Improvement in the Design & Manufacturing of Twin Worm Self Locking Technique and applications

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Abstract
In most of the gear drives, when the driving torque is suddenly reduced as a result of power off, torsional vibration, power outage or any mechanical failure at the transmission input side, 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.
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.Worm gears are one of the few gear systems that can be made self locking, but at the total of efficiency, they rarely exceed 40% efficiency, when made self locking. The experiment is carried in the laboratory with sufficient small prototype model.

Keywords:- Self Locking; Efficiency; Input shaft; Output shaft.

1. Introduction

This innovation in gearing is a combination of two worm screws that gives self locking characteristics or to operate as fast acting brake when power is shut off.
The Mating worm self-locking system is a simple dual worm system that not only provided self locking with maximum efficiency, but also exhibited a new phenomenon called deceleration locking. The mating worm gear drive is quite simply constructed, two threaded rods, or worm screws are meshed together. Each worm is wound in a different direction and has a different pitch angle. For proper mesh the worm axes are not parallel but slightly skewed. But by selecting proper and different pitch angles, the drive exhibit self locking or combination of self locking and deceleration locking as desired. Some of the solutions for these, there is no principle or accuracy in manufacturing these gears but for solution on that the twin worm gear drive is quite simply constructed two thread rod or worm screw meshed together. Each worm is wound in different direction and has different pitch angle for proper mesh the worm axes are not parallel but slightly screwed. By selecting proper and different pitch angle the drive wheel exhibits self locking or combination of self locking and deceleration on desired. 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 shown 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 total of efficiency, they rarely exceed 40% efficiency, when made self locking.
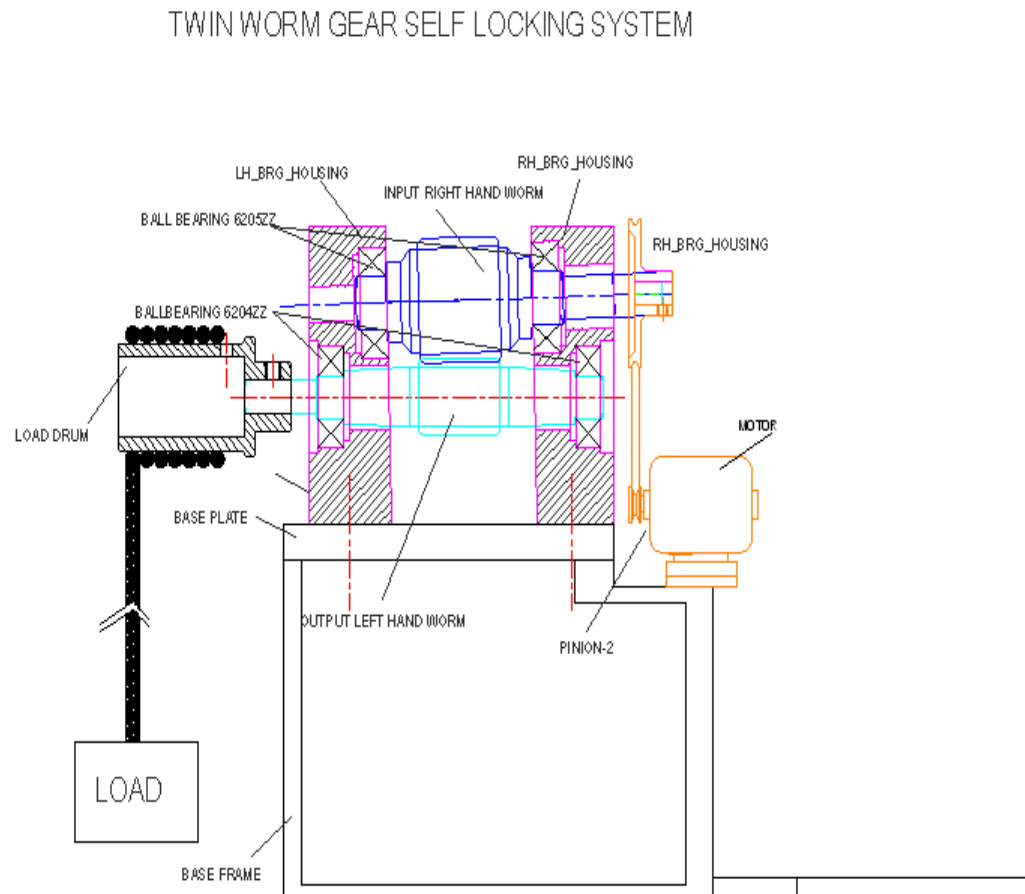
TWIN WORM GEAR SELF LOCKING SYSTEM

Fig. 1.1 Twin worm gear self locking system

2. Design Procedure

System design mainly concerns with various physical constrains, deciding basic working principle, space requirements, arrangements of various components etc.

Following parameters are looked upon in system design.

1. Selection of system based on physical constraints. The mechanical design has direct norms with the system design hence system is designed such that distinctions and dimensions thus obtained in mechanical design can be well fitted in to it.
2. Arrangement of various components made simple to utilize every possible space.
3. Ease of maintenance and servicing achieved by means of simplified layout that enables quick decision assembly of components.
4. Scope of future improvement.

2.1 Approach to Mechanical Design of 'Twin Worm System'

Selection of appropriate material .

- Assuming an appropriate dimension as per system design.
- Design check for failure of component under any possible system of forces.

Our present model is an demonstrative set up in order to show the motion and power transmission capabilities of the proposed 'Twin Worm System'
Thus selecting a drive motor as follows,
Transmission of power to the input shaft is done by means of motor of the following specifications

2.1.1 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) =1.19N-m
Reduction ratio of belt drive = 5
Tdesign = 5.96 N-m

2.2.2 Design of Input Shaft:
The right hand worm gear is the input gear of the system , it has worm threads of 2 module which are machined with right hand of operation with helix angle of 2 degrees. The root clearance is appropriately provided , as well as the pressure angle is maintained at 14.5 degree .It is mounted at an angle of 1.5 degrees to the horizontal, this angle is the opening angle between the two worm shafts .The shaft is held between two medium duty ball bearings i.e.single row deep groove ball bearings with metallic seals at either end.
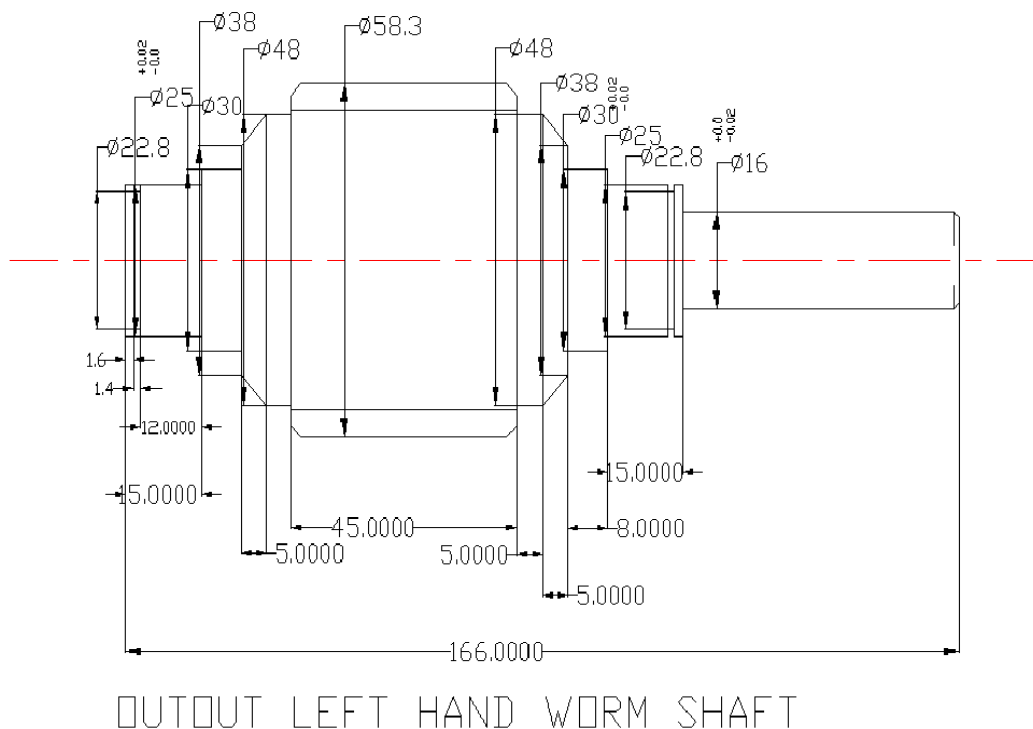

Figure 2.1 Design of Input shaft

2.2.3 ASME Code for Design of Input 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 form various relation.

τ_{max} = 0.18 × Sut
= 0.18 x 800
= 144 N/mm^2

OR
τ_{max} = 0.3 × Syt
=0.3 x 680   =204N/mm^2

Considering minimum of the above values;
$\Rightarrow$ τ_{max} = 144 N/mm^2
This is the allowable value of shear stress that can be induced in the shaft material for safe operation
In selection of ball bearing the main governing factor is the system design of the drive i.e. the size of the ball bearing is of major importance hence we shall first select an appropriate ball bearing first select an appropriate ball bearing first taking into consideration convenience of mounting the planetary pins and then we shall check for the actual life of ball bearing .
P = X Fr + Yfa.
Where;
P=Equivalent dynamic load, (N)
X=Radial load constant
Fr= Radial load (H)
Y = Axial load contact
Fa = Axial load (N)
In our case;
Radial load F_R = RA
= 135N
Axial load (F_a)
$F_a = 0$
Now,   P x Fr =1 x 135
=135
L= (C/P)^p
here, p=3, for single row ball bearings

L = 60 n L h/ 10^6 .
= 60 x1000 x4000 / 10^6
= 240 mrev

Therefor,   C = 842.68N

As required dynamic of bearing is less than the rated dynamic capacity of bearing.
 So bearing is safe.

2.2.4 Design of Output Shaft:

The left hand worm gear is the output gear of the system , it has worm threads of 2 module which are machined with left hand of operation with helix angle of 5 degrees. The root clearance is appropriately provided , as well as the pressure angle is maintained at 14.5 degrees . The shaft is held between two medium duty ball bearings 6204zz ie, single row deep groove ball bearings with metallic seals at either end and it is placed parallel to the horizontal.

In selection of ball bearing the main governing factor is the system design of the drive i.e. the size of the ball bearing is of major importance ; hence we shall first select an appropriate ball bearing first select an appropriate ball bearing first taking into consideration convenience of mounting the planetary pins and then we shall check for the actual life of ball bearing .

P = X Fr + Yfa.

Where ;

P=Equivalent dynamic load ,(N)

X=Radial load constant

Fr= Radial load(H)

Y = Axial load contact

Fa = Axial load (N)

In our case;

Radial load F_R = RA

                    = 100N

Axial load (F_a)

$F_a = 0$

Now,   P x Fr =1 x 100

              =100

But, L= $(C/P)^p$

here, p=3, for single row ball bearings,

$L = 60 \ n \ L \ h/10^6$

   $= 60 \ x1000 \ x4000 \ /10^6$

   $= 240$ mrev

 Therefor,   C = 464.1N

As required dynamic of bearing is less than the rated dynamic capacity of bearing.

So bearing is safe.
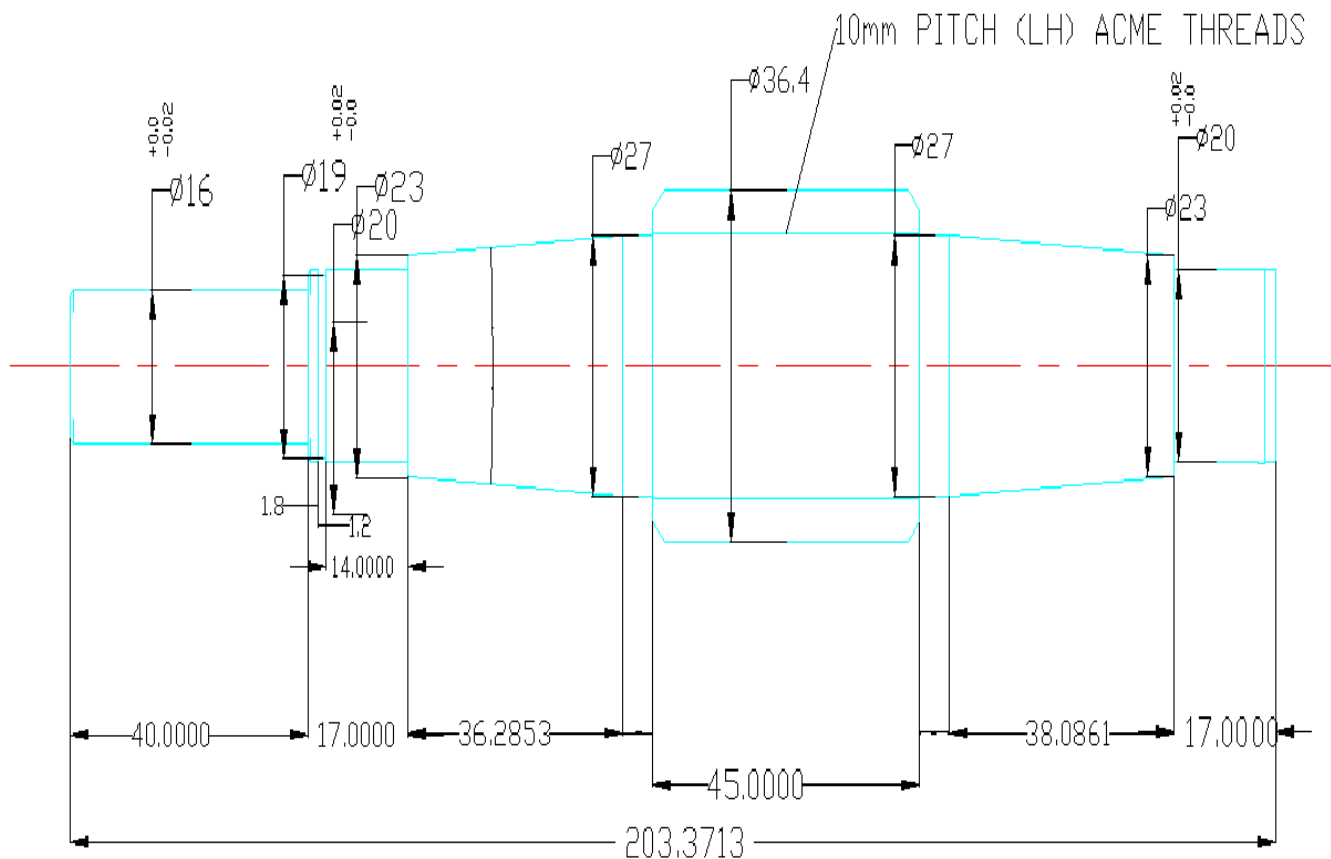
Figure 2.2 Design of Output shaft

2.2.5 ASME codes for Design of Output 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

Then shear stress calculated form various relation.

τ_{max} = 0.18 x Sut
            = 0.18 x 650
            = 117 N/mm^2

OR

τ_{max} = 0.3 x Syt
            =0.3 x 480
            =144 N/mm^2

considering minimum of the above values ;

$\Rightarrow$ τ_{max} = 117 N/mm^2

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

$\Rightarrow \quad \tau_{max} = 87.75 \ N/mm^2$

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

T= 5.96 x10^3 N-mm

Assuming 25% overload.
$\Rightarrow$ T design $= 1.25 \ x \ T$
$= 7.46 \ x10^3$

2.2.6 Check for Torsional Shear failure of shaft:
Assuming minimum section diameter on input shaft = 16 mm
$\Rightarrow$ d = 20 mm
Td $= \pi/20 \ x \ \tau_{act} \ x \ d^3$

$\Rightarrow \tau_{act} = \dfrac{20 \ x \ Td}{\pi \ x \ d^3}$

$= \dfrac{20 \ x \ 7.46 \ x \ 10^3}{\pi \ x \ (16)^3}$
$\Rightarrow \tau_{act} = 49.39/mm^2$

As, $\quad \tau_{act} < \tau_{th}$
$\Rightarrow$ I/P shaft is safe under tensile load.

2.2.7 Design of load drum hub:

Load drum hub can be considered to be a hollow shaft subjected to torsional load.
As Per ASME Code;
$\Rightarrow \tau_{max} = 108 \ N/mm^2$

Check for torsional shear failure:-
$T = \dfrac{\pi \ x \ \tau_{act}}{16} \ x \ \left(\dfrac{Do^4 - Di^4}{Do} \right)$

$7.46 \ x \ 10^3 = \dfrac{\pi \ x \ \tau_{act}}{16} \ x \ \left(\dfrac{32^4 - 16^4}{32} \right)$

$\Rightarrow \tau_{act} = 1.237 \ N/mm^2$
As; $\quad \tau_{act} < \quad \tau_{all}$
$\Rightarrow$ Hub is safe under torsional load.

2.2.8 Design of Screw:

d = Nominal /outer diameter (mm) = 32.36mm
dc = core / inner diameter (mm) = 22.36mm
dm = mean diameter (mm) = 19mm
M_t = W x (dm/2) tan (Ø + ∝)
Where, W= Axial load
Ø = friction angle
∝ = Helix angle

2.2.9 Helix angle:

$$\tan \propto = \frac{L}{\pi \ dm}$$

For the single start sq. thread lead is same as pitch=10

$$\tan\propto = 10/ \ \pi \ x \ 27.36$$
$$\propto = 6.63$$

2.2.10 Friction Angle:

Coefficient of friction under different conditions like average quality of material & workmanship & average running conditions average coefficient of friction taken for starting is 0.18 and for running 0.13.
$\Rightarrow$
μ = tan Ø
0.18 = tan Ø
$\Rightarrow$ Ø = 10.2

Assuming that load of 800 kg is carried by the drum of 120 mm diameter , then the resultant Torque,
T= 8000X 60 =480000N-mm------------(A)
M_t = W X27.32/2 x tan (10.2 + 6.63)
M_t = 4.13 X W N-mm --------(B)
Equating (A) & (B)
W =116.22KN.

3. Nomenclature

P = X Fr + Yfa.
Where;
P=Equivalent dynamic load, (N)
X=Radial load constant
Fr= Radial load (H)
Y = Axial load contact
Fa = Axial load (N)
d = Nominal /outer diameter (mm) = 32.36mm
dc = core / inner diameter (mm) = 22.36mm
dm = mean diameter (mm) = 19mm
M_t = W x (dm/2) tan (Ø + ∝)

Where, W= Axial load

Ø = friction angle

∝ = Helix angle

3.1. Result Table:

Sr. No.	Loading		Unloading		Mean Speed
	Weight (gm)	Speed (rpm)	Weight (gm)	Speed (rpm)	
1.	500	856	500	862	858
2.	600	826	600	828	827
3.	700	682	700	686	684
4.	800	643	800	647	645
5.	900	614	900	616	615
6.	1000	579	1000	583	582

Table 3.1. Measured different speed for different weight

4. Verification and Validation

The results are implemented highly loaded self locking system. The efficiency   89 % is getting experimentally for light model considering different losses. On the basis of this efficiency other heavy twin worm self locking system can be designed also implemented in order to optimization of design and enhance the life of the product which again can be a different project altogether.

5. Results and Discussion:

1) Average Speed: $N = N_1 + N_2/2$

$\qquad\qquad\qquad = 826 + 828/2$

$\qquad\qquad\qquad = 827$

2) Output Torque:-

$\quad T_{dp} \quad$ = Weight in pan x Radius of Pulley

$\qquad\quad = (0.6 \times 9.81) \times 12.5$

$\qquad\quad = 73.575$ N.mm

$\quad T_{dp} \quad = 0.0737575$ N.mm

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

$\quad P_{i/p} \qquad = 2 \pi N T_{i/p}/60$

$\qquad\qquad = 2 \times \pi \times 827 \times 0.08/60$

$\quad P_{i/p} \qquad = 6.92$ watt

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

$\quad P_{o/p} \qquad = 2 \pi N T_{o/p}/60$

$\qquad\qquad = 2 \times \pi \times 827 \times 0.0737575/60$

$\quad P_{o/p} \qquad = 6.18$ watt

5) Efficiency:-

= Output Power / Input Power

= 6.18 x 100/6.92

= 89.30 %

$\Rightarrow$ Efficiency of transmission of orbital mechanism at 90 degree = 89.30%

6. Conclusion

Efficiency of self locking gear box from patent paper by Devid W. Pessen gives up to 92% we have to calculate it from the actually by,

Efficiency:-

$$= \frac{output\ power}{Input\ power}$$

is 89.30 % .Which it gives to nearer to theoretical efficiency.

Also, twin worm self locking system reduces the hole gear box of the lifting mechanism which, reduces the cost and total size. Therefore it is very useful in lift and also in industrial application.

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