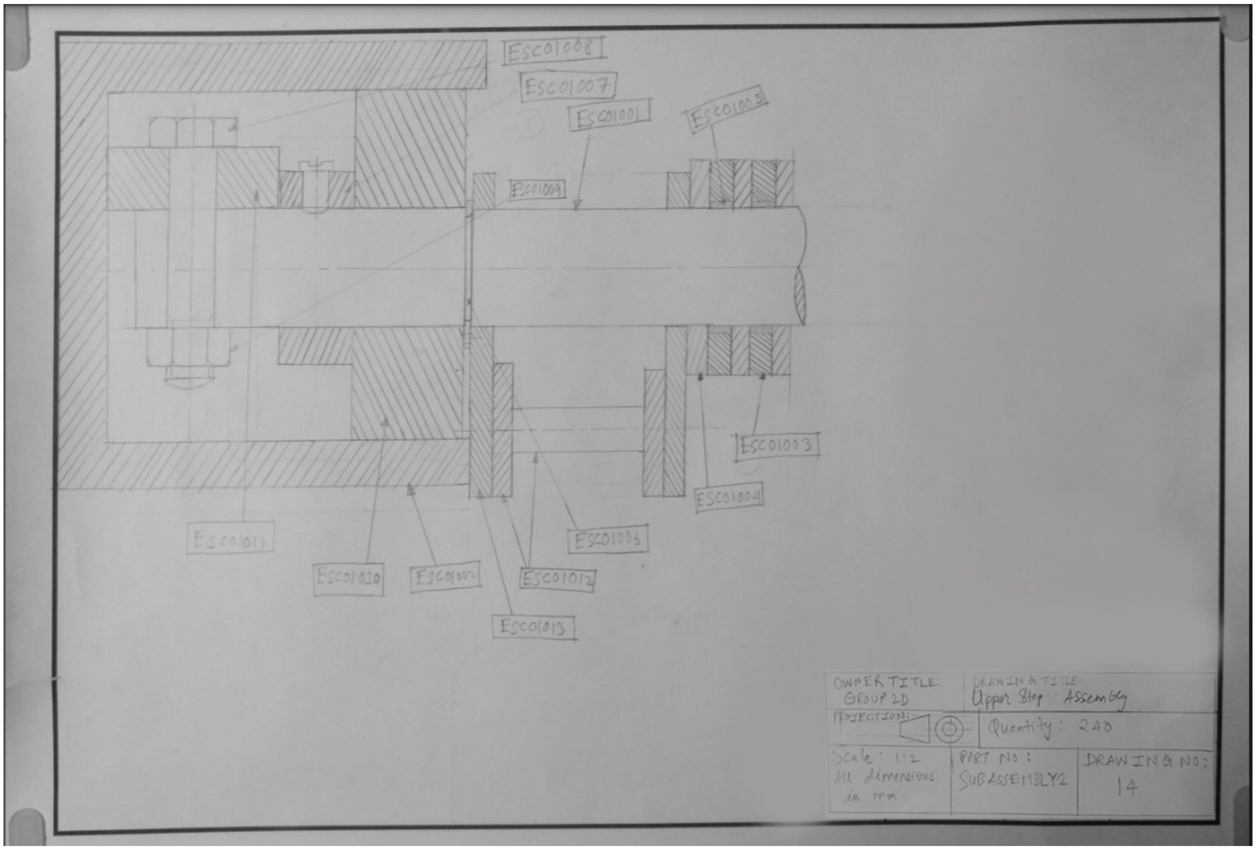
31	Shaft 3	ESC03012	C50	1	12
32	Key	ESC03013	C50	2	Gib Head Key 25 IS 2293-1963
33	Key	ESC03014	C50	2	Gib Head Key 55 IS 2293-1963
34	Key	ESC03015	C50	3	Gib Head Key 70 IS 2293-1963
35	Shaft collar for shaft 1	ESC03016	Structural Steel St 37	2	Light Series Type A Bore25H8
36	Shaft collar for shaft 2	ESC03017	Structural Steel St 37	2	Light Series Type A Bore56H8
37	Shaft collar for shaft 3	ESC03018	Structural Steel St 37	3	Light Series Type A Bore70H8
38	Bearing for shaft 1	ESC03019	-	2	7305 B
39	Bearing for shaft 2	ESC03020	-	2	7311 B
40	Bearing for shaft 3	ESC03021	-	2	NU 2314
41	Chain	ESC03022	-	1	R160

Subassembly Drawings :

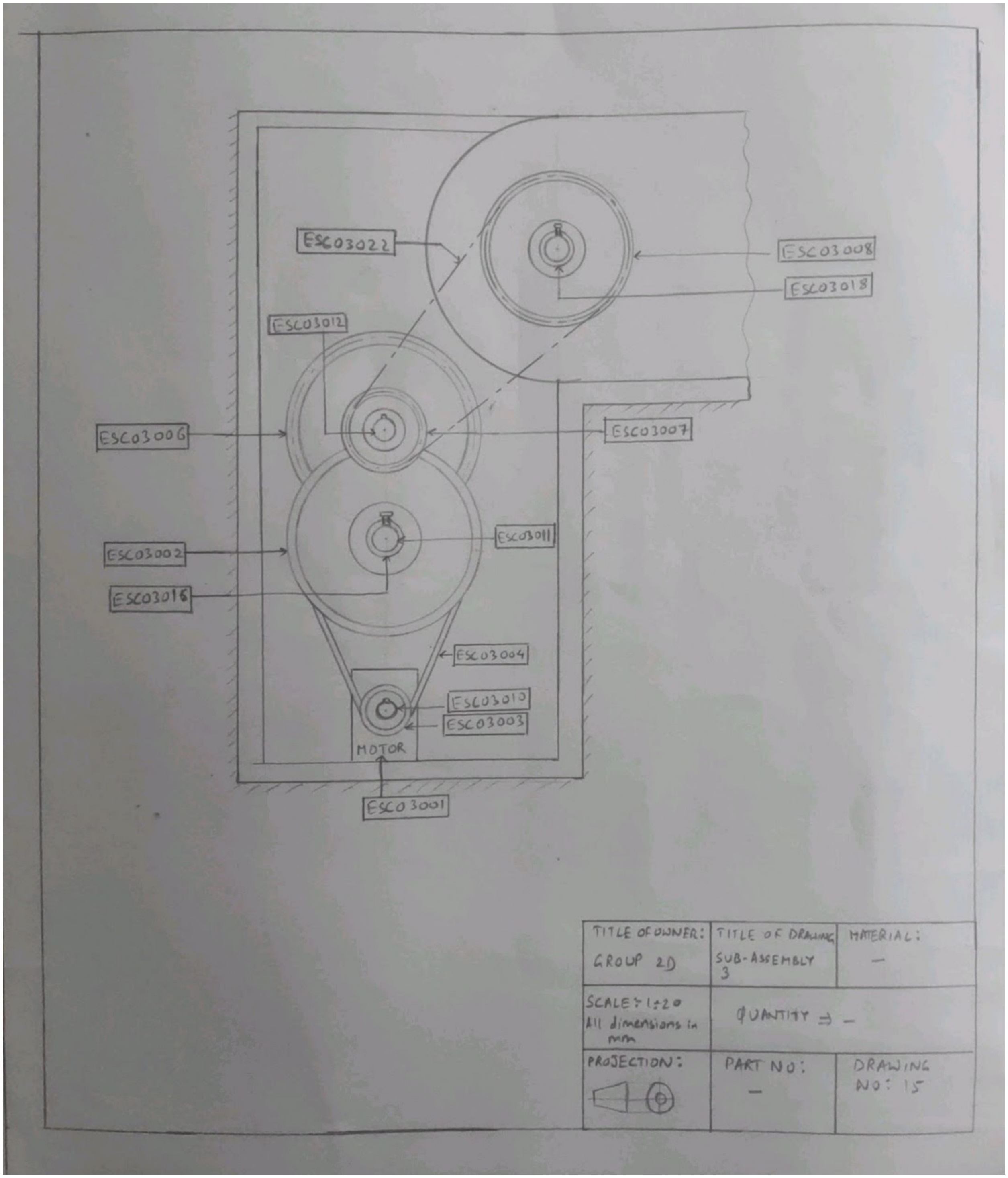
Step Assembly(Sub-Assembly 01, Drawing 13)



Upper Pivot Step Assembly (Sub-Assembly 02, Drawing 14)

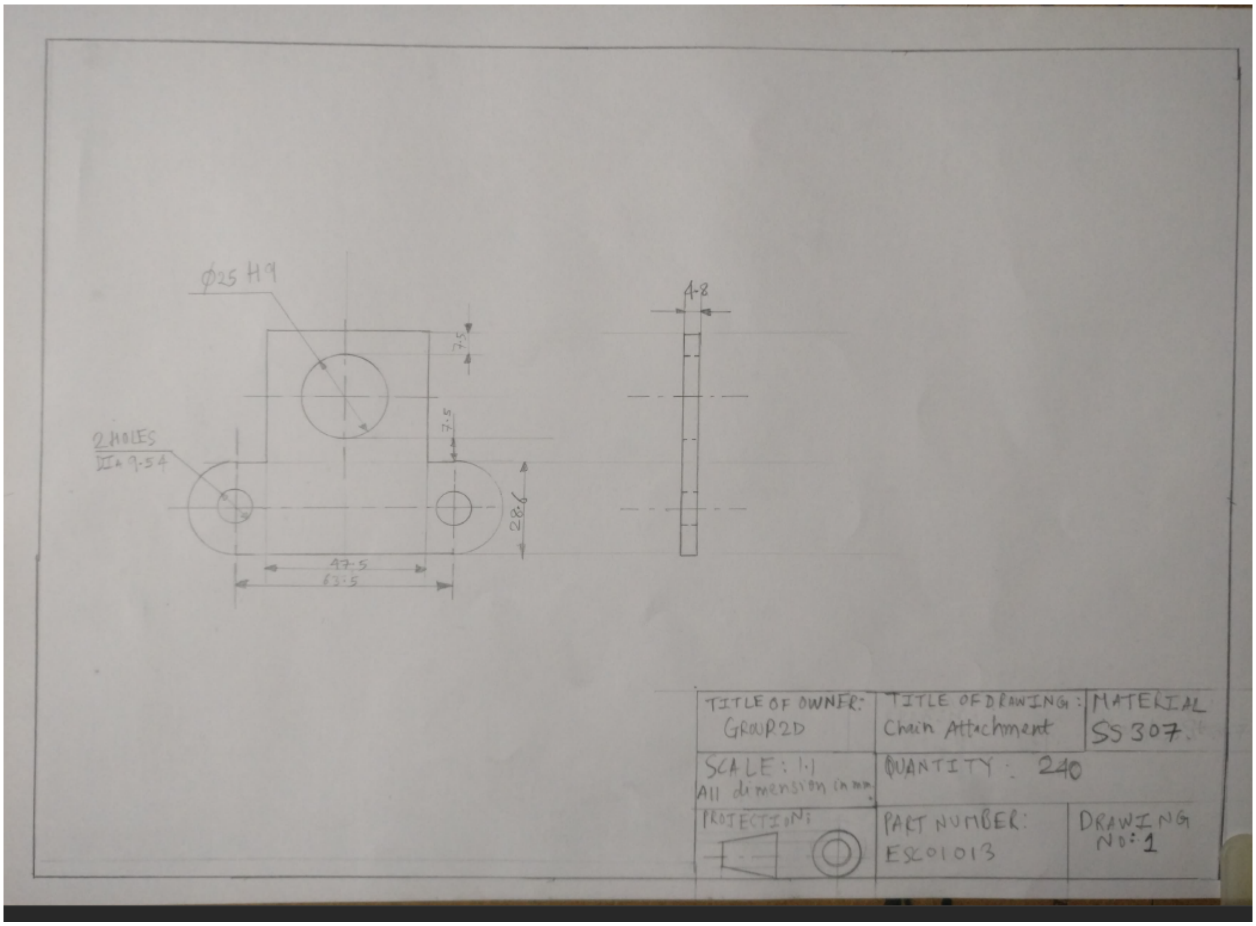


Transmission Assembly(Sub-Assembly 03,Drawing 15)

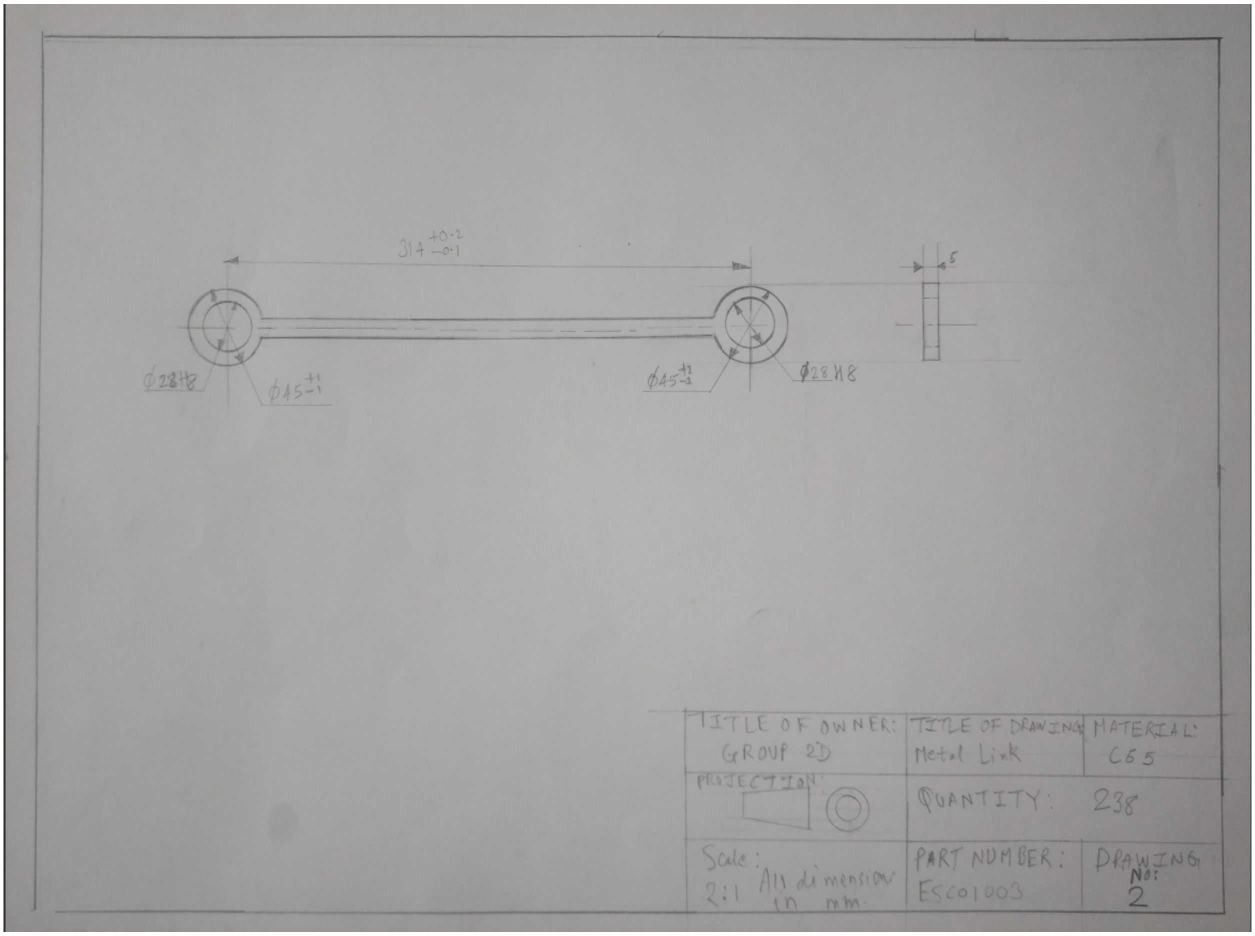


13. COMPONENT DRAWINGS

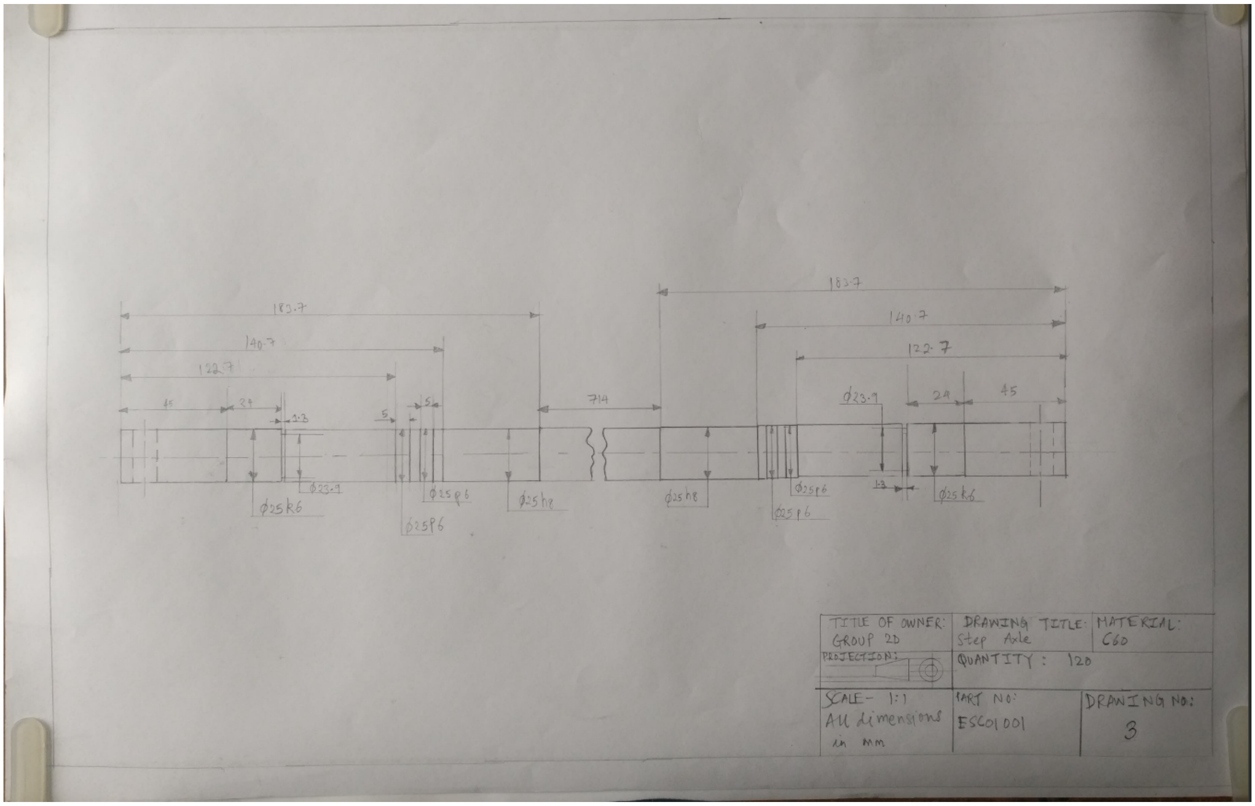
Step Attachment(ESC01013, Drawing 1)



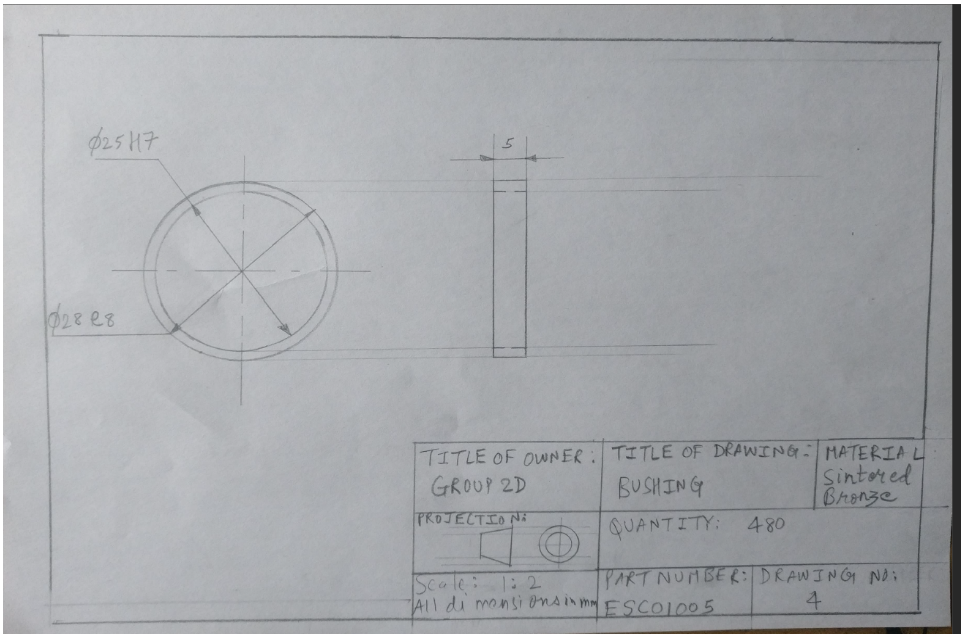
Metal Linkage (ESC01003, Drawing2):



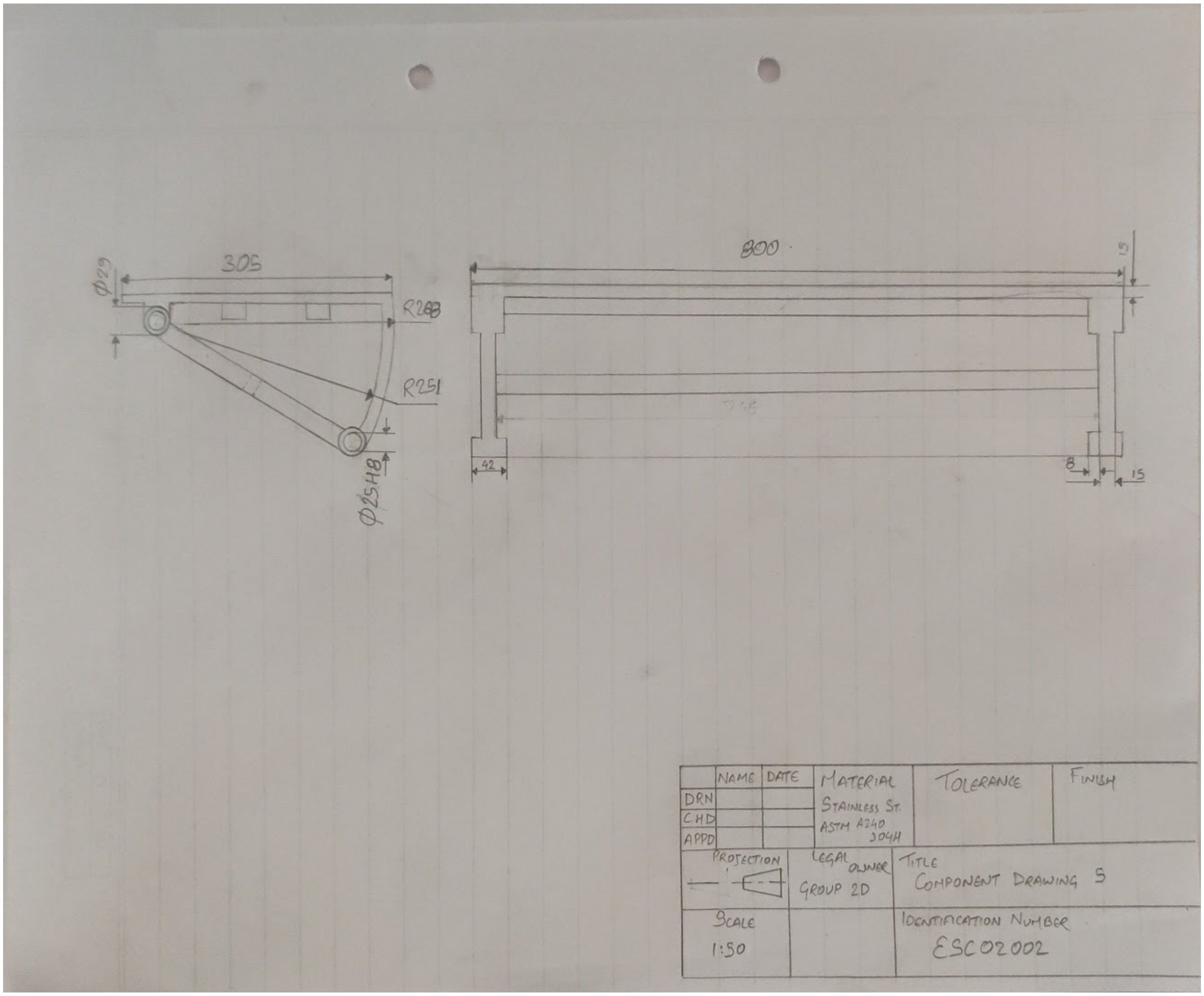
Step Axle :(ESC01001, Drawing 3)



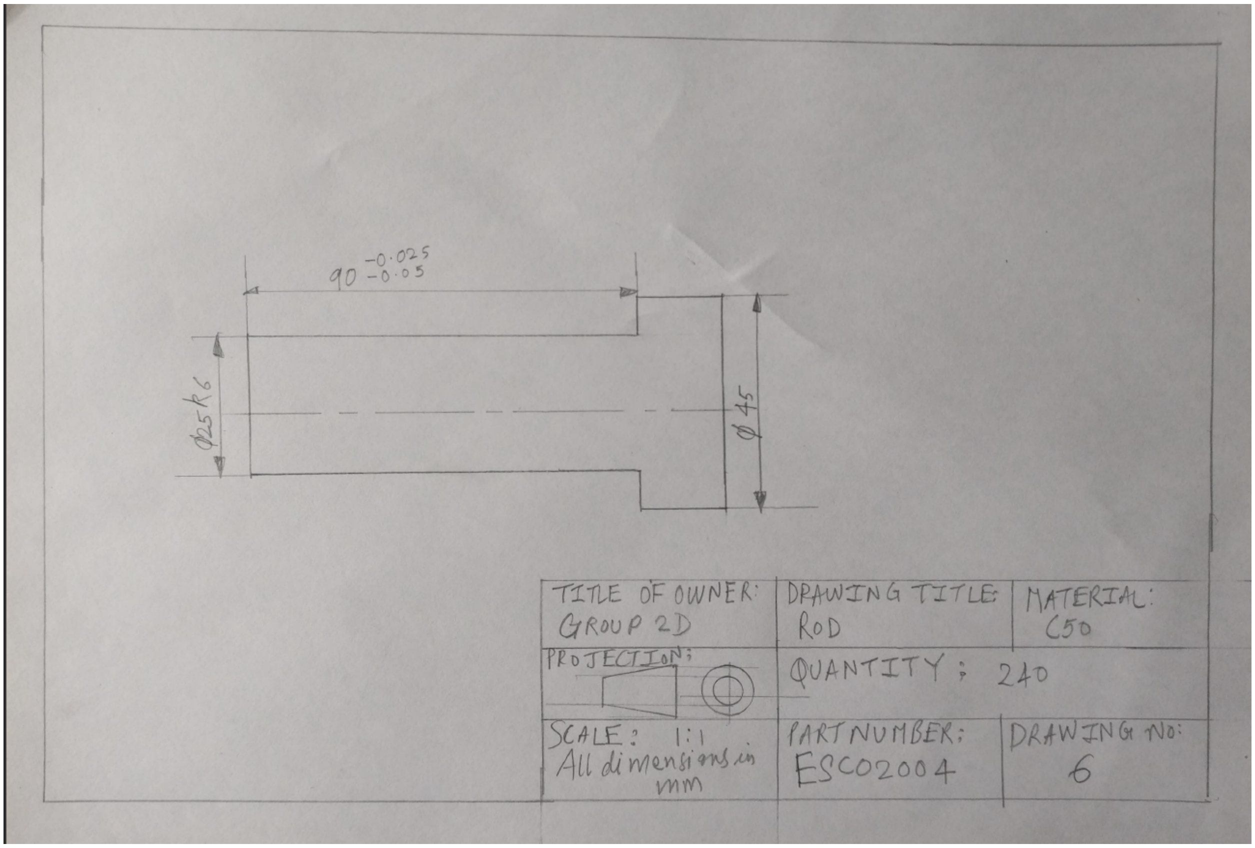
Bushings (ESC01005, Drawing 4):



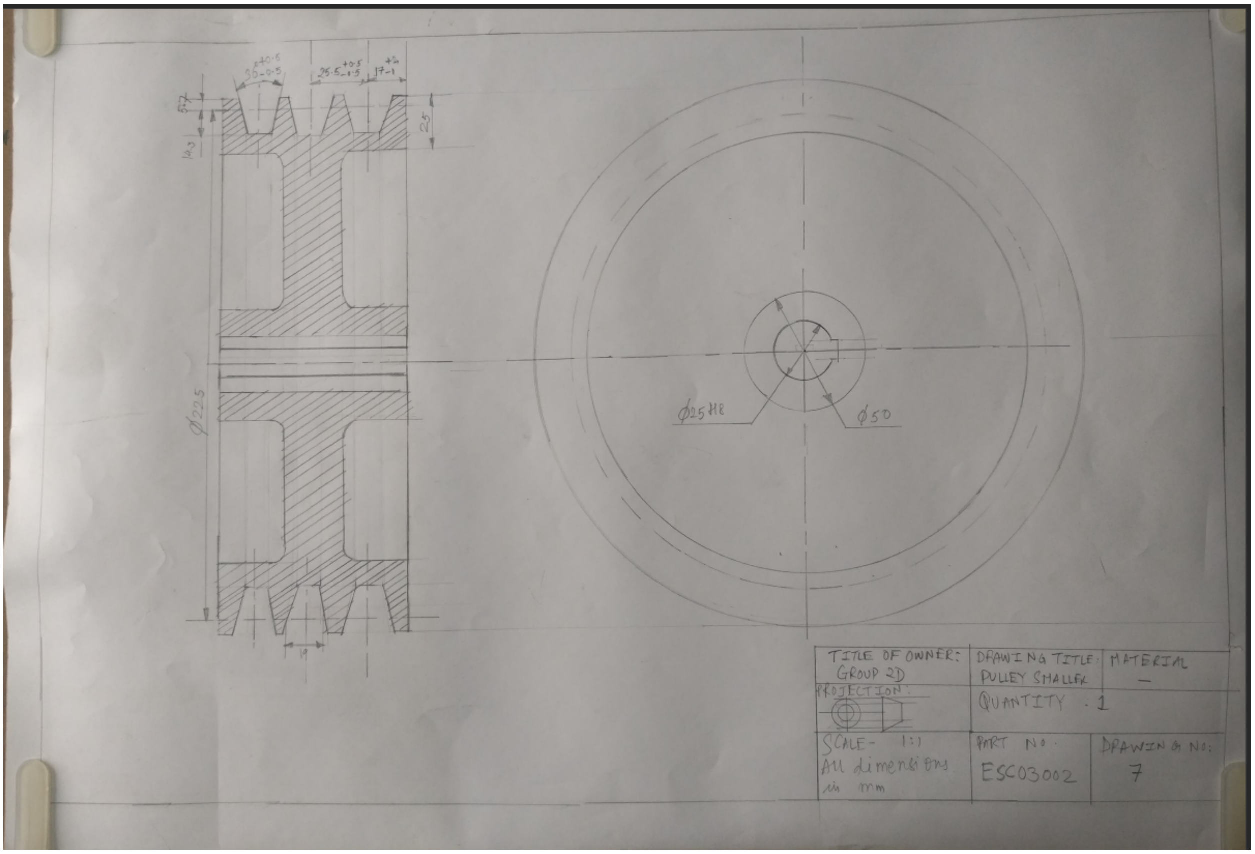
Step Frame(ESC02002.Drawing 5)



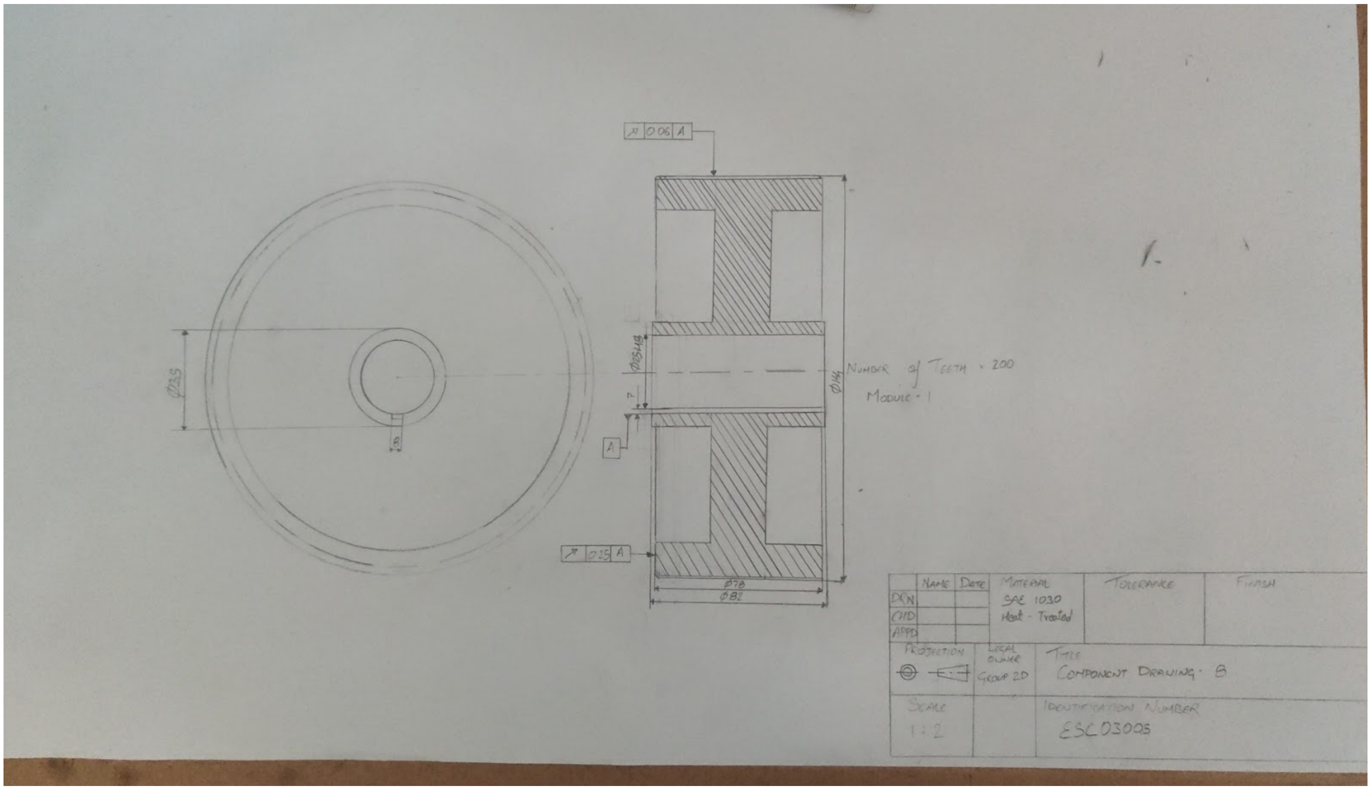
Rod(ESC02004, Drawing 6):



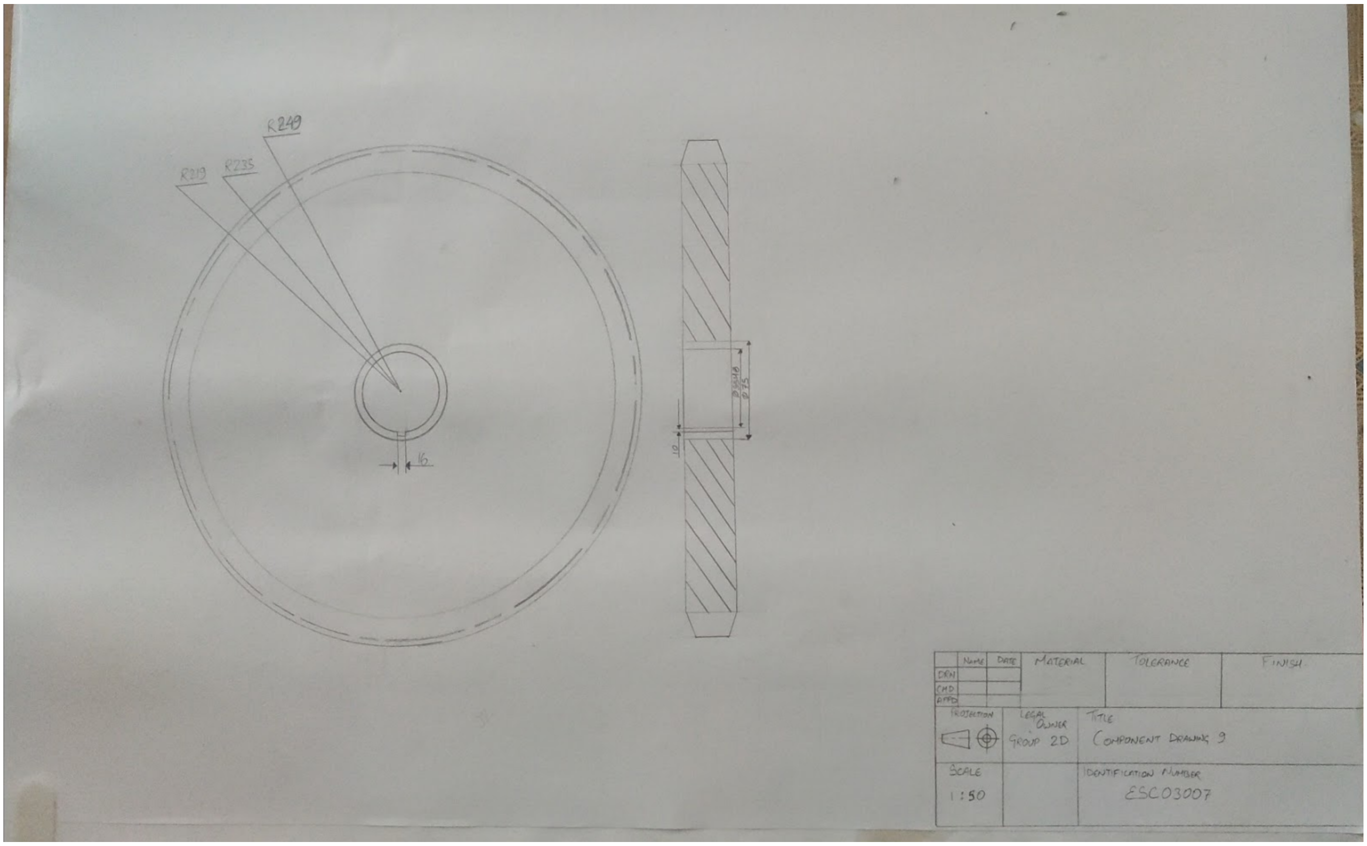
Smaller Pulley (ESC03002, Drawing 7):



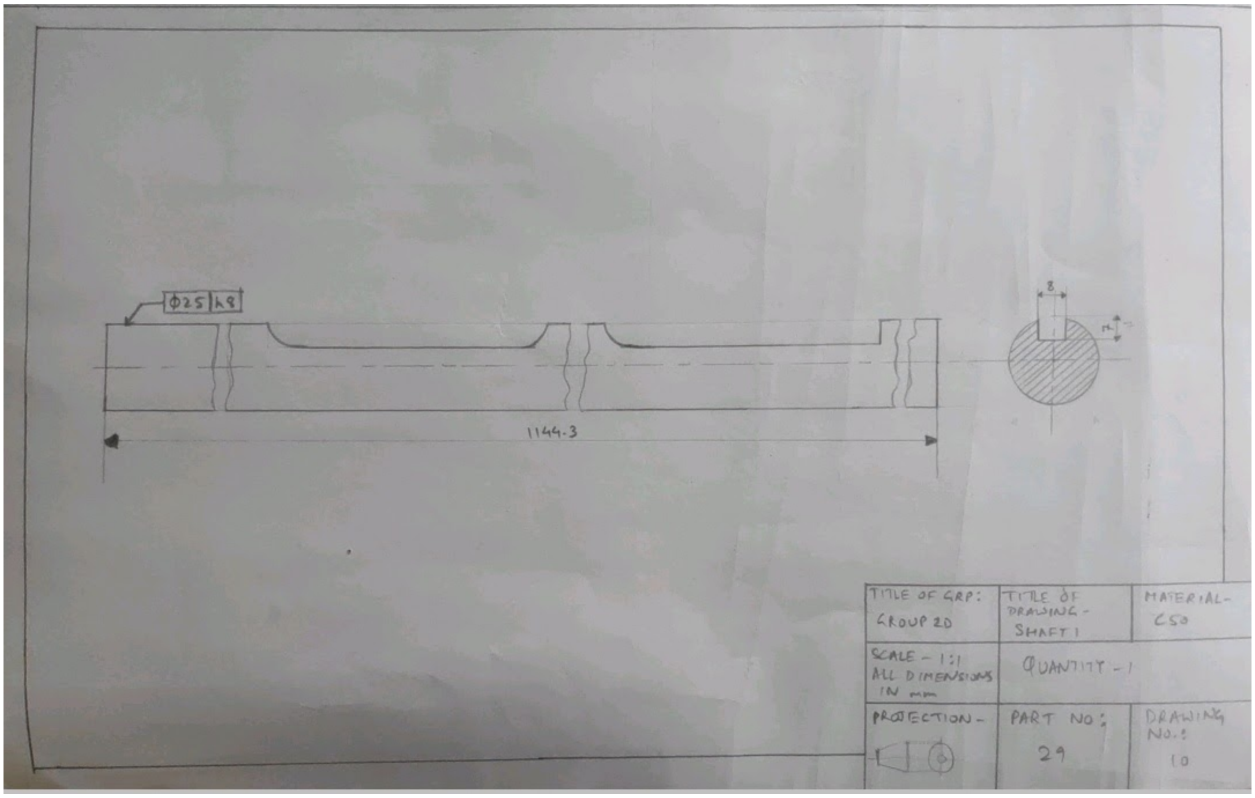
Helical pinion (ESC03005,Drawing 8)



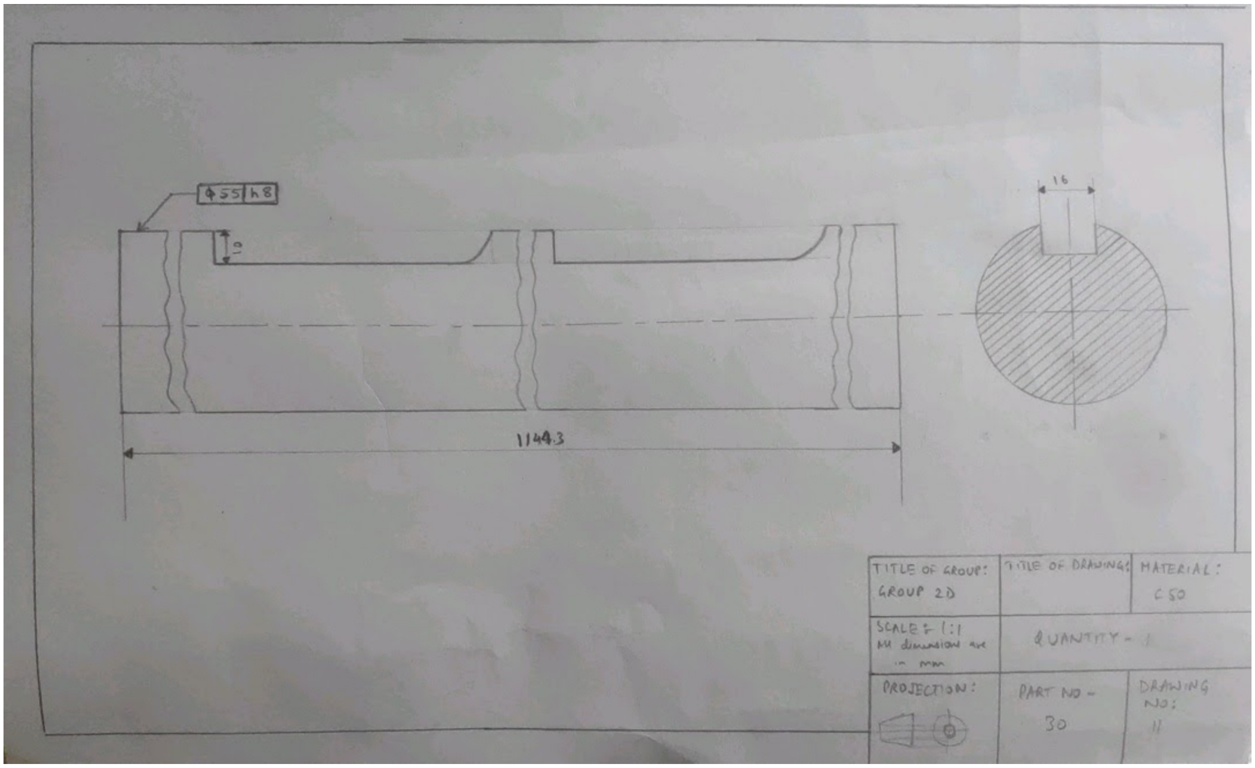
Sprocket 1(ESC03007.Drawing 9)



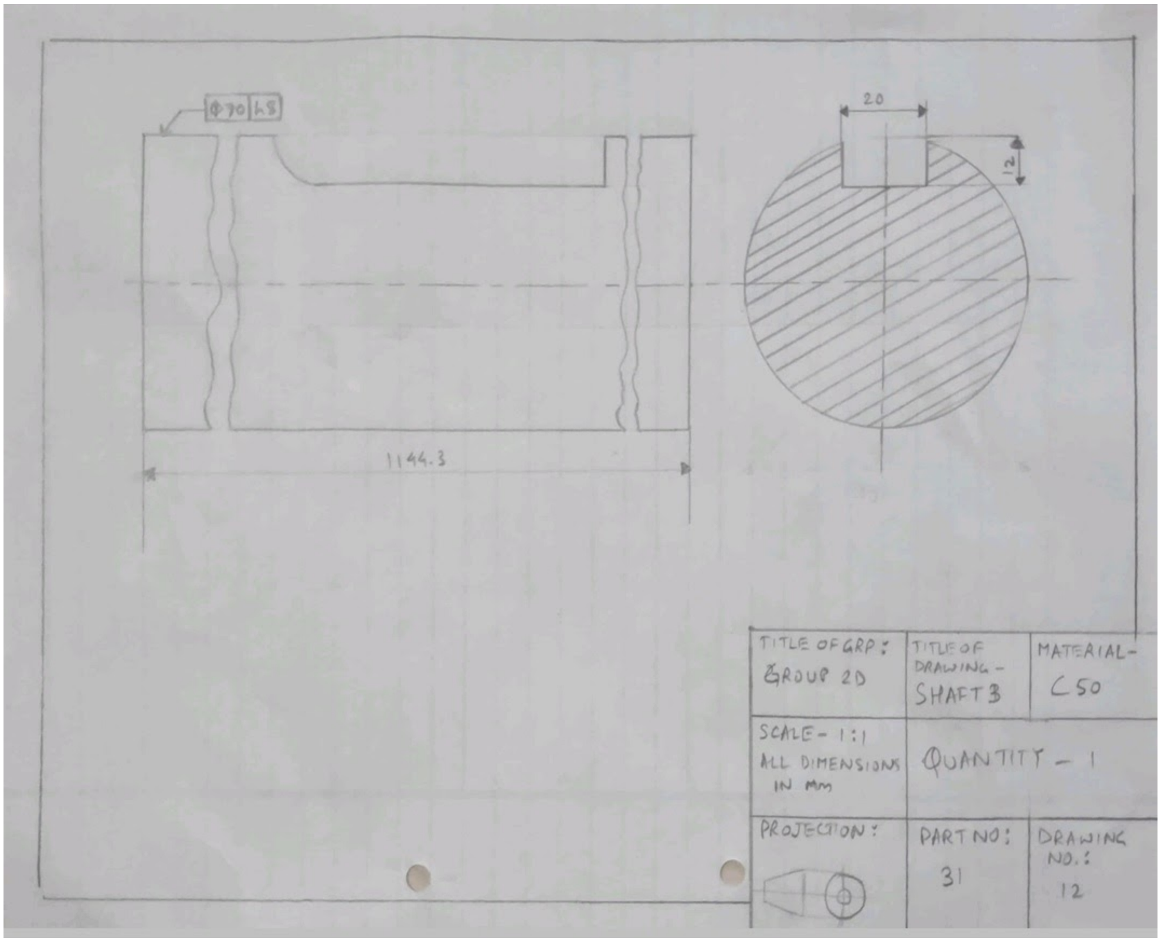
Shaft 1(ESC03010,Drawing 10)



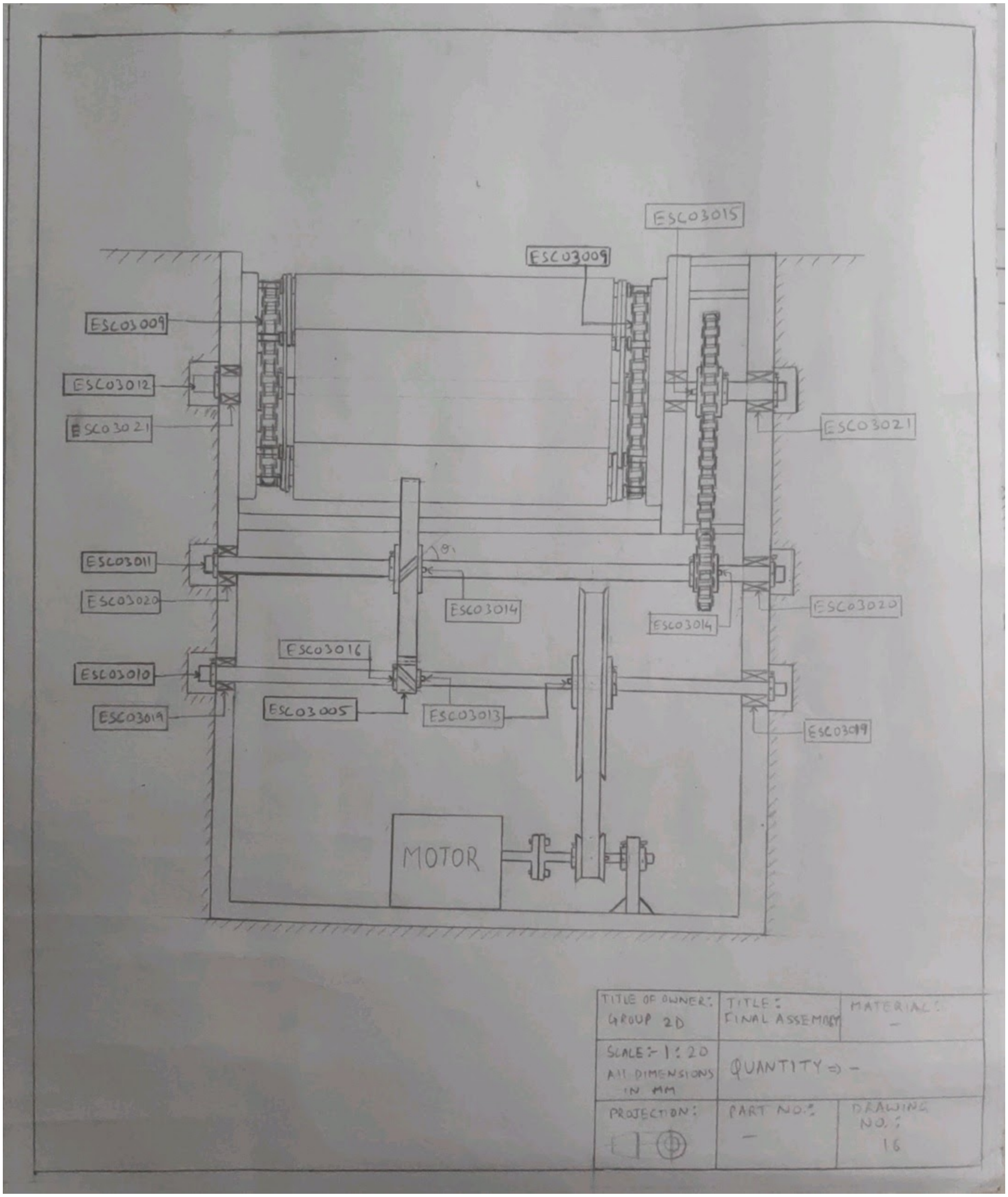
Shaft 2(ESC03011,Drawing 11)



Shaft 3(ESC03012,Drawing 12)



14. Final Assembly Drawing



15. SPECIFICATIONS

Overall height of the escalator = 7500 mm

Overall width of the escalator = 1101 mm

Overall horizontal traversal = 12990.38mm

Power requirement = 15.4hp

Maximum people flow rate = 7490 people/hr

Speed of step = 0.65m/s

REFERENCES :

[1] Schindler "Planning Guide for Escalators and Moving Walks."

[2] Tsubaki "Catalogue 2 | Attachment Chain"

[3] PSG College of Technology. "Design Data – Data Book of Engineers." May 2013 Edition.

[4] Faigle "Rollers and Components – For Escalators and Moving Walks."