IJESRT

INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

DESIGN, DEVELOPMENT AND ANALYSIS OF MULTI-UTILITY ZERO SLIP GRIPPER SYSTEM BY APPLICATION OF MATING WORM SYSTEM

Pooja D. Shintre*, Sambhaji S. Gaikwad
* Mechanical Design, Sinhgad Institute of Technology, Lonavala.
Associate Professor, Sinhgad Institute of Technology, Lonavala.

ABSTRACT
Material handling is the primary activity of every manufacturing organization. It has been estimated that at least 15 to 25% of the cost of the product is attributable to material handling activities. In case of machine tools like lathe or vertical machining centres it is desired to handle heavy jobs, which is conventionally done manually using chain blocks. This method is time consuming, unsafe and takes a lot of labour time adding to unproductive time of machine. Thus there is a need of a modified work handling device in the form of jaw capable to handle heavy pipes as well as plates with equal efficiency. The jaw system incorporates a twin worm drive that is 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 will exhibit either self-locking .

KEYWORDS: Dual worm system, material handling, self locking system, worm screw and Zero slip system.

INTRODUCTION
Unlike many other operations, material handling adds the cost of the product and not its value. It is therefore important first to eliminate or at least minimize the need of material handling and second to minimize the cost of handling. Other than the cost factor the safety of the operator during the material handling operation is also a matter of concern.

In case of machine tools like lathe or vertical machining centers it is desired to handle heavy jobs, which is conventionally done manually using chain blocks. Thus there is a need of a modified work handling device in the form of jaw capable to handle heavy pipes as well as plates with equal efficiency.

PROBLEM STATEMENT
The clamping in the case of the below twin jaw gripper that is presently used is a function of the pull force applied to the pull tie rod attached to one of the jaw arms but sometimes the tie rod pull force may reduce and become insufficient to

Grip the object in the jaw owing to the slack in the pull chain attached to the tie rod, this may lead to slipping of the gripped object further leading to accident that may lead damage to work-piece / property or human life.

OBJECTIVE

The objective of the project is to design the zero-slip gripper system to fool proof and secure as compared to the earlier device mentioned above resulting in maximum safety of work-piece or human life.

PROPOSED SYSTEM

Approach to mechanical design of 'twin worm system'

In design the of parts we shall adopt the following approach;
Selection of appropriate material.

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

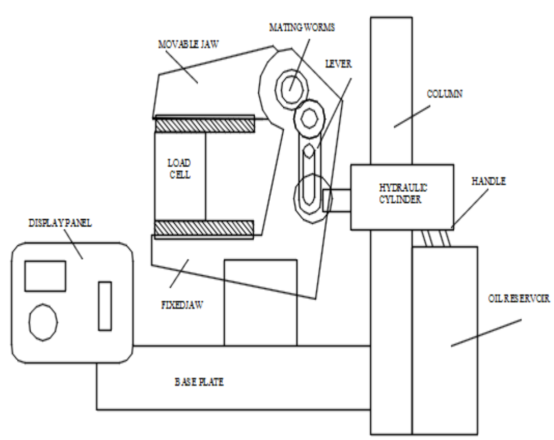
Our present model is a demonstrative set up in order to show the motion and power transmission capabilities of the proposed 'twin worm system'.

Set-up:

The set-up consist mainly of the following
Components:

- Fixed jaw
- Movable jaw
- Mating worm
- Lever
- Column
- Hydraulic cylinder
- Handle
- Oil reservoir
- Base plate
- Display panel

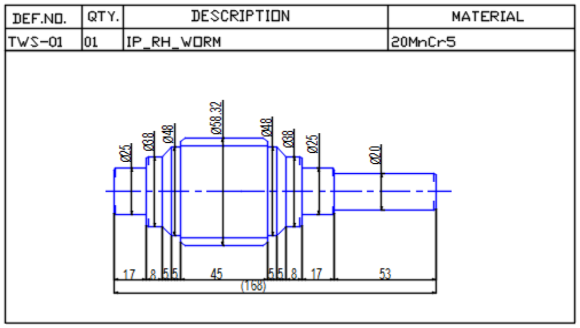
Standard parts are selected from PSG Design Data/Manufacturer Catalogue [6]



LAYOUT OF TEST RIG FOR ZERO SLIP GRIPPER

Here the hydraulic cylinder is used to operate both the jaws (movable and fixed jaws). The display panel is used, which gives the value of the wt. lifted by the jaw.

Design of input shaft



Material selection: - ref: - PSG (1.10 & 1.12) + (1.17)

MATERIAL	ULTIMATE TENSILE STRENGTH N/mm^2	YIELD STRENGTH N/mm^2
20MnCr5	800	680

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.

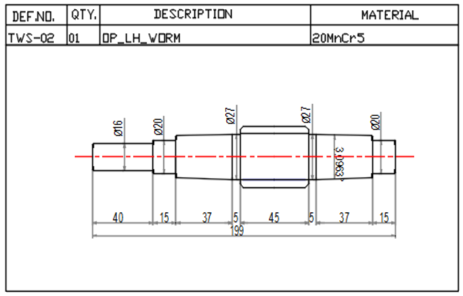
According to ASME code permissible values of shear stress may be calculated from various relations

$fs_{max} = 0.18\ f_{ult}$
$= 0.18 \times 800$
$= 144$N/mm2
OR
$fs_{max} = 0.3\ f_{yt}$
$=0.3 \times 680$
$=204$ 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. The system is operated by means of an handle of length 160 mm and the force exerted by foot is 200 N hence torque exerted by the foot pedal is given by
T= 200 x 160 =32 x10^3 N-mm

Design of output shaft



Material selection: - ref: - PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN 9	650	480

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
= 0.18 x 650

= 117 N/mm^2
OR
fs_{max} = 0.3 fyt
=0.3 x 480
=144 N/mm^2

Considering minimum of the above values
$\Rightarrow$ fs_{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%
$\Rightarrow$ fs_{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= 32 $x10^3$ N-mm
Assuming 25% overload.
$\Rightarrow$ T design = 1.25 x T
= 40 $x10^3$ N-mm

Check for torsional shear failure of shaft.
Assuming minimum section diameter on input shaft = 16 mm
$\Rightarrow$ d = 16 mm

$$Td = \frac{\pi \times fs_{act} \times d^3}{16}$$

$$fs_{act} = \frac{16 \times Td}{\pi \times d^3}$$

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

$$fs_{act} = \frac{16 \times 40 \times 10^3}{\pi \times 16^3}$$

$\Rightarrow fs_{act} = 49.75/mm^2$

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

$\Rightarrow$ I/P shaft is safe under torsional load.

Design (selection of ball bearing)

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.

Ball bearing selection: - series 60

ISI NO	Brg. Basic Design No (SKF)	d	D1	D	D2	B	Basic capacity	
							C kgf	Co Kgf
15B C00	6002	15	17	32	30	9	2550	4400

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 $F_R = RA$

$= 50N$

Axial load (F_a)

$F_a = 0$

P = (x) X Fr

P = 1 x 50 +0

P = 50 N

$$L = \left(\frac{C}{P}\right)^a$$

Here, a=3, for single row ball bearings

$$L = \frac{60\ nLh}{10^6}$$

$$L = \frac{60 \text{ x } 1000 \text{ x } 4000}{10^6}$$

L = 240 mrev

C = 310 kgf

AS required dynamic capacity of bearing is less than the rated dynamic capacity of bearing. Hence Bearing is safe.

Design of load drum hub:

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

Material selection:-

Designation	Ultimate Tensi strength N/mm^2	Yield streng N/mm^2
EN 9	600	420

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

As Per ASME Code;
$\Rightarrow fs_{max} = 108\ N/mm^2$
Check for torsional shear failure:-

$$T = \frac{\pi\ X\ fs\ act}{16}\ X\ \frac{Do^4 - Di^4}{Do}$$

$$4.95\ X\ 10^3 = \frac{\pi\ X\ fs\ act}{16}\ X\ \frac{32^4 - 14^4}{32}$$

$\Rightarrow fs_{act} = 0.77\ N/mm^2$

As; $fs_{act} < fs_{all}$, Hub is safe under torsional load.
Design of screw
Design Check
d = Nominal /outer diameter (mm) = 32.36mm
d_c = core / inner diameter (mm) = 22.36mm
d_m = mean diameter (mm) = 27.36mm

$$Mt = W\ X\ \frac{dm}{2} \tan(\phi + \alpha)$$

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

Helix angle:-

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

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

$$\tan\alpha = \frac{10}{\pi\ x\ 27.36}$$

$$\alpha = 6.63$$

Friction Angle:
Ref: - R.S.Khurmi (Table17.5)
Coefficient of friction under different conditions

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

μ = tan Ø
0.18 = tan Ø
⇒ Ø = 10.2
Assuming that load of 800 kg is carried by the drum of 120 mm diameter, then the resultant torque
T= 800X 60 =48000N-mm ………..(A)
M_t = W X27.36/2 x tan (10.2 + 6.63)
M_t = 4.13 X W N-mm …………(B)
Equating (A) & (B)
W =11.622K N

Material selection for screw:
Ref :- (PSG – 1.12)

Designation	Tensile Strength N/mm^2	Yield Strength N/mm^2
20Mn Cr5	800	680

Direct Tensile or Compressive stress due to an axial load:-

i.) Compressive shear stress :-

$$fc_{\ act} = \frac{W}{\frac{\pi}{4}\ X\ dc^2}$$

$$fc_{\ act} = \frac{11.622\ X\ 10^3}{\frac{\pi}{4}\ X\ 22.36^2}$$

$fc_{\ act} = 29.597\ N/mm^2$

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

ii.) Torsional shear stress :-

$$T = Mt = \frac{\pi \times fs_{act} \times dc3}{16}$$

$$48 \times 10^3 = \frac{\pi \times fs_{act} \times (22.36)3}{16}$$

$fs_{\ act} = 21.867\ N/mm^2$

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

Stresses due to buckling of screw
According to Rankine formula,
Where,

$$W_{cr} = \frac{f_c\ X\ A}{1 + a(\frac{L_e}{k})^2}$$

Where; W_{cr} = Crippling load on screw (N)
A = Area of c/s at root (mm^2)
a = constant
L_e = Equivalent unsupported length of screw (mm) Decided by end conditions.
k = Radius of gyration = dc/4 (mm)
f_c = Yield stress in compression (N/mm^2)
L_e = 0.5L; as both end of screw are considered to be fixed (Ref. PSG Design Data Pg. No. 6.8)
L_e = 0.5x 45
= 22.5mm

$$W_{cr} = \frac{29.5969\ X\ A}{1 + a(\frac{L_e}{k})^2}$$

$$W_{cr} = \frac{29.5969\ X\ (\frac{\pi}{4} X\ 22.36^2)}{1 + \left(\frac{1}{7500}\right)\left(\frac{22.5}{\frac{22.36}{4}}\right)^2}$$

W_{cr} = 11596.91N
W_{cr} = 1182.15 kg

As, The critical load causing buckling is high as compared to actual compressive load of 11.84 kg, the screw is safe in buckling .

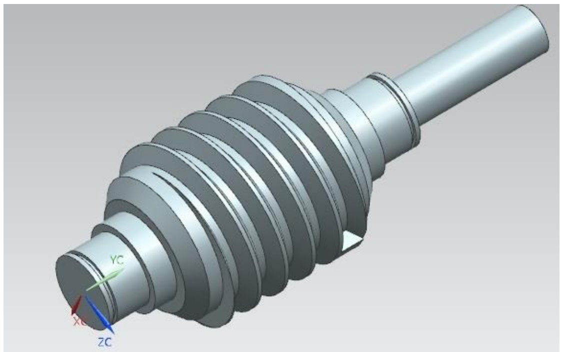
With this, the design of major components of multi-utility zero slip gripper system is complete.

ISSN: 2277-9655
(I2OR), Publication Impact Factor: 3.785

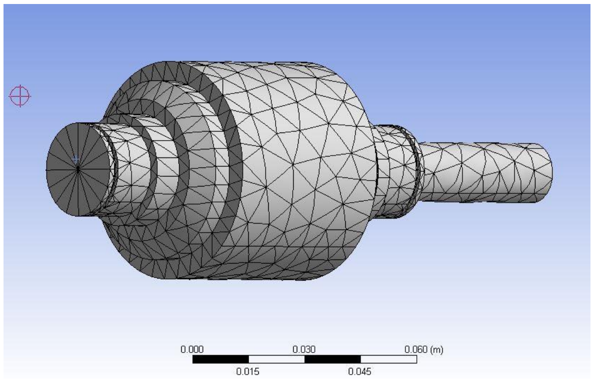
ANALYSIS OF MAJOR COMPONENT

Analysis of input R.H worm

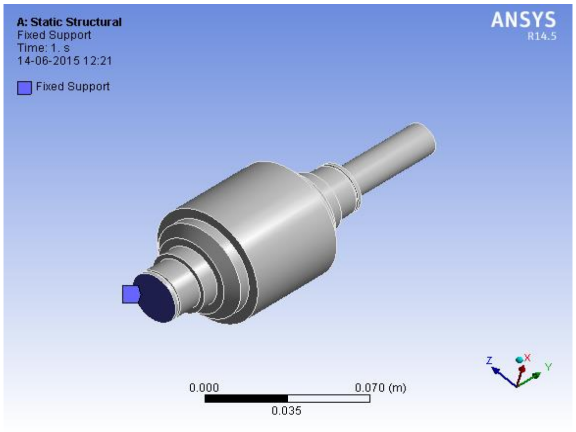
a) Modeling of i/p worm

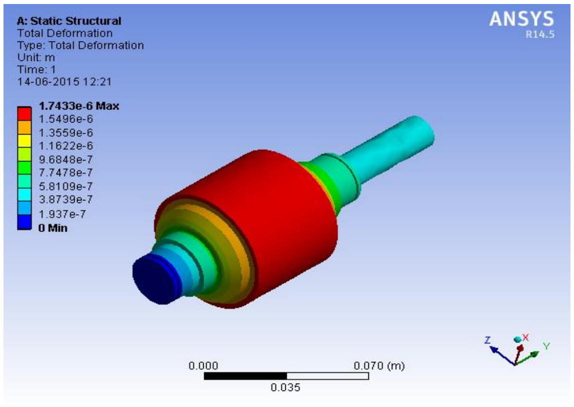


b) Meshing of i/p worm



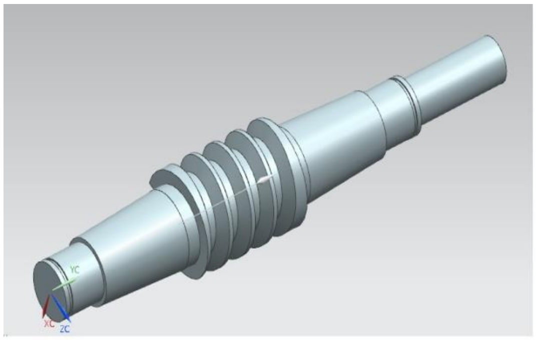
c) Static structural analysis



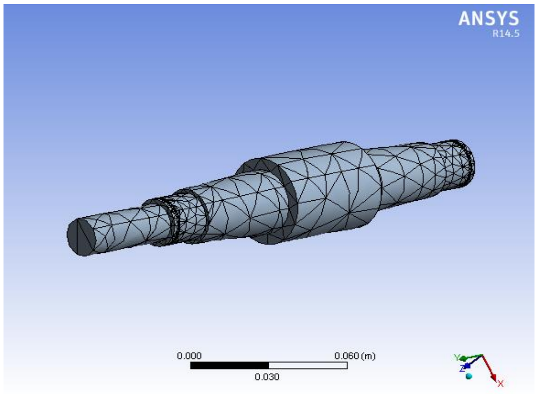


Analysis of output L.H worm

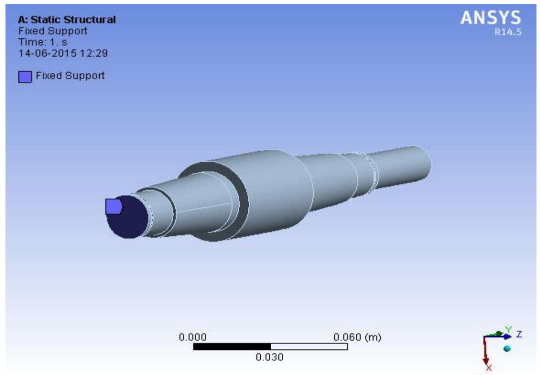
a) Modeling of o/p worm

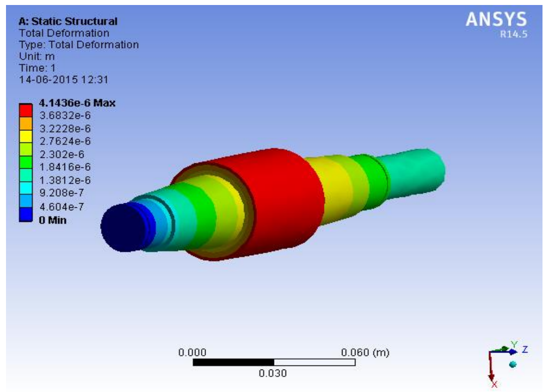


b) Static structural analysis



c) Static structural analysis





With this, the analysis of major components of multi-utility zero slip gripper system is complete.

CONCLUSION

From this new designed gripper system we get following benefits.

a) Maximum gripping force from the gripper system
b) Self locking characteristics.
c) Reduced power consumption.
d) Reduced cost.
e) Secure and safe clamping.

ACKNOWLEDGMENT

Thanks to Prof. S.M.Gaikwad for his valuable contribution in this work.

REFERENCES

[1] Wiktor W Panjuchin Self locking Dual worm gear & tools to produce it US Patent55,22,278 June 1996.
[2] Alex Kapelevich and Elias Taye Applications for Self-Locking Gears Spline & Gear Dec 2008
[3] Tudor Deaconescu Pneumatic Muscle actuated gripper IMECS2011MARCH 2011
[4] Brian Carlisle Ken Goldberg a Pivoting Gripper for Feeding Industrial Parts IEEE Trans. on Systems, Man, And Cybernetics, 4 2012
[5] Madhusmita Senapati V.Balakrishnan, J.Srinivas3 Gripper Considerations for In-Vessel Tile Handling Tasks in a Fusion Reactor Vessel International Journal of Research in Advent Technology, Vol.2, No.7, July 2014
[6] O.Dennis Mullen Improving a Gripper End Effect Pacific Northwest National Laboratory Journal 9, 2003
[7] Earl H. Sigman; Timothy E. Abblett Positive grip lifting mechanism US Patent 4,360,110
[8] PSG Design data book.
[9] SKF bearing catalogue.