Design and Modeling of 3d of Handling Mechanism for High Payload

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***Abstract* -A higher payload handling ability of a mechanism demands strength, easy response for accurate movement of load. The system must be flexible enough so that it can lift the defined payload with minimum deformation. Various mechanisms for handling payload are analyzed briefly and suitable mechanism is selected on the basis of specified environment. The design calculations are performed with standard design procedures for each part of the mechanism and values are derived. Linear motion of the mechanism is achieved using v-groove caster and lead screw combination. Lifting of the load is achieved by Scissor Jack and rotation of the payload is done by using U-groove caster wheels. These three motions provide a flexibility to position the payload to any position with the envelope defined for handling the payload. The self-locking capability of the lead screw prevents the reverse motion of the mechanism due to the payload weight acting on the mechanism and causing its displacement itself.**

***Index Terms* – Handling mechanism, V-groove caster wheels, Scissor jack design.**

I. INTRODUCTION

A handling mechanism is one in which the functions vary from manipulating the part safely without its falling and also to position it at any place within the specified envelope. As the payload to be handled increases, the flexibility of the mechanism decreases as more rigid elements needs to be used to withstand higher statc load due to own dead weight of the payload. Also increased payload calls for higher inertia and movement of such a high standstill mass requires application of large initial forces.

The motion of the higher payload also poses problem with limited movements desired. The higher payload when Translated or rotated from a resting position requires higher forces to stop at a particular position increasingly calling for precise braking systems as well. Here in this paper design calculations for such a handling mechanism are described and a higher payload of around 1000 kg i.e., 1 ton can be handled by the proposed mechanism and can be moved in 3 directions independently controlled.

II. CONCEPT PROPOSAL

The handling mechanism is to be such that the payload lifted can be translated, vertically raised or lowered and also rotated with respect to the base plate. Thus there needs to be 3 DOF for the handling mechanism which must be satisfied by the mechanism developed. The concept starts with analyzing various mechanical elements used in industrial applications for high payload handling capability.

The handling mechanism is designed in such a way that payload can have linear motions in X, Y and Z directions and rotational motion along Z direction. The main criteria is it should be able to handle load carrying capacity of about 1000 kg.

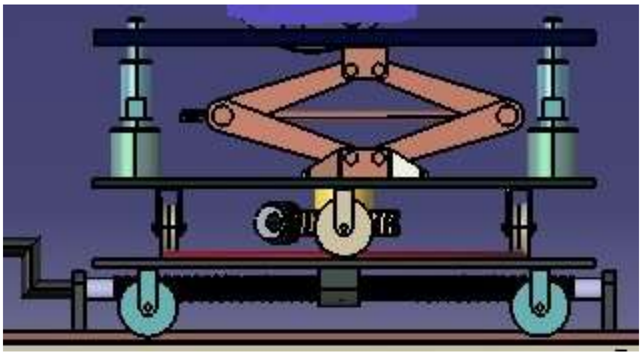

Fig. 1 Handling Mechanism

III. DESIGN CALCULATIONS

A. Design of lead screw for linear motion

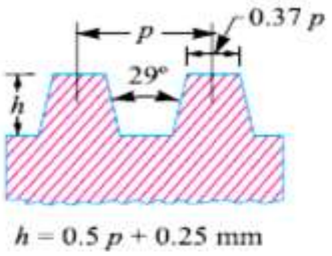

Fig. 2 ACME Thread

Type of thread: ACME or trapezoidal shown in Fig 2
Where,
α = helix angle, μ = coefficient of friction between nut and lead shaft, w = axial load in N = according to requirement 13000N, P = external force
According to standard,
Required pitch (p) = 6 mm, Major diameter of shaft (do) = 30 mm, Mean diameter = $d_0 - \frac{P}{2}$ = 30 - $\frac{6}{2}$ = 27, μ= tan ø,
ø= friction angle tan α = $\frac{P}{\pi d} = \frac{6}{\pi * 27}$ =0.0707.
For ACME threads,

$$R_N = \frac{W}{\cos\beta} \qquad \text{where, } \beta = 14.5$$

Friction force,

$$F = \mu \times \frac{W}{\cos\beta} = \mu_1 \times W \qquad \text{where, } \mu_1 = \tan \phi_1 = 0.155$$

From the equation

$$P = 13000 \times \left[\frac{\tan\alpha + \tan\phi_1}{1 - \tan\alpha \tan\phi_1}\right] = 13000 \times \left[\frac{0.0707 + 0.155}{1 - 0.0707 \times 0.155}\right] = 2966.66 \text{ N}$$

Torque required to overcome friction between the screw and nut, from eq.2

$$T_1 = P \times \frac{d}{2} = 2966.66 \times \frac{27}{2} = 40049.22 \text{ N-m}$$

Stress calculation: Direct stress (tensile or compressive) = $\frac{W}{A_c} = \frac{13000}{\frac{\pi}{4}(23.5)^2}$ = 29.975 N/mm^2

Torsional shear stress: $\tau = \frac{16\,T}{\pi\, d_c{}^3} = \frac{16 \times 40049.22}{\pi (23.5)^3}$ = 369.34 N/mm^2

Shear stress due to axial load: $\tau_{(screw)} = \frac{13000}{\pi\,(92)\,(23.5)\,(2.22)}$ = 0.862 N/mm^2 and $\tau_{(nut)} = \frac{13000}{\pi\,(8)\,(30)\,(2.22)}$ = 7.766 N/mm^2
Where,
W = Axial load on the screw, n = Number of threads in engagement, d_c = Core or root diameter of the screw, d_o = Outside or major diameter of nut or screw, and, t = Thickness or width of thread = 0.37×P = 0.37×6 = 2.22 mm.

B. Design calculation of Scissor jack

The links screw and pins are made from SS 304L or 304 for which the permissible stresses are 200 MPa in tension and 100 MPa in shear. The bearing pressure on the pins is limited to 20N/mm^2.
Assumption: The pitch of the square threads as 6 mm and the coefficient of friction between threads as 0.20,Lifting load = 13 KN and Maximum lifting distance = 300 mm.
Force analysis: Force analysis of scissor jack is shown in Fig 12. All links are of same in length, width and thickness. Considering force F applied on top of the jack and free body diagram of force applied at top of jack is shown in Fig.3.
From the fig. α = 30 °

$\Sigma F_X = 0$, $F1 \sin\alpha - F2 \sin\alpha = 0$, $F1=F2$, $\Sigma F_Y = 0$, $F1 \cos\alpha + F2 \cos\alpha = F$,

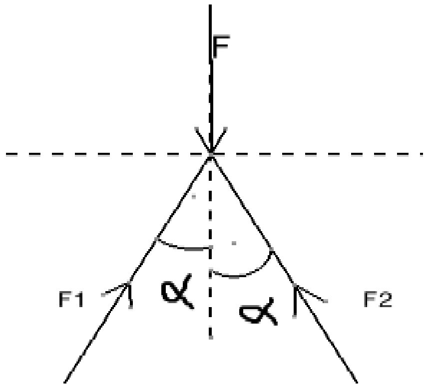

Fig. 3 Force on top joint of jack and its free body diagram

$F1=\frac{F}{\cos\alpha} = \frac{13000}{2\cos 30} = 7505.5$ N

So, $F1=F2=7505.5$ N

Now, angle is decreased at maximum raising height of jack, so maximum force is decreased. Since the maximum loading force will act at the minimum raising height of jack the design stresses will be analysed at point.

From the fig. $\Sigma F_Y = 0$, $F1 \cos\alpha - F3 \cos\alpha = 0$, $F1=F3$ and $\Sigma F_X = 0$, $F1 \sin\alpha + F3 \sin\alpha - Fs = 0$, $Fs = 2\, F1 \sin\alpha = 7505.5$N. All four link are symmetric so we can write the following equation F1 = F2 = F3 = F4 = F1' = F2' = F3' = F4'

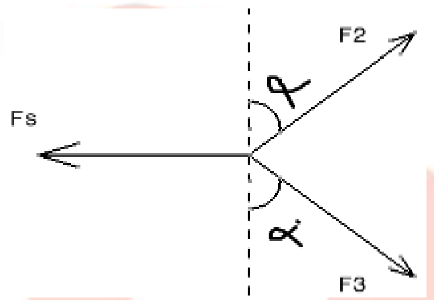

Fig. 4 Force on middle joint of jack

From free body diagram, $\sin 60 = \frac{150}{L}$, L (length of link) =173.20 mm.

Design of screw: Length of center screw is to be at-least twice the length of the arms. L_{s1} = 141.41 mm and L_s= 322.83 mm. Each nut carries half the total load on the jack and due to this, the link is subjected to tension while the square threaded screw is under pull as shown in below Fig 4.26. The magnitude of the pull on the square threaded screw is given by

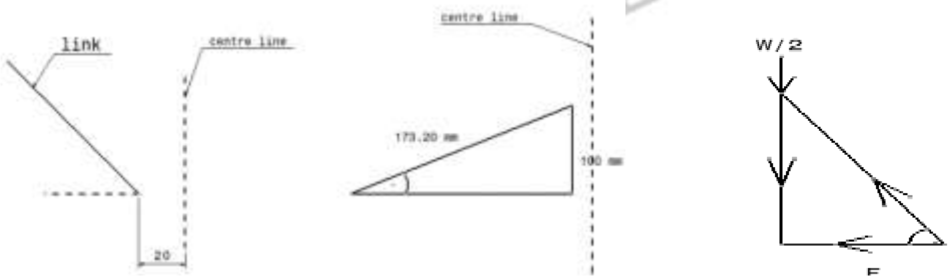

Fig. 5 Distance between center line to link joint and Load diagram on half portion of jack

$\theta = 35.26°$, screw length between two nut $L_s = 2L_{s1} + 40$ mm

$F=\frac{W}{2\tan\theta} = \frac{13000}{2 \times \tan 35.26} = 9193.882$ N

Since a pull acts on the other nut therefore total tensile load on screw Ft = 2F= 18387.76 N

d_c=core diameter of the screw and p= pitch of thread = 6 mm

We know that load on the screw, $Ft = \frac{\pi}{4} d_c^2 \times \sigma_t$, $18387.76 = \frac{\pi}{4} d_c^2 \times 200$,

Therefore, dc = 10.81 mm, I take dc= 15 mm. Nominal (or) outer diameter of screw, do = dc + p = 20 + 6 = 21 mm

Mean diameter of screw, d_m = do – p/2 = 26 – 6/2 = 18 mm

Helix angle, $\tan\alpha = \frac{p}{\pi dm}$, $\alpha = 6.05°$

Angle of friction, $\tan \Phi = \mu$, $\Phi = 11.310$ where, $\mu=0.2$

Effort required to rotate screw, $P = Ft \tan(\alpha + \Phi)$, $P = 5750.58$ N

Torque required to rotate the screw, $T = P \times dm/2$, $T = 51.75$ Nm
Shear stress in the screw due to torque, $\tau = \frac{16\,T}{\pi\, dc^3} = \frac{16\times 51755.23}{\pi\times 15^3} = 78.09\ \text{N/mm}^2$
Direct tensile stress in screw, $\sigma_t = \frac{Ft}{\frac{\pi}{4}dc^2} = 104.049\ \text{N}/mm^2$
Maximum principle tensile stress $\sigma_{tmax} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{{\sigma_t}^2 + 4\tau^2} = 145.85\ \text{N}/mm^2$
Maximum shear stress, $\tau_{max} = \frac{1}{2}\sqrt{{\sigma_t}^2 + 4\tau^2} = 93.83\ \text{N}/mm^2$
Maximum stresses are within safe limit, therefore the design of square threaded screw is satisfactory

Design of nut: Assuming that the load W1 is distributed uniformly over the cross-sectional area of the nut, therefore bearing pressure between the threads (p_b), N= number of threads in contact with the screw
$p_b = \frac{Ft}{\frac{\pi}{4}[\,do^2 - dc^2]\,n}$, $20 = \frac{18387.76}{\frac{\pi}{4}[\,21^2 - 15^2]\,n}$, $n = 5.4 \sim 6$
Thickness of the nut, t = n x p = 36 mm
Width of nut, b = 1.5 do = 31.5 mm ~32mm
To control the movement of the nuts beyond 322.8 mm (the maximum distance between the center lines of nuts), rings of 8 mm thickness are fitted on the screw with the help of set screws.
Therefore, Length of screwed portion of the screw = 322.8 + t + 2 × Thickness of rings = 322.8 + 36 + 2 × 8 = 374.8 mm.

Design of pins in the nuts:
Let d1 = Diameter of pins in the nuts. Since the pins are in double shear, therefore load on the pins (Ft), Therefore, $9193.882 = 2 \times \frac{\pi}{4}\, d1^2 \times \tau$, d1 = 8mm ~ 9 mm. Diameter of pin head = 1.5 x d1 = 13.5 mm. Thickness of pin head = 4 mm
Design of links: Due to the load, the links may buckle in two planes at right angles to each other. For buckling in the vertical plane (i.e. in the plane of the links), the links are considered as hinged at both ends and for buckling in a plane perpendicular to the vertical plane, it is considered as fixed at both ends.
We know that load on the link = F / 2 = 9193.882 / 2 = 4596.94 N
Assuming a factor of safety = 1.5, the links must be designed for a buckling load of
$W_{cr} = 4596.94 \times 1.5 = 6895.41$ N
Let, t1 = Thickness of the link, and b1 = Width of the link. Assuming that the width of the link is three times the thickness of the link, i.e. b1 = 3 t1,
Therefore cross-sectional area of the link, $A = t1 \times 3t1 = 3\ {t_1}^2$
Moment of inertia of the cross section of the link, $I = \frac{1}{12} \times t_1 \times b1^2 = 2.25\ t1^4$

We know that the radius of gyration, $K = \sqrt{\frac{I}{A}} = 0.866\ t1$
Since for buckling of the link in the vertical plane, the ends are considered as hinged, therefore equivalent length of the link,
$L = l = 173.20$ mm and Rankin's constant, $a = \frac{1}{7500}$
According to Rankin's formula, buckling load (W_{cr}),
$6895.41 = \frac{\sigma_c \times A}{1 + a(\frac{L}{K})^2}$

$6895.41 = \frac{200 \times 3\ {t_1}^2}{1 + \frac{1}{7500}(\frac{173.20}{0.866\ t1})^2}$, $600t1^4 - 6895.41\ t1^2 - 36752.53 = 0$, t1= 3.9 mm

Considering t1 = 5 mm, b1= 15 mm
Now let us consider the buckling of the link in a plane perpendicular to the vertical plane. Moment of inertia of the cross-section of the link, $I = \frac{1}{12} \times b1 \times t1^3 = 0.25\ t1^4$
Therefore cross-sectional area of the link, $A = t1 \times 3t1 = 3\ {t_1}^2$

Radius of gyration, $K = \sqrt{\frac{I}{A}} = 0.29\ t1$
Since for buckling of the link in a plane perpendicular to the vertical plane, the ends are considered as fixed, therefore Equivalent length of the link, L = l / 2 = 173.20 / 2 = 86.6 mm
Again according to Rankin's formula, buckling load,
$w_{cr} = \frac{\sigma_c \times A}{1 + a(\frac{L}{K})^2} = \frac{200 \times 3t1^2}{1 + \frac{1}{7500}(\frac{86.6}{0.29\ t1})^2} = 10168.11$ N
Since this buckling load is more than the calculated value (i.e. 9193 N), therefore the link is safe for buckling in plane perpendicular to the vertical plane.
∴ t1 = 5 mm; and b1 = 15 mm

IV. CONCLUSION

According to require motion of object or pay load, design the mechanism which can move pay load/object in three degree of motion. Main part of mechanism is v groove caster wheel, u groove caster wheel and toggle jack. V groove wheel for linear motion, u groove for rotary motion and toggle jack for lifting motion. Also lead screw use as for actuator and worm wheel use as accurate motion in rotation direction. Base one the movement of mechanism, theoretical design calculation done and we found that all value of stresses are below the allowable permissible value. Base on the theoretical calculation generate the model of mechanism using CATIA V5 modelling Software.

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