Emergency braking mechanism for an elevator using hydraulic and pneumatic actuation

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Abstract: The proposed idea aims at replacing the current safety mechanisms used by the elevators (i.e., governors) which are rope-based safety mechanism to rope independent, actuated by pneumatic and hydraulic cylinders. The proposed safety mechanism is lightweight and mounted on the top of the elevator. The mechanism involves a pneumatic cylinder actuating a hydraulic master cylinder using a class 1 lever for mechanical advantage. The hydraulic cylinder will pressurise the brake fluid and supply it to the callipers mounted on the elevator body. Callipers will cling to the guide rails upon actuation, thus stopping the elevator in very short distance. The effective force to be stopped by the callipers was calculated and parts were designed accordingly. Analyses of parts are done by considering maximum forces acting on hydraulic calliper mount and lever arm for varying thickness. Through calculations and analyses, the proposed system was found to be safe and reliable.

Keywords: safety mechanism; elevator; pneumatic cylinder; hydraulic master cylinder; class 1 lever; guide rail; hydraulic calliper mount; lever arm.

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Biographical notes: Nitin Rohatgi is final year BTech Mechanical Engineering student of VIT Chennai Campus. During the stay of four years in VIT, he along with Krutarth Mehta and Parijat Sarkar has registered one patent and involved in research activities and projects. This work was carried out by them and guided by Dr. T. Christo Michael.

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1 Introduction

In the age of globalisation and to meet the burgeoning demands for space, high rise buildings have now become a must for any hub to develop. Elevator is the means of transport for passengers up and down the building and, to cope with the demands of the high rise buildings, speeds of the elevators have increased. Along with high speeds, the primary aim while designing any elevator system should be its safety system. As the height of the building increases, the lengths of the ropes needed to support the elevator also increase thus adding extra weight and manufacturing costs. With elevators being a part and parcel of today's life, a reliable emergency braking system is what allows the passengers to use the elevator. A passenger needs to be sure before boarding the elevator that if worst comes to worst, his/her life is not in danger. Keeping this in mind, lots of efforts have been put into devising an efficient and reliable mechanism. Year after year, owing to so many advancements in science and engineering, we witness ever improving braking mechanism. Along the same line, we propose a rope independent stopping mechanism. The proposed mechanism eliminates the usage of ropes in stopping the elevator in case of failures.

A governor based apparatus is currently being used by the elevators as the emergency stopping mechanism (Yunpu and Yixing, 2014). The governor mechanism itself uses a rope which will be under no tension during normal operating condition of the elevator. There are flyweights attached to a spring kept under tension. When the cables break, the governor's sheave starts rotating quickly and due to buildup of centrifugal force the flyweights move outwards. The hooked ends of the flyweights will cling to the ratchets mounted to a stationary cylinder surrounding the sheave. This helps in stopping the governor and the cable attached to it which in turn stops the elevator cabin (Yunpu and Yixing, 2014). Another mechanism used during emergency braking, safety gear mechanism, is employed in the Shanghai Tower's elevator. Al-Kodmany (2015) presents that, whenever the governor reaches its tripping speed, the rope sheave is braked by the sliding brake and engages the safety gear. Some sky scrapers are employing the concepts of rope-less elevators which employ the use of Permanent Magnet Linear Synchronous Motor (PMLSM) for the driving source. Yamaguchi et al. (1996) show a setup, where, in case of power supply interruption, the current will be induced by a secondary permanent magnet and brake force will be generated to stop the elevator in case of an emergency. Zhang et al. (2013) show another setup for rope less elevators. In this setup, in case of power failure, an emergency braking mechanism which employs the use of brake discs and brake pads, will stop the elevator. The brake fluid, which forces the brake pads to move, is pressurised by a variable displacement piston pump. However, as hydraulic systems are slower compared to pneumatic systems, it would take time to build up the pressure of the brake fluid, required to stop the elevator.

The system proposed by us in this paper delves more into protecting the passengers as it can work when the elevator's speed exceeds the specified limits and even when there is no rope attached to the elevator i.e., in the case of wire rope failure. Also, our system uses a series of pneumatic and hydraulic system and avoids usage of pumps, thus rendering it efficient and fast, as no time would be lost in building up the required pressure. In other words, even in the worst case of the breakage of all the ropes, the proposed system would work, and stop the elevator.

2 Methodology

2.1 Alternate mechanism

The existing emergency stopping mechanism of an elevator employs the usage of governors. To remove the need for a governor and to install an emergency braking mechanism which can be operational in a circumstance when all the ropes break, two different mechanisms to stop the elevator are discussed. First, stopping the elevator by solenoid actuation and the other one where the elevator is stopped by using pneumatic and hydraulic actuation. Song and Lee (2015) show the forces produced by solenoid when it is actuated. Both ultimately stopping the lift through hydraulic callipers mounted on the elevator top. In this system i.e., actuation by solenoid, an iron billet was attached to the lever arm on one side and hydraulic cylinder on the other. A solenoid was kept at a distance '*d*' from the lever arm. As soon as, the accelerometer sensor detects the free fall, current is passed through the solenoid coil and a magnetic field is generated outside the coil. Song and Lee (2015) show that the coil attracts the iron billet attached to the lever arm thus actuating the lever arm. Other end of the lever arm will be connected to the hydraulic cylinder. As the iron billet gets attracted, it actuates the hydraulic cylinder thus pressurising the brake fluid present in the reservoir. Smith (1978) illustrates how the brake fluid stored in the reservoir passes to the callipers by using steel braided hoses. The calliper clamps to the guide rails thus stopping the elevator. But, based on the calculations of force generated by a solenoid, it has failed to generate enough force needed to stop the elevator. So an alternative idea was considered to stop the elevator i.e., actuation by using pneumatic cylinder instead of solenoid actuation.

2.2 Proposed mechanism

The system consists of elevator body on top of which the assembly of the emergency braking system will be mounted (see Figures 1 and 2). The emergency braking mechanism's assembly will comprise a pneumatic cylinder with a reservoir storing compressed air, a class 1 lever with a mechanical advantage, a hydraulic master cylinder, a reservoir containing braking fluid, braking callipers, steel braided brake hose, pressure control valve and 3/2-way solenoid valve (see Figures 1 and 2). The pneumatic cylinder would be connected to a compressed air reservoir through a 3/2-way solenoid valve. Situm et al. (2007) discuss a high speed 3/2 way solenoid valve. The paper also discusses the actuation time of the solenoid valve with respect to various voltage supplies and input frequencies. The air pressure provided to the cylinder would be monitored by a pressure control valve. The plunger of the pneumatic cylinder will be connected through a clevis joint to a class one lever arm, and on the other hand of the lever arm a hydraulic master cylinder will be connected. Röske (1997) shows factors which are decisive for determining the lever arm length. The input for the master cylinder would be from a reservoir containing the brake fluid. The hydraulic brake callipers would be connected to the master cylinder through steel braided brake hoses which would carrying the pressurised brake fluid from master cylinder to the brake callipers. The brake callipers will be mounted on top of the elevator body over the guide rails (see Figure 3). Blau (2001) shows the properties of carbon ceramic materials. Durak and Yurtseven (2016) have discussed an experimental setup where they coated a guide rail with three different brake lining materials, and obtained the coefficient of friction and wear of those materials under different working conditions. In our proposed system, the guide rails would be coated with a carbon ceramic material having high coefficient of friction and high thermal resistance.

Figure 1 Isometric view of assembly (see online version for colours)

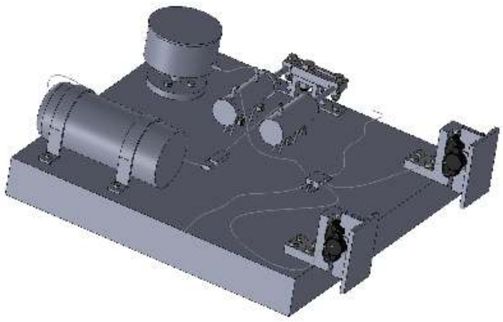


Figure 2 Top view of assembly

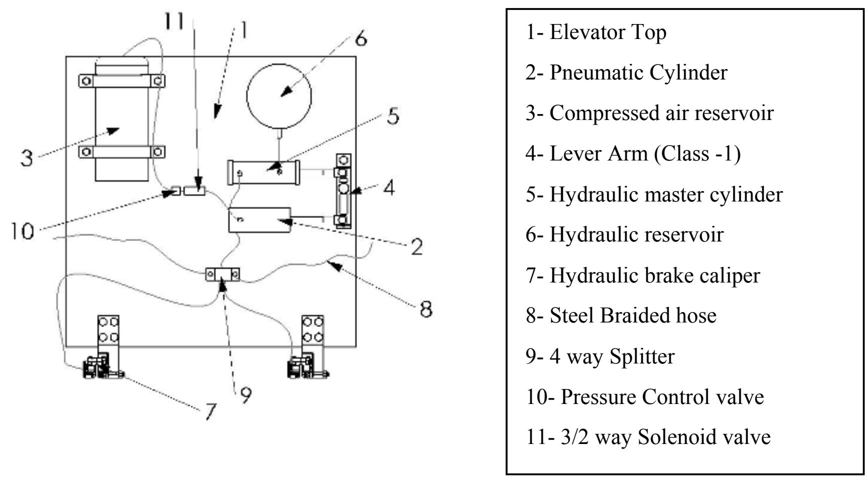
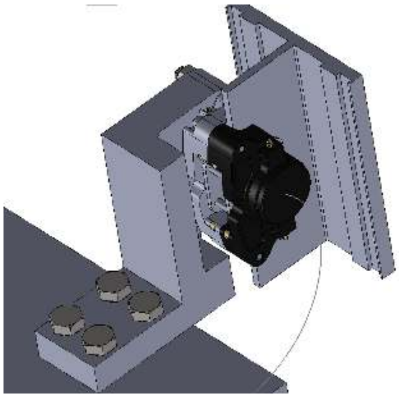


Figure 3 Assembly of hydraulic calliper and guide rail (see online version for colours)



In case of failure an accelerometer sensor mounted on the elevator body would detect the free fall of the elevator and would actuate the 3/2-way solenoid valve and the compressed air with metered pressure would be supplied to the pneumatic cylinder. The compressed air would push the piston to move out actuating the lever with which it is connected through a clevis pin. This in turn would actuate the master cylinder with 3.5 times the supplied force, as the force is multiplied by the lever arm. The hydraulic master cylinder will pressurise the brake fluid into the steel braided brake hoses towards the brake callipers. Once the brake fluid is in the brake calliper, the pressurised fluid will cause the pistons in the calliper to move outwards resulting in clamping of brake pads on the guide rail (see Figure 3) and thus stopping the elevator.

The principle of the proposed stopping mechanism is based on Pascal's law. The multiplication of forces throughout the system helps achieve a great amount of force with just a meagre input force, which renders our system very efficient and cost-effective. To make the system light weight, instead of using compressors on the elevator body, use of compressed air stored in pressure vessels is proposed. To take in account the minimal air leakage from the air reservoirs, the reservoirs will be filled with greater pressure and monitored using pressure control valves. These compressed air reservoirs can maintain the required amount of pressure, and Chen et al. (2016) discusses the prospects of Compressed Air Energy Storage Systems (CAES), in which compressed air is stored in reservoirs and then used to produce electricity via turbines when required. Thus, storing compressed air in a reservoir is a good option, as it would eliminate the need of mounting an air compressor on top of the elevator thus saving the weight.

Based on the following calculations, hydraulic brake callipers produces sufficient clamping force for elevator to stop upon application of brakes, along with an adequate amount of brake fluid being displaced which allows the brake pads to clamp on the guide rails. The analysis done on ANSYS (ANSYS 14.5) warrants the claim by proving the design's robustness under the application of forces acting on parts as in the real life situations. Thus based on calculations and analysis results obtained, it is safe to replace the existing system by the proposed system.

2.3 Analysis methodology

The analyses were carried out in ANSYS static structural. The geometry files were designed in SolidWorks (2016) and then analysed. Fine meshing of the imported geometries was carried out and boundary conditions were applied based on results obtained from calculations. For the hydraulic calliper mount four bottom holes were fixed and forces were applied on the other two holes, where the calliper is mounted (see Figure 4). Total deformation (see Figure 6) and von mises stress (see Figure 7) were obtained for the hydraulic calliper mount. The analysis of the lever arm was also done in a similar way by fixing the fulcrum and applying the force on the holes (see Figure 5), where hydraulic and pneumatic master cylinders will be attached by using clevis pin. The forces were obtained from calculations and applied on the holes. The total deformation (see Figure 10) and von Mises stress (see Figure 11) were obtained for lever arm. Lever arm and hydraulic calliper clamp were analysed for varying thickness in ANSYS and graphs were plotted from the results obtained. Figure 8 shows the graph of deformation versus thickness of the calliper mount. Figure 9 shows the graph of von Mises stress versus thickness of the calliper mount. Figure 12 shows the graph of deformation versus thickness of the lever arm. Figure 13 shows the graph of von Mises stress versus thickness of the lever arm.

Figure 4 Boundary conditions for hydraulic calliper (see online version for colours)

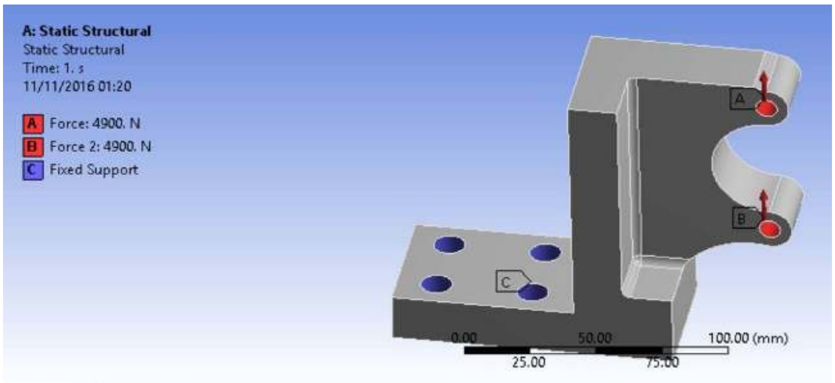


Figure 5 Boundary conditions for lever arm (see online version for colours)

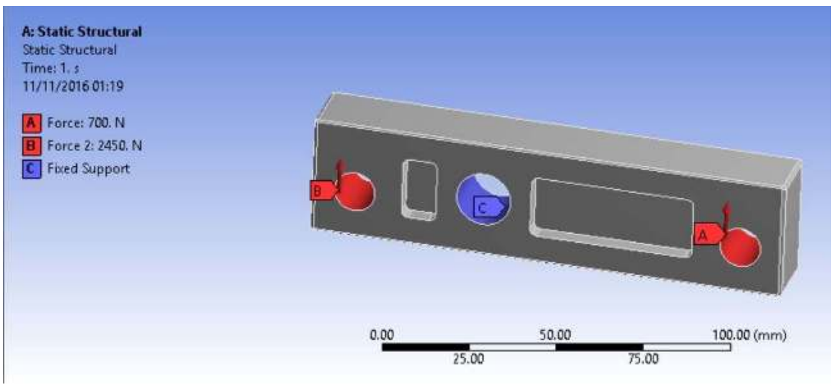


Figure 6 FEM (deformation) of hydraulic calliper mount in ANSYS (see online version for colours)

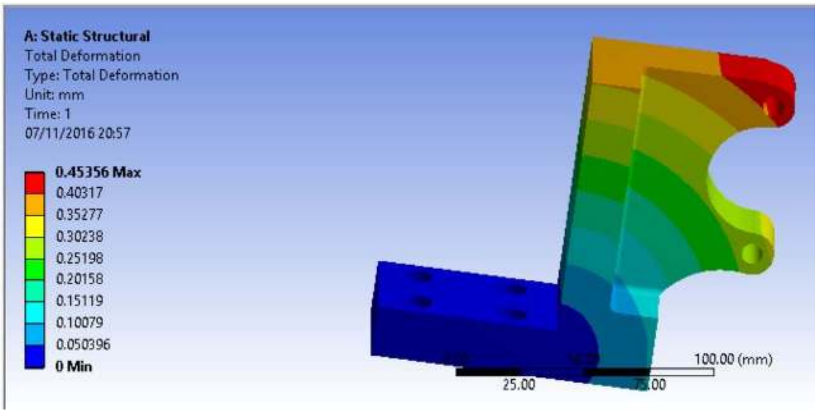


Figure 7 FEM (von Mises stress) of hydraulic calliper mount in ANSYS (see online version for colours)

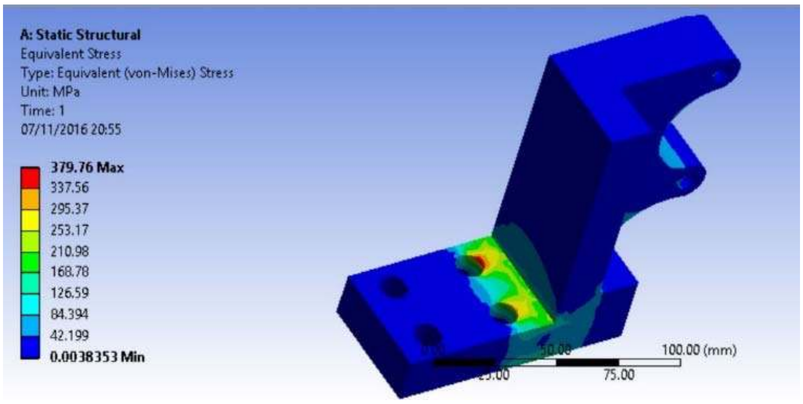


Figure 8 Graph showing deformation versus thickness of hydraulic calliper mount

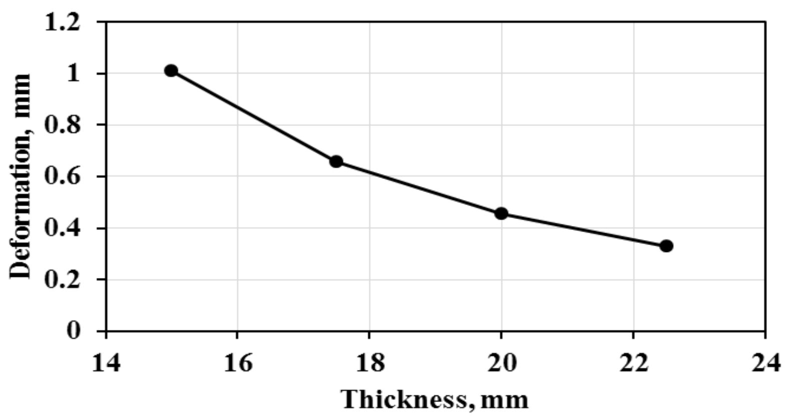


Figure 9 Graph showing von Mises stress versus thickness of hydraulic calliper mount

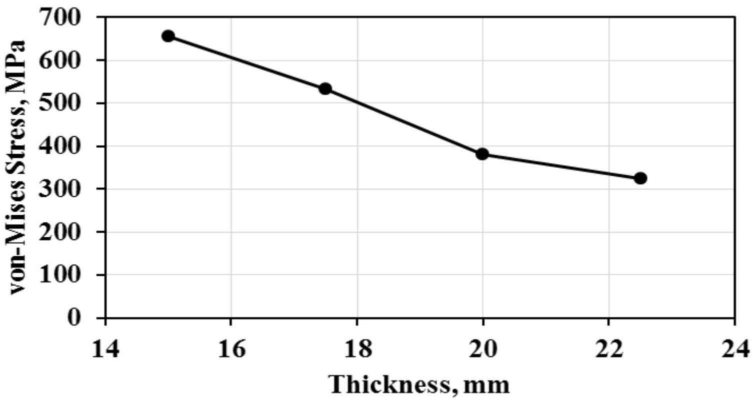


Figure 10 FEM (deformation) of lever arm in ANSYS (see online version for colours)

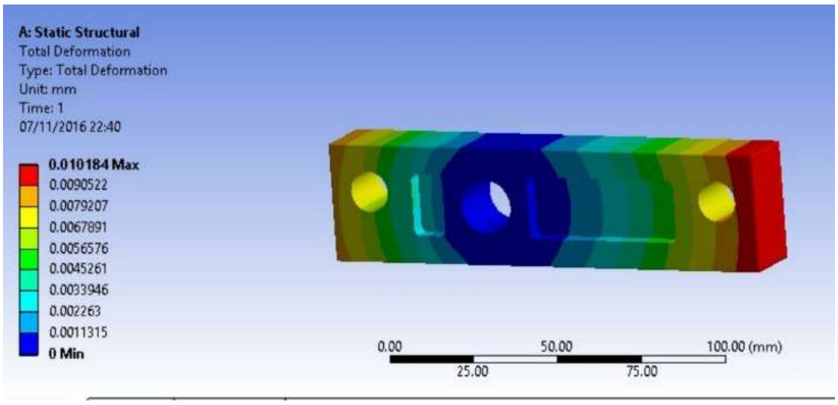


Figure 11 FEM (von Mises) of lever arm in ANSYS (see online version for colours)

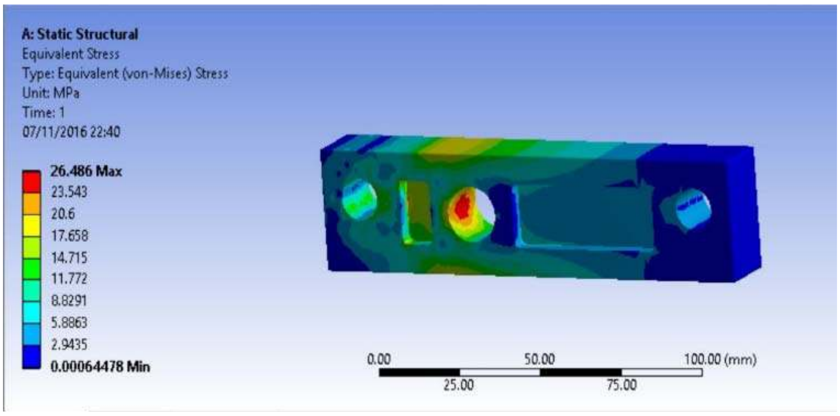


Figure 12 Graph showing deformation versus thickness of lever arm (Class 1)

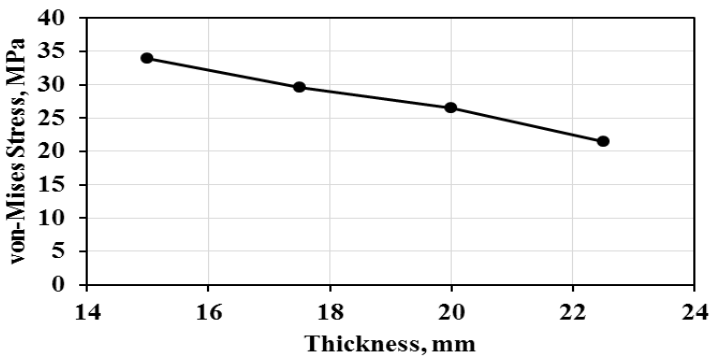
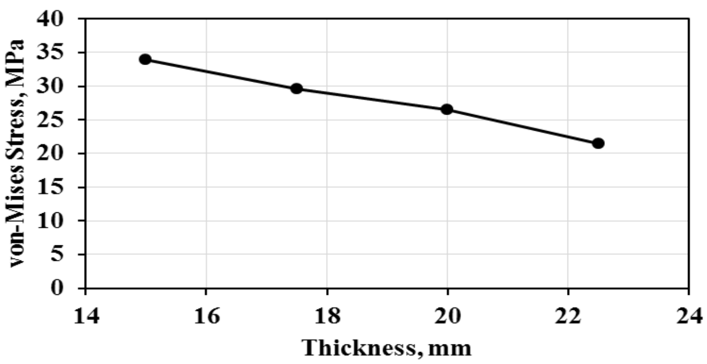


Figure 13 Graph showing von Mises stress versus thickness of lever arm (Class 1)



3 Calculations

3.1 Calculations for mechanism by using solenoid actuation

Richmond (2016) gives the formulae to calculate the amount of magnetic field at point *P* which is *D* away from the edge of the solenoid of length *L* and radius *R* and *N* number of turns. The values assumed for calculations have been discussed in Table 1.

$$B = \frac{\mu In}{2}\left[\frac{D+L}{\sqrt{(D+L)^2+R^2}} - \frac{D}{\sqrt{D^2+R^2}}\right] \quad (1)$$

The force on an iron rod as it's pulled into solenoid is given by

$$F = \frac{\pi R^2 B^2}{2\mu}\left[\frac{\mu'}{\mu} - 1\right] \quad (2)$$

Table 1 Assumed values of parameters used for calculations

Serial No.	*Parameters*	*Values*
1	Elevator cabin weight	2500 kg
2	Maximum velocity of elevator under normal operating condition	1.2 m/s
3	Pressure inside the air compressor reservoir	8 bar
4	Pneumatic cylinder bore	1.5”
5	Mechanical advantage of lever arm	3.5
6	Hydraulic cylinder bore	1.5”
7	Number of pistons in a hydraulic caliper	2
8	Number of callipers	4
9	Coefficient of friction between hydraulic calliper pads and guide rail (Blau, 2001)	0.6
10	Diameter of solenoid	4”
11	Number of turns in solenoid	1000
12	Current supplied to the solenoid	15 A
13	Dimensions of cylindrical iron core	1.8” (Radius) * 10” (Length)
14	Steel, high strength alloy ASTM A514 yield strength	690 MPa

3.2 Calculations for emergency braking mechanism

Puhn (1985) gives the formula for calculating the braking force. The values assumed for calculations have been discussed in Table 1.

$$Ap = \frac{\pi * D1^2}{4} \tag{3}$$

$$Pf = P1 * Ap$$

$$Amc = \frac{\pi * D^2}{4}$$

$$P = \frac{Pf * PR}{Amc} \tag{4}$$

$$Acal = \frac{\pi * d^2}{4}$$

$$Cf = P * Acal * n * n1$$

$$FRclamp = \mu * Cf * n2 \tag{5}$$

3.3 Calculations for elevators’ velocity, acceleration, stopping distance and energy conversation

Equations of motion are given below:

$$v = u + g \times t$$

$$s = ut + \frac{1}{2}at^2$$

$$v^2 - u^2 = 2as$$

$$E = mgh + \frac{1}{2}mv^2 \tag{6}$$

$$S = \frac{v^2}{2\mu g} \tag{7}$$

4 Results and discussions

4.1 Calliper mount

From Figure 9 and Figure 13 it can be inferred that with increase in thickness the stress (von Mises) concentration decreases. Also from data shown in Table 2, the maximum stress build up is 655.31 MPa for the mount having thickness of 15 mm. The analysis results shown in Figures 6 and 7 are for 20 mm thickness calliper mount. The calliper mount is mounted on the top of the elevator body with the help of four bolts. The mount being cantilever, the maximum stress (379.76 MPa) is experienced at the fixed end in Figure 7 in the red region. Rest portion of the calliper mount experiences minor stresses.

From Table 2, it is evident that increasing the thickness of the calliper mount from 15 mm to 22.5 mm the total deformation decreases as shown in Figure 8. The forces were applied on the two holes which will be holding the calliper in upward direction. The maximum deformation in a cantilever is at the free side which is evident from Figure 6. In Figure 6 the maximum deformation is 0.45356 mm in the red region. The end which will be fixed to the elevator body with four bolts shows no deformation.

Table 2 Stress and deformation variation for calliper mount with varying thickness

Calliper Mount		
Thickness (mm)	*Stress (MPa)*	*Deformation (mm)*
15	655.31	1.01
17.5	532.1	0.65595
20	379.76	0.45356
22.5	323.05	0.3297

4.2 Lever arm

For analysis purpose at extreme conditions the lever arm is made fixed in all the directions at the fulcrum i.e., its moment and linear motion along all the three axes are restricted. Forces of 700 N and 2450 N are applied by the pneumatic piston and hydraulic piston on either side of the fulcrum, at the junctions of lever arm and clevis, respectively. The maximum deformation and stress concentration obtained are 0.01485 mm and 33.987 MPa when the thickness of the lever arm is 15 mm. With increase in thickness to 20 mm the stress concentration and maximum deformation have been reduced to 26.486

MPa (see Figure 11) and 0.01018 mm (see Figure 10), respectively. Thus thickness varies inversely with stress concentration and deformation, as evident from Figures 12 and 13 and as the data suggests in Table 3. The maximum deformation of the lever arm is obtained at the farthest end on the pneumatic side (see Figure 10). This is because more bending moment acts on this side because of greater length. Maximum stress concentration (von Mises) of the lever arm is obtained at the circumference of fulcrum lever arm junction (see Figure 11). The reason being moment is about this surface. Fillets and blind pockets have been incorporated for better stress distribution and reduction in weight, respectively.

Table 3 Stress and deformation variation for lever arm (Class 1) with varying thickness

Lever arm (Class 1)		
Thickness (mm)	*Stress (MPa)*	*Deformation (mm)*
15	33.987	0.01485
17.5	29.584	0.012151
20	26.486	0.01018
22.5	21.492	0.0079608

5 Conclusions

The conclusions are summarised as follows:

- Through the calculations and analyses, the proposed design was found to be safe.
- Thickness of the hydraulic calliper mount was chosen to be 20 mm and that of the lever arm to be 17.5 mm. They were chosen such that they do not undergo much deformation during application of the loads and also keeping in mind the maximum yield strength of the chosen material i.e. high strength alloy steel ASTM A514.

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Nomenclature

Magnetic field	B
Current	I
Magnetic permeability	μ_o
Number of turns	n
Radius of solenoid	R
Length of solenoid	L
Force on iron rod when pulled in solenoid	F
Magnetic permeability of a medium	μ'
Initial velocity of elevator cabin	u
Final velocity of elevator cabin	v
Acceleration due to gravity	g
Coefficient of friction	μ
Stopping distance	s
Area of piston	A_p
Diameter of piston	$D1$
Pedal force	Pf
Pedal ratio	Pr
Bore diameter	D
Area of cross section of master cylinder	A_{mc}
Bore diameter calliper	d
Number of pads	$n1$
Number of callipers	$n2$
Clamping force	Cf
Clamping force in friction	FR_{clamp}
Pressure inside the air compressor reservoir	$P1$