DESIGN AND ANALYSIS OF WORM PAIR USED IN SELF-LOCKING SYSTEM WITH DEVELOPMENT OF MANUAL CLUTCH

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Abstract

Mostly in material handling equipment gear drives are used. In these systems when the driving torque is suddenly decreased due to power off or any mechanical failure at the input shaft of the system, then gears will be rotating either in the same direction driven by the system inertia, or in the opposite direction driven by the resistant output load due to gravity, spring load, etc. The latter condition is known as back driving. However, there are also solutions in gear transmission that prevent inertial motion or back driving using self-locking gears without any additional devices. Self-locking is the ability of gear system which constitute a drive which gives the input gear the freedom to rotate the output gear in either directions but the output gear locks with input when an outside torque attempts to rotate the output in either direction. Worm gear pair is also a self-locking gear pair but it is having very low efficiency around 40%, when made self-locking. In our project, self-locking property is obtained by using worm pair which is in mesh with each other. This system is simple in construction. The efficiency around 90% can be obtained as compared to 40% of conventional worm gear system.

Keywords: *worm, efficiency, self-locking*

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1. INTRODUCTION

The term self-locking as applied to gear systems denotes a drive which gives the input gear the freedom to rotate the output gear in either directions but the output gear locks with input when an outside torque attempts to rotate the output in either direction. This characteristic is often sought after by designers who want to be sure that the loads on the output side of the system cannot affect the position of the gears. Worm gears are one of the few gear systems that can be made self-locking, but at the expense of efficiency, they seldom exceed 40% efficiency, when made self-locking. Worm pair self-locking system is an innovation in gearing combines two worm screws to give self-locking characteristics, or to operate as fast acting brake when power is shut off. The Mating worm self-locking system as shown in Fig.1 is a simple dual worm system that not only provided self-locking with over 90% efficiency.

The worm pair drive is quite simply constructed, two threaded rods, or worm screws are meshed together. Each worm is wound in a different direction and has an different pitch angle. For proper mesh the worm axes are not parallel but slightly skewed. (If both worms had the same pitch angle, a normal, reversible drive would have resulted similar to helical gears.) But by selecting proper and different pitch angles, the drive will exhibit self-locking or combination of self-locking as desired.

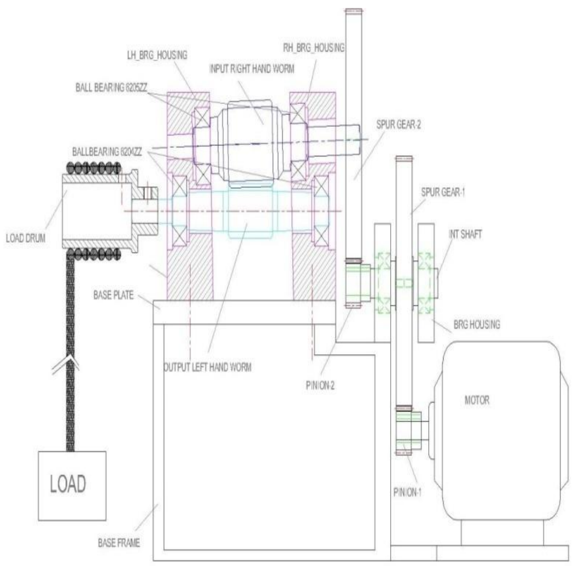

Fig -1: Worm pair system used in load lifting device

2. DESIGN PROCEDURE

2.1 Selection of Drive Motor

Electric motor of following specification is used as the source of power in drive;
230 volt, 50Hz, 0.5 Amp
Power =50 watt (1/15HP)
Speed= 0 to 9000rpm
TEFC construction
Commutator Motor
Input Torque at motor = 60x50 / (2 x Π x 400) = 1.19 N-m
Reduction ratio of gear drive = 32.57
T_{design} = 38.74 N-m

2.2 Design of Input Shaft

2.2.1 Material Selection

Table -1: Material for Input shaft

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
40Cr1	1100	900

2.2.2 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 fult = 198 N/mm^2

fs_{max} =0.3 fyt = 270 N/mm^2

Considering minimum of the above values;

$\Rightarrow fs_{max}$ = 198 N/mm^2

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

Shaft is provided with notch for locking; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow fs_{max}$ = 148.5 N/mm^2

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

T= 38.74 x10^3 N-mm

Assuming 25% overload.

$\Rightarrow$ T design = 1.25 x T = 48.47 x 10^3 N-mm

Check For Torsional Shear Failure of Shaft

Assuming minimum section diameter on input shaft = 16 mm

$$\text{fsact} = \frac{16\times \text{Td}}{\pi \times \text{d3}} = 60.22 \text{ N/mm}^2$$

As $fs_{act} < fs_{all}$ $\Rightarrow$ Input shaft is safe under torsional load

2.3 Design of Output Shaft

2.3.1 Material Selection

Table -2: Material for Output shaft

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
40Cr1	1100	900

2.4 Design of Load Drum Hub

Load drum hub can be considered to be a hollow shaft subjected to torsional load.

2.4.1 Material Selection.

Table -3: Material selection for Load Drum

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
30C8	500	400

2.4.2 As Per ASME Code;

$\Rightarrow fs_{max}$ = 81 N/mm^2

Check for torsional shear failure:-

$$\text{T} = \frac{\pi \times \text{fsact}}{16} \times \frac{\text{Do}^4 - \text{Di\^{}4}}{\text{Do}}$$

$\Rightarrow fs_{act}$ =8.03 N/mm^2

As; $fs_{act} < fs_{max}$ $\Rightarrow$ Hub is safe under torsional load

2.5 Design of Spur Gear Pairs for First Stage & Second Stage Reduction

Power = 01/15 HP = 50 watt
Speed = 1000 rpm
b = 10 m
Tdesign = 0.48N.m
Sultpinion =Sult gear = 400 N/mm2
Service factor (Cs) = 1.5
dp = 16.5
T =T design = 0.48N-m

$$\text{T} = \text{Pt} \times \frac{\text{dp}}{2} \Rightarrow \text{Pt} = 58.18 \text{ N.}$$

$$\text{Peff} = \frac{\text{Pt} \times \text{Cs}}{\text{Cv}}$$

Neglecting effect of Cv as speed is very low

Peff = 87.27 N--------(A)

Lewis Strength equation

Sb = m x b x fs xY

Where;

$$\text{Y} = 0.484 - \frac{2.86}{\text{Z}}$$

Pinion and gear both are of same material and with same number of teeth hence

$$Sb = m \times b \times fs \times Yp$$
$$= m \times 10m \times Sult/3 \times 0.198$$
$$= 263.99\ m2 \text{----------(B)}$$

Equation (A) & (B)

$$263.99\ m2 = 87.2 \Rightarrow m= 0.57$$

Selecting standard module =1.5 mm

This is done according to the geometry of the flywheel drum standard gears and pinions of following details are selected. Hence a larger module is selected which gives maximum tooth depth. Hence the gear selected as below

2.5.1 Gear Data

2.5.1.1 First Pair

No. of teeth on Flywheel gear = 74
No. of teeth on pinion = 10
Module = 1.5mm

2.5.1.2 Second Pair

No. of teeth on Flywheel gear = 44
No. of teeth on pinion = 10
Module = 1.5mm

2.5.2 Design Check for First Stage Reduction

Check for beam strength

$$\mathrm{Fs} = \frac{\mathrm{Sb}}{\mathrm{Peff}} = \frac{1.5\mathrm{x}1.5\mathrm{x}263.99}{87.27} = 6.8$$

Since Factor of safety Fs = 6.8 design is safe

Check for wear strength

Sw = b x Q x $d`_p$ x K
Q = $2xZ_p/(Z_g-Z_p)$ = 0.3125
$d`_p$ = mxZ_p = 15
K = 0.16 x (BHN/100)^2 = 1.96 (BHN = 350)
Sw = 137.81 N
Fs = Sw/Peff = 1.57
Design is safe

2.5.3 Design Check for Second stage reduction

Check for beam strength

$$\mathrm{Fs} = \frac{\mathrm{Sb}}{\mathrm{Peff}} = \frac{1.5\mathrm{x}1.5\mathrm{x}263.99}{87.27} = 6.8$$

Since Fcator of safety Fs = 6.8 design is safe
Check for wear strength
Sw = $bxQxd`_pxK$
Q = $2xZ_p/(Z_g-Z_p)$ = 0.5882
$d`_p$ = mxZ_p = 15
K = 0.16x(BHN/100)^2 = 1.96 (BHN = 350)
Sw = 258.07 N
Fs = Sw/Peff = 2.95
Design is safe

2.6 Design of Screw

Assume
Pitch – L = 10 mm
C – Centre to center distance = 50 mm
For input worm – Lead angle $\gamma_1 = 3^0$
For Output worm – Lead angle $\gamma_2 = 5^0$
From geometry we have

2.6.1 For Input Worm

$$\tan\propto_1 = L / \pi d_1$$

d = Nominal /outer diameter (mm) = 60.73 mm
dc = core / inner diameter (mm) = 48.00 mm
dm = mean diameter (mm) = 54.36 mm

2.6.2 For Output Worm

$$\tan\propto_2 = L / \pi d_2$$

d = Nominal /outer diameter (mm) = 36.38 mm
dc = core / inner diameter (mm) = 27.00 mm
dm = mean diameter (mm) = 31.69 mm

2.6.3 Design Check for Input Worm

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

Where , W= Axial load
Ø = friction angle
∝ = Heilx angle

2.6.4 Friction Angle

Table -4: Coeff. of friction under different condition

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

⇒μ = tan Ø
0.18 = tan Ø ⇒Ø = 10.2

Assuming that load of 500 kg is carried by the drum of 100 mm diameter, then the resultant torque T= 5000X 50 = 250000 N-mm

$$M_t = WX\ 31.7/2 \times \tan (10.2 + 5 \text{ --------(A)}$$

$$M_t = 4.3\ X\ W\ \text{N-mm --------(B)}$$

Equating (A) & (B)

W = 58.13 KN ------ Theoretical load carrying capacity

2.6.5 Material Selection

Table -5: Material Selection for Screw

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
40Cr1	1100	900

Direct Tensile or Compressive stress due to an axial load:

$fc_{act} = W / ((\pi/4) \times dc^2)$
$fc_{act} = 58.13 \times 10^3 / ((\pi/4) \times 27^2)$
$\Rightarrow fc_{act} = 101.52$ N/mm^2

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

Torsional shear stress:

$T = M_t = \pi/16 \times fs_{act} \times dc^3$
$250 \times 10^3 = \pi / 16\ fs_{act} \times (27)^3$
$fs_{act} = 64.68$ N/mm^2
As $fs_{act} < fs_{all}$; the screw is safe in torsion.

2.7 Design and Development of Manual Clutch

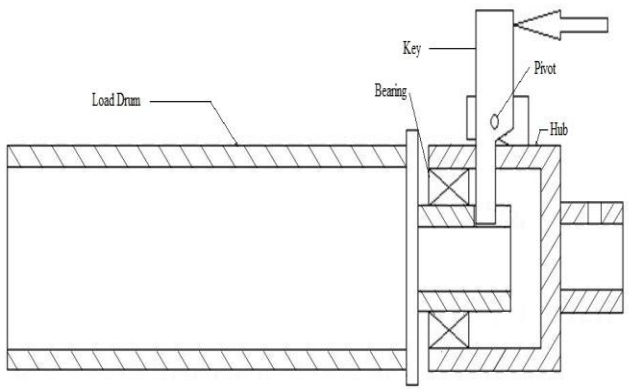

Fig -2: Manual clutch

2.7.1 Construction

This system is as shown in figure. This consists of a bearing, a load drum, a hub and a metallic key which is welded to the hub with the help of a pivot as shown figure.

2.7.2 Working

In case of self-locking systems the occasion can arise where it becomes desirable to quickly release the load. This could be accomplished by introducing a clutch between the pulley and worm but such clutches are generally expensive and space consuming. This invention comprises of a modification of a self-locking mating worm drive.

In our system we have used a load drum which is connected to the output worm of the system. This load drum is connected to the output shaft through a hub. This load drum is fitted to the hub with the help of a bearing. Due to this bearing the load drum can rotate freely relative to the hub. This relative motion of drum can be interrupted with the help of a key or lever. The load drum is provided with a slot at its end towards hub. The key is welded to the hub and it is provided with a fulcrum. When this key is straight then it's one end is engaged with the load drum due to this position the load drum gets fixed with the hub and relative motion between them is avoided. Now when there is a situation when it is required to release the lifted load we can release it by applying some axial force at the free end of the key towards left side as shown in figure. When we apply force at its free end then its another end gets disengaged from the slot provided in the load drum and it is now free to rotate relative to the hub. Thus the self-locking property can thus be removed at will.

3. EXPERIMENTAL RESULTS AND VALIDATION

3.1 Conventional 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. By tabulating these values and by referring various empirical relations for this system efficiency of this system can be determined. The experimental data, sample calculations along with their respective graphs are given below,

3.1.1 Observation Table

Table -6: Observation Table for conventional system

Sr. No.	Load (gm)	Effort (gm)	Load (N)	Effort (N)
1	1500	75	14.715	0.73575
2	2000	95	19.62	0.93195
3	2500	130	24.525	1.2753
4	3000	160	29.43	1.5696
5	4000	210	39.24	2.0601
6	4500	235	44.145	2.30535
7	5000	250	49.05	2.4525

Sample Calculation (For 4000 Gm Load)
Let L = Radius of the wheel
r = Radius of the load drum
W = Load lifted
P = Effort applied to lift the load
T = Number of teeth on the worm wheel

If the worm is single threaded (i.e., for one revolution of the wheel, the worm pushes the worm wheel through one tooth) then for one revolution of the wheel the distance moved by the effort = $2\pi L$

The load drum will move through = 1/ T revolution

Distance, through which the load will move = 2πr / T

V.R = (Distance moved by P) / (Distance moved by W) = (2πL / (2πr / T)) = LT / r

In general, if the worm is n threaded then, V.R = LT / nr
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 = 4000 gm = 4000/1000x9.81=39.24 N

Similarly Effort P = 210 gm = 210/1000x9.81 = 2.0601 N
Now for our system V.R. (Velocity Ratio) is 37.14
M.A. = W/P = 39.24/2.0601 = 19.04

Therefore Efficiency of the system η = M.A. / V.R.
= 19.04/37.14
= 51.18 %

3.1.2 Result Table

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

Table -7: Observation Table for conventional system

Sr. No.	Load (N)	Effort (N)	Efficiency	Avg. Efficiency
1	14.71	0.735	53.850	52.78 %
2	19.62	0.931	56.684	
3	24.52	1.275	51.779	
4	29.43	1.569	50.484	
5	39.24	2.060	51.286	
6	44.14	2.305	51.558	
7	49.05	2.452	53.850	

From above result table we get the efficiency of the conventional worm gear system as 52.78%.

3.2 Mating Worm Pair System

Test on Mating Worm Pair system are conducted in order to determine the efficiency of the system. For calculating efficiency of the system we need to determine the input and output power of the system. By making suitable arrangements as discussed above we have conducted test on this system for various loads and the experimental data and results are as follows,

3.2.1 Observation Table

Input Shaft

These readings are taken on the output shaft of the motor in order to calculate the input power to mating worm pair system. The arrangement for setup is as discussed above. And the readings observed are as follows,

Table -8: Observation table for input shaft

Sr. No.	W in gm	W in N	RPM N
1	100.00	0.98	46.07
2	200.00	1.96	45.69
3	300.00	2.94	45.31
4	400.00	3.92	44.95
5	500.00	4.91	44.42
6	600.00	5.89	43.84
7	700.00	6.87	43.27
8	800.00	7.85	42.78
9	900.00	8.83	42.27
10	1000.00	9.81	41.55

Output Shaft

These readings are taken on the output shaft of the mating worm pair 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,

Table -9: Observation table for Output shaft

Sr. No.	W in gm	W in N	RPM N
1	0.00	0.00	38.90
2	100.00	0.98	38.60
3	200.00	1.96	38.20
4	300.00	2.94	37.60
5	400.00	3.92	37.20
6	500.00	4.91	36.90
7	600.00	5.89	36.70
8	700.00	6.87	36.50
9	800.00	7.85	36.30
10	900.00	8.83	36.00
11	1000.00	9.81	35.80

Sample Calculations (For 500 Gm Load)

For Input Power
Input Torque:-
T_{ip} = Weight in pan x Radius of Dynamometer Pulley
= (0.5 x 9.81) x 36
= 176.6 N.mm
T_{ip}= 0.1766 Nm

Input Power:- ($P_{i/p}$)

$$\mathrm{Pi/p} = \frac{2\pi \mathrm{NTip}}{60}$$

$$= \frac{2 \times \pi \times 44.42 \times 0.1766}{60}$$

$P_{i/p}$ =0.82 watt

For Output Power
Output Torque:-
T_{op} = Weight in pan x Radius of Dynamometer Pulley
= (0.5 x 9.81) x 36
= 176.6 N.mm
T_{op}= 0.1766 Nm

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

$$Po/p = \frac{2\pi NTop}{60}$$
$$= \frac{2 \times \pi \times 36.90 \times 0.1766}{60}$$

$P_{o/p}$ =0.6882 watt

Efficiency:- (η)
$\eta = P_{o/p} / P_{i/p}$
= 83.07 %

3.2.2 Result Table

Table -10: Result table for mating worm pair system

Sr. No.	W in gm	Speed ($N_{o/p}$)	Torque ($T_{o/p}$)	Power ($P_{o/p}$)	Efficiency (η)	Avg Efficiency (η_{Avg})
1	100	38.60	0.035	0.143	83.79	84.046
2	200	38.20	0.071	0.283	83.61	
3	300	37.60	0.106	0.417	82.98	
4	400	37.20	0.141	0.551	82.77	
5	500	36.90	0.177	0.683	83.07	
6	600	36.70	0.212	0.815	83.72	
7	700	36.50	0.247	0.945	84.35	
8	800	36.30	0.283	1.074	84.85	
9	900	36.00	0.318	1.199	85.17	
10	1000	35.80	0.353	1.325	86.15	

3.3. Comparison of Efficiencies Of Conventional Worm Gear And Mating Worm Pair System

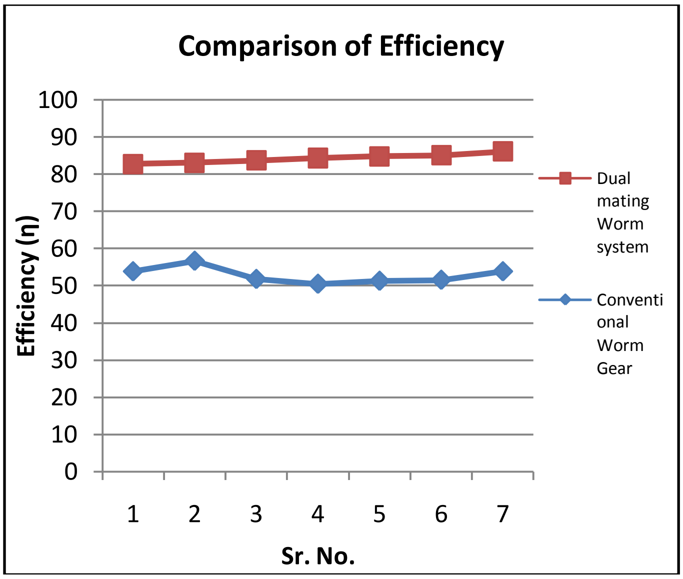

Chart -1: Graph of efficiency comparison of conventional & Worm pair system

We have plotted the efficiencies of conventional worm gear system and worm pair system. We have plotted total seven readings for comparison of both the systems. From graph we can see that the efficiency of conventional worm gear system lies in between 50% to 60% with an average efficiency of 52.78% while the efficiency of mating worm pair system lies in between 80% to 90% with an average efficiency of 84.04%.

4. CONCLUSIONS

From above results and observation we can conclude that the mating worm pair system also exhibits a self-locking ability as that of conventional worm gear system. Also the efficiency of the mating worm pair system is also greater than that of conventional worm gear system. So with having some further modifications related to dimensions of worms such as its helix angle, lead angle and other parameters we can replace this conventional worm gear system with new worm pair self-locking system.

5. FUTURE SCOPE

The present set up is manually operated, hence this limits the load carrying ability of the system, withan appropriate gearing we can increase the mechanical advantage thereby improving the load carrying ability.

The present system has an open casing thereby the need of external lubrication may arise, with appropriate modifications we can make cast iron casing to retain the lubricating oil.

By employing some improved processes and fine workmanship in manufacturing we can further achieve improved quality of parts and which may affect the results in positive manner.

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