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Design, Simulation and Analysis of Self-Locking Mobile Hoist

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ABSTRACT

In most of the industries and construction fields weight lifting is usually carried out with the help of machines like chain hoists, forklifts and even electric cranes. A majority of these machines require an external power to operate which is provided mainly in the form of electric or pneumatic drive. These machines would be difficult to use in cases where such drive mode is remotely accessible. This paper deals with the design, simulation and analysis of a self-locking mobile hoist which can be used in situations where there is unavailability of the above-mentioned power drives. The machine is designed to lift a weight of 60 Kg. By incorporating a worm and worm wheel in the design, the effort required for lifting the weight could drastically be reduced. Worm and worm wheel also helps in making the design self-locking. Modelling of the machine was done using Solid Works software and the simulation of it for the designated weight was carried out on Fusion 360. Various components of the machine were designed based on the load and static analysis was carried out for the prominent components. The proposed design was able to lift a weight of 60 Kg and was simulated satisfactorily.

Keywords: Effortless, Mechanical drive, Self-locking, Weight lifting, Worm and worm wheel.

1. INTRODUCTION

Lifting operations are inherent in many construction sector occupations. They can be done by hand or with lifting equipment. Both manual lifting and mechanical lifting operations are available today, each of which is used according to the different requirements. Basically, a lifting operation is concerned with the lifting and lowering of a load. A 'load' is the item or items being lifted which could include a person or people. A lifting operation can be done manually or with lifting equipment. Manual lifting, holding, putting, carrying or moving of loads are often referred to as 'manual handling.' There are several risk factors which can increase injury from manual lifting. These may arise due to various reasons, like the load being too heavy, the load being too large, the load being unstable, difficulty in grasping etc. In order to avoid these kinds of difficulties, mechanical lifting operations are generally preferred. A wide range of lifting equipment are available in the market today. These may include equipment like hoist, crane, power shovel, forklifts etc. Each of this equipment has numerous advantages, as well as disadvantages associated with it. But in situations where the power from the above-mentioned systems would seem uneconomic, manual cranes are utilized.

Weight lifting is an inevitable process in different fields such as construction, mining, manufacturing etc. Even simple tasks like drawing water from deep wells or trenches, a water drawing mechanism is needed. Conventionally, the lifting mechanisms uses hydraulic or pneumatic power to operate. These mechanism is achieved by the use of link, folding support in crisscross pattern known as a Pantograph. However generally they are designed to lift fairly light loads and require a hydraulic circuit. Another commonly used lifting mechanism in various industries are forklifts. General forklift can be defined as a tool capable of lifting hundreds of kilograms of weight. The forklift operator drives forward until the forks push under the cargo, and can then lift the cargo several feet in the air by operating the forks. But this also needs an external power source to operate. An alternative design, consists of a simple mechanical device used to raise element or object from ground level to a certain height to perform a specific work with maximum load and minimum efforts. The design was developed keeping in mind that the lift can be operated by mechanical means so that the overall cost of the scissor lift is reduced. This design makes the lift more compact and much suitable for medium scale work. [1, 3]

But most of these mechanisms are field specific. And moreover, the amount of effort and energy expended is very high in the case of manual lifting using simple pulleys and ropes. Also, the safety of these devices is to be ensured. In recent times, we’ve seen some great advancements in the technology of lifting. But most of these technologies rely on some form of electric, pneumatic or hydraulic power sources to operate, of which electric operated mechanisms are the widely used one. So, in remote areas where power supply is yet to be established, such mechanisms cannot be used or implemented. This calls for an alternative mechanism which can be operated manually. But the effort required would be very high as the entire operation has to be performed manually. So, there is a need for a device or an equipment which serves the basic purpose of lifting the freight using less effort without compromising on the safety of the users.

The design consists of foot pedals which are manually driven by the user sitting in the seat. The force of the pedaling is transferred from the foot pedals to the worm shaft through a chain. This chain drive is similar to that seen in the ordinary bicycles. The movement of the chain causes the sprocket fixed onto the worm shaft to rotate in the same direction as that of pedaling. This in turn rotates the worm. The rotary motion of the worm makes the gear meshed with it to rotate slowly. Being keyed to the main shaft, this gear transfers this rotation to the main shaft. A drum of suitable size is designed and welded to the main shaft in such a manner that with the rotation of the main shaft the drum also rotates in the same direction. A rope of suitable strength is wound over the drum. One end of this rope is attached to the drum and the other end slides over a simple pulley. To the end of this rope a tray for carrying/lifting the weight is attached. This entire system is fixed on a movable frame so that the equipment can be transported to the desired location with ease. Worm gear reducers are quiet, compact, and can have large reduction ratios in a single stage. The ideal ratio range for worm gearing is 5: 1 to 75: 1. This is the general range for most catalog reducers. Ratios of 3: 1 to 120: 1 are practical and have applications that are very successful. For ratios below 3: 1. Worm gearing is not a practical solution for most applications, and other forms of gearing should be considered. Worm gearing for ratios above the ranges mentioned are generally more practical as part of a multistage reduction. In service, worm gears survive large overloads and high shocks. When properly applied, worm gearing can offer excellent performance and cost savings. Worm gearing has an inherent 200% overload (i.e. 3x rating) capacity in its rating. Other forms of gearing do not have this built-in service factor. Therefore, when sizing a worm gear set. A lower service factor than normal can be used. [4]

Running a worm gear set with the gear (worm wheel) as the input member is commonly called ‘back driving’. Back drive efficiency of a worm gear set is lower than its forward drive efficiency. By varying design, the back-drive efficiency can be reduced to zero, as in a self-locking or irreversible gear set. If the gear tries to drive the worm, internal friction causes the mesh to lock. No matter how much torque is applied to tile gear shaft, mesh friction increases proportionally, preventing rotation. This is the same principle that keeps a nut and bolt from unscrewing under an applied tension load.

Fig.1 Basic overview of the design

Fig.2 Worm and worm wheel integrated with the shafts

Fig. 3 Drum and the pedals

2. DESIGN OF WORM AND WORM GEAR

The required calculation for the design of worm and worm gear is performed. The figures and calculations are shown below.

Abbreviations

i = Gear ratio
N_w = Speed of the worm (rpm)
M_x = Module (mm)
〈 = Normal pressure angle
P_c = Axial pitch (mm)

2.1 Selection of standard dimensions of worm and worm gear

Self-Locking property of worm and worm gear is utilized for locking the vertical movement at a particular height, so, no: of start of worm is taken as 1.

Assuming, Worm be made of hardened steel and worm gear of bronze

i = 1:20
N_w =120 rpm
m_x = 4 mm

Standard dimensions are taken from K. Mahadevan data book

Standard dimensions of worm are

Axial pitch = 12.566 mm
Pitch circle diameter = 40 mm
Tip diameter = 48 mm
Root diameter = 30.4 mm
Lead Angle =5.71°

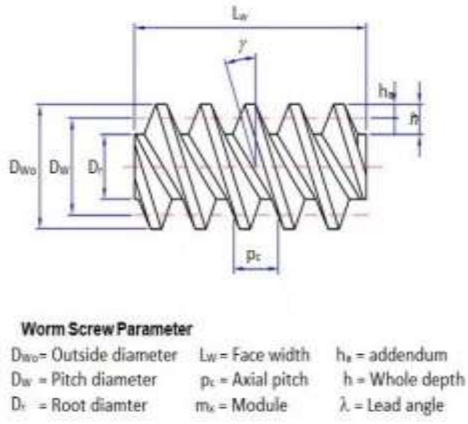

Fig. 4 Dimensions of worm

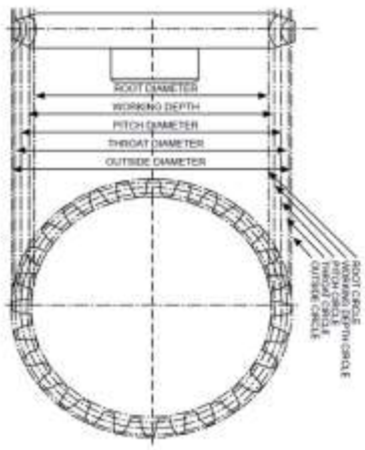

Fig. 5 Dimensions of worm gear

Actual centre distance: $x = \frac{l}{2\pi}[cot\ cot\ \lambda\ + i]$

$$= \frac{12.566}{2\pi}[cot\ cot\ 5.71\ + 20] \qquad (1)$$

The centre distance obtained after calculation is 60 mm

Face length of worm:

$$L_W = (4.5 + 0.02T_W)p_c \qquad (2)$$
$$= (4.5 + 0.02 \times 1) \times 12.566$$
$$= 56.8\ mm$$

Depth of tooth:

$$h = 0.686 \times p_c \qquad (3)$$
$$= 0.686 \times 12.566 = 8.62 \text{ mm H8.8 mm}$$

Addendum:

$$h_a = mx = 4 \qquad (4)$$

For worm gear, Pitch circle diameter of gear:

$$D_G = m_x \times T_G \quad (5)$$
$$= 4 \times 20 = 80 \text{ mm}$$

Outside diameter of gear:

$$D_{Go} = D_G + 1.0135p_c \quad (6)$$
$$= 80 + 1.0135\times12.566 = 92.74 \text{ mm}$$

Throat diameter:

$$D_t = D_G + 0.636p_c \quad (7)$$
$$= 80 + 0.636 \times 12.566 = 88 \text{ mm}$$

Root diameter:

$$D_r = D_t - 2\times h \quad (8)$$
$$= 88 - 2\times8.8 = 70.4 \text{ mm}$$

Face width:

$$b = 2.38p_c + 6.35 \quad (9)$$
$$= 2.38 \times 12.566 + 6.35$$
$$= 36.26 \text{ mm} \approx 37 \text{ mm}$$

3. DESIGN OF SHAFTS

3.1 Load Calculation of Main Shaft

Tangential force on the worm:

$$W_T = \frac{2 \times torque\ on\ worm}{pitch\ circle\ diameter\ of\ worm} = \frac{2\times3000}{40} = 150\ N \quad (10)$$

Radial force on the worm:

$$W_R = W_A . tan\phi \qquad (\phi = 14.5) \quad (11)$$

$$W_A = \frac{W_T}{tan tan \lambda} = \frac{150}{0.1} = 1500 \text{ N} \quad (12)$$

$$W_R = 1500 \times tan\ tan\ (14.5^0) = 387.92\ N$$

The diameter of the shaft:

$$d^3 = \frac{16}{\pi\times[\tau]} \sqrt{\left(K_b M_b\right)^2 + (K_t M_t)^2} \quad (13)$$

$$= \frac{16}{\pi\times[\tau]} \sqrt{\left(1.5 \times M_b\right)^2 + (1 \times M_t)^2}$$

The weights, tensions, and bending moments acting on the shafts are found out. The diameter of the shaft after calculating the equation gives 24.89 mm

Therefore, the standard diameter of the shaft = 25 mm

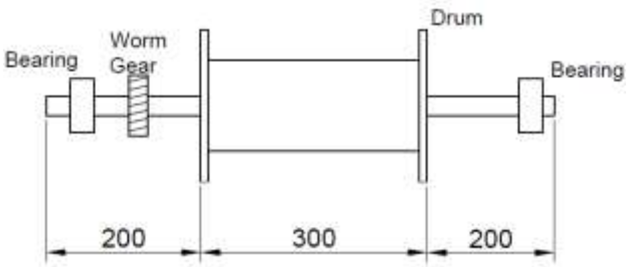

Fig.6 Main shaft dimensions

3.2 Load Calculation of Worm Shaft

Tangential force:

$$W_T = 150 \text{ N}$$

Axial force:

$$W_A = 1500 \text{ N}$$

Radial force:

$$W_R = 387.92 \text{ N}$$

Worm shaft diameter:

$$d^3 = \frac{16}{\pi\times[\tau]} \sqrt{\left(K_b M_b\right)^2 + (K_t M_t)^2} \quad (14)$$

$$= \frac{16}{\pi \times [\tau]} \sqrt{(1.5 \times M_b)^2 + (1 \times M_t)^2}$$

$$d = 22.28\ mm$$

Therefore, the standard diameter of the shaft = 25 mm

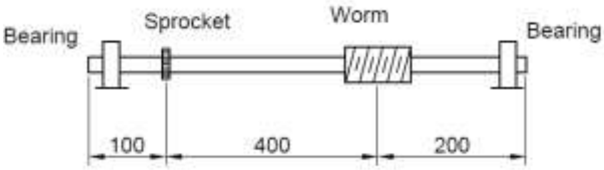

Fig. 7 Worm shaft dimensions

4. DESIGN OF BEARINGS

4.1 Design of Main Shaft Bearing

The standard diameter of the main shaft is 25 mm. So, SKF 6005 deep groove ball bearing is used for design.

Standard dimensions for 6005 deep groove ball bearing,

d = 25 mm

D = 47 mm
B = 12 mm
r = 0.6 mm
Static capacity $C_o = 6.55\ kN = 6550\ N$
Dynamic capacity $C = 11.9\ kN = 11900\ N$

Equivalent load:

$$P = (XF_r + YF_a)S \tag{15}$$
$$P = (1.5 \times F_r + 1 \times F_a)S$$

The equivalent load acting on the bearing after solving the equation is 441.67N

Dynamic capacity:

$$C = (\frac{L}{L_{10}})^{\frac{1}{k}} \times P \tag{16}$$
$$= (\frac{L}{L_{10}})^{\frac{1}{3}} \times 441.67$$

Dynamic capacity after calculation gives 767.62 N, which is in the rated dynamic capacity of SKF 6005 ball bearing.

Probability of survival:

$$\frac{L}{L_{10}} = \left[\frac{ln\left(\frac{1}{P}\right)}{ln\left(\frac{1}{P_{10}}\right)}\right]^{\frac{1}{b}} \tag{17}$$
$$\frac{L}{L_{10}} = \left[\frac{ln\left(\frac{1}{P}\right)}{0.1053}\right]^{\frac{1}{1.34}}$$

The probability of survival P gives 99%.

4.2 Design of Worm Shaft Bearing

SKF 6005 deep groove ball is used for 25 mm shaft diameter

Equivalent load:

$$P = (XF_r + YF_a)S \tag{18}$$

$$P = (1.5 \times F_r + 1 \times F_a)S$$

The equivalent load acting on the bearing after solving the equation is 415.62 N

Dynamic capacity:

$$C = (\frac{L}{L_{10}})^{\frac{1}{k}} \times P \tag{19}$$
$$= (\frac{L}{L_{10}})^{\frac{1}{3}} \times 415.62$$

Dynamic capacity after calculation gives 1961.51 N, which is in the rated dynamic capacity of SKF 6005 ball bearing.

Probability of survival:

$$\frac{L}{L_{10}} = \left[\frac{ln\left(\frac{1}{P}\right)}{ln\left(\frac{1}{P_{10}}\right)}\right]^{\frac{1}{b}} \tag{20}$$
$$\frac{L}{L_{10}} = \left[\frac{ln\left(\frac{1}{P}\right)}{0.1053}\right]^{\frac{1}{1.34}}$$

After equating the values probability of survival P gives 99%.

5. Design of frame

Fig.8 Frame

The entire unit is supported on the frame. The machine is designed to lift a maximum of 60 Kgs up to a 2-storey height (6-7m). So, the frame should be strong enough and it should maintain its position during the lifting operation. Counter weights must be provided to get a total weight of more than 120Kgs (without considering the weight of the man operating the machine).

6. DESIGN OF CHAIN

Specification

Chain number	= ISO 08B
Pitch	= 12.7mm
Speed ratio	= 1:2
No. of teeth on smaller sprocket	= 18
No. of teeth on bigger sprocket	= 36
Length of the chain	= 1244.9mm
PCD of smaller sprocket	= 73.16mm
PCD of bigger sprocket	= 145.71mm

Velocity of chain: $v = \frac{P \times Z_1 \times n_1}{1000}$ = 0.457 m/s (21)

Tangential force: $F = \frac{1000 \times P}{v}$ = 80.9 (22)

Allowable working load per strand:

$$F_w = \frac{F_u}{FS \times K_s} = 2550.6\text{N} \quad (23)$$

FS – Factor of safety (tables)= 7

Actual factor of safety:

$$(FS)_a = j \times \left(\frac{F_u}{F+F_c+F_s}\right) \quad (24)$$

where,

$$F_c = \frac{w' \times v^2}{g} = 2.715 \quad , \; F = 80\ N \quad (25)$$

$$F_s = K_2 \times 2w' \times C_1 = 344.31\ \text{N} \quad (26)$$

$$(FS)_a = 1 \times \left(\frac{17845.2}{80+2.715+344.31}\right) = 41.8\ \text{N}$$

$(FS)_a > FS$ (tables), the design is safe.

7. DESIGN OF DRUM

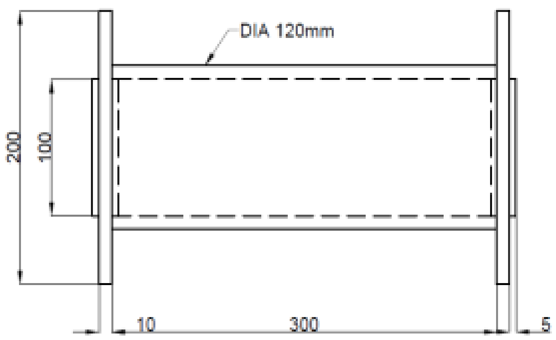

Fig.9 Drum

Volume, $V = 18967.35\ cm^3$

Density of C 40 Steel $= 7.87\ \frac{g}{cc}$

$$Weight = V \times d = 14.92 \approx 15\ kg \quad (27)$$

8. ANALYSIS OF THE DESIGN

Simulation of the design was done using AutoCAD Fusion360 software. Various static parameters (stress, displacement, safety factor) on the different components of the design were analyzed.

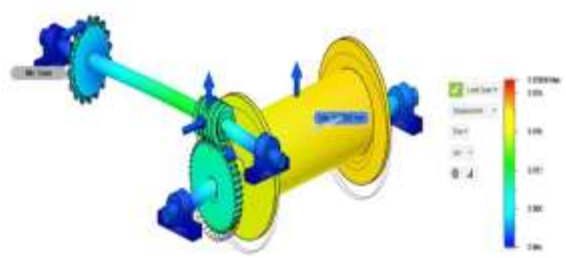

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Fig. 10 Displacements on different elements

Fig. 11 Safety factor of the design

Fig. 12 Stresses acting on different elements.

For the load given, the hoist was able to lift the weight with minimum effort and from the simulation it was found out that the design is not expected to bend or break with the current analysis criteria. The material chosen for most of the components was steel which has an yield strength of 207 MPa. The simulation showed that the actual minimum factor of safety is around 5.32. The maximum value of the stress was found on the worm and worm gear mating surface which was 35 MPa. The maximum displacement was observed at the location of the drum which was around 0.02mm.

9. CONCLUSION

In this paper, a self-locking mobile hoist was designed and simulated for the purpose of lifting a weight of 60kg to a certain height. Design of various components was based on the load acting on them. The simulation and the static analysis of the design was carried out on FUSION 360. It was found out that by using a higher grade steel, the factor of safety could further be increased. Due to some technical constraints, the analysis performed in the design was limited to static. By performing an additional dynamic analysis, a greater amount of details regarding the design can be acquired and the design will be ready to be fabricated.

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