DESIGN, ANALYSIS AND FABRICATION OF ROTARY ASCENDING LIFT SYSTEM

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Abstract

The Rotary Ascending Lift System is an inventive vertical transportation solution that combines simplicity, adaptability, and efficiency. This innovative system utilizes a screw rod mechanism for vertical movement and employs worm gear and worm wheel for power transmission. It will be operated using an Electric Motor, making it a versatile and accessible choice for various applications. At its core, this lift system relies on a precision-engineered screw rod mechanism. When activated by an Electric Motor, the screw rod rotates, enabling the lift system to ascend or descend along its threaded shaft. This design ensures a controlled and secure vertical motion, enhancing passenger safety and comfort. To optimize power transfer from the Electric Motor to the screw rod, a set of gears with a worm and a worm wheel are meshed seamlessly and integrated into the system. These gears amplify the torque applied to the screw rod, guaranteeing a smooth and reliable vertical movement. The standout feature of this Rotary Ascending Lift System is its accessibility and adaptability. This versatility along with elimination of hydraulic system makes it suitable for a broad spectrum of applications, from warehouses to construction sites and beyond. In conclusion, the Electric Motor-Powered Rotary Ascending Lift System presents an innovative approach to vertical transportation. Its integration of a screw rod mechanism and worm gears for efficient power transmission, and the ease of operation with an Electric Motor with an option of converting it into manual operation using a crank during emergency

situations, promising improved mobility and convenience, even in challenging environments.

INTRODUCTION

- In today's industrial landscape, efficient and reliable vertical transportation systems play a vital role in optimizing productivity and ensuring seamless operations. This project report focuses on the design of a lift system that combines the power of worm gear system with screw rod to achieve a maximum working height of around 12ft with a load capacity of 150kgs. The objective of this project is to create a robust and versatile lifting solution that meets industry standards.

- The lift system's design utilizes a combination of worm gears and screw rod to achieve the desired lift. The unique design of worm gears results in a perpendicular arrangement of the input and output shafts, providing reliable torque transfer and precise control. The incorporation of screw rod allows for linear motion, enabling the lift system to efficiently raise and lower heavy loads.

- To ensure the strength and durability of the lift system, a careful selection of materials has been made. The design incorporates the usage of EN8 steel, Phosphor Bronze, ASTM-A500, Grey Cast Iron (FG-150), SAE 2320 (Case hardened) Alloy Steel and Structural Mild Steel. EN8 steel, renowned for its balance of strength and machinability along with high wear resistance, which makes it a suitable material for screw rod. Phosphor Bronze is known for its usage in worm wheel because of its softness and its ability to adjust with the plastic deformation and Case-Hardened Alloy Steel for worm gears is known for its surface endurance and energy absorption, reinforces power transmission components to withstand heavy loads and ensure long-term performance. Additionally, ASTM-A500 for columns, known for its high strength, corrosion resistance along with its weldability further enhances the lift system's durability and reliability.

- The significance of this project lies in its potential impact on diverse industries. Construction sites, warehouses, manufacturing facilities, and logistics centres are just a few fields that could greatly benefit from a versatile and reliable lifting solution. By optimizing the lift system's design, operations in these sectors can be streamlined, leading to improved productivity and enhanced workplace safety.

- **OBJECTIVES**
- The objective of this project is to design and build a lift pod that utilizes a combination of screw rod and worm gears for a maximum vertical working height of 11.5ft as the best alternative for podiums and ladders.
- Simulate and analyze the lift pod’s mechanical behavior.
- Fabrication of the lift system prototype and conduct comprehensive performance testing.

METHODOLOGY

Problem statement: Designing a Lift System with Screw Rod and Worm Gears System for a load capacity of 150kg and reach maximum working height of about 12ft in less than 1 minute using an Electric Motor as power source.

Solution –

• **Needs Assessment and Goal Definition:** Identify the specific requirements and objectives of the vertical transportation solution. Understand the target applications, load capacity, height requirements, and other key parameters. Define the project goals and constraints.

• **Conceptualization and Design Phase:** Brainstorm and develop conceptual designs for the lift system. Consider various mechanisms, materials, and configurations. Evaluate the feasibility of different power transmission systems and choose the screw rod mechanism with worm gear and worm wheel for vertical movement.

• **Engineering and Detailed Design:** Develop detailed engineering plans and specifications based on the selected concept. This includes precise dimensions, materials, load-bearing capacity calculations, safety features, and integration of the screw rod mechanism with worm gears and worm wheel.

• **Material Selection:** Choose appropriate materials for durability, strength, and weight considerations. Iteratively refine the design based on results.

• **Power Transmission System Integration:** Integrate the worm gear and worm wheel power transmission system with the screw rod mechanism. Ensure compatibility and efficiency in converting rotary motion from the Electric Motor into controlled vertical movement.

• **Safety and Compliance Checks:** Conduct thorough safety assessments and ensure compliance with relevant industry standards and regulations.

• **Capacity & Compatibility Testing:** Validate the lift system's maximum working height and load-bearing capacity & verify the compatibility and efficiency of the system with an Electric Motor.

PRODUCT DESIGN

SCREW ROD DESIGN –

σ_y for EN8(Hardened) = 465 MPa   FOS(n) = 2

$\sigma_{allow} = \frac{\sigma_y}{n}$

= 232.5 MPa

$\sigma_{max} = \sigma_{direct} + \sigma_{bending}$

$\sigma_{max} = \frac{W}{\pi d2/_4} + \frac{Mb\ X\ y}{I}$ (M_b = W x e)

$232.5 = \frac{2700}{\pi d2/_4} + \frac{2700X650X32}{\pi d3}$

$232.5\ \pi d^3$ = 1000d + (2700x650x32)

d = 42.5331 mm

According to the standards from Design Data Handbook, nearest Minor Diameter = 51mm

The supposed to be dimension as per stresses induced or involved in the

Nominal Dia(d)	Bolt Dia.	Nut Dia.(D)	Minor Dia.(d_1)	Core Area(A_c)
52mm	52mm	52.5mm	44mm	1521mm^2

Checking for buckling of screw rod:

Rankine Method:

Length(l) = 1160mm Effective Length(l_e) = 2 x 1160mm (∵One free & one fixed end)

σ_c = 232.5MPa   E= 190x10^3 MPa   Area(A) = $\frac{\pi * d_1^2}{4}$ a = $\frac{\sigma_c}{\pi^2 * E}$

Radius of Gyration(K) = $\sqrt{\frac{I}{A}} = \frac{d_1}{4}$ Eccentricity (e) = 650mm

$$P_{cr} = \frac{\sigma c.A}{\left(1+\frac{ey}{K}\right)\left(1+a\left(\frac{l}{K}\right)^2\right)}$$

Diameter	Force
44mm	455.44 N
48mm	683.21 N
54mm	1174.38 N
58mm	1623.92 N
64mm	2522.01 N
68mm	3294.70 N

Pitch (P) = 14mm   tanα = $\frac{l}{\pi d_2}$ Ø = $\tan^{-1}$μ   μ =0.17(Grease)

Torsional Moment (M_{ts}) = $F_t * \frac{d_2}{2}$ Torsional Shear Stress(τ) = $\frac{M_{ts} * r}{J}$

$r = \frac{d_1}{2}$ $J = \frac{\pi * d_1^4}{32}$ $\sigma_{max} = \frac{\sigma_c}{2} + \tau_{max}$ $\tau_{max} = \frac{1}{2} * \sqrt{\sigma_c^2 + 4\tau^2}$

Single Start

(l)=14mm

tanα = $\frac{14}{\pi \text{ x } 71}$ = 0.0628

α = 3.590

F_t = W x $\left(\frac{\tan\alpha+\mu}{1-\mu.\tan\alpha}\right)$

= 2700 x $\left(\frac{0.0628+0.17}{1-(0.17 * 0.0628)}\right)$

= 635.22N

To raise the load

M_t = W x $\frac{d}{2}(\tan(\phi + \alpha))$

M_t = 24773.64N-mm

To lower the load

M_t = W x $\frac{d}{2}(\tan(\phi - \alpha))$

M_t = 11173.46N-mm

Check for self-locking of screw rod –

Since tanα = 0.0628 < μ = 0.14 the screw rod is self-locking

Efficiency of screw rod

$\eta = \frac{\tan\alpha}{\tan(\alpha+\phi)}$

$\eta = \frac{0.0628}{\tan(3.590+9.65)}$

η = 0.2667 = 26.67%

Torsional Shear Stress(τ) = 0.44 N/mm^2

Maximum Principal Shear Stress(τ_{max}) = 0.61 N/mm^2

Principal Normal Stress ($\sigma_{\max}$) = 1.03 N/mm^2

NUT DESIGN –

Material – Grey Cast Iron (FG150 ~ ASTM 20)

D = 78.5 mm Weight on Nut = 340 kgs

σ_t = 150MPa σ_c = 600MPa

Width across flats = 1.5(D) + 3mm = 120.75mm

Chosen Width across flats = 138mm (Column Size Constraints)

Thickness of nut (T) = 0.9D to D = 78mm

l_n = i x P ➔ $i = \frac{l_n}{p} = \frac{78}{12}$ = **5.571**

i ~ 6 [Next even no.]

l_n = 6x14 = **84mm**

$i = \frac{4W}{\sigma_b' \pi (d^2 - d_1^2)}$

σ_b' = **0.3629 MPa**

Take FOS = 3.5 for σ_t Take FOS = 6 for σ_c

$\sigma_{t\ nut} = \frac{150}{3.5}$ = 42.86 MPa $\sigma_{c\ nut} = \frac{600}{6}$ = 100 MPa

$\tau_{nut} = 0.5\ \sigma_{t\ nut} = 21.43$ MPa

Tearing Strength: (W) = $\frac{\pi}{4}(D_1^2 - D^2)\ \sigma_{t\ nut}$

D_1 = <u>79.14mm</u>

Since we already have higher width of nut, it is safe.

Crushing strength: W = $\frac{\pi}{4}(D^2{}_2 - D^2{}_1)\ \sigma_{c\ nut}$

D_2 = **<u>79.41mm</u>**

∴ The design is safe.

<u>INNER COLUMN DESIGN –</u>

Material – IS4923/ASTM-A500

B = H = 150mm b = h= 138mm Thickness = 6mm

σ_y = 315 MPa n=2 E = 190 GPa σ_{allow} = 157.5 MPa

Length of Inner Column = 1600mm

Length after extension = 2887mm

Effective Length(l_e)= 2887x2 = 5774mm

Area(A) = (150x150) – (138x138) = 2900 mm^2

$\sigma_{max} = \sigma_a + \sigma_b = \frac{W}{A} + \frac{M_b}{y}$ x y

y = $\frac{B}{2} = \frac{150}{2}$ = 75mm

I = $\frac{BH^3 - bh^3}{12}$ = **<u>10174166.67 mm^4</u>**

$\sigma_{max} = \frac{3600}{2336} + \frac{(3600 * 650) * 75}{10174166.67}$ = **<u>18.79 MPa</u>**

Check for buckling of column:

<u>Rankine's Buckling:</u>

$$P_{cr} = P_{cr} = \frac{\sigma_c \text{ x A}}{\left(1+\frac{ey}{K}\right)\left(1+a\left(\frac{l}{K}\right)^2\right)}$$

P_{cr} = 17054.93 N

Parameter	Dimension
Outer Depth(H) = Outer Breadth(B)	150mm
Inner Depth(h) = Inner Breadth(b)	138mm
Thickness(t)	6mm
Length of Column	1600mm
Cross-section Area of Column	2900mm^2
Critical Load(P_{cr})	17054.93N

OUTER COLUMN DESIGN –

Material – IS4923/ASTM-A500

B = H = 180mm b = h= 168mm Thickness = 6mm

σ_y = 315 MPa n=2 E = 190 GPa σ_{allow} = 157.5 MPa

Length of Outer Column = 1550mm

Length after extension = 1550mm

Effective Length(l_e)= 1550 x 2 = 3100mm

Area(A) = (150x150) – (138x138) = 4176 mm^2

$\sigma_{max} = \sigma_a + \sigma_b = \frac{W}{A} + \frac{M_b}{y} \text{ x } y$

$y = \frac{B}{2} = \frac{180}{2} = 90\text{mm}$

$I = \frac{BH^3 - bh^3}{12}$ = **21097152 mm^4**

$\sigma_{max} = \frac{3100}{4176} + \frac{(3100 * 650) * 90}{21097152}$ = **9.06MPa**

Check for buckling of column:

Rankine's Buckling:

$P_{cr} = P_{cr} = \frac{\sigma_c \text{ x A}}{\left(1+\frac{ey}{K}\right)\left(1+a\left(\frac{l}{K}\right)^2\right)}$

P_{cr}= 45083.74 N

Parameter	Dimension
Outer Depth(H) = Outer Breadth(B)	180mm
Inner Depth(h) = Inner Breadth(b)	168mm
Thickness(t)	6mm
Length of Column	1550mm
Cross-section Area of Column	$4176mm^2$
Critical Load(P_{cr})	45083.74N

INNER & OUTER COLUMN AVAILABLE STANDARDS –

PROPERTIES OF APL APOLLO STRUCTURA (SHS) IS : 4923 : 1997/EN 10219-1 : 2006*/ASTM A-500												
SHS	Thickness	Sec Area	Unit Wt	Moment of Inertia		Radius of Gyration		Elastic Modulus		Torsional Constants		Outer Surface Area per m
				Ixx	Iyy							
D X B	t	A	W			rxx	ryy	zxx	zyy	J	B	
mm	mm	cm^2	kg/m	cm^4	cm^4	cm	cm	cm^3	cm^3	cm^4	cm^3	m^2
100 x 100	5	18.36	14.41	271.1	271.1	3.84	3.84	54.22	54.22	441.84	441.84	0.374
	6	21.63	16.98	311.47	311.47	3.79	3.79	62.29	62.29	511.8	511.8	0.369
113.5 x 113.5	4.8	20.28	15.92	393.3	393.3	4.4	4.4	69.3	69.3	637.45	637.45	0.429
	5.4	22.6	17.74	432.58	432.58	4.38	4.38	76.23	76.23	708.69	708.69	0.426
132 x 132	4.8	23.83	16.71	634.39	634.39	5.16	5.16	96.12	96.12	1018.3	1018.3	0.503
	5.4	26.59	20.88	700.11	700.11	5.13	5.13	106.08	106.08	1134.25	1134.25	0.5
150 x 150	4	22.95	18.01	807.82	807.82	5.93	5.93	107.71	107.71	1273.46	1273.46	0.579
	5	28.36	22.26	982.12	982.12	5.89	5.89	130.95	130.95	1569.09	1569.09	0.574
	6	33.63	26.4	1154.91	1154.91	5.84	5.84	152.7g	152.7g	1856.18	1856.18	0.569
	7	38.78	30.44	1299.44	1299.44	5.79	5.79	173.26	173.26	2134.99	2134.99	0.564
	8	43.79	44.38	1443	1443	5.74	5.74	192.4	192.4	2405.78	2405.78	0.559
180 x 180	4	27.75	21.78	1421.74	1421.74	7.16	7.16	157.97	157.97	2224.31	2224.31	0.699
	5	43.36	26.97	1736.87	1736.87	7.11	7.11	192.99	192.99	2747.93	2747.93	0.694
	6	40.83	32.05	2036.52	2036.52	7.06	7.06	226.28	226.28	3259.23	3259.23	0.689
	7	47.18	33.03	2321.04	2321.04	7.01	7.01	257.89	257.89	3758.53	3758.53	0.684
	8	53.39	41.9	2590.73	2590.73	6.97	6.97	287.86	287.86	4246.16	4246.16	0.679
220 x 220	4	34.15	26.61	2639.14	2639.14	8.79	8.79	239.92	239.92	4099.49	4099.49	0.859
	5	42.36	33.25	3238.02	3238.02	8.74	8.74	294.37	294.37	5076.22	5076.22	0.854
	6	50.43	39.59	3813.36	3813.36	8.7	8.7	346.67	346.67	6034.73	6034.73	0.849
	7	58.38	45.83	4365.55	4365.55	8.65	8.65	396.67	396.67	6974.82	6974.82	0.844
	8	66.19	51.96	4894.99	4894.99	8.6	8.6	445	445	7897.48	7897.48	0.839
250 x 250*	4	38.95	30.57	3907.3	3907.3	10.02	10.02	312.58	312.58	6045.4	6045.4	0.979
	5	48.36	37.96	4805.01	4805.01	9.97	9.97	384.4	384.4	7494.83	7494.83	0.974
	6	57.63	45.24	5672	5672	9.92	9.92	453.76	453.76	8920.44	8920.44	0.969
	7	66.78	52.42	6508.73	6508.73	9.87	9.87	520.7	520.7	10322.7	10322.7	0.964
	8	75.79	59.5	7315.65	7315.65	9.82	9.82	585.25	585.25	11702.07	11702.07	0.959

BASE TROLLEY DESIGN –

Material – Mild Steel

σ_y = 315MPa FOS(n) = 1.5

Allowable Strength (σ_a) = 210 N/mm^2 Weight on base = 390 kg Force on base(F) = 3900 N

Dimensions:

B = H = 50mm b = h = 42mm Thickness (t) = 4mm

Length of Tube (L_t) = 1100mm Area (A_b) = 736 mm^2

Distance from Neutral Axis (c) = $\frac{B}{2}$ = **25mm**

Moment of Inertia (I) = $\frac{BH^3 - bh^3}{12}$ = **261525.33 mm^4**

Total Stress (σ_{tot}) = $\frac{F}{A} + \frac{(F * L_t) * c}{I}$ = **415.39 N/mm^2**

Number of Tubes taking on Load (n) = 2

Stress after distribution = $\frac{\sigma_{tot}}{n}$ = **207.70 N/mm^2**

∴ The design is safe.

BASE PLATE DESIGN –

Material – Mild Steel

σ_y = 315MPa FOS(n) = 1.5

Allowable Strength (σ_a) = 210 N/mm^2

Force (F) = 2000 N

Factored Axial Load (C_f) = 3900 N

Factored Moment (M_f) = $F * e$ = 1300 N-m

Resistance Factor of Concrete (Φ_c) = 0.65

Resistance Factor of Steel (Φ_s) = 0.9

Yield Strength of Base Plates (F_y) = 300 N/Mm^2

Compressive Strength of Concrete (F_c) = 20 N/Mm^2

Width of Column (b) = 150mm

Depth of Column (d) = 150mm

Width of Base Plate (B) = 250mm

Depth of Base Plate (D) = 250mm

Resistance Factor for Anchor Bolts (Φ_{ab}) = 0.67

Yield Strength of Anchor Bolts (F_u) = 465 N/Mm^2

C/C of Anchor Bolts In X-Direction (d'x) = **<u>210mm</u>**

Edge Distance from Face of Base Plate (E) = $\frac{B*d'X}{2}$ = **<u>20mm</u>**

Cantilever Parallel to Plate Side (m) = $\frac{D-d}{2}$ = **<u>50mm</u>**

Thickness of Plate (t) = $\sqrt{\frac{2*\phi_c*0.85*F_c*m^2}{\phi_c*F_y}}$ = **<u>14.30mm</u>**

No. Of Concrete & Sandwich Plates (n) = 2

Minimum Thickness of a Plate (t_{min}) = $\frac{t}{2}$ = **<u>7.15mm</u>**

Chosen Plate Thickness (T) = <u>10mm</u>

<u>PLATFORM DESIGN –</u>

Material – Mild Steel

σ_y = 315MPa FOS(n) = 1.5

Allowable Strength (σ_a) = 210 N/mm^2

Weight on base = 200 kg   Force on base = 2000 N

Dimensions:

B = H = 50mm   b = h = 42mm   Thickness (t) = 4mm

Length of Tube (L_t) = 660mm   Area (A_b) = 736 mm^2

Distance from Neutral Axis (c) = $\frac{B}{2}$ = **<u>25mm</u>**

Moment of Inertia (I) = $\frac{BH^3-bh^3}{12}$ = **<u>261525.33 mm^4</u>**

Total Stress (σ_{tot}) = $\frac{F}{A} + \frac{(F*L_t)*c}{I}$ = **128.90 N/mm²**

Number of Tubes taking on Load (n) = 2

Stress after distribution (σ_{dis}) = $\frac{\sigma_{tot}}{n}$ = **64.45 N/mm²**

∴ The design is safe.

POWER TRANSFER DESIGN –

Mass (M) = 390 kgs = 0.39 tones

Velocity (v) = 4m/min

Gear Stages (n) = 1

Motor Service Factor (C_v) = 0.67 (Considering IS3938 & IS3177)

Duty Factor (C_{df}) = 1.7

Efficiency (E) = 90%

Derating Factor (D_f) = 1

Power (P) = 0.32 kW

Power Considered for Calculation (P) = **0.4 kW**

kW = $\frac{MVC_vC_{df}}{6.12\ E} X \frac{1}{C_{amb}}$

P = $\frac{2\pi NT}{60000}$

Here, 0.32 Kw is not sufficient considering the torque required for the screw rod. Therefore, 0.55kW motor is obtained (0.75hp) which offers sufficient torque according to the design.

$$T = \frac{0.55\ X\ 60000}{2\ X\ \pi\ X\ 960}$$

T = **5470.24 N-mm**

Motor RPM	960	RPM
HP	0.75	HP
kW	0.55	kW
Shaft Dia.	19	mm

KEY DESIGN –

Output Shaft Diameter (d) = 40mm

Key material – EN8 Steel

Ratio of gear (i) = 5

Input Torque = 45000 N-mm

M_t = Ratio of gear * Input Torque = **32821.45 N-mm**

According to standards, for d = 38 to 44mm

Width of key (b) = **12mm** Height of key (h) = **8mm** Length of key (l) = **28 to 140mm**

Depth of keyway in shaft = **5mm** Depth of keyway in hub = **3.3mm**

Bearing Stress (σ_b')

σ_y = 385 MPa

Factor of safety (n) = 2

σ_{allow} = 192.5 MPa

$\sigma_b' = \frac{4*M_t}{h*l*d}$

When l = 100mm

σ_b' = **4.103 MPa**

When l = 40mm

σ_b' = **10.257 MPa**

Shear Stress (τ_s)

τ = 192.5 MPa

Factor of safety (n) = 2

τ_{allow} = 96.25 MPa

$\tau_s = \frac{2*M_t}{b*l*d}$

When l = 100mm

τ_s = **1.368 MPa**

When l = 40mm

τ_s = **3.419 MPa**

SHAFT DESIGN –

Output Shaft Design

Highest input torque possible (T_{maxi}) = 5470.24 N-mm

Highest output torque (T_{maxo}) = T_{maxi} * I = 32821.45 N-mm

Shear stress of screw rod shaft (τ_{ed}) = $\frac{\sigma_a}{2}$ = 116.25 N/mm^2

Minimum Diameter (D_{min}) = $\sqrt[3]{\frac{16*T_{max}}{\pi*\tau_{ed}}}$ = 15.32 mm

Diameter considered for manufacturing (D) = **40 mm**

Input Shaft Design

Highest input torque possible (T_{maxi}) = 5470.24 N-mm

Highest output torque (T_{maxo}) = T_{maxi} * I = 32821.45 N-mm

Shear stress of screw rod shaft (τ_{ed}) = $\frac{\sigma_a}{2}$ = 86.25 N/mm^2

Minimum Diameter (D_{min}) = $\sqrt[3]{\frac{16*T_{max}}{\pi*\tau_{ed}}}$ = 9.31 mm

Diameter considered for manufacturing (D) = **24 mm**

WORM GEAR DESIGN –

Center Distance (a) = 64.5mm n_1 = 960RPM i=6

$i = \frac{n_1}{n_2} = \frac{z_2}{z_1} = \frac{\pi d_2}{P_z}$

$6 = \frac{960}{n_2}$ n_2 = 160 RPM $d_1 = \frac{a^{0.875}}{1.5}$ = 25.54mm

$a = \frac{d^1+d^2}{2}$ = 2(64.5) – 25.54 = 103.45mm = d_2

$d_1 = 3\pi m \qquad 25.54 = 3\pi m$

Calculated module (m) = 2.71mm

$i = \frac{\pi d_2}{P_z} = \frac{\pi d_2}{\pi z_1 m_x} = \frac{d_2}{m z_1}$

$d_2 = imz_1 \qquad d_2 = 18z_1$

z_1	1	2	3	4	5	6
d_2	18	36	54	72	90	108

d_2 = 90mm (Since this is the closest)

$d_1 = 2a - d_2$ = 129-90 = 39mm

(d_1 = 24mm according to Indian standard)

$\gamma = \tan^{-1}\left(\frac{m z_1}{d_1}\right) = \tan^{-1}\left(\frac{3x5}{39}\right) = 21^{o}$

DIMENSIONS OF WORM

Parameter	Formula	Dimension
Module(m)		3mm
Lead Angle(γ)		21^0
No. of starts (z_1)		5
Pitch Diameter (d_1)		39mm
Face length (L_1)	$(4.5+0.02z_1)\pi m$	43.36
Depth of tooth (h_1)	$0.623\pi m$	5.8716mm
Addendum (h_{a1})	$0.286\pi m$	2.6954mm
OD of worm (d_{a1})	$d_1 + 2h_{a1}$	44.39mm
Dedendum(h_{f1})	$(2.2\cos\gamma - 1)m$	3.160mm
Root dia. of worm (d_{r1})	$d_1 - 2h_{f1}$	32.68mm
Diametral Quotient	$\frac{d_1}{m}$	13

DIMENSIONS OF WORM WHEEL

Parameter	Formula	Dimension
No. of teeth (z_2)		30 teeth
Pitch Diameter (d_2)	$m \times Z_2$	90mm
Face Width (b)	$2.15\pi m + 5$	25.26 (41.5mm for manf.)
Addendum (h_{a2})	$m[2\cos\gamma - 1]$	2.6mm
OD of worm wheel (d_{a2})	$d_2 + 2h_{a2}$	95.2 mm
Dedendum (h_{f2})	$m[1 + 0.2\cos\gamma]$	3.56mm
Root dia. of worm wheel (d_{r2})	$[1+0.2\cos\gamma]$	82.88mm

Checking for input capacity

Material for worm wheel – Phosphor Bronze

(σ_{02}= 55MPa)

$Y = \pi y_2$

$y_2 = 0.154 - \frac{0.912}{z_2} = 0.124$ (20^0 Full Depth)

$Y = \pi \times 0.124 = 0.3884$

$V_m = \frac{\pi d_2 n_2}{60000} = \frac{\pi x 90 x 160}{60000} = 0.75$ m/s

Considering dynamic effect, $C_v = \frac{6}{6+V_m} = \frac{6}{6+0.5105} = 0.89$

Tangential Load on tooth (F_{t2}) = $\sigma_{02}.m.b.Y.C_v$

F_{t2} = **2362.34 N**

$F_d = \frac{F_{t2}}{C_v}$ = **2659.23N**

Wear Load (F_w) = d_2bK

K form T(23-100) – Hardened steel + phosphor bronze & $\gamma \geq 25^0$

K = 0.827

F_W = 90x41.5x0.827 = **3088.85N**

Since $F_W > F_d$, the design is safe

Power Capacity based on strength

$P = \frac{F_{t2} n_2 r_2}{9550 x 1000} = \frac{2362.34 \ x \ 160 \ x \ 45}{9550 \ x \ 1000}$ = **1.78 kW**

According to AGMA standards, IP based on wear,

$P = \frac{n_1 x K x Q x C_v}{i}$

$Q = \frac{i}{i+2.5} = \frac{6}{8.5} = 0.71$

C_v – Velocity factor based on speed of worm

$C_v = \frac{2.3}{2.3+V_w+3\frac{V_w}{i}}$ $V_W = \frac{\pi d_1 n_1}{60000} = 1.961$ m/s

$C_V = \frac{2.3}{2.3+1.961+3*\frac{1.961}{5}} = 0.439$

K – Pressure const. for a = 64.5mm from AGMA Standards

K= 0.0184 kW/RPM

$P = \frac{960}{6}$ x 0.0184 x 0.71 x 0.439= **0.912 kW**

Input power based on heat dissipation

According to AGMA standards,

$P = \frac{3650\ a^{1.7}}{i+5}$ (a in meters)

$P = \frac{3650(\frac{64.5}{1000})^{1.7}}{6+5}$ = **3.142 kW**

Least value of power input to be chosen

∴ Input power capacity based on wear = **0.912 kW**

Efficiency of the gear system

$\gamma = 21^0$ Pressure angle (α_n) = 20^0

$\eta = \frac{tan\gamma[\cos\alpha_n.\cos\gamma - \mu.\sin\gamma]}{cos\alpha_n.sin\gamma+\mu.cos\gamma}$

$V_r = \frac{\pi d_1 n_1}{60000*cos\gamma} = \frac{\pi*39*960}{60000*cos(21)}$ = **2.1006 m/s**

$\mu = \frac{0.0422}{V_r^{0.28}}$ for $0.2<V_r<2.75$ m/s

$\mu = \frac{0.0422}{(2.1006)^{0.28}}$ = **0.0343**

$$\eta = \frac{\tan(21)[(cos20*cos21)-(0.0343*sin21)]}{(cos20*sin21)+(0.0343*cos21)}$$

$$\eta = \frac{\tan(32)[(0.87727)-(0.012292)]}{(0.33675)+(0.032021)} = 0.9006$$

η = **<u>0.9006 = 90.06 %</u>**

Heat Balance

Heat Generated (H_g) = $\frac{\mu F_n V_r}{1000}$

$$F_n = \frac{F_t}{cos\gamma . cos\alpha_n} = \frac{2362.34}{cos21.cos20} = \underline{\mathbf{2693.48\ N}}$$

$$H_g = \frac{0.0343\ x\ 2693.48\ x\ 2.1006}{1000}$$

H_g = **<u>0.194 kW</u>**

Since the equipment is not continuously in action and heat generated is very low, no artificial cooling is required.

CHAIN STANDARDS –

Chain Ref.		Technical Details (mm)													Connecting Links					
Renold Chain No.	ISO Ref.	Pitch (inch)	Pitch (mm)	Inside Width MIN	Roller Diam. MAX	Plate Height MAX	Pin Diam. MAX	Pin Length MAX	Conn. Pin Length Spring Clip MAX	Conn. Pin Length Cotter Type MAX	Conn. Pin Length Scrl Type MAX	Tensile Strength (Newtons) MIN	Transverse Pitch NOM	Weight kg/m	No. 4	No. 107	No. 11	No. 26	No. 12	No. 30
European (BS) Standard - Simplex																				
		A	A	B	C	D	G	H1					K							
R 957*	06B-1	0.375	9.525	5.900	6.35	8.16	3.28	13.30	14.30	14.50	15.96	920	-	0.42	✓	✓	✓	✓	✓	✓
R 1230	81	0.500	12.700	3.600	7.75	9.90	3.66	9.90	10.90	11.40	12.40	840	-	0.30	✓	✓	✓	✓	✓	✓
R 1224	82	0.500	12.700	2.400	7.75	9.90	3.66	8.20	-	-	-	840	-	0.27	✓	✓	-	-	-	-
R 1249	83	0.500	12.700	4.900	7.75	10.36	4.09	12.90	14.00	-	15.65	1360	-	0.48	✓	✓	-	✓	✓	✓
R 1248H	84	0.500	12.700	4.900	7.75	11.10	4.09	14.80	15.6	-	16.30	1630	-	0.58	✓	✓	-	✓	✓	✓
R 1278	08B-1	0.500	12.700	7.850	8.51	11.71	4.45	16.80	18.10	-	21.20	1840	-	0.71	✓	✓	-	✓	✓	✓
R 1595	10B-1	0.625	15.875	9.850	10.16	14.66	5.08	19.50	20.60	-	22.50	2310	-	0.96	✓	✓	-	✓	✓	✓
R 1911	12B-1	0.750	19.050	11.70	12.07	16.10	5.72	22.40	36.60	-	25.95	2990	-	1.20	✓	✓	-	✓	✓	✓
R 2517	16B-1	1.000	25.400	17.50	15.88	20.70	8.27	35.70	43.60	38.90	41.00	6120	-	2.75	✓	✓	✓	✓	✓	✓
R 3119	20B-1	1.250	31.750	19.56	19.05	25.90	10.17	41.85	-	45.10	46.80	9690	-	3.70	✓	✓	✓	✓	✓	✓
R 3825	24B-1	1.500	38.100	25.40	25.40	33.30	14.63	52.90	-	59.30	60.00	16310	-	7.22	✓	✓	✓	-	✓	✓
R 4431	28B-1	1.750	44.450	30.99	27.94	37.20	15.90	64.90	-	71.00	71.80	20390	-	9.34	✓	✓	✓	-	✓	✓
R 5031	32B-1	2.000	50.800	30.99	29.21	41.50	17.81	65.10	-	72.80	73.00	25500	-	10.05	✓	✓	✓	-	✓	✓
R 6338	40B-1	2.500	63.500	38.20	39.37	52.20	22.89	82.00	-	90.70	92.60	36200	-	16.00	✓	✓	✓	-	✓	✓
R 7645	48B-1	3.000	76.200	45.80	48.26	63.80	29.24	99.00	-	108.20	109.10	57100	-	25.40	✓	✓	✓	-	✓	✓

CHAIN & SPROCKET DESIGN –

Chain drive and standards of R-1278 copy data

CHAIN DRIVE DESIGN

N=0.55kW n_1=960rpm n_2=960rpm

Pitch of chain

P $\leq 25\left(\frac{900}{n_1}\right)^{2/3}$

$\leq 25\left(\frac{900}{960}\right)^{2/3}$

$\leq$ 23.95mm

Number of teeth on the sprockets

$$\frac{n_1}{n_2} = \frac{960}{960} = 1$$

From table 21.60 for n_1/n_2 = 1, select number of teeth on the smaller sprocket z_1 = 21

Now, $\frac{n_1}{n_2} = \frac{z_2}{z_1}$

$$\frac{960}{960} = \frac{z_2}{21}$$

Number of teeth on larger sprocket z_2 = 21

Pitch Diameter

$$d = \frac{p}{sin\left(\frac{180}{z}\right)}$$

Pitch diameter of input sprocket d_1 -

$d_1 = \frac{p}{sin\left(\frac{180}{z_1}\right)} = \frac{12.7}{sin\left(\frac{180}{21}\right)}$ = 85.21

Pitch diameter of output sprocket d_2 -

$d_2 = \frac{p}{sin\left(\frac{180}{z_2}\right)} = \frac{12.7}{sin\left(\frac{180}{21}\right)}$ = 85.21

Velocity

$$v = \frac{pz_1n_1}{60000} = \frac{12.7\ X\ 21\ X\ 960}{60000} = 4.27\ m/sec$$

Required Pull

Power $N = \frac{F_\theta . v}{1000k_l k_s}$

K_l = load factor = 1.75

K_s = service factor = 0.15

$$0.55 = \frac{F_\theta X 4.27}{1000X1.75X0.15}$$

$$F_\theta = 24.61\ N$$

Allowable Pull

$F_r = \frac{F_u}{n_0}$, where n_0 = working factor of safety

From table for n_1 = 960 rpm and p = 12.7 take working factor of safety as 11

$n_0 = 11$

$F_r = \frac{F_u}{n_0} = \frac{1840}{11} =$ **167.27 N**

Number of strands

$i = \frac{F_\theta}{F_r} = \frac{24.61}{167.27} = \mathbf{0.15 \sim 1}$

Check for actual factor of safety

Actual factor of safety $n_a = \left[\frac{F_u}{F_\theta + F_{cs} + F_s}\right].i$

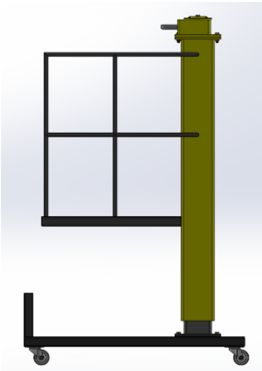
$$F_\theta = \frac{1000XN}{v} = \frac{1000\ X\ 0.55}{4.27} = 93.74$$

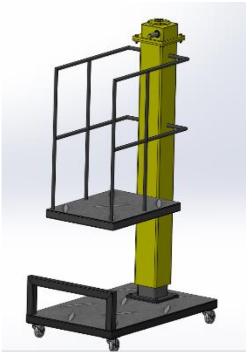
$$F_{cs} = \frac{wv^2}{g} = \frac{25X4.27^2}{9.81} = 45.67\ N$$

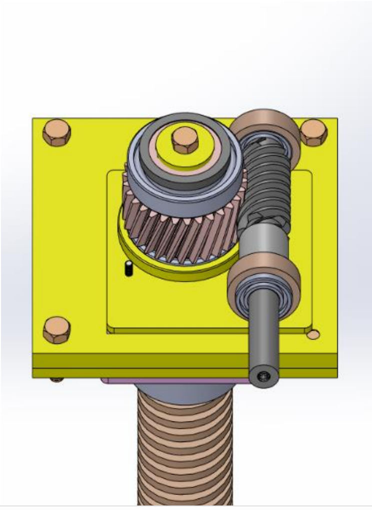
$$F_s = k_{eq}wC = 6X25XC = 13.53\ Nn_a = \left[\frac{1840}{93.74 + 45.67 + 13.53}\right]X1 = 12.03$$

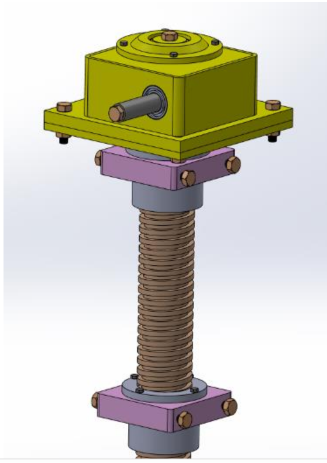
Since $n_a > n_o$, the selection of the chain is safe.

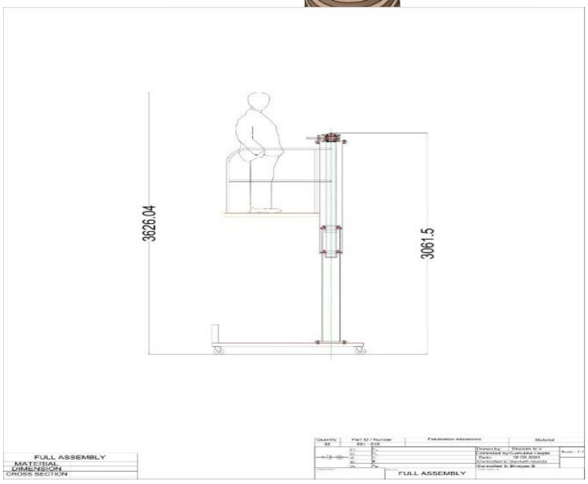
FINAL ASSEMBLY DRAWINGS AND MODELS

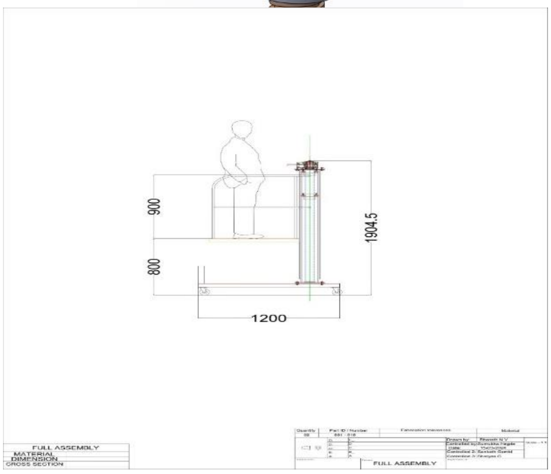












CONCLUSION

In conclusion, this project report has presented the design and development of a lift system with a maximum lift height of 12ft and a load capacity of 150kgs. The primary objective was to create a reliable and efficient lifting solution as a replacement for ladders and podiums.

The design process involved a thorough analysis of the requirements, safety considerations, and technical specifications. Several design iterations were implemented, focusing on structural integrity, stability, and ease of use. Through meticulous engineering and integration of various mechanisms, we have successfully achieved the desired lift height and load capacity.

However, it is important to note that the current design is considered a baseline, and further optimization is still underway. The project team is committed to enhancing the lift system's performance, efficiency, and overall user experience. This ongoing optimization includes exploring opportunities for weight reduction, energy efficiency improvements, and enhanced control systems.

The project has been a significant learning experience, requiring interdisciplinary collaboration, extensive research, and innovation. The lift system's potential applications are vast, ranging from warehouses and construction sites to industrial facilities and beyond. The optimized lift system will not only contribute to increased productivity and safety but also minimize physical strain and streamline operations.
In conclusion, the design of the lift system represents a crucial milestone, showcasing the team's expertise in engineering and problem-solving. The project's success has laid a solid foundation for further optimization and the eventual deployment of an exceptional lifting solution that meets the demands of various industries. With continuous refinement, the lift system has the potential to revolutionize vertical transportation and provide significant value to a wide range of sectors.

WHAT IS THE INNOVATION IN THE PROJECT?

This invention deals with solutions for vertical transportation system. The Rotary Ascending Lift System is a novel vertical transportation solution featuring a screw rod mechanism powered by an Electric Motor. It utilizes worm gears for efficient power transmission, ensuring controlled and secure vertical movement. Its adaptability makes it suitable for diverse applications, from warehouses to construction sites and beyond. The system offers improved mobility and convenience, with the option for manual operation using a crank during emergencies. Overall, it presents an innovative approach to vertical transportation, promising enhanced safety and efficiency.

This equipment saves cost in terms of maintenance by eliminating hydraulic systems. Saves set-up time unlike traditional podiums and brings in a lot of safety when compared to the ladders. Instead of using manpower for set-ups, this can be solely used by the operator. Power consumption is very low for load transportation when compared to the movement speed. A revolutionary integration of screw rod and worm gear has offered a lot of advantages in terms of power transmission and load carry capacity with minimal wear. Number of forklifts required in the warehouses can be reduced by replacing it with the rotary ascending lift system when the purpose is checking the requirement, data collection or adjustments.

CLAIMS

1. The Rotary Ascending Lift System has a unique design of the screw rod which has the capacity to provide both strength to the structure and transfer power for the lifting mechanism.
2. The Rotary Ascending Lift System has a Worm Gear Box with a unique design of being able to reach the maximum height within one minute.
3. The Rotary Ascending Lift System has a unique design for self-breaking and self-locking, eliminating the usage of brake system for the motor.
4. The Rotary Ascending Lift System has a unique design that it can be operated both manually and with a motor.
5. The Rotary Ascending Lift System can carry a load of 150 kg to a maximum working height of 13ft.
6. The Rotary Ascending Lift System can also carry pallets that are used in warehouses.
7. The Rotary Ascending Lift System is designed in such a way that it can pass through a standard door and also through commercial lifts.

Scope For Future Work

- It is important to note that the current design is considered a baseline, and further optimization is still underway. The project team is committed to enhancing the lift system's performance, efficiency, and overall user experience. This ongoing optimization includes exploring opportunities for weight reduction, energy efficiency improvements, and enhanced control systems.
- The project has been a significant learning experience, requiring interdisciplinary collaboration, extensive research, and innovation. The lift system's potential applications are vast, ranging from warehouses and construction sites to industrial facilities and beyond. The optimized lift system will not only contribute to increased productivity and safety but also minimize physical strain and streamline operations.

- The design of the lift system represents a crucial milestone, showcasing the team's expertise in engineering and problem-solving. The project's success has laid a solid foundation for further optimization and the eventual deployment of an exceptional lifting solution that meets the demands of various industries. With continuous refinement, the lift system has the potential to revolutionize vertical transportation and provide significant value to a wide range of sectors.