Design and Development of Zero Slip Lifter Mechanism

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Abstract –In machine tool applications it is often required to lift heavy work-pieces using conventional winches or chain blocks which may require two or more workers for loading the work-piece on to machine which is rather contradictory to the one man-multi machine concept. Hence there is a need of zero slip lifting devices which will be operated by the machine operator himself without any other aid or assistance. Self-locking characteristics is required in case of this device in order to be sure that the loads on the output side of the system cannot change position without input from the operator to prevent damage to the machine table if the work-piece drops on to it . Worm gears are one of the few gear systems that can be made self-locking, but at the expense of efficiency, they seldom exceed 45% efficiency, when madeself-locking. Proposed system will be designed by developing twin worm system in form of external threaded worm and internal threaded ring with view to develop a zero slip winch system to be operated using 12 Volt DC PMDC motor .Objectives of project are design of mathematical model of twin worm system with internal threaded ring system for maximum load lifting capacity , maximum factor of safety & maximum efficiency for reduced power consumption. The experimental validation part of the lifting force developed by the twin worm system be validated using test-rig developed and performance characteristics of torque, Power and efficiency Vs Speed will be plotted.

Index Terms – Material Handling Equipment's, Conventional method, Self Locking System, Twin worm gear.

1. INTRODUCTION

Material handling equipments are used for the movement and storage of material within a facility or at a site. Classification of material handling equipments is as follows.

- TransportEquipment.
- PositioningEquipment.
- Unit Load FormationEquipment.
- StorageEquipment.
- Identification and ControlEquipment.

Various types of accidents are happens during handling and storing of materials, improper handling and storing of materials can cause costly injuries, by using proper material handling equipment's we can avoid accidents.The term Self Locking System plays vital role in safety of working.

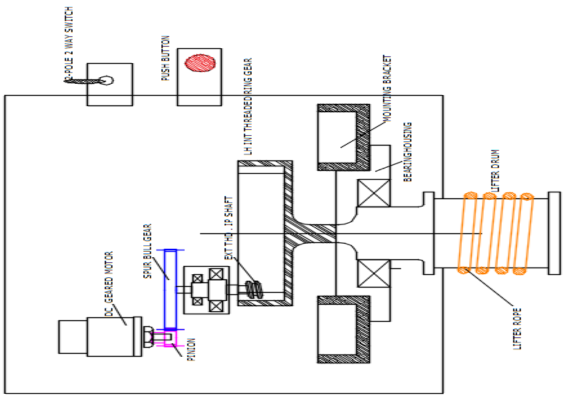
For self locking system worm gears are used, but they hardly ever exceed 45% efficiency, when made self locking. Thus we can conclude that if worm gear drives when used for lifting applications with self locking as the main objective for safety considerations the drives are extremely in-efficient; hence there is a requirement of special purpose drive that will provide better transmission efficiency in self locking condition. Design Development and Testing of Zero Slip Lifter Mechanism is a novelty over the conventional Worm gear box arrangement, by selecting proper, and different, pitch angles, the drive will exhibit either self-locking, or a combination of self-locking and deceleration locking characteristics, as desired and drive efficiency can range up to 90%.

2. RELATED WORK

R.D.Ankush et.al [1] Uses Manual Clutch to improve self locking property and efficiency,Prof. P.D.Kadam et.al [2] uses Twin worm external meshing gears for designing of worm drive,Chandrasekaran. et.al [3] obtains optimalsolution of Worm gear drive design problem with ACO,Alinezamabadi et.al [4] focus on Contact pattern and worm wheel teeths,Y.K. Mogalet.al[5] uses Multi-objective Optimization Approach for Design of Worm and Worm Wheel Based on Genetic Algorithm ,Prof.A.R.Gurav et.al [6] Design and Analysis of Dual Worm Self Locking System by using 3D drafting through UnigraphixNx 08, both gears are analyzed through ANSYS, ClaudiuValentinSuciu et.al [7] deals with the modeling and simulation of ascrew-worm gear mechanical transmission,Zheng Yong et.al [8] Introduced the importance of transmissionerror (TE) measurement in brief.

Shunqi Yang et.al[9] Designs worm gear for remotely operated vehicle (ROV)

3. PROPOSED MODELLING



Past research and experiences had indicated that when conventional method uses for load lifting application introduces following disadvantages,

- It gives 45% efficiency
- Large horse power motor required
- Increases production cost
- Also required large space

Efficiency of worm gear depends on friction angle and Lead angle; it is Possible to self lock of worm when Lead angle is less than of friction angle.

Hence design a special purpose system having lead angle in the range of 1° to 2°, which gives maximum transmission efficiency, reduces power consumption, and reduces production cost.

The set-up is a novelty over the conventional Worm gear box arrangement .The Zero slip worm drive is simply constructed, Two threaded elements are meshed together., Each worm is rotate in a different direction and has a different pitch angle. For proper mesh, the worm axes are not parallel, but slightly skewed, Proposed system will be designed by developing twin worm system in form of external threaded worm and internal threaded ring with view to develop a zero slip which system to be operated using 12 Volt DC PMDC motor for loading lifting application. For application in machine tool work loading for a vertical machiningcenter.

METHODOLOGY

- Theoretical Work
  - Study the conventional method used for load lifting application
  - Overview on disadvantages of system
  - Study the problems comes with conventional system
- Analytical Work
  - Design of parts of zero slip lifter mechanism
- Numerical Work
  - 3D modeling of different parts using CREO
  - Analysis of parts using ANSYS
- Experimental Work
  - Trial of system
  - Take readings and calculation
  - Draw graph based on result

4. DESIGN PROCEDURE

4.1. Selection of drive motor

Assume that 40 Kg load is to be lift

Effort to be applied to lift load of 40kg

= Wtan (Ø+μ)

= 9.81x 40 x tan (0.05 +1.8) = 12.6 N

Torque required by motor = Force x radius of lifter drum

= 12.6 x (50)

Torque required by motor = 630 N-mm = 63 kg-cm

As we are using reduction (74/12) = 6.16 maximum torque to be imported by motor = 10.41 kg-cm

Thus selecting motor as per following specifications:

The drive motor is 12 VDC permanent magnet motor with

Gearbox, Supply voltage range: 6-12 VDC, Gear ratio: 50:1, Rotational speed 60 rpm, Supply current: 600mA max, Torque: 11 Kg-cm /1.0971N-m, Length: 63mm (Exclusive shaft) Shaft: 6mm dia., Gearbox dia.: 37mm, Gearbox length: 25mm, Reduction ratio = 6.16, T design = 6.75N-m.

4.2. Design of RH Shaft

4.2.1 Material Selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
EN 24	800	680

Table: - 1 Material for input shaft

4.2.2 ASME Code for Design of Shaft

Since the loads on most shafts in connected machinery are not constant, it is necessary to make proper allowance for the harmful effects of load fluctuations.

According to ASME code permissible values of shear stress may be calculated from various relations.

$$fs_{max} = 0.18\ f_{ult} = 144\ N/mm^2$$

$fs_{max} = 0.3\ f_{yt} = 204\ N/mm^2$

Considering minimum of the above values;

$$\Rightarrow fs_{max} = 144\ N/mm^2$$

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Assuming 25% overload.

$\Rightarrow$T design = 1.25 x T = 8.43 x 10^3N-mm

Check for Torsional Shear Failure of Shaft

Assuming minimum section diameter on input shaft = 16 mm

$\Rightarrow fs_{act}=\frac{16\times Td}{\pi\times d^3}$ = 10.48 N/mm^2

As $fs_{act}<fs_{all}$

$\Rightarrow$Rh threaded shaft is safe under torsional load

4.3. Design of Lifter Drum Shaft

4.3.1 Material Selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
EN 9	600	480

Table: - 2 Material for Lifter Drum Shaft

4.3.2 ASME Code for Design of Shaft

$\Rightarrow fs_{max}$= 108 N/mm^2

Assuming minimum section diameter on input shaft = 20 mm

$\Rightarrow fs_{act}=\frac{16\times Td}{\pi\times d^3}$ = 5.36 N/mm^2

$fs_{act}<fs_{all}$

$\Rightarrow$lifter drum shaft is safe under torsional load at gear mounting side.

4.4. Design of Internal Threaded LH Worm Gear

4.4.1 Material Selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
20 MnCr1	1000	800

Table: - 3 Material for Internal Threaded LH Worm Gear

4.4.2 ASME Code for Design of Shaft

$\Rightarrow fs_{all}$ = 180 N/mm2

Assuming minimum section diameter D=148mm,d = 120 mm

$\Rightarrow$ = fs act = $\frac{16\times 6.75\times 10^3\times 148}{\pi(148^4-120^4)}$ = fs $_{act}$ = 0.018 N/mm^2

$fs_{act}<fs_{all}$

$\Rightarrow$LH internal threaded worm is safe under torsional load at the threaded end.

4.5. Design of Spur Gear Pair for Reduction

b = 10 m

Tdesign = 1.0971N.m

Sultpinion =Sult gear = 400 N/mm2

Service factor (Cs) = 1.5

dp = 111

T=T design = 1.0971N-m

$$\text{Maximum load} = \frac{\text{Maximum Torque}}{\text{Radius Of gear}}$$

$\Rightarrow \frac{1\times 10^3}{55.5} \Rightarrow$Pt =18 N.

$$P_{eff}=\frac{Pt\times Cs}{Cv}$$

Neglecting effect of Cv as speed is very low

$$P_{eff} = 27\ N\text{-------- (A)}$$

Lewis Strength equation

$$Sb = m \times b \times fs \times Y\times\pi$$

Where;

$$Y=0.484-\left(\frac{2.86}{12}\right)$$

Sb = m x b x fsxYp

$$= m \times 10m \times S_{ult}/3 \times 0.2456$$

= 327.47 m2---------- (B)

Equate (A) & (B)

$$327.47\ m2 = 27 \Rightarrow m= 0.2872$$

Selecting standard module =1.5 mm this is done according to the geometry of the flywheel drum standard gears and pinionsof following details are selected. Hence a larger module is selected which gives maximum tooth depth. Hence the gear selected as below, 4.5.1 Gear data

No. of teeth on Flywheel gear = 74

No. of teeth on pinion = 12

Module = 1.5mm

4.5.2Check for beam strength

$$Fs=\frac{Sb}{Peff}$$

$$=\frac{327.47\times1.5\times1.5}{166.67}=4.4$$

Since Factor of safety Fs = 4.4 design is safe

4.6. Design of Screw

Self-locking characteristic of the system is given by the relation

$$\text{Tan}\,\gamma 1=\frac{\mu}{S1}$$

$\gamma 1$ = Lead angle, μ= Coefficient of friction, S1= Factor of safety.

Coefficient of friction for a range of materialcombinations

	Dry	Lubricated
steel-steel	0.05...00.6	0.015

Considering factor of safety (S1) = 1.6

$\Rightarrow \gamma 1 = 1.8°$

Thus adopting following specifications of thread angle for screw:

Lead angle on output threaded ring = 1.8 °, pitch (L) = 10mm

$$\tan\gamma 1=\frac{L}{\pi dm}$$

$$\Rightarrow dm= 101.28 \text{ mm}$$

d = Nominal /outer diameter (mm) = 101.28 mm

dc = core / inner diameter (mm) = 81 mm

dm = mean diameter (mm) = 96.28 mm

4.6.1 Design Check for Input Worm

$$M_t = W \times (dm/2) \tan(Ø+\gamma)$$

Where, W= Axial load, Ø = friction angle, γ = Lead angle

4.6.2 Friction Angle

Condition	Average coefficient of friction	
	Starting	Running
Average quality of material & workmanship & average running conditions	0.18	0.13

Table -5: Coefficient of friction under different condition

$\Rightarrow$ μ= tan Ø

$\Rightarrow$ 0.18 = tan Ø $\Rightarrow$ Ø = 10.2

Assuming that load of 40 kg is carried by the drum of 100 mm diameter, then the resultant torque

T= 40000X 50 = 2000000 N-mm

Mt = Wx96.28/2 x tan (10.2 + 1.8) -------- (A)

Mt = 10.23xW N-mm -------- (B)

Equating (A) & (B)

W = 195.50 KN ------ Theoretical load carrying capacity

4.6.2 Material Selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
40Cr1	1100	900

Table -6: Material Selection for Screw

Direct Tensile or Compressive stress due to an axial load:

$\Rightarrow fc_{act} = W / A$

$$\Rightarrow fc_{act}=\frac{W}{\frac{\pi}{4}x\, dc^2}$$

$$\Rightarrow fc_{act}=\frac{195.50x\,10^3}{\frac{\pi}{4}x\,81^2}$$

$$\Rightarrow fc_{act} = 37.93 \text{ N/mm2}$$

As $fc_{act} < fc_{all}$; Screw is safe in compression.

Torsional shear stress:

$T = Mt = \pi/16 \times fs_{act} \times dc^3$

$200 \times 10^3 = \pi / 16\, fs_{act} x (81)^3$

$fs_{act} = 19.16$ N/mm^2

As $fs_{act} < fs_{all}$; the screw is safe in torsion.

5. EXPERIMENTAL RESULTS AND VALIDATIONS

5.1Conventional Worm Gear System

The tests are carried out on the conventional system i.e. Worm and Gear system. In these tests the values of efforts required to lift different amounts of loads are calculates and find out system efficiency.

5.1.1Observation table

Sr. No.	Load (gm)	Effort (gm)	Load (N)	Effort (N)
1	2000	120	19.62	1.1772
2	2500	130	24.52	1.2753
3	5000	140	49.05	1.3734
4	7500	160	73.57	1.5696
5	10000	170	98.10	1.6677
6	12500	190	122.62	1.8639

Table -7: Observation Table for conventional system

Sample Calculation (For 5000 Gm Load)

Let L = Radius of the wheel (5.89)

r = Radius of the load drum (6.36)

W = Load lifted

P = Effort applied to lift the load

T = Number of teeth on the worm wheel (100)

$$\text{V.R} = \frac{(\text{ Distance moved by P })}{(\text{ Distance moved by W })}$$

$$\Rightarrow \frac{2\pi L}{\frac{2\pi r}{T}} \Rightarrow \frac{LT}{r}$$

$$\Rightarrow \quad 92.61$$

M.A = W / P

η = M.A / V.R

Load is taken in gm but for our calculations we have to convert it into N.

Load W = 5000 gm = (5000/1000)x9.81=49.05 N

Similarly Effort P = 140 gm = (140/1000) x9.81 = 1.3734 N M.A. = W/P = 49.05/1.3734 = 35.71

Therefore Efficiency of the system η = M.A. / V.R.

= 35.71/92.61

= 38.56 %

5.1.2Result table

Results in terms of efficiency for different amount of loads and efforts are calculated according to the sample calculations and are tabulated as given below,

Sr. No.	LoadW (N)	Effort P (N)	M.A (W/P)	V.R	Efficiency (M.A/V.R)
1	19.62	1.1772	16.67	92.61	17.99
2	24.52	1.2753	19.22	92.61	20.76
3	49.05	1.3734	35.71	92.61	38.56
4	73.57	1.5696	46.87	92.61	50.61
5	98.10	1.6677	58.82	92.61	63.51
6	122.62	1.8639	65.78	92.61	71.03

Table -8: Observation Table for conventional system

5.2 Zero Slip Lifter Mechanism

Test on Zero Slip Lifter Mechanism system are conducted in order to determine the efficiency of the system. For calculating efficiency of the system we need to determine the output power of the system.

These readings are taken on the output shaft of the Zero Slip Lifter Mechanism system in order to calculate the output power of the system. The arrangement for setup is as discussed above. And the readings observed are as follows,

Sr. No.	W in gm	W in N	RPM
1	2000	19.62	11
2	2500	24.52	10.8
3	5000	49.05	9.3
4	7500	73.57	9.1
5	10000	98.10	8.9
6	12500	122.62	8.4

Table -9: Observation table for Output shaft

Sample Calculations for 10000 gm. Load

Output speed = 8.9 rpm

Output torque= Weight in pan x Radius of pulley

= 10x9.81x 50= 4905N-mmT_{op} = 4.905 N-m

Output Power :-($P_{o/p}$)

$$P_{o/p}=\frac{2\pi NTop}{60}$$

$$P_{o/p}=\frac{2 \times \pi \times 8.9 \times 4.905}{60}$$

$$P_{o/p}=4.571 \text{ watt}$$

As $P_{i/p}$ = 6 Watt

Efficiency:- (η)

$\eta = P_{o/p} / P_{i/p} = 76.18$ %

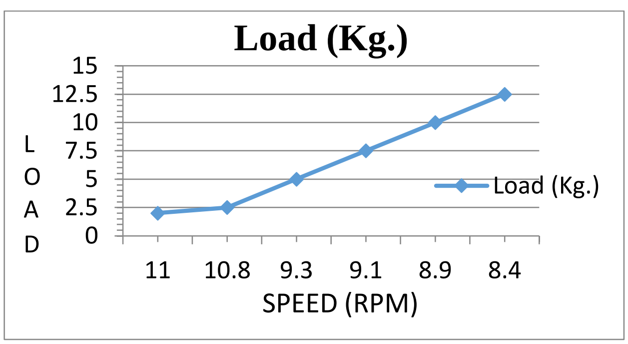
5.2.1 Result Table

Sr. No.	Load (gm)	Speed ($N_{o/p}$)	Torque ($T_{o/p}$)	Power ($P_{o/p}$)	% Efficiency (η)
1	2000	11	0.981	1.1300	18.83
2	2500	10.8	1.22625	1.3868	23.11
3	5000	9.3	2.4525	2.3884	39.80
4	7500	9.1	3.67875	3.5056	58.42
5	10000	8.9	4.905	4.5714	76.18
6	12500	8.4	6.13125	5.3933	89.88

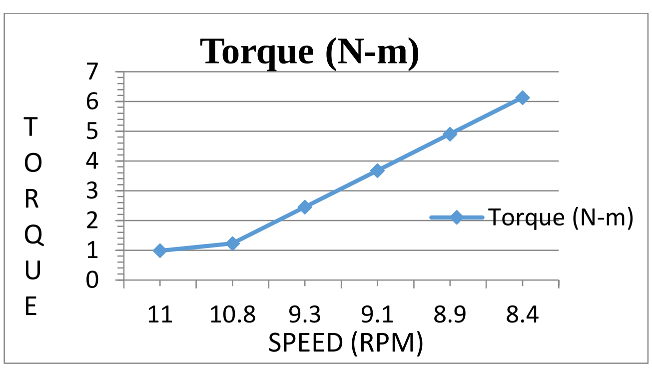
Table -10: Result table for Zero Slip Lifter Mechanism sys.

6. GRAPHICAL REPRESENTATION

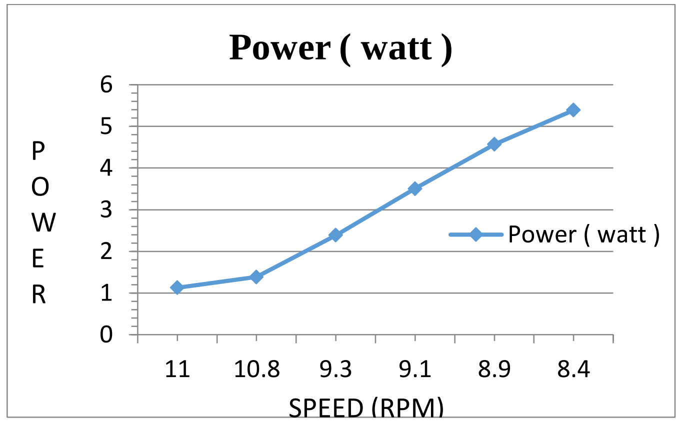
6.1 Load vs. Speed



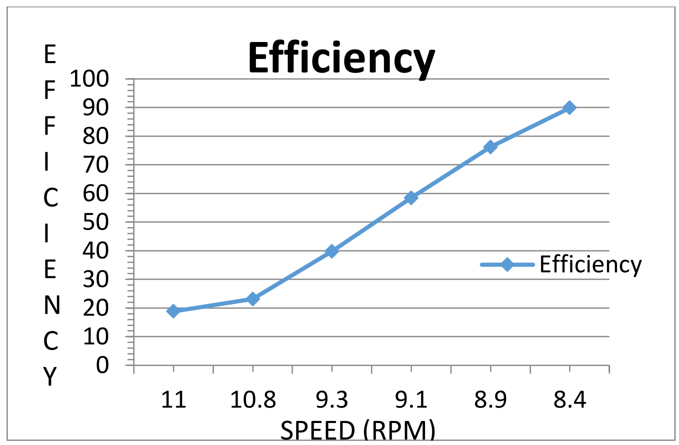
6.2 Torque vs. Speed



6.3 Power vs. Speed



6.4 Efficiency vs. Speed



7. CONCLUSION

The device shows load lifting ability in vertical upward direction with instantaneous self locking also shows load lowering ability in vertical downward direction with instantaneous self locking .There is trivial deceleration of the load drum with increase in load. Device shows increase in power output with increase in load with trivial drop in load speed also shows increase in transmission efficiency with increase in load with trivial drop in load speed maximum efficiency being 90% which is greater than that of conventional worm gear system.

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