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Quantitative Design Analysis of an Electric Scissor Lift

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Abstract

In the industry, as a lifting system, the electrical scissor lift system is commonly used. this system is mainly preferred to do Load lifting and lowering activities, cleaning services, and maintenance-repair activities. In this paper, quantitative design analysis of several principal components of an electric scissor lift will be undertaken. A brushless dc geared-motor will be used to drive the rear wheels of the vehicle within an industrial area with front-wheel steering. The scissor lift will be operated via a motor-ballscrew arrangement. A systematic and detailed evaluation of the design will be carried out with paying particular attention to several aspects such as actuator selection for wheel drive unit and lift motion, quantitative design of the wheel drive gearbox, gearbox bearing selection, qualitative design of scissor lift and ballscrew unit selection, stress analysis of scissor pin joints and ballscrew connections, and vehicle power requirements and the motor control.

Keywords: Electrical scissor lift; quantitative design; brushless motor; motor-ballscrew; gearbox; stress analysis.

1. Introduction

For purposes of lifting a load or providing at unreachable heights, lifting systems are generally used [1, 2]. Currently, in industry, many elevating systems are designed to be used for numerous purposes. Load lifting and lowering activities, cleaning services, and maintenance-repair activities are examples of a range of services and multi-purpose applications where these systems can be used. Lifting systems can be classified as scissor lifts, articulated lifts and telescopic lifts [3, 4].

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Scissors lifting systems are the most popular lifting systems in the industry. Figure 1 illustrated an example of a Scissors lifting system. Each platform has a height capacity and certain carrying. These systems are especially used for cleaning and maintenance. The platforms are elevated by opening the scissors which are connected to each other in the scissors lifting systems [2, 3]. At the base, there is a mobile cart with electrical DC motors for driving the wheels and giving the energy for levitation. A suitable design of the electric scissor lift parts is offered considering several practical factors. A brushless dc geared-motor will be used to drive the rear wheels of the vehicle within an industrial area with front-wheel steering. The scissor lift will be operated via a motor-ballscrew arrangement. A systematic and detailed evaluation of the design will be carried out with paying particular attention to several aspects such as actuator selection for wheel drive unit and lift motion, quantitative design of the wheel drive gearbox, gearbox bearing selection, qualitative design of scissor lift and ballscrew unit selection, stress analysis of scissor pin joints and ballscrew connections, and vehicle power requirements and the motor control. To undertake a 'preliminary' analysis for the gear design, British Gear Association (BGA) handbook is used [5]. When carrying out the pin design a NASA reference publication [6] is being used. In establishing the ratings for the bearings, the SKF bearing selection tools is being used [7]. Material selection plays a key role in designing a machine and also influence on several factor such as durability, reliability, strength, resistance which finally leads to increase the life of scissor lift [8].

Figure 1: An Example of the scissors lifting system.

2. Scope of the design

To undertake a quantitative design analysis of several components of an electrical Scissor is to lift up to a height of 8m and carrying capacity of 150kg. The drive speed of the vehicle is 1.5m/s and the platform size 2.5m×1.0m with the total weight 1000kg. It is required to use BLDC geared-motor to drive the rear wheels of the vehicle and a motor ball-screw should be used to operate the scissor.

3. Scissor part

3.1. Scissor arm

It is essential to select the following parameters for the scissor lift arm structure shown in Figure 2 to achieve the required scissor height (H):

- The arm lengths (L): It is better to select L less than 2.5m which is the total length of the vehicle is 2.5m, so let L (2.25m).
- Vehicle height (hv): Let assume its value = 0.5m.
- Scissor arms height (h): After assuming the value of the hv, it is possible to find its maximum value.
- The number of levels (n): It is required to assume it's initial value to be able to find(θmax). Let n=4.
- The minimum angle (θmin) and a maximum angle (θmax): The value of the θmax is calculated from the previous step (56.44°). It essential to select reasonable values for (θmin) where the maximum value of force needed to lift the platform and the minimum height of the scissor are largely depends on the value of θmin. Let assume its value equal to 5°. For this value of θmin, the minimum scissor height will be 1.284m. This value is essential to be small because it is better to be compact.

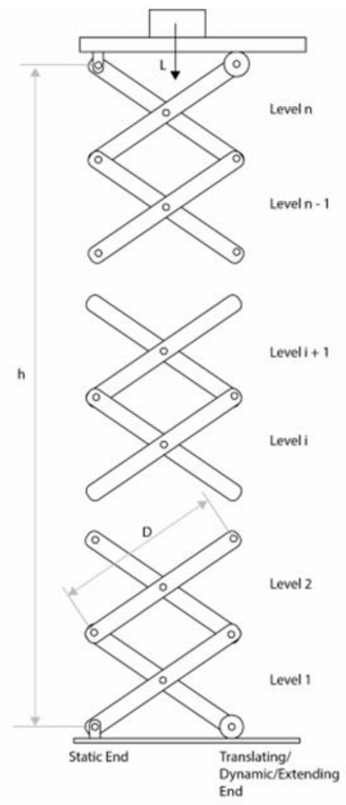

Figure 2: Scissor lift arm structure.

3.2. Force analysis

To do the force analysis, it is required to know the following parameters:

- Scissor lift capacity Mc= 150 kg
- Platform mass, assume its value =25kg.
- Safety factor (F.S), let be =1.2
- Scissor arm mass Mb, assume its value = 35kg , so Wb = 343.35 [N].

The total mass (MT) will be 210 kg. Assume that the total weight (WT) will be equally distributed to the two sides of the scissor, where each side will bear W, so W= (WT /2).

Total mass MT = (F.S) [Mc + Mb] =1.2 [150+25] =210 kg.

Total weight WT = MT *g = 210* 9.81 =2060.1 [N] , So W = (WT /2) = 1030.05 [N]

The force analysis will be used to know the force on each joint which is important for stress analysis for the pin joint. In addition, to select the ball-screw and the actuator.

3.3. Ball-Screw selection

In establishing the ratings for the bearings, the SKF bearing selection tools is being used [7]. The position of the ball-screw needs to be selected. In addition, the way in which the ball-screw will be fixed should be designed. The easiest way to fix the ball-screw is to put it in the button but the force needed will be the largest (16×Wb+4×W)/tanθ for each side. The value of force will be reduced when going up (the smallest value will be in the top) but the problem will be how to fix the ball-screw; so the decision will be is to put the ball-screw in the middle (between level 2 and level 3). The value of the force for one side in this location is F=[(4×Wb+2×W)/tanθ]. The maximum value needed is when the value of θ is the minimum (5°) so the value of the force will be F= 39.25 [KN], so to get the total force needed, FT = 2×(force of one side) = 78.5 [KN].After calculating the value of the total force needed, the selection of the ball-screw can be achieved. It is required to select a ball-screw can bear more than 78.5 [KN] so the ball-bearing in Figure 3 and Figure 4 has been chosen from SKF.

Designations	SL Basic load ratings dynamic C_a	SL Basic load ratings static C_{oa}	TL Basic load ratings dynamic C_a	TL Basic load ratings static C_{oa}	Support bearing Recommended thrust support bearings	Recommended support pillow block
	kN					
SL/TL 25×20R	22,8	51,5	12,6	25,8	PLBU 25/FLBU 25	BUF 25
SL/TL 25×25R	22,3	50,6	12,3	25,3	PLBU 25/FLBU 25	BUF 25
SL/TL 32×20R	25,4	65,2	14,0	32,6	PLBU 32/FLBU 32/FLRBU 3(*)	BUF 32
SL/TL 32×32R	26,1	69,3	14,4	34,7	PLBU 32/FLBU 32/FLRBU 3(*)	BUF 32
SLD/TLD 32×32R	26,1	69,3	14,4	34,7	PLBU 32/FLBU 32/FLRBU 3(*)	BUF 32
SL/TL 32×40R	12,6	29,8	6,9	14,9	PLBU 32/FLBU 32	BUF 32
SL/TL 40×20R	41,3	128,8	22,8	64,4	PLBU 40/FLBU 40	BUF 40
SL/TL 40×40R	51,7	130,5	28,5	65,3	PLBU 40/FLBU 40/FLRBU 4(*)	BUF 40
SL/TL 50×50R	92,9	235,1	51,2	117,6	PLBU 50/FLBU 50/FLRBU 5(*)	BUF 50

Figure 3: SKF ball bearing selection.

Also, the following information is required :

Designations	Nominal diameter	Lead	Dimensions Shaft			Nut								
	d_0	P_h	d_1	d_2	L Max	L	L_1	L_3	L_7	D_1 g9	D_4 js12	D_5 H13	D_6	M_2 6H
	mm													
SL/TL 25×20R	25	20	24,29	21,7	4 700	66,8	18	17,6	15	48	60	6×6,6	73	M6
SL/TL 25×25R	25	25	24,38	21,5	4 700	78,2	27	18,7	15	48	60	6×6,6	73	M6
SL/TL 32×20R	32	20	30,06	27,4	5 700	67,4	18	17,9	15	56	68	6×6,6	80	M6
SL/TL 32×32R	32	32	31,066	28,4	5 700	80,3	41	13	15	56	68	6×6,6	80	M6
SLD/TLD 32×32R	32	32	31,066	28,4	5 700	80,3	41	13	15	50*	65	6×9	80	M6
SL/TL 32×40R	32	40	29,58	26,9	5 700	54,8	17	12,2	15	53*	68	6×6,6	80	M6
SL/TL 40×20R	40	20	37,77	35,1	5 700	87,3	38	18	15	63	78	6×9	95	M6
SL/TL 40×40R	40	40	38,34	34,2	5 700	110,8	44	21,6	25	72	90	6×11	110	M8×1
$\boxed{\text{SL/TL 50×50R}}$	50	50	49,14	43,4	5 700	134	60	25,5	25	85	105	6×11	125	M8×1

Figure 4: The selected SKF ball bearing specifications.

The dynamic load for this ball-screw is 92.9 [KN] which is larger than 78.5 [KN]. Now, it is required to calculate the ball-screw practical efficiency to find the required torque value.

$T = F\ P_h / 2000\ \pi\ \eta_p$

$\eta_p = 0.9\ \eta$

where η = theoretical efficiency, T = input torque [Nm], F = maximum load of the cycle [N], Ph = lead [mm], and ηp = practical efficiency.

$H = 1 / [1 + [\pi\ d_o / P_h]\ \mu]$

Where μ = 0.006 for SD/SDS, BD/BDS, SX, SL, SN, SND, BX, BN, TL, PN, PND.

From the table, Ph =50mm and d0 =50mm , So $\eta = 0.98$, But $\eta p = 0.9\ \eta = 0.88$

So $T=(F \times Ph)/(2000\ \pi\ \eta p\ ) = (78.5 \times 1000 \times 50)/(2000\ \pi \times 0.88) = 710$ [N.m]

3.4. Actuator selection

It is required to calculate the rotational speed for the ball-screw to find the required power.

The power required to lift the scissor up and down is 2.556 [KW]

Torque T= 710 [N.m]. Speed n = 34.3 [rpm].

To get a motor with high torque and low speed, a reduction gearbox is required.

Motor selection: The selected motor has the following characteristics:

Power P= 3 [KW], Torque T= 15 [N.m], Speed n = 2000 [rpm] , 24V BLDC motor

Gear selection: The selected gearbox was "Low backlash planetary gearbox" which has a wide ratio from i= 3 to i=100.

3.5. Material selection for the arms

Let alloy steel (1010) will be used for the arms with rectangular tube as shown in Figure 5 with the following dimensions:

Arm mass calculations : Density (ρ) = (Mass (M))/(Volume (V))

Volume (V) = Area (A) × arm length (L)

Area (A) = Outer Area (Ao) – Inner Area (Ai)

Ao=A×B = 3.2 ×10^(-3) [m2] , Ai=(A-2C)×(B-2C) = 1.2 ×10^(-3) [m2] , A=2 ×10^(-3) [m2]

Volume (V) = Area (A) × arm length (L) = 4.5 ×10^(-3) [m2]

For Alloy steel 1010 Density (ρ) = 7868 kg/m3 , Mass (M)= Density (ρ) × Volume (V) = 35 kg

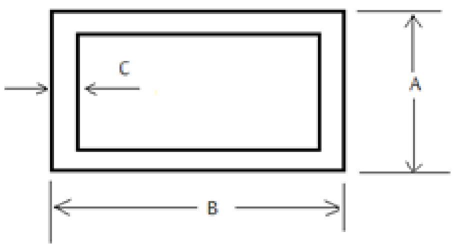

Figure 5: Cross-section area of the scissor arm.

Where A = 4 cm, B = 8 cm, C = 1 cm, and Arm Length = 2.25 m.

3.6. Stress analysis of the pin joints and the ball screw

The analysis will be for the joints that have the maximum load which are the lowest pairs; also, the worst case will be considered which is θ = θmin =5 °, this will make Fx reaches to its maximum value. In addition, assuming all the weight will be on one side (W=WT) where Fy will be maximum.

Fx= (4Wb+2WT)/tan θ and Fy=4Wb+WT/2 Totally F = 110.03 [KN]

The stress on the pin is " Shear (τ)" = F/A Where A= (πd^2)/4 , d = pin diameter

After the calculations , the pin diameter d = 0.04m = 4 cm.

3.7. Ball-screw mounting

Figure 6 is showing the element that will be used, where the connection with the scissor will be by the " Nut" and the " Buf 50".

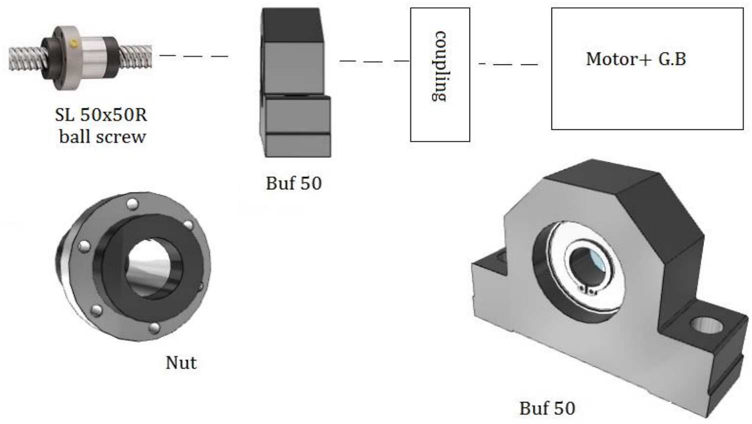

Figure 6: Ball-screw mounting.

4. Vehicle part

Weights and measures should be expressed in either SI (MKS) or CGS as primary units. (SI units are encouraged.).

4.1. Force analysis

To be able to choose an appropriate motor to drive the scissor, the forces on the vehicle should be analyzed. Assuming that there is a slope in the ground (α max =10°). M= total mass = (scissor mass) + (lift capacity) = 1000+210 = 1210 kg, F = Applied force, Fr = Force due to friction = Kr×N, Kr = Rolling resistance coefficient, N = force normal on the surface, v= Linear velocity = 1.5 m/s, a= linear acceleration, T = applied torque, P = Power applied, Assume that the vehicle needs 5 sec to reach to the maximum speed (t=5 sec), The value of the Rolling resistance coefficient (Kr) depends mainly on two main factors, "Tire type" and "ground type". For "Harden-rubber tire" and "Concrete ground" Kr = 0.035, Tire dimensions: The tire that will be used is 406×125 mm so the tire radius: r=d/2=406/2= 20.3mm

4.2. Differential efficiency

It is important to consider the differential efficiency which is required to change the axis of rotation. The differential has two stages, the efficiency per stage =0.96. Total efficiency η =0.96×0.96=0.92 = Po/Pi , Po = 4.26 [KW], Pi = 4.26/0.92 = 4.6 [KW] Because the input power to the differential is the same output power from the gearbox, the (Po) for the gearbox is known = 4.6 [KW] also the To=575.2 [N.m] and the rotational speed =70.7[rpm] as shown in Figure 7.

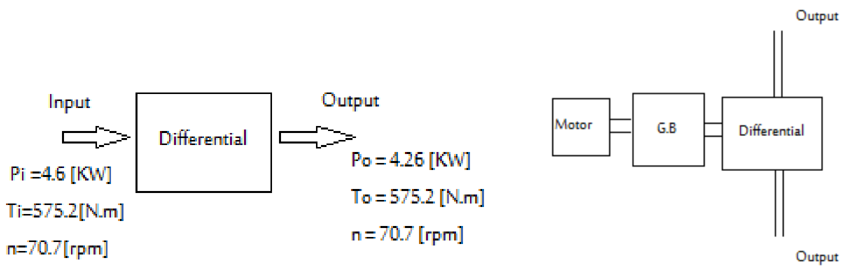

Figure 7: (a) Differential efficiency (b) Differential, gearbox, and motor connection.

4.3. Actuator selection

It is required to select a motor that has enough power to drive the vehicle. The additional factor which is also important is the gearbox efficiency. Assume that this gearbox has two stages and the efficiency per stage is 0.98, for two stages η=0.98×0.98=0.96.

Pi =Po/η = 4.6/0.96=4.8 [KW] This value needed to move the vehicle so the motor should be selected according to this value.

The motor selected was: 48 V BLDC motor , P = 5.5 [KW], T = 36 [N.m] , n= 1500 rpm as shown in Figure 8.

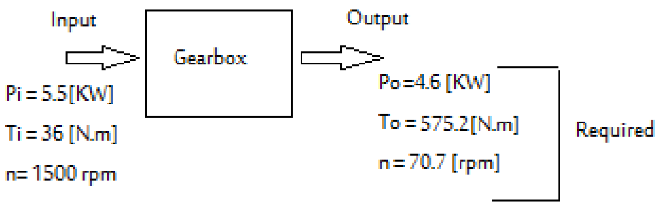

Figure 8: Actuator selection.

4.4. Gearbox design

To undertake a 'preliminary' analysis for the gear design, British Gear Association (BGA) handbook is used [5]. The Input requirements: Nominal input power, Pn =5.5 KW, Nominal input speed, n = 1500 rpm.

Transmission ratio, i = 1500 / 70.7 = 21.22 rpm

Gear Life (hours), NL = 24x7x52x5= 43680 (based on 5 years life)
Required Reliability 95%

To specify the ratio split, overall Gear Ratio, i=21.22, should use 2 stages.

For the design torque, because there is no load histogram so the application factor, KA must be chosen.

KA = 1.3, for the driven machine (moderate shock) and the prime mover is electrical motor.

1st stage: TD1= Tn. KA, TD1 $=36 \times 1.3 = 46.8\ [N.m]$

2nd stage: TD 2= TD1. i1. η1 $= 46.8 \times 5.2 \times 0.98 = 238.5\ [N.m]$
For the gear life, the number of pinion load cycles, for 1st stage: NL = 43680×60×1500 =3.9×109, for 2nd stage: NL = 43680×60×(1500/5.2) =7.56×108

The safety factor is based on 5% failure (95 % reliability), estimated KA, with good accuracy

The material selected for the gear is "Surface Hardened, Alloy steel, 708 A42"

The surface fatigue limit σHlim=1160 N/mm2, Bending fatigue limit σFE = 680 N/mm2

It is not specified; gears will require grinding after heat treatment to remove any distortion.

Choose grade accuracy = 5 [Q5].

Helical gears are recommended, it is run quieter

The Permissible surface stress is given by:

$\sigma_{HP} = \sigma_{Hlim}.Z_N.(M_Q)^{0.5}$

B by making the assumption that some pitting is allowed,

By assuming it is better than average quality control and small-batch manufacture

All surface hardened gears, choose $M_Q = 1.15$

Permissible bending stress is given by

$$\sigma_{FP} = \sigma_{FE}.Y_N.Y_X.M_Q$$

where YN life factor for strength (accounting for higher fatigue strength with finite life region less than 3x106 load cycles). For NL = 3.9×109 and 7.56×108, YN = 1.0.

For the 1st stage, assume module size MN = 2, and for the 2nd stage, assume module size MN = 3

YX = 1.0 for the 1st and 2nd stages.

So, for the 1st stage, σ FP = 680×1.0×1.0×1.15=782N /mm2

for the 2nd stage, σ FP = 680×1.0×1.0×1.15=782N

Face-width ratio should be between 0.4d1 and 0.8d1 are desirable.

Choose c = b/d1 = 0.6.

For the 1st stage : TD = 46.8 N.m , n= 1500 rpm, Accuracy grade = Q5, KV = 1.05 (for spur gears)

KVhelical = $1 + 0.6(KV - 1) = 1 + 0.6(1.05 - 1) = 1.03$

For the 2nd stage TD = 238.5 N.m, n= 288.5rpm, Accuracy grade = Q5, KV = 1.02 (for spur gears)

KVhelical = $1 + 0.6(KV - 1) = 1 + 0.6(1.02 - 1) = 1.012$

The KHβ face-load distribution factor for the two stages are:

For the 1st stage, KHβ = 2

This value is larger than 1.6, and some lead correction will be required. For the purpose of this calculations, Assume KHβ = 1.6 can be achieved.

For the 2nd stage , KHβ = 1.6

The estimation of d1 for the two stages are:

$$d_1 = 700.\sqrt[3]{\left[\frac{S_H}{\sigma_{Hper}}\right]^2.\frac{T_D}{c}.\frac{u+1}{u}.K_V.K_{H\beta}}$$

For the 1st stage

$$d_1 = 700 \times \sqrt[3]{\left[\frac{0.98}{1244}\right]^2 \times \frac{46.8}{0.6} \times \frac{5.2+1}{5.2} \times 1.03 \times 1.6} = 31.95 \quad mm$$

For the 2nd stage

$$d_1 = 700 \times \sqrt[3]{\left[\frac{0.98}{1269}\right]^2 \times \frac{238.5}{0.6} \times \frac{4.1+1}{4.1} \times 1.012 \times 1.6} = 54.71 \quad mm$$

Checking for minimum module strength.

$$m_n = \frac{2T_D 10^3}{c{d_1}^2}.\frac{S_F}{\sigma_{FP}}.Y.K_{F\beta}.K_V$$

For the 1st stage,

Y = YF.YS.Yβ = 2.9 for helical, and KFβ = 0.9 KHβ = 1.44

$$m_n = \frac{2\times 46.8\times 10^3}{0.6\times(31.95)^2}\times\frac{1.35}{782}\times 2.9\times 1.44\times 1.03 = 1.135$$

According to mn standard, mn= 1.25

For the 2nd stage

Y = YF.YS.Yβ = 2.9 for helical, and KFβ = 0.9 KHβ = 1.44

$$m_n = \frac{2\times 238.5\times 10^3}{0.6\times(54.71)^2}\times\frac{1.35}{782}\times 2.9\times 1.44\times 1.012 = 1.94$$

According to mn standard , mn= 2

The nearest number of teeth on pinion:

$$z_1 \le \frac{d_1}{m_n}\cos\beta$$

For the 1st stage ,

$$\sin\beta = \frac{\pi.m_n}{b}.\varepsilon_\beta = \frac{\pi\times 1.25}{0.6\times 31.95}1.2$$

$$\beta = 14.23 \approx 14$$

z1 is the number of teeth and β helix angle:

$z1 = \frac{31.95}{1.25} \times \cos 14 = 24.8$, z1 ≈ 25 teeth, wheel teeth $z2 = i1\times z1 = 5.2 \times 25 = 130$

For the 2nd stage

$$\sin\beta = \frac{\pi.m_n}{b}.\varepsilon_\beta = \frac{\pi\times 2}{0.6\times 54.71}1.2$$

$$\beta = 13.3 \approx 13$$

z1 is the number of teeth and β helix angle, $z1 = \frac{54.71}{2} \times \cos 13 = 26.7$, z1 ≈ 27 teeth wheel teeth z2 = i2× $z1 = 4.1 \times 27 = 110.7$ ≈ 111

New 2nd stage transmission ratio i2 = $\frac{111}{27} = 4.11$

New transmission ratio i = $5.2 \times 4.11 = 21.372$.

If the z2 of the 2nd stage has been chosen 110 instead of 111, i2 = 4.074 , thus i = 5.2× $4.07 = 21.2$, (Initial 21.22 :1) .

The preliminary gear specification details are shown in Table 1.

Table 1: Preliminary gear specification details.

	1^{st} Stage	2^{nd} Stage
Type of Gear	Helical, Ext	Helical, Ext
No of teeth : Pinion, z_1	25	27
Wheel, z_2	130	110
Module, m_n	1.25	2
Helix Angle, β	14°	13°
Reference Diameter, d_1	31.95	54.71
Face-width, b	19.17	32.83
Centre distance, a		
Stage ratio, u_i	5.2	4.074
Overall ratio, u	21.2	

4.5. Power requirement

The power requirements are show in Table 2.

Table 2: Power requirements.

Scissor lift motor	Vehicle motor
Power : 3 KW	Power : 5.5 KW
Torque : 15 N.m	Torque : 36 N.m
Speed: 2000 rpm	Speed: 1500 rpm
Voltage : 48 V BLDC	Voltage : 48 V BLDC
Required power : 2.556 [KW]	Required power : 4.8 [KW]

By neglect the winding resistance and the friction:

P electrical = voltage (V) × $Current\ (I)$

For scissor motor, $I = \frac{P}{V} = \frac{2556}{48} = 53.25$ [A]

For vehicle motor, $I = \frac{P}{V} = \frac{4800}{48} = 100$ [A]

These motor does not work together. The maximum current will be consumed will be when the vehicle moving; so, the batteries selection will be respect to vehicle motor.

The battery has been chosen is “8A8DLTP-DEKA, AGM Deep Cycle Battery,12V, 250Ah at C/100 Hr Rate LTP Terminal” so, to get 48V, four 12V batteries will be connected in series.

4.6. Motor control

The control of the three-phase BLDC Motor is sensor less as shown in Figure 9. There are six different positions for the rotor can be defined from the polarities of two coil currents with one coil left unconnected. To make the rotor turning, the currents would be switched in a way that the currents pull the rotor from the current position to the next position. By a switching sequence, each position of the rotor is associated with a configuration of coil currents. The coils are controlled by a PWM block to provide the effective voltage value; these coils are powered through fast switches (power MOSFETs). There is one terminal connected to ground, one terminal is connected to the power supply (circle), and one terminal is left open (terminal name U, V, W) for each commutation step. For a given motor load with a given supply voltage. permanent connection to power supply and ground will turn it with the maximum speed and drives the maximum current through the coils of the motor that is possible for a given motor load with a given supply voltage.

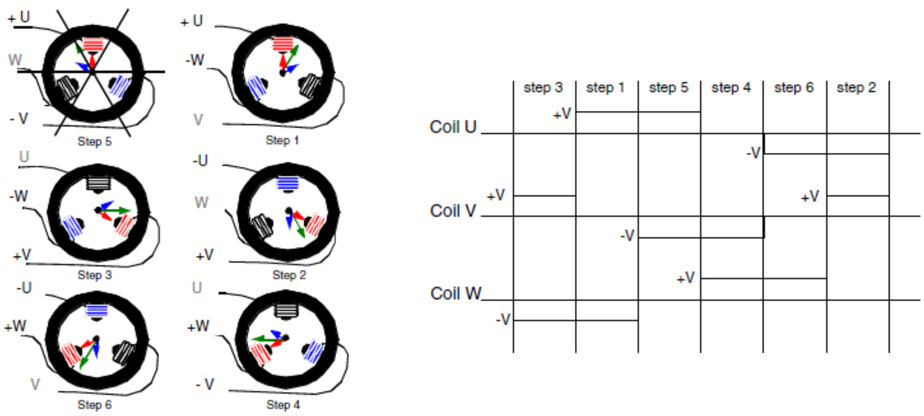

Figure 9: The control of the three-phase BLDC Motor.

5. Conclusion

The aim of this paper was to quantitatively design analysis of several principal components of an electric scissor. A detailed evaluation of the electric scissor principal components design has been discussed in terms of the mechanical and electrical design requirements. In conclusion, the primary aims and objectives of this paper have been successfully accomplished. Since the electric scissor basic components design was effectively developed and all the analyses, showed expected results. However, in order to improve the efficiency of the existing design and due to the aspiration of the scope of this research, further adjustments may be performed in future.

6. Recommendations

- Using gear design software such as the ones provided by Dontyne Systems to improve the calculation speed and the analysis for rapid design and product development.
- Test the system by a simulation software such as Ansys to make it ready for the prototyping.

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