Design of a Motor Driven Household Vertical Lift Elevator Chair

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Abstract

Design of motor driven household elevator chair" is a theoretical design of a hoisting system which will carry old and physically impaired people between various floors of a multistoried building. These people due to their disability can't use conventional staircase. The system is a modification of an elevator driven by rope and pulley in which the cabin is replaced by a chair. Thus the system will carry only one person at a time. Also as compared to conventional elevators which are used between several floors, this system is used only for one to two floors hoisting. Thus the system is cheaper than conventional elevator and can be afforded and used by individual families.

Keywords: Elevator Char, Sheave diameter, Brinel hardness number of the gears, damping ratio, Proximity sensors, Velocity factor.

I. INTRODUCTION

There are many dangerous areas in a home, but for seniors, the staircase is the most dangerous area in a home. Each year, falls on the stairs represent the leading cause of hospitalizations, as well as accidental deaths for those who are over 65. Since a fall can be such a serious and potentially life changing event, it is very important to prevent them whenever possible. Also if a person is suffering from physical disability which may be permanent like polio etc. or temporary such as fracture or those arising with age like gout, arthritis etc. can have difficulties in reaching upstairs by the conventional stair case and this will restrict their world to the ground floors only. This requires development of a device which can carry these people to different floors and increase their accessibility. At the same time they must be of low cost so that average individual families can afford it.

- Stair lifts [1]
- Elevators [2]

Based upon the driving mechanism the elevators are classified as:

A. *Hydraulic elevators*

The mechanism of working of hydraulic elevator is shown below in Figure 1

B. *Elevators using ropes and pulley*

The mechanism of working of Elevator using ropes and pulley is shown below in Figure 2

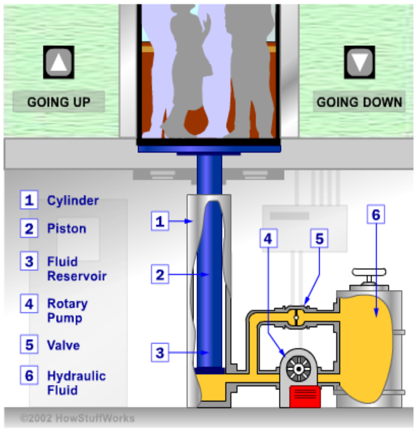


Fig 1: Hydraulic elevator

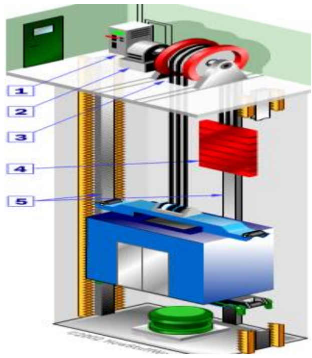


Fig 2: elevator using rope and pulley

C. An elevator chair: An ultimate solution

Our need is to design a hoisting system which can carry one person at a time and also the system should be cheap. Hydraulic elevators are not suited for our purpose since the mechanism is costly to install and run at individual household levels. The elevators with rope and pulley drive are close to our requirements but needs certain modifications to suits the purpose ideally. Since the system has to carry one person only at a time, the car must be replaced by chair. Rolling contacts are used in place of sliding to reduce friction. Also a motor of smaller KW rating is used since the chair carries one person only. Other features are very much similar to elevator using rope and pulley system. [3] This modified form of elevator driven by rope and pulley is named as ELEVATOR CHAIR and is an ideal solution for the problem.

As compared to other hoisting methods the elevator chair offers several benefits for the given design problem:

- They are cheaper to install and run than other methods as especially compared to hydraulic system.
- Elevators chair moves vertically and hence requires lesser space than stair lift. It requires a maximum cross section of 2ft*2ft while the stair lift runs along the entire staircase thus blocking a large portion of it causing difficulty for other people moving up and down through staircase.
- An elevator chair can be installed anywhere for example inside or outside of the house while stair lift need to be installed along with the staircase.
- Curved staircase causes problem in stair lift installation and adds to the cost while there is no such problem in case of an elevators.

1) *Step I Design of wire rope, sheave and selection of fasteners and power and torque calculations,*

Type of rope.	Feature of rope.	Applications.
6x7	Standard course laid rope.	Used as haulage ropes in mine haulage, tramways etc.
6x19	Standard hoisting rope.	Hoisting ropes in mines, ore docks, cargos etc.
6x37	Extra flexible hoisting rope.	Hoisting rope in steel mill ladle, cranes, high speed elevators.
8x19	Extra flexible hoisting rope.	Hoisting rope.

Table 1: Standard rope types and their uses [4]

Nominal diameter.	8, 10,12,14,16, 18, 20, 22, 25.
Average weight. (N/m)	$3.4xd^2$
Tensile strength of wire /1000-1250 N/mm^2	$35.5xd^2$
Tensile strength of wire/1250-1400 N/mm	$44.50xd^2$
Approximate wire diameter. (d_w)	0.050xd
Approximate wire areas. (A)	$0.35xd^2$
Preferred sheave diameter.	31xd
Minimum sheave diameter.	21xd

Table 2: Specifications of 8x19 type of hoisting rope: [5]

Note: Here d is in cm.

Ropes are subjected to the following stresses:

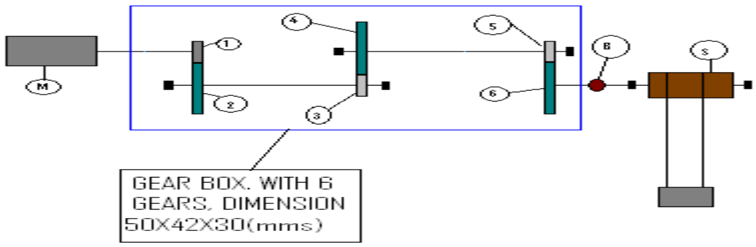
- Directed stress due to weight hoisted and weight of rope.
- Bending stresses when the ropes wind around sheave or drum.
- Starting stresses.

Rope fasteners: The various rope fasteners are shown in the figure and their efficiencies are given [6];

Type of fastener	Efficiencies
Wire rope socket with zinc	1.00
Thimble with four or five wire trucks	0.90
Special offset thimble with clips	0.90

Regular thimble with clips	0.85
Three bolt wire clamps	0.75

2) *Step 2 Gear design:*



Gear Box Layout

M=Motor, 1, 2, 3, 4, 5, 6 = Gears, B=Brakes, S=Sheave or pulley.

We require a constant speed of rotation of 25 rpm of the sheave shaft. [3]This speed should remain constant for all values of torque acting on the pulley. Thus we require a constant speed drive i.e. motor. The constant speed motor used commonly is a synchronous motor whose speed depends only upon supply frequency and number of poles and is given by

$$N_S = \frac{120f}{p}$$

Where f is the supply frequency= 50 Hz, P is the number of poles.

For P=2, N=3000 rpm, P=4, N=1500 and so on. Now we have to select a motor having number of poles such that the cost of motor and gear box in order to obtain speed reduction should be minimum. The cost of motor is least for P=2, N=3000 rpm but the cost of gear box reducing the speed from 3000 rpm to 25 rpm will be very high. Hence P=2 is not feasible. Increasing the number of pole pair, cost of motor rise by 1.5 times for every set of pole pair. Hence too low speed motor will not justify the coat. SO WE TAKE A FOUR POLE MOTOR HAVING A SPEED OF 1500 RPM AND WILL USE A GEAR BOX TO REDUCE IT TO 25 RPM.

Considering the losses in transmission, the motor is selected for rated power = 500watts

The gear box consists of six gears. The speed of shaft on which sheave is mounted comes out be 25rpm from previous calculations. This total speed reduction from 1500rpm to 25 rpm is obtained in three stages.

Gear 1-2 Gear ratio4:1
Gear 3-4 Gear ratio4:1
Gear 5-6 Gear ratio4:1

Total gear ratio is 64:1.i.e the speed is reduced from 1530 rpm to 25 rpm, Gear 6 rotates at 24 rpm (shaft – sheave shaft)

Results for the calculation of all gear set.

SR NO	Values	Gear set 1-2	Gear set 3-4	Gear set 5-6
1	Power transmitted	900watt	900watt	900watt
2	Pitch line velocity	4.71m/s	1.17m/s	0.29m/s
3	F_t	192N	768N	3072N
4	Diameters	D_1 = 6cm,D_2= 24cm	D_3= 6cm, D_4=24cm	D_5 6cm, D_6 =24cm
5	Modules	2.5mm	3mm	5mm
6	Numbers of teeth's	Z_1=24, Z_2 = 96	Z_3 =20, Z_4 =80	Z_5 = 12, Z_6 = 48
7	Form factor y	Y_1 =0.364, Y_2 =0.456	Y_3 =0.34, Y_4 =0.448	Y_5 =0.245, Y_6 =0.424
8	Face width for safe bending stress	15mm	25mm	50mm
9	Dynamic load error value e	0.025,class 2 gear	0.025 class 2 gear	0.025,class 2 gear
10	C value	105.450KN/m	105.450KN/m	118KN/m
11	F_i(N)	1243	1008	542
12	F_d(N)	1434	1776	3614
13	Beam strength for weaker of the two gear (N)	Pinion 1 is weaker f_b =2293>1.5F_d So safe	Pinion 1 is weaker f_b =4284>1.5F_d So safe	Pinion 1 is weaker f_b =10290>1.5F_d So safe
14	Checking for failure	F_{es} = 490Mpa K_W=1543KN/m^2 F_W = 2222N F_W>F_d so ok also f_b > F_W so ok	F_{es} = 490Mpa K_W =1543KN/m^2 F_W = 3703N F_W>F_d so ok also f_b > F_W so ok	F_{es} = 490Mpa K_W =1543KN/m^2 F_W = 7406N F_W>F_d so ok also f_b > F_W so ok

Dimensions of various gear set [7]:

SR NO	Gear dimensions	Gear set1-2	Gear set3-4	Gear set 5-6
1	Addendum	a=m=2.5mm	a=m=3mm	a=m=5mm

2	Dedendum	d=1.25m=3.125mm	d=1.25m=3.75mm	d=1.25m=6.25mm
3	Clearance	0.25m=0.625mm	0.75mm	1.25mm
4	Total depth (a+d)	5.625 mm	6.75mm	11.25mm
5	Working depth	5.625-0.625=5mm	6mm	10mm
6	Circular pitch P_C	πxm=7.85mm	9.42mm	15.71mm
7	Backlash	For m=2.5mm And backlash=0.08mm	For m=3mm And backlash=0.10mm	For m=5mm And backlash=0.15mm
8	Width of space	$\frac{7.85+0.08}{2}$=3.965mm	4.76mm	7.93mm
9	Tooth thickness	3.965-0.08=3.165mm	4.66mm	7.78mm
10	Diameter of addendum circle	Gear1 D_1+2a=0.06+2a=0.065m Gear2 D_2+2a=0.24+2a=0.245m	Gear3 D_3+2a=0.066m Gear4 D_4+2a=0.246m	Gear5 D_5+2a=0.07m Gear6 D_6+2a=0.25m
11	Diameter of Dedendum circle	Gear1 D_1-2d=0.06-2×0.003125 =0.05375m Gear2 D_2-2d=0.24-2×0.003125 =0.23375m	Gear3 D_3-2d=0.0525 m Gear4 D_4-2d=0.2325 m	Gear5 D_5-2d=0.0475 m Gear6 D_6-2d=0.2275m

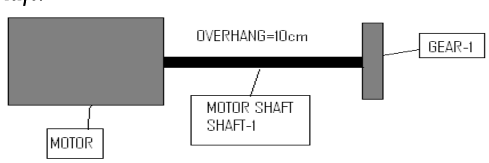
3) *Step 3: Shaft design:*

There are 4 shafts which are to be designed. These are:
- Motor shaft i.e. shaft 1 (rotating clockwise when viewed from right)
- Auxiliary shaft 2 (rotating anticlockwise when viewed from right)
- Auxiliary shaft 3 (rotating clockwise when viewed from right)
- Main shaft or sheave shaft 4 (rotating anticlockwise when viewed from right)

[4] Let us assume that all gears are transmitting power horizontal plane i.e. the tangential component of tooth load is horizontal and the normal component be vertical. All the shafts will be transmitting torque and subjected to bending moment. Hence the design will be for combined loading. To find out the maximum bending moment we draw BMD in both horizontal and vertical plane and find out the resultant maximum BM. In drawing FBD of the beams in horizontal plane let us take force acting into the paper positive and that acting out of the paper negative. In drawing FBD of the beams in vertical plane let us take force acting upward positive and acting downward negative. The upward direction is positive direction for FBD by convention.

The weight of gears is neglected since its values are very small as compared to other forces acting on the shaft.

a) *PART A: Design of Motor Shaft:*



As shown above, the shaft behaves like a cantilever beam

The shaft should be designed for maximum torsional moment to be transmitted, not the useful torsional moment. Maximum torsional moment will be induced in the shaft when the driver is starting from stationary position. The tooth load acting at the pitch line at the moment will be corresponding to the stalling load.

Maximum tangential load = $F_O = \frac{F_t}{C_V} = \frac{192}{0.39}$=493N.

Where CV is the dynamic factor.

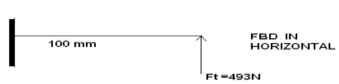
Torsional moment acting on the shaft = power/angular velocity

= 900/160

= 5.625N-m

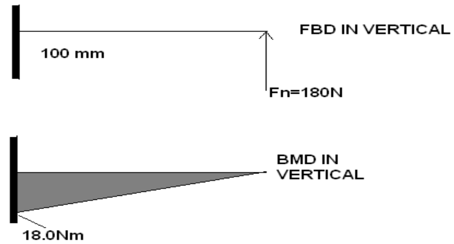
Now, F_n=F_t x tan20=493xtan20=180N.

Drawing the FBD in horizontal plane:



Maximum value of BM occurs at the support O and is given by M_b=49.3Nm

Drawing the FBD in vertical plane:



Maximum value of BM occurs at the support O and is given by M_b=18.0Nm

Hence the resulting BM is maximum at O given by, $M_b=\sqrt{49.3^2+18^2}$= 52.5 Nm

$$(f_s)_{\text{Max}} = \frac{16}{\pi d^3}\left[\sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right],\ (f_t)_{\text{Max}} = \frac{16}{\pi d^3}\left[K_b M_b + \sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right]$$

Where K_b *and* K_s are shock and fatique factor, For moderate shock $K_b = K_s$= 1.5, Design stress f_s = 0.3×elastic limit in tension Or f_s =0.18×ultimate tensile strength whichever is smaller

Design stress, f_t= 0.6×elastic limit intension Or, f_t= 0.36× ultimate tensile strength whichever is smaller

Due to keyways there is a reduction of strength of shaft by 25%.

The shaft is 0.35 percent carbon steel with ultimate tensile strength = 410Mpa, Elastic limit = 240Mpa

f_s = 0.3×0.75×240=54Mpa (smaller is taken) *or* 0.18×0.75×410=55.35Mpa

Similarly f_t= 0.6×0.75×240=108 Mpa (smaller is taken) or 0.36×0.75×410=110.7Mpa

Finding the diameter of shaft

$$\text{Shear stress } d^3 = \frac{16}{\pi f_s}\left[\sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right] \text{ or}$$

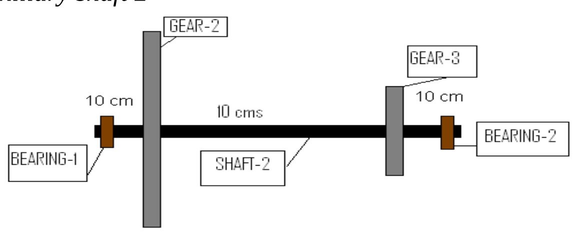
$$d^3 = \frac{16}{\pi \times 54 \times 10^6}\left[\sqrt{(1.5 \times 52.48)^2 + (1.5 \times 5.625)^2}\right],\ \text{d = 1.95cm}$$

$$\text{Tensile stress } d^3 = \frac{16}{\pi \times f_t}\left[K_b M_b + \sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right] \ or$$

$$d^3 = \frac{16}{\pi \times 108 \times 10^6}\left[1.5 \times 52.48 + \sqrt{(1.5 \times 52.48)^2 + (1.5 \times 5.625)^2}\right],\ d = 1.97cm$$

Hence taking a suitable factor of safety, diameter of 2.5cm is taken for motor shaft. So motor shaft is 0.35% carbon steel with diameter of 2.5cm and overhang of 10cm.

b) PART B: Design of auxiliary shaft 2



The shaft acts like simply supported, with support at two bearings

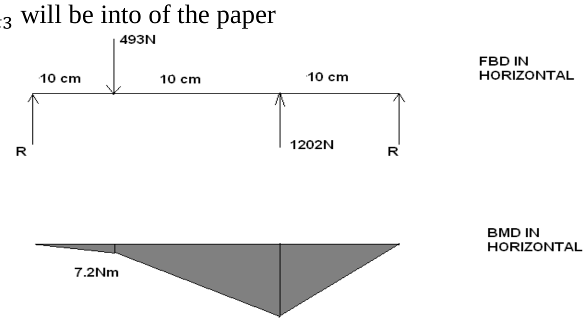
- For gear 2; F_t=F_O=493N, F_N =180N
- For gear 3 F_t=F_O=$\frac{768}{0.64}$ = 1202N (as C_V = 0.64), F_n=F_t x tan20=1202xtan20=437N.

The BMD can be drawn for the shaft in vertical and horizontal planes respectively.

Drawing the FBD in horizontal plane:

Taking the rotation of the shaft anticlockwise when viewed from right side (see block diagram)

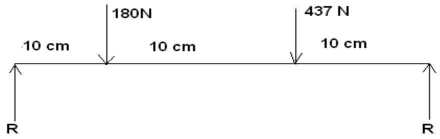
F_{t2} Will be out of the paper and F_{t3} will be into of the paper



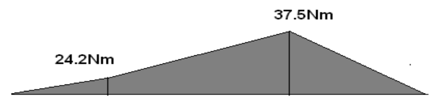
Maximum value of BM occurs at the point D and is given by M_b=63.7Nm

Drawing the FBD in vertical plane:

F_{n2} And F_{n3} will be downward.



FBD IN VERTICAL



BMD IN VERTICAL

Maximum value of BM occurs at the point D and is given by M_b=37.5 Nm

Total bending moment at D, $M_b=\sqrt{63.7^2+37.5^2}$=74 Nm, Torque transmitted = $\frac{power}{\omega}=\frac{900}{40}$= 22.5N-m

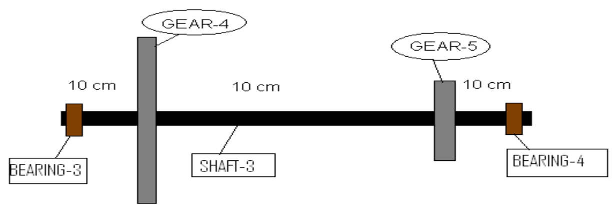
Taking the same material for shaft-2 as shaft-1 and same shock and fatigue factor we find,

Shear stress, $d^3=\frac{16}{\pi f_s}\left[\sqrt{(K_b\times M_b)^2+(K_s\times M_t)^2}\right]$, d = 2.22 cm

Tensile stress, $d^3=\frac{16}{\pi\times f_t}\left[K_bM_b+\sqrt{(K_b\times M_b)^2+(K_s\times M_t)^2}\right]$, d= 2.2cm

Hence a standard value of diameter of 2.5cm is taken for shaft 2 also, which is made up of 0.35% carbon steel.

c) PARTC: Design of auxiliary shaft 3:



The shaft acts like simply supported beam, with support at two bearings

For gear 4; F_t=F_O=1202N, F_N =437N,

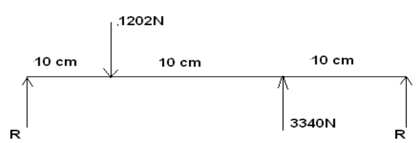
For gear 3; F_t=F_O=$\frac{3072}{0.92}$ = 3340N (as C_V = 0.92)

F_n=F_t x tan20=3340xtan20=1215N.

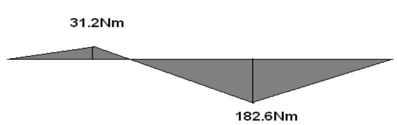
The BMD can be drawn for the shaft in vertical and horizontal planes respectively.

Drawing the FBD in horizontal plane:

Taking the rotation of the shaft clockwise when viewed from right side (see block diagram) F_{t4} Will be out of the paper F_{t5} will be into of the paper



FBD IN HORIZONTAL

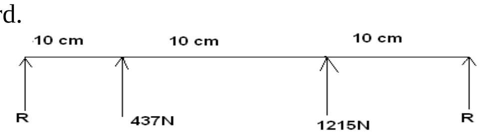


BMD IN HORIZONTAL

Maximum value of BM occurs at the point D and is given by M_b=182.6Nm

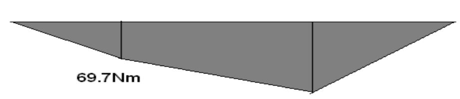
Drawing the FBD in vertical plane:

F_{n4} And F_{n5} will be upward.



FBD IN VERTICAL

BMD IN VERTICAL



Maximum value of BM occurs at the point D and is given by M_b=95.6Nm

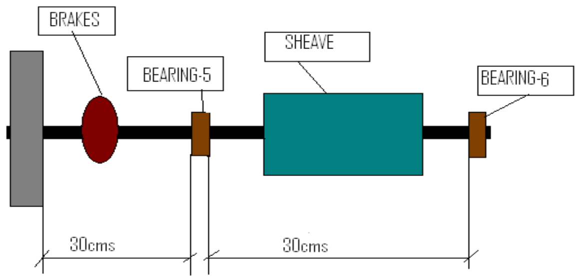
Total bending moment at D, $M_b=\sqrt{182.6^2+95.6^2}$ =206.11N-m, Torque transmitted = $\frac{power}{\omega}=\frac{900}{10}$= 90N-m

Shaft-3 is made up of nickel steel with ultimate strength = 700Mpa and elastic limit = 504Mpa and taking the same shock and fatigue factor of 1.5 we find,

– Shear stress, $d^3=\frac{16}{\pi f_s}\left[\sqrt{(K_b\times M_b)^2+(K_s\times M_t)^2}\right]$ d = 2.99cm
– Tensile stress, $d^3=\frac{16}{\pi\times f_t}\left[K_bM_b+\sqrt{(K_b\times M_b)^2+(K_s\times M_t)^2}\right]$ d= 2.97cm

Hence a standard value of diameter of 3.0 cm is taken for shaft 3.

d) PART D: Design of sheave shaft-4:



MAIN SHAFT OR SHEAVE SHAFT

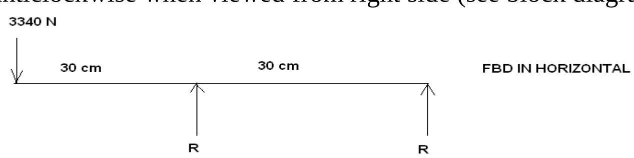
The shaft acts like simply supported beam with an overhang on one side and support at two bearings. The weight of brakes can be neglected in design since its value is negligible as compared to other loads [8] (electromagnetic power off brakes with torque rating less than 300Nm has a mass between 10-30 lb.)

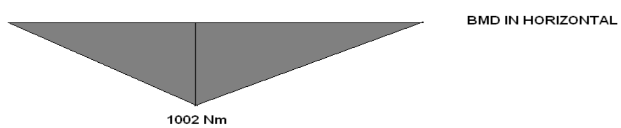
For gear 6; $F_t=F_O$=3340N, $F_{N=}$1215N

The BMD can be drawn for the shaft in vertical and horizontal planes respectively.

Drawing the FBD in horizontal plane:

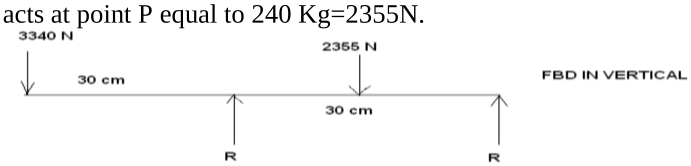
Taking the rotation of the shaft anticlockwise when viewed from right side (see block diagram) F_{t6} Will be out of the paper.

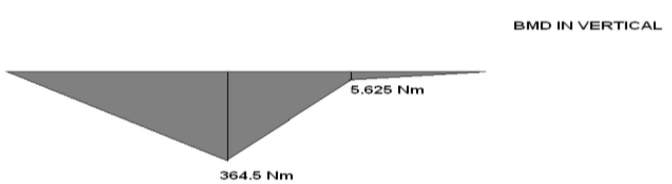




Maximum value of BM occurs at the point A and is given by M_b=1002Nm

Drawing the FBD in vertical plane: F_{n6} will be downward and weight of chair with full load i.e. 100+50=150Kg plus the weight of counter weight 90Kg acts at point P equal to 240 Kg=2355N.





Maximum value of BM occurs at the point A and is given by M_b=364.5Nm

Total bending moment at A, $M_b=\sqrt{1002^2+364.5^2}$=1066N-m

Torque transmitted = $\frac{power}{\omega}=\frac{900}{12.5}$= 360N-m

Shaft-4 is also made up of nickel steel with ultimate strength = 700Mpa and elastic limit= 504 Mpa and taking the same shock and fatigue factor of 1.5 we find,

- Shear stress, $d^3 = \frac{16}{\pi f_s}\left[\sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right]$ d = 5.14 cm
- Tensile stress, $d^3 = \frac{16}{\pi \times f_t}\left[K_b M_b + \sqrt{(K_b \times M_b)^2 + (K_s \times M_t)^2}\right]$ d= 5.06cm

Shaft no.	Material	Diameter
Shaft 1:	0.35% carbon steel.	25 mm
Shaft 2:	0.35% carbon steel.	25 mm
Shaft 3:	nickel steel	30 mm
Shaft 4:	nickel steel	60 mm

Hence a standard value of diameter of 6cm is taken for shaft 4.

II. CONCLUSION

The following procedures have been followed in the design of elevator chair. Requirements of the elevator chair such as weight of chair, weight of the person to be lifted, weight of the counter weight, chair dimensions and design. Selection of standard factor of safety for elevator rope design ='8'.Selection of type of rope best suited for elevators and its strength and dimensional specifications. Calculating the various stresses induced in the rope, number of ropes required and checking for the given factor of safety. Selection of a proper sheave type from the standard tables. Selection of a suitable rope fastener to attach rope to the chair and the counter weight. Calculating torque and power required and selecting a suitable motor. Design of gears in gearbox to carry out suitable speed reduction. Calculating the dimension of the shaft from stress requirement and selecting a suitable dimension from the standard tables. This study researches the modification of homes, where people spend a large part of their lives, focusing on the homes of the disabled, who are likely to spend more time at home. Specifically, the study explores the current state of the use of modified homes, problems with home modification, and why many of modified homes are in need of re-modification

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