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"Design and Analysis of Parking Elevator having Combined Mechanism"

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Abstract:** Growing population is a significant challenge of all over the world, which increasing the number of vehicles day by day, that's reason more parking spaces are required. Parking elevators are a solution of the problem. Various types of parking elevators are available in market like 2 - Post Type, 4 - Post Type, Puzzle Type, Rotary Type, Pit Type, etc. These elevators can be operated using a variety of mechanisms like Chain sprocket, Hydraulic, Lever, Lead screw, Rope pulley. An innovative mechanism for the said application has been thought of, and its design has been validated in the present work. The said mechanism comprises of a lead-screw based assembly to transmit the required torque to lift the elevator platform. Various components are checked for failure criteria based on exploration of a variety of available literature. The design calculations are further cross – verified using CAE analysis. A 3D model has been generated and Analyzed for different load conditions. The results verify the safety of the proposed design.
***Keyword:** Parking elevator, Combine mechanism, Lead screw, Rope pulley, 3-D Modelling, Analysis*

I. INTRODUCTION

Parking elevator is a mechanical system designed to minimize the area and/or volume required for parking cars. E.g., a multi-storey parking garage. A car parking elevator provides parking for cars on multiple levels, stacked vertically to maximize the number of parking spaces with minimization of land usage. Parking elevator are available in different types and it depends on number of cars to be parked at a time and arrangement of parking elevators. The different types are 2 - Post Type, 4 - Post Type, Puzzle Type etc. These elevators can be operated using variety of mechanisms to moved platform which lifts car in vertical direction. Presently, a single mechanism like hydraulic, lead screw etc. is used in most of the parking elevator, but in this work combine mechanism is selected and used to get their advantages for efficient elevator mechanism. Here, lead screw and rope pulley mechanisms are used together to get a combine mechanism which is used to lift a car in vertical direction.

In this paper, design calculation and selection of various components like lead screw, nut, rope, pulley, motor, H-beam (which is use to fabricate frame of elevator) are presented. This parking elevator is designed for lifting time 40 sec., load lifting capacity 3 tons and lift up to 2.1 meter. All the components are verified for safety with overloading criteria. Here different research papers are studied related to lead screw to get clear understanding for type of threads, geometry of thread (different angle, friction coefficient), number of starts and self-locking of lead screw. Different research papers related to rope and pulley are also studied for the concepts like best material combination for rope pulley mechanism, selection for wire core, bending in rope, contact of rope pulley and life of rope.

II. BACKGROUND

As lead screws have different types of threads, selection is based on requirements and application. H. H. R. El-Sayed et al [1] investigated that when externally pressurized, the performance of the Acme screw is better than that of the square type. Load factor increases with respect to increase in helix angle, due to this reason Acme thread is better than square thread. H. A. KHATAAN et al [2] examined standard power screws with thread angle is 14.5° for Acme and 0° for square type, and concluded that the running performance characteristics of the square power screw-nut system may be deduced from the Acme system by introducing the effect of the change of the thread angle. V. Karmarkar et al [3] showed that square threads have best efficiency but are weaker at the root, also they are costly for manufacturing, thicker at the root of thread so they can carry higher load. Vaishali Kumbhar et al [4] reported that Acme threads are stronger than equivalent screw with square threads and for fatigue point of view, Acme threads are preferred. Chances of sudden failures in square threads are higher, so they are not recommended in risky or corrosive environment. K. W. Hollander et al [5] investigated that although a high lead angle can lead to a high efficiency, it can also lead to a system that is "back-drivable". A back-driveable lead screw is a bad idea for a car jack, but is desirable in a wearable robot. For the lead angles

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in which back-drive will occur, is described in following equation, tan α ≥ μ Lead angle α, and coefficient of friction μ. M. Zhang et al [6] studded that the increment of preload alleviates the thread wear through reducing the relative slip between the threads of the bolt and the nut. Thereby, the anti-loosening ability of bolted joints is improved.

Rope is selected based on its type and its core type. D. Zhang et al [7] represented the paper that shows that the fracture of the rope working with steel pulley occurs at positions where wear is most severe, whereas the fracture of the rope working around nylon pulley occurs at the position where it is smooth and complete. The primary fracture form that matched with nylon pulley is fatigue fracture, but the use of the nylon pulley substantially decreases wear in the external wires. The fatigue life of the rope working with nylon pulley is more than twice that of the rope working with steel pulley. S. Takehara et al [8] investigated the motion of the contact part of the pulley and expressed in detail the normal contact and frictional forces between a rope and pulley. It was determined that the rope slips when the ratio of tension is low, and the bending elastic modulus of the rope is large. S. A. Velinsky et al [9] concluded the fiber core is assumed to act in a linear elastic manner with contact pressure loading from the adjacent wire strands, such fiber-core ropes are found in water well service and elevators.

R. Luo et al [10] showed that plate girders are often manufactured with corrugated webs usually of a trapezoidal or other type. The corrugated profile in webs provides a kind of uniformly distributed stiffening in the transverse direction of a girder. L. St-Pierre et al [11] compared plate girders with stiffened flat webs, a girder with a trapezoidal corrugated web enables the use of thinner webs, thus for less cost a higher load-carrying capacity is achieved. J. He et al [12], Y. A. Khalid et al [13] and G. Nirupama et al [14] proposed the synergistic use of partially encased concrete and composite girders with corrugated webs to avoid buckling of corrugated webs and compression flange under large combined shear force and bending moment in hogging area.

III. DESIGN AND CALCULATION

A. Lead Screw

Calculation of lead screw based on lifting capacity 3 ton, lifting height 2.1 meter, lead screw material is EN 8 steel, it is medium carbon steel material and some required material properties are Ultimate tensile strength is 550MPa, Yield tensile strength is 485MPa, Modulus of Elasticity 200GPa.

The axial load (force) is W, tensile in nature and the area which carries the force is the core cross section of diameter d_c, and this diameter is calculated by tensile stress equation, Vaishali Kumbhar et al. [4], A. Panchani et al. [15], V. Bhandari [16]

1) Tensile stress

$$W = \frac{\pi}{4} * d_c^2 * \frac{\sigma_t}{F.O.S}$$

Where, W = 30000 N, σ_t = 550 MPa , F.O.S = 10

$$\therefore d_c = 26.35 \text{ mm}$$

According to standard IS: 4694 – 1968 ACME thread

According to calculation core diameter is 26.35 mm. Selection of thread dimension is done according to standard table, where nearest value of core diameter 27.5 mm but using this value of core diameter, lead screw is not safe in shear and compressive load, so considering next value according to table,

d_{mim} = 29.5 mm, d_{maj}= 36 mm, P = 6mm

2) ACME thread profile

$$h = 0.5P+0.25 \text{ mm} = 3.25 \text{ mm}$$
$$\text{Width} = 0.37P = 2.22\text{mm}$$

3) Mean diameter

$$d = d_{maj.} - \frac{P}{2} = 33\text{mm}$$

4) Helix angle

$$\tan\alpha = \frac{L}{\pi d}$$

Here, L= lead of thread and it depends on number of start.

L = n *P, in this n = number of start = 2

$$\tan\alpha = 0.11,\ \alpha = 6.27$$

For ACME thread,

$$2\beta = 29^\circ,\ \beta = 14.5^\circ$$

5) Co-efficient of friction

The coefficient of friction depends upon various factors like material of screw and nut, quality of lubrication, etc. The coefficient of friction, with good lubrication and average workmanship, may be assumed between 0.10 and 0.15.

$$\mu = \tan \varnothing = 0.12$$

But, for ACME thread

But in case of trapezoidal or Acme thread, the normal reaction between the thread of screw and thread of nut is increased because the axial component of this normal reaction must be equal to the axial load.

$$\mu_1 = \tan \varnothing_1 = \frac{\mu}{\cos \beta} = 0.123, \ \varnothing_1 = 7.012$$

6) Self-locking

Self-locking screw will hold the load in place without any application of torque. It does not need a brake to hold the load.

The condition of self-locking for a lead screw is Co-efficient of friction should be greater than and equal to Helix angle.

$$\varnothing_1 \geq \alpha$$

In this application the above condition is satisfied, co-efficient of friction $\varnothing_1 = 7.012$ is greater than to helix angle $\alpha = 6.27$.

7) Torque

To find the torque (T_1) required to rotate the screw and to find the shear stress (τ) due to this torque.

Load applied on the thread $P = W \tan(\alpha + \phi)$, $= W\left[\frac{\tan\alpha + \tan\varnothing_1}{1 - \tan\alpha * \tan\varnothing_1}\right]$, P = 7081.74 N

$$T_1 = P * \frac{d}{2}, = 116848.71 \text{ N-mm}$$

8) Shear stress

$$\tau = \frac{16 * T_1}{\pi * d_c^3}, \ = 23.18 \text{ N/mm}^2$$

9) Direct compressive stress

$$\sigma_c = \frac{W}{\frac{\pi}{4} * d_c^2}, = 43.89 \text{ N/mm}^2$$

10) Maximum principal stress

$$\sigma_{c(max.)} = \frac{1}{2}\left[\sigma_c + \sqrt{\sigma_c^2 + (4 * \tau^2)}\right], = 53.86 \text{ N/mm}^2$$

11) Maximum shear stress

$$\tau_{(max.)} = \frac{1}{2}\left[\sqrt{\sigma_c^2 + (4 * \tau^2)}\right], = 31.92 \text{ N/mm}^2$$

12) Maximum distortion energy theory

$$\sigma_v = \sqrt{(\sigma_1^2 - \sigma_1, \sigma_2 + \sigma_2^2)}$$

σ_1 *and* σ_2, are calculated by Mohr's circle method

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \mp \sqrt{\left(\frac{\sigma_x + \sigma_y}{2}\right)^2 + \tau_{xz}}$$

Here, $\sigma_x = \sigma_c = 43.8$ N/mm², $\sigma_y = \sigma_b = 0$, $\tau_{xz} = 23.18$ N/mm², (calculated earlier using principal stress theory)

$$\sigma_1 = 53.78 \text{ N/mm}^2 \qquad , \sigma_2 = , 9.98 \text{ N/mm2}$$

So ultimately, von-mises stress according to maximum distortion energy theory

$$\sigma_v = 47.34 \text{ N/mm}^2$$

13) Buckling

When the screw is longer than 8 times the root diameter it must be considered as a column. Bucking effect considered under purely axial loads is represented by Rankine-Gordon.

$$W_{cr} = A_c * \sigma_y \left[1 - \frac{\sigma_y}{4 * C * \pi^2 * E} * \left(\frac{L}{K}\right)^2\right]$$

Where, A_c = 679 mm²

σ_y = 485 MPa

C = factor to allow the column end support

Both end are fixed, so C=4

L = 2100 mm, E = 200 GPa, K = 0.25*dc

$$W_{cr} = 75588.45 \text{ N}$$

B. Nut

For nut using AISI 1026 Steel which is low carbon steel material, ultimate tensile strength is 490MPa, Yield tensile strength is 415MPa, Modulus of Elasticity 200GPa. To calculate the height of nut (h), bearing pressure is required..

Bearing pressure on the nut,

$$P_b = \frac{W}{\frac{\pi}{4} * [d_m^2 - d_c^2] * n}$$

Where n = Number of threads in contact with screwed spindle,

$$n = 9.97 \approx 10 \text{ Thread}$$

Height of nut

h = n *p = 60 mm

Thickness of screw

$t = \frac{P}{2}$ = 3mm

Stresses in the screw and nut

$$\tau_{(screw)} = \frac{W}{\pi * n * d_c * t} = 10.79 \text{ N/mm}^2$$

$$\tau_{(nut)} = \frac{W}{\pi * n * d * t} = 9.65 \text{ N/mm}^2$$

Now, the inner diameter (D1) is found by considering the tearing strength of the nut.

$$W = \frac{\pi}{4} * [D_1^2 - d_m^2] * \frac{\sigma_t}{F.O.S}$$

$$D_1 = 45.5 \text{ mm} \approx 46 \text{ mm}$$

The outer diameter (D2) is found by considering the crushing strength of the nut collar.

$$W = \frac{\pi}{4} * [D_2^2 - d_1^2] * \frac{\sigma_t}{F.O.S}$$

$$D_2 = 53.8\text{mm} \approx 54 \text{ mm}$$

The thickness (t_1) of the nut collar is found by considering the shearing strength of the nut collar.

$$W = \pi * D1 * t_1 * \tau$$

$$t_1 = 8.66 \text{ mm} \approx 10 \text{ mm}$$

C. Rope

The following standard procedure was used designing a wire rope, using IS: 3938-1967 standard, referred by Haideri et al. [17] For elevator application, most commonly 6*19 type wire ropes used. According to application mechanism, class 1 is selected and it depend on running hours per day, minimum 0.5 hour.

Design load = 2.5 * load to be lifted *factor of safety

Considering the above wire rope type and mechanism class, standard Factor of safety = 4 and load = 30000/4 = 7500 N (In this application 4 wire ropes are used to distribute entire load) Design load = 2.5 *7500* 4 = 75000 N

Selection of wire rope diameter “d”:-

For design load of 75000 N, for diameter of rope 10mm selected from the standard table.

Cross section area “A”:- $A = 0.4 * \frac{\pi}{4} * d^2$, A = 31.41 mm^2

So wire diameter “d_w”:- $d_w = \frac{d}{1.5 * \sqrt{i}}$, Here i =number of strands * number of wires in each strand,

$$d_w = \frac{10}{1.5 * \sqrt{6 * 19}}$$

$$d_w = 0.624 \text{ mm}$$

Maximum static load,$F_{max} = \frac{S * A}{FOS}$ = 9862.74 KN

Substituting this value in equation = $\frac{2 * F_{max}}{S_u * d * D}$ = 0.0069,

Based on this value, form the graph, Number of bend is 50,000 for 6*19 type wire rope, assume approximate 10 bend per day, tentative life of this wire rope, = 50000/ 10 = 5000 days, so wire rope life is 13.5 years.

$$\text{Bending Load: - } F_B = \frac{A * E * d_w}{D} = 21777.6 \text{ N}$$

$$\text{Breaking strength of the wire rope } F_u = 510 * d^2 = 51000 \text{ N}$$

D. Pulley

Using the selected type of wire rope selection D/d ratio from **IS: 3938 – 1967**and using the ratio "D" is calculated, for the selected wire rope 6*19F for class 1

$$\frac{D}{d} = 18,\ D = 10 * 18,\ D = 180\ mm$$

Based on this, other dimension of the pulley are selected from the table

a = 40 mm,
b = 30 mm
c = 7 mm
h = 25 mm
l = 10 mm

E. H-beam

Columns fail by buckling when their critical load is reached. Long columns can be analysed with the Euler's column formula.

$$W = \frac{n * \pi^2 * E * L}{L^2}$$

Where, W = allowable load = 30000 N = 30000*10 (hear, 10 = F.OS), n = factor accounting for the end conditions, one end free: n = 0.25, E = modulus of elasticity = $200*10^9$ pa, L = length of column = 3 m, I = Moment of inertia

Based on this all value, found moment of inertia,

$$I = 1.3678 * 10^{-6}\ m^4 = 136.78\ cm^4$$

H beam having closes Moment of Inertia to the calculated value h beam UB 203 * 133 *25 is selected from the industrial standards table.

F. Motor Selection

$$\text{Screw efficiency:-}\ \eta_1 = \frac{\tan\alpha * (1 - \mu * \sec\theta * \tan\alpha)}{(\mu * \sec\theta + \tan\alpha)}$$

Here, $\alpha = 6.27°$, $\theta = 14.5°$, $\mu = 0.12$

$$\eta_1 = 46.34\%$$

Form application, rope and pulley diameter ratio D/d = 180 / 10 = 18, from rope and pulley diameter ratio v/s power transmit efficiency graph $\eta_2 = 92\ \%$

Combined efficiency of these two mechanism $\eta_1 * \eta_2 = 42.63\ \%$

This mean approximate overall efficiency is 42 % (by considering all power transmit loss of elevator in motor, rope, pulley, lead screw).

Means approximate 58 % loss in power transfer.

Power required to lift a car,

P = work / time (Work = F * d = 63000 Nm)

P = 63000 / 40 = 1575 W

Based on this all these values,

$$\text{Motor Power} = \frac{\text{Reqired Power}}{\eta_{mechanisum}} = \frac{1575}{0.45} = 3714\ W$$

3714 W For this select 5 HP (3730W) Motor.

IV. 3-D MODELLING AND ANALYSIS

A. 3-D Modelling

Based on calculation and available standard data create 3-D model of various components are created for proper visualization in software. These models are used for simulation and stress analysis and the design calculations are further cross verified.

The primary structural component for this parking elevator is H beam which is modelled as shown in showing fig. 1 and table 2.

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TABLE 1. H-beam Parameters

H- beam Parameters	Total Depth 'H'	Flange Width 'B'	Thickness of Web 't_w'	Thickness of Flange 't_f'	Root Radius 'r'	Area of section 'A'
Value	152.4mm	152.2mm	5.8 mm	6.8 mm	7.6 mm	29.25cm^2

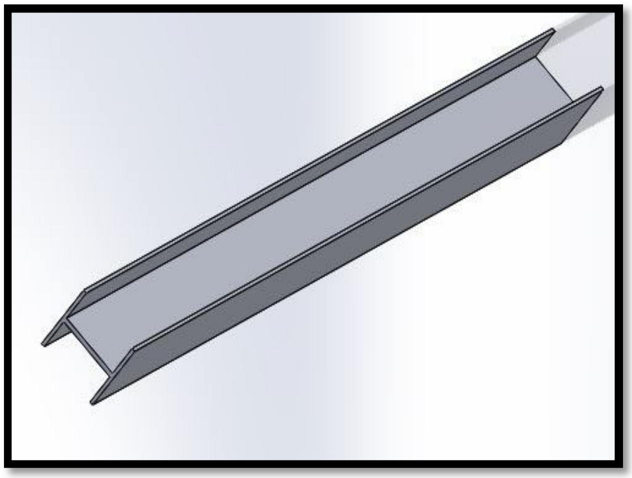

Fig 1. 3-D Model of H-beam

Another important component in the parking elevator unit is the lifting platform. Its model is created based on the data shown in table 2 and its model is shown in fig 2.

TABLE 2. Platform Parameters

Platform Parameters	Plate Thickness	Plate Support	Length	Corrugated Distance
Value	1.5 mm	3 mm	3500 mm	100 mm

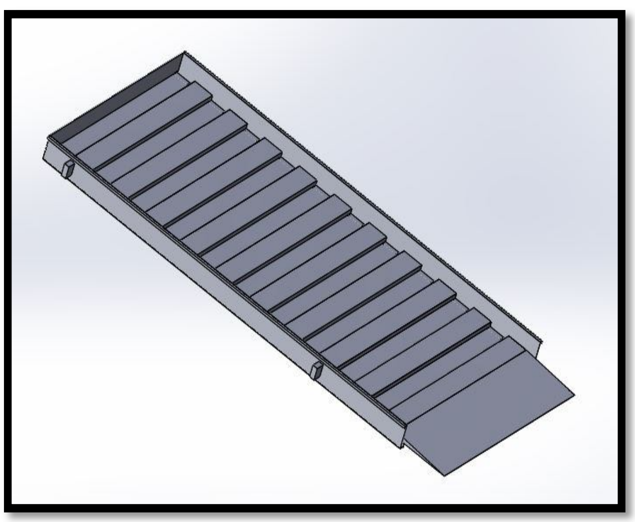

Fig. 2 3-D Model of Platform

B. Analysis

Based on analysis done in CSE software different types of results were attained. The results include deformation, stress, strain, etc. The figure shown below represent the results of H-beam that include total deformation, maximum principal stress and Von-Mises stress respectively.

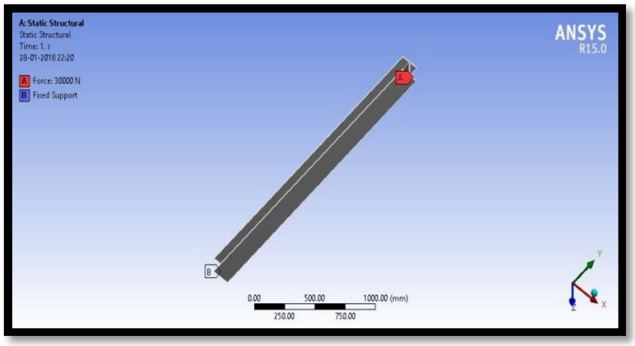

Fig. 3 Load apply on h-beam

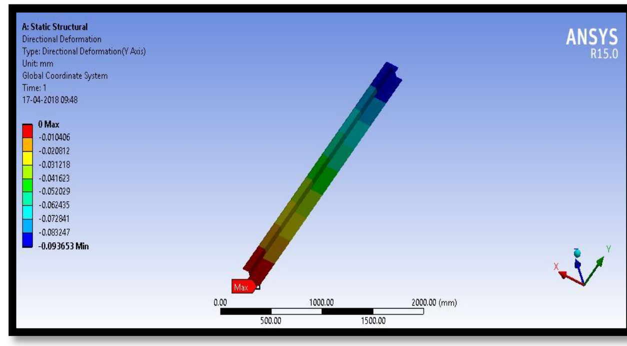

Fig. 4 Total Deformation of H-beam

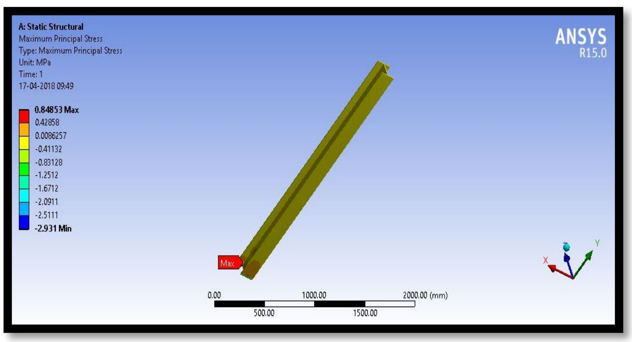

Fig. 5 Maximum Principle Stress of H-beam

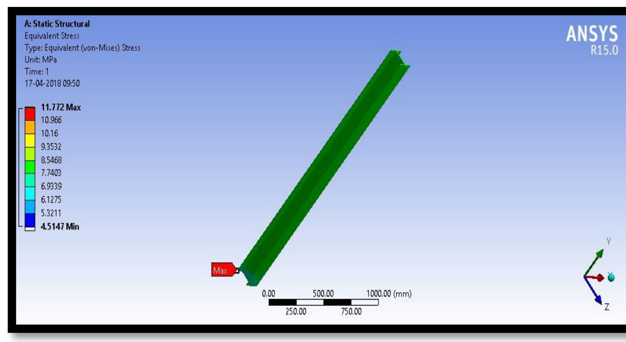

Fig. 6 Von-Mises Stress of H-beam

The following figures show the total deformation, maximum principal stress and Von-Mises stress respectively for the platform.

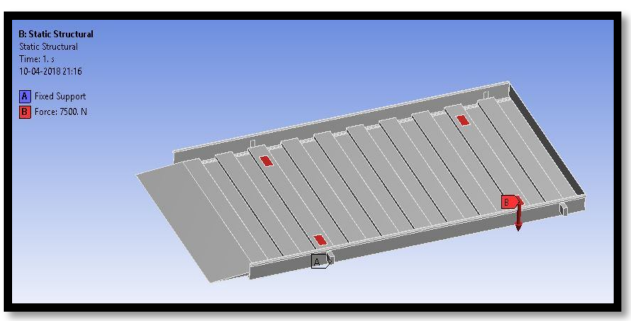

Fig. 7 Load apply on Platform

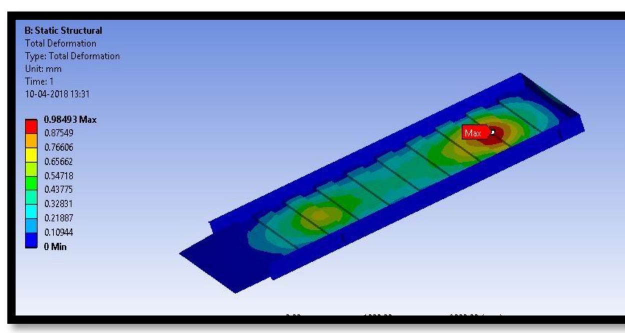

Fig. 8 Total Deformation of Platform

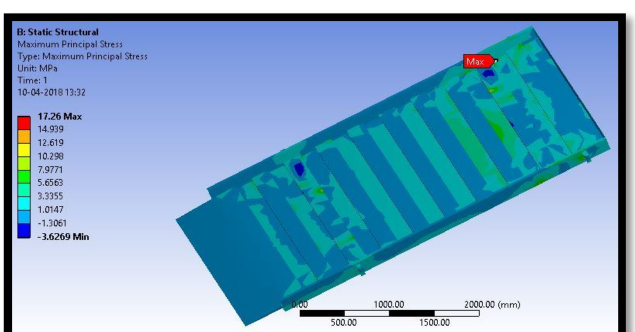

Fig. 9 Maximum Principle Stress of Platform

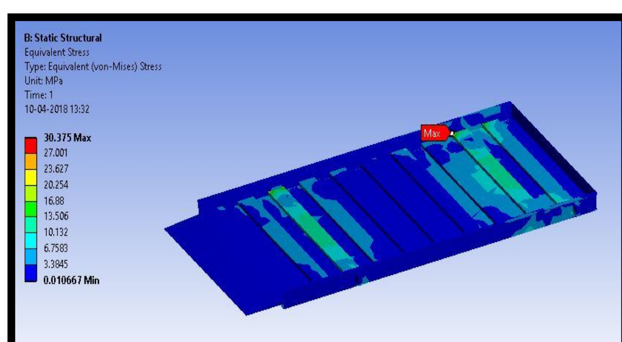

Fig. 10 Von-Mises Stress of Platform

V. RESULTS AND DISCUSSION

A. From the design calculation, theoretical of failure for lead screw occurs at Maximum principal stress 53.86 N/mm², Maximum shear stress 31.92 N/mm² and Von-mises 49.54 N/mm² for 36 mm major diameter ACME thread lead screw.

B. Based on calculations wire rope, the Rope diameter achieved is 10 mm and the approximate life of the said rope is 13.5 years. For each rope Bending Load is 21777.6 N and breaking strength of the wire rope is 51000 N. Pulley is directly selected from the standard data based on this wire rope.

C. H-beam is selected based on moment of inertia calculated by Euler's column formula and this moment of inertia is compared with industrial available beam and closest H-beam is selected. The analysis of H-beam gives the following results: total deformation, Maximum Principle Stress is 0.84 N/mm² and Von-Mises Stress is 11.77 N/mm².

D. Corrugated type platform is selected for lighter weight. The analysis of corrugated type platform gives the following results: total deformation is 0.98 mm, Maximum Principle Stress is 17.26 N/mm² and Von-Mises Stress is 30.37 N/mm².

E. Overall system efficiency is 42 %, select 5 HP (3730W) Electrical Motor.

VI. CONCLUSION

It is conclude from the results that selected and designed components for the parking elevator unit are safe under applied load, they have suitable life-span and fulfil all the basics requirements.

1) For lead screw maximum principal stress, shear stress and Von-Mises stress are less than the permissible stresses, and it is safe under maximum load. From Rankine-Gordon equation permissible critical load is higher than applied load so found to be lead screw is safe under buckling.

2) For 10 mm diameter wire rope life is 13.5 years. For each rope, Bending Load is 21777.6 N and breaking strength of the wire rope is 51000 N. For application, permissible applied load is 30,000N.

3) It was conclude from analysis of H-beam that selected dimensions are safe. The analysis show negligible deformation, Maximum Principle Stress and Von-Mises stress are below permissible limit for applied load.

4) Selected Corrugated platform is safe applied under stress, and it is verified through in analysis.

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