Design and Fabrication of Two Wheel Drive Forklift

Dheerendra singh[1], Dhanmoni Daimary[2], Hemant Sangwan[3]

[1,2,3]*Mechanical Engineering Department, Third Year Student, MIT-ADT University, Pune, Maharashtra, India,*
Mechanical Engineering Department, MIT-ADT University , Pune, Maharashtra, India

---

Abstract- *As we all know material handling has always been an important element in our day-to-day life. Moving raw material, finished goods, semi- finished goods from one place to other are now done by forklift. Forklift are used for carrying heavy loads, loading and unloading of trucks, ships etc. in ware housing and construction sites. Varieties of forklifts are present which run though electricity and fuels like liquid petroleum gas, diesel, gasoline etc. are used in large industries and are expensive. So we decided to design and fabricate miniature two wheel drive forklift which is cost effective and can be used in small industries, factories, ware house etc.*

1. INTRODUCTION

Forklift is a very useful and time saving vehicle similar to a small truck used to lift and move material over short distance are now widely use in industries, factories, construction sites to carry hefty building materials, for loading and unloading of trucks and ships and can also be serve as snow plows to clear off the way covered by snow.

Forklift have a power operated forked platform attached at the front that is used to rise and lower for insertion under a cargo to lift or move it. Some forklift allow the operator to sit while driving and operating machine while some require the operator to stand.

1.1 Field of Use

Forklift are use for moving materials from one place to another. Forklift are use in commercial areas such as factories, construction sites industrial workshop etc.

1.2 Literature Survey

By taking the reference of an actual forklift we have designed and fabricated miniature two wheel drive forklift which is cost effective. The construction of structure is made by using metal frame, lead screw, chain sprocket, C-channel, nuts, bearings etc.

2. WORKING

Forklift works on semi electric techniques where power is transfer from one form to another by using mechanical as well as electrical components. In this project we have worked on the transfer of rotational motion to linear motion by using a dc motor which is constructed by using a gearbox of gear ratio 2:1. As we run the motor the chain drive which is coupled from gearbox to lead screw by using sprocket starts running and provides rotational motion to the lead screw. Lead screw is mounted on the guide column, the nut is fastened into the lead screw and roller is bolted to the nuts. And fork is attached to the roller which provide linear motion to the fork.

Fig. 1 Frame of forklift

3. Design

The design of forklift is derived as follows.

CHEMICAL ANALYSIS	
C%	Carbon 0.38 – 0.43
Mn%	Manganese 0.75 – 1.00 max
P%	Phosphorus 0.035 max
S%	Sulphur 0.040 max
Si%	Silicon 0.20 – 0.35
Cr%	Chromium 0.80 – 1.10
Mo%	Molybdenum 0.15 – 0.25

Material- AISI 4140

3.1 Total load in Newton –

Total Load (W) = Mass (m) × Acceleration due to gravity (g)= 100×9.81
= 981N

3.2 Permissible compressive stress (σ_c)-

$\sigma_c = S_{yt}/FS = 415/3$
$=83\ N/mm^2$

3.3 Core diameter of screw (d_c) –

$\sigma_c= W/[\pi/4\times d_c^2]$
$83= 981/(\pi/4\times d_c^2)$
$d_c= 3.880$ mm

Nominal Diameter d(mm)	Pitch p(mm)
22,24,26,28	5
30,32,36	6
40,44	7
48,50,52	8

3. 4 Core diameter of screw (d_c) $d_c= d - p = 22 – 5 = 17$ mm

3.5. Mean diameter of screw (d_m) – $d_m = d – 0.5p = 22 – 0.5(5) = 19.5$ mm

3.6 Helix angle (α) - $\tan \alpha = l/(\pi d_m) = 5/(\pi(19.5))$ $\alpha = 4.666^o$

3.7 Friction angle (?) – $\tan\emptyset = \mu = 0.35$ $\emptyset = 19.29^o$ Since $\emptyset > \alpha$ the screw is self- locking.

3.8 Torque required to lift & lower the load – $M_t = (Wd_m)/2 \times[(\mu\sec\theta\pm\tan\alpha)/(1\pm\mu\sec\theta\tan\alpha)]$ For Lifting – $M_t= 4366.06$ Nmm For lowering – $M_t = 2558.005$ Nmm

3.9 Check for shear & compressive stress failure- 1. $\tau = (16M_t)/(\pi d_c^3) = (16(4366.06))/(\pi(17)^3) = 4.5282\ N/mm^2$ 2. $\sigma_c= W/[\pi/4\times d_c^2] = 981/(\pi/4\times 172) = 4.321\ N/mm^2$

3.10 Checking for buckling failure –

3.10.1 Moment of Inertia (I) – $I = \pi/64 \times d_c^4 = \pi/64 \times (17)^4 = 4099.82\ mm^4$

3.10.2 Cross sectional area (A) – $A = \pi/4 \times d_c^2 = \pi/4 \times 17^2 = 226.98\ mm^2$

3.10.3 Radius of gyration (K) – $K = \sqrt{(I/A)} = \sqrt{(4099.82/226.98)} = 4.249$ mm

3.10.4 Slenderness ratio (l/K) – $l/K = 1000/4.249 = 235.84$

3.10.5 Critical slenderness ratio – End fixity coefficient (n) = 0.25 $S_{yt}/2 = (n\pi^2 E)/(l/K)^2$ $415/2 = (0.25\pi 2\times 210\times 10^3)/(l/K)^2$ $l/K = 49.94$

3.10.6 Critical load on buckling (P_{cr}) – $P_{cr}= S_{yt}A[1- S_{yt}/4n\pi E\ (l/K)^2$ $P_{cr}= 415 \times 226.98\ [1- 415/(4 \times 0.25 \times \pi^2 \times 210 \times 10^3)\ (49.94)^2]$ $P_{cr}= 47109.57$ N

3.10.7 Factor of safety for buckling Failure – $FS = P_{cr}/W = 47109.57/981 = 48.0219$ so the design is safe against buckling failure.

3.11. Selection of Motor – · Required torque of motor (M_t) = 4366.06Nmm · Required speed of motor (N) = 70 - 100 rpm · Type of supply to motor = 12 Volt D.C. Torque required to raise maximum load in Kg-cm = $(M_{tRAISE})/(9.81 \times 10) = 4366.06/(9.81 \times 10)$

= 44.506 kg-cm Required speed of motor shaft in rpm
= 70-100 rpm after searching in the market we found no motor of this specification so we choose a motor with 100 rpm 100 Kg-cm torque motor

3.1. Gear ratio (G) –
Motor output = $(2\pi NM_t/60)$ = 33.066w
Motor torque = 3.2Nm(given)
Required torque = 4.3Nm
So gear ratio should be = 2:1
Torque at driven shaft will be 2*(3.2) = 6.4Nm

3.13 Selection of Battery –
From battery available range in market we assume battery capacity 12V, 7.2 Ah
Battery capacity = V × I
= 12 × 7.2
= 86.4 Watt
The output power will be less so we will have to use two batteries in a series.
Totak capacity of both batteries combined = 2 x 84.4 = 168.8
But Consider deep cycle of battery is 80% of total capacity of battery
Running capacity of battery = 0.80 × required battery capacity
= 135.04 Watt

3.14 Design of Fork –
· Outer face height (D) = 50.8 mm
· Outer face width (B) = 50.8 mm
· Inner ace height (d) = 44.8 mm
· Inner face width (b) = 44.8 mm
· Length of fork (L) = 600 mm

Moment of Inertia of fork (I) -

$I= 1/12\ [BD^3-bd^3]$
$I= 1/12\ [6659702.81-4028209.56]$
$I = 219291.10\ mm^4$

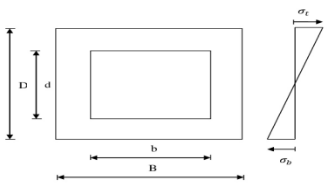


Fig.2 dimensions of forklift

3.14.1 Consider Two Fork with Point Load – Moment of Inertia for two fork (I) – I = I_1+I_2 = 219291.10 + 219291.10 I = 438582.2 Bending Moment (M_A) - M_A = -W × L = - 981 × 600 M_A = - 588600 Nmm Deflection (y_max) - y_{max} = $(WL^3)/3EI$ = $(981×600^3)/(3×210×10^3×438582.2)$

y_max = 0.766 mm

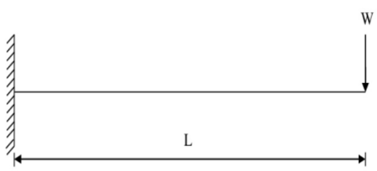


Fig.3 Point load on fork

3.14.2 Consider Two Fork with Uniform Distributed Load – Moment of Inertia for two fork (I) – I = I_1+I_2 = 219291.10 + 219291.10 I = 438582.2 mm^4 Find W/mm – W/mm = W/L = 981/600 W/mm = 1.6355 N/mm Bending Moment (M_A) - M_A = $(-W×L^2)/2$ = $(- 1.655×600^2)/2$ M_A = -294300 Nmm

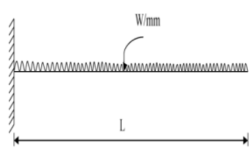


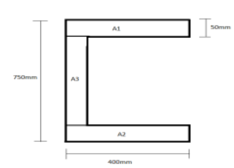
Fig.4 Uniform distribution diagram

Deflection (y_max) – y_{max} = $(WL^4)/8EI$ = $(1.6355×600^4)/(8×210×10^3×438582.2)$ y_{max} = 0.2875 mm

3.15 Selection of bearing – · Bearing life = 12000-20000 Hrs · Radial load (P) = 2450 N · Speed (n) = 50 rpm · Assume L_{10h} = 12000 Hrs . **Bearing life (L_{10}) –** L_{10}= (60 × n × L_{10h})/10^6 L_{10}=(60 ×50×12000)/10^6 L_{10} = 36 million rev.

3.16 Load capacity (C) – C = $P(L_{10})^{(1/3)}$ C = $981(36)^{(1/3)}$ C = 3239.19 N Using standard table of bearing selection, C = 9950 N C_0 = 4150N d = 12mm, D = 37mm, B = 12mm But, from available bearing range in market, we are assuming suitable bearing – d = 12mm, D = 28mm, B = 8mm

3.17 Center Of Gravity- $Part_{ABC}$ = (517.13, 400, 25) $Part_{DEF}$ = (536.42, 400, 25) $Part_{GHIJ}$ = (900, 400, 25) $Part_{MKL}$= (825,485,547.5) CG of whole frame lies on the coordinates (768.5_X, 442.5_Y,275_Z) above coordinates are with respect to origin that lies on the upper left corner of component A. CG of the part is lying on the part B and as low as 225mm from the ground so the design could not topple, additional trolly wheels are fixed on part A and B to give mobility to the machine.



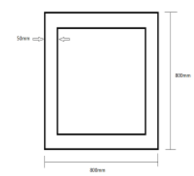
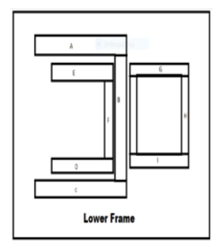


Fig.5 Lifting fork Fig.6 Box section.



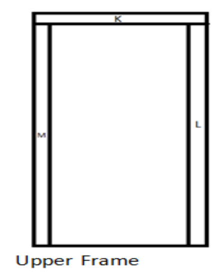


Fig.7 Upper frame Fig.8 Lower frame

3.18. Power output: - **P = τ * ω** = 17 * (100 * 2 π)/60 = 177.93 watts

4. CONCLUSION

The proposed design of forklift require less space for working and parking and is eco-friendly as it does not produce ant toxic gases. The maximum weight it can carry is 100kg to maximum height of 1200mm. It provide safety to the operator while handling the material.

ACKNOWLEDGMENT

We would like to thanks Prof. AnilKumar Sathe for his guidance. His patience and technical expertise who helped use overcome obstacles during this project.

References

1) Sachin Uglae et al, "Design and Structural Analysis of Mechanical Forklift using ANSYS Software", International Journal of Research in Advent Technology, E-ISSN: 2321-9637, Vol. 2, No. 5, May 2014, pp-234-237

2) R. S. Krishna et al, "Design and buckling strength evaluation of a lifting beam", International Journal of Engineering Research and Science & Technology, ISSN: 2319-5991, Vol. 3, No. 4, November 2014, pp-184-189

3) RS Khurmi, J.K Gupta (2005) a text book of machine design.

4) S. S. Ratan, Theory of Machine, Tata McGraw-Hill, 3rd edition, 2009

5) V. B. Bhandari, "Design of Machine Element", Tata McGraw-Hill, 3rd edition, 2010.