DOI: https://doi.org/10.53555/nnmce.v2i4.336 **Publication URL: http://nnpub.org/index.php/MCE/article/view/336+**

DESIGN AND FABRICATION OF EPICYCLIC INTERNAL GEAR PUMP

Omkar S. Pawar[1], Sushil C. Suryavanshi[2], Suyog B. Khamkar[3], Paresh D. Shinde[4], Prof. A.S. Raut[5]
*[*1,2, 3&4]Student in Mechanical Engineering Department of Rajendra Mane college of Engineering and Technology, Ambav (Ratnagiri) Mumbai University, India*
[5]Assistant Professor in Mechanical Engineering Department of Rajendra Mane college of Engineering and Technology, Ambav (Ratnagiri) Mumbai University, India,
*[*1]Email: omkar.pawar657@gmail.com, [2]Email: sushilsuryavanshi7@gmail.com*

Corresponding Author: -
***Email:** omkar.pawar657@gmail.com*

Abstract: -
This paper gives outlines Epicyclic internal gear pump where one sun gear is meshed with three planet gears to achieve variable discharge rate as per requirement. This paper describes techniques for the design, construction, and testing of a Epicyclic internal gear pump. In many applications it is required to drive the actuators hydraulic cylinder or hydraulic motors at variable speed. This is only possible by variable discharge from a variable displacement pump (this pump has very high cost approx Rs.90000/-) so it is not possible to use it. One method employed is to use a pump of higher discharge capacity. But higher capacity means higher cost and higher power consumption. Hence there is need of special pump system at low cost so that the requirement of variable discharge is met easily without much cost and set up.

Keywords: *"Epicyclic gear, Crescent pump, Variable discharge."*

I. INTRODUCTION

This system comprises of three internal gear crescent pumps mounted in parallel around epi-cyclic gear train i.e. the sun gear of the drive train drives the planet gears mounted on the input shaft of each gear pump. The input to all three gear pumps come from a common tank whereas the output from the gear pumps is delivered to a common manifold thus it is possible to get maximum discharge when needed.

The minimum output available is that of one pump. Maximum output available is that of three pumps. This is possible as each of pumps is capable of being de-coupled from circuit.

II. PRINCIPLE

The system entitled ― EPICYCLIC INTERNAL GEAR PUMP works on principle of epicyclic gear train. The basic idea is to get maximum discharge as well as minimum discharge as per requirement. It is possible as each of pumps is capable of being de-coupled from circuit.

III. WORKING PROCEDURE

Basic working of Internal Gear Pump:

It is rotary flow positive displacement pump. It is more advantageous due to low speed and inlet pressure requirement. It consist of one external gear and one internal gear that meshes with each other and with (as shown in fig. b)or without crescent shaped partition (as shown in fig. a.).

When the gears disengage on the inlet side liquid comes into the pump and forces out discharge port by the meshing of the gears.

Applications:

Petrochemical, marine, terminal unloading, asphalt, chemical and general industries applications for transfer, lubrication, processing and low-pressure hydraulic fluids.

Advantages:

Compact size, low speed and inlet pressure requirement.

Disadvantages:

It can be damage while pumping large suspended solids.

It can handle small suspended particles but it causes gradual wear & loose performance.

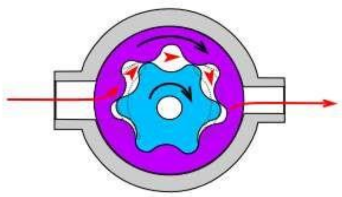


Fig. a

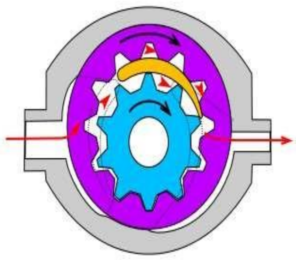


Fig. b

IV. PROJECT SETUP

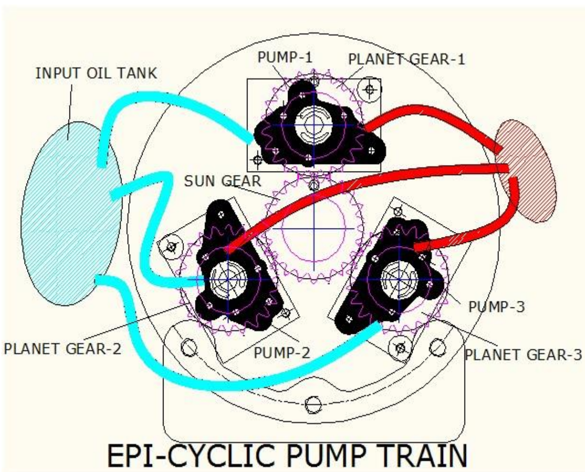


Fig. 1 Detail information of each component of the pump.

Fig. Project Setup (Side view)

Fig. Project Setup (Front view)

V. GENERAL ASSUMPTION IN DESIGN OF MACHINE

MOTOR SELECTION

Thus, selecting a motor of the following specifications
Single phase AC motor
Commutator motor
TEFC construction
Power = 1/15hp=50 watt
Speed= 0-6000 rpm (variable)

DESIGN OF BELT DRIVE

Selection an open belt drive using V-belt;
Reduction ratio = 5
Planning a 1 stage reduction;
A) Motor pulley (ϕ D1) =20mm
B) Main shaft pulley (ϕ D2) =100mm
INPUT DATA
Input Power = 0.05KW
Input Speed =1000 rpm
Centre Distance = 210 mm
Max Belt Speed = 1600 m/min = 26.67 m/s
Groove Angle (2 β) = 400
Coefficient of Friction between belt and pulley = 0.25
Allowable Tensile Stress = 8 N/mm2

Section of belt section
Ref Manufacturers Catalogue

C/S SYMBOL	USUAL LOAD OF DRIVE (KW)	NOMINAL TOP WIDTH (W mm)	NOMINAL THICKNESS T mm	WEIGHT DER METER Kgf
FZ	0.03 - 0.15	6	4	0.05

$$\text{Sin} \propto \;= \frac{O_2M}{O_1O_2} = \frac{R_2-R_1}{x} = \frac{D_2-D_1}{2x} = \frac{100-20}{2*210}$$

$\Rightarrow \;\propto\; = 10.980$

Angle of lap on smaller pulley i.e. motor pulley;

$$\theta o = 180 - 2\propto$$
$$= 180 - 2(10.98)$$
$$\theta = 158.04$$
$$\Rightarrow \theta = 2.75^{o}$$

Now;
Mass of belt/meter length = 0.05 kgf
⇒Centrifugal Tension (T_c) = Mv2
⇒T_c = 0.05 (26.67) 2
⇒ T_c = 35.56 N

Max Tension in belt (T) = f_{all}* Area
= 8 *20
=160N/mm2

A) Tension in Tight side of belt= T1 = T-T_c
=160-35.56
$\boxed{T1 = 124.4 N}$

B) Tension in slack side of belt = T2

$$2.3 \log \frac{T_1}{T_2} = \theta * \mu * \text{cosec } \beta$$
$$= 0.25 * 2.8 * \text{cosec } 20$$
$$\log \frac{T_1}{T_2} = 0.86$$
$$\Rightarrow \frac{T_1}{T_2} = 7.75$$
$$\Rightarrow T2 = 16 \text{ N}$$

POWER TRANSMITTING CAPACITY OF BELT;
P = (T1 - T2) v
= (124.24 - 16) 26.67
P= 3.13 KW
⇒ Belt can safely transmit 0.05 kw power SELECTION OF BELT.
Selection of belt ‘FZ 6 * 600’ from std manufacturers catalogue
MAKE: HELICORD

RESULT TABLE

1.	BELT SELECTED	FZ 6 * 600
2.	Tight side Tension	T1 =124.24 N
3.	Slack side Tension	T2 = 16 N
4.	Motor pulley did. (ϕ D1)	D1 =20 MM
5.	Pulley (a) diameter (ϕD2)	D2 =100MM

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.
Assuming minimum section diameter on input shaft = 16 mm

$$\Rightarrow d = 16 \text{ mm}$$
$$T_d = \Pi / 16 * (f_s)_{act} * d3$$
$$\Rightarrow (f_s)_{act} = \frac{16 * T_d}{\pi * d^3}$$
$$= \frac{16 * 4.84 * 10^3}{\pi * 16^3}$$
$$\Rightarrow (f_s)_{act} = 6.01 \text{ N/mm2}$$
$$\text{As} \quad (f_s)_{act} < (f_s)_{all}$$

⇒ I/P shaft is safe under torsional load

DESIGN OF KEY
Selecting parallel key from standard data book for given application.

For Shaft Diameter	Above Upto	17 22
Key cross section	Width Height	6 6

Material of key ‘EN9’ S_{ult} = 520 N/mm2
S_{yield}= 340 N/mm2
⇒ $(f_s)_{all}$= 85 N/mm2
⇒ $(fs)_{all}$= 170 N/mm2
⇒ Selecting parallel key;
6* 6*30

Check for direct shear failure of key: -

$$T = L * d/2 * t/2 * (f_s)_{act}$$
$$\Rightarrow 4.84 \text{ x } 103 = \frac{30 \text{ x } 6\text{x } 20 \text{ x } (f_s)_{act}}{2}$$
$$\Rightarrow (f_s)_{act} = \frac{4.84 \text{ x } 103 \text{ x } 2}{30*6*20}$$
$$\Rightarrow (f_s)_{act} = 2.68 \text{ N. mm2}$$
$$\text{As } (f_s)_{act} < (f_s)_{all}$$

⇒ Key is safe under shear load

Check for crushing failure of key

$$T = L * d/2 * t/2 * (f_s)_{act}$$
$$7.66\text{x}103 = 30 * 20/2 * 6/2 * (f_s)_{act}$$
$$\Rightarrow (f_s)_{act} = \frac{4.84 \text{ X } 103\text{X } 2 \text{ X } 2}{30*20*6}$$
$$\Rightarrow (f_s)_{act} = 5.367 \text{ N. mm2}$$
$$\text{As } (f_s)_{act} < (f_s)_{all}$$

⇒ Key is safe under crushing load.

DESIGN OF PLANETARY SPUR GEAR BOX

The multi spindle drilling attachment has an drive train in the form of planetary gear system comprising of the central sun gear and three planet gears which drive the three individual spindles on which drill chucks are mounted. The following dimensions are assumed for the gear drive train,

Sun gear
Module = 1.5 mm
No. of teeth = 32
Planet gear
Module = 1.5 mm
No. of teeth = 32

Power = 0.5 HP = 375-watt Speed = 480 rpm b = 10 mm T_{design} = 4.84 N. m
(S_{ult}) pinion=(S_{ult}) gear = 600 N/mm2
Service factor (C_s) = 1.5

$$\text{Now; } T = (P_t) * \frac{d_p}{2}$$
$$\Rightarrow (P_t) = 201.6 \text{ N.}$$
$$(P_{eff}) = \frac{P_t * C_s}{1.5} = \frac{750*1.5}{1.5}$$
$$\text{Now; } C_v = \frac{3}{3+v}$$
$$v = \Pi \text{ DN} = \frac{\Pi \text{ x } 48 \text{ x } 10^{-3} \text{x } 480}{60} = 1.2 \text{ m/sec}$$
$$\Rightarrow C_v = 1.2$$
$$\Rightarrow P_{eff} = \frac{201.6 * 1.5}{1.2} * 1.5$$
$$\boxed{P_{eff} = 378} \quad \text{--------(A)}$$

Lewis Strength equation

$$WT = S*b*y*m$$
Where ;
$$y = \frac{0.484 - 2.86}{Z}$$
$$\Rightarrow y_p = \frac{0.484 - 2..86}{32} = 0.394$$
$$\Rightarrow s_{yp} = 0.394$$

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

$$s_{yp} = s_{yg} = 236.4$$
$$WT = (s_{yp}) * b * m$$
$$= 236.4 * 10m * m$$
$$WT = 2364 \text{ m2} \quad \text{----------(B)}$$
Equation (A) & (B)
$$2364 \text{ m2} = 378$$
$$\Rightarrow m = 0.39$$

Selecting standard module =1.5 mm

GEAR DATA
No. of teeth =32
Module = 1.5 mm
Addendum diameter = 51 mm
Deddendum diameter = 44.25 mm

DESIGN OF PLANET SPINDLES

MATERIAL SELECTION: - Ref: - PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm2	YEILD STRENGTH N/mm2
EN 24	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.

$$= 0.18 \times 800$$
$$= 144 \text{ N/mm2 OR}$$
$$(f_s) \text{ max} = 0.3\ f_{yt}$$
$$=0.3 \times 680 = 204 \text{ N/mm}$$

Considering minimum of the above values;

$$\Rightarrow (f_s) \text{ max} = 144 \text{ N/mm2}$$

DESIGN OF INPUT SHAFT.

MATERIAL SELECTION: - Ref.:- PSG (1.10 & 1.12) + (1.17)

	ULTIMATE TENSILE STRENGTH N/mm2	YEILD STRENGTH N/mm2
EN 24	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.

$$= 0.18 \times 800$$
$$= 144 \text{ N/mm2 OR}$$
$$(f_s)_{max} = 0.3\ f_{yt}$$
$$=0.3 \times 680 \quad =204 \text{ N/ mm2}$$

Considering minimum of the above values;

$$\Rightarrow (f_s)_{max} = 144 \text{ N/mm2}$$

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

$$\Rightarrow (f_s)_{max} = 108 \text{ N/ mm2}$$

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

TO CALCULATE INTERMEDIATE SHAFT TORQUE

$$\text{POWER} = \frac{2\,\Pi\,NT}{60}$$

Motor is 50-watt power, run at 5000 rpm, connected to intermediate shaft by belt pulley arrangement with reduction ratio 1:5

Hence input to input shaft = 1000 rpm

$$\Rightarrow T = \frac{60*P}{2\pi N} = \frac{60*50}{2*\pi*1000}$$
$$\Rightarrow T = 0.48 \text{ N-m}$$

$$\Rightarrow T_{design} = 0.48 \text{ N-m}$$

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

But as per manufacturing considerations we have an H6h7 fit between the pulley and shaft and to achieve this tolerance boring operation is to be done and minimum boring possible on the machine available is 16mm hence consider the minimum section on the shaft to be 16mm

Assuming minimum section diameter on input shaft = 16 mm

$$\Rightarrow d = 16 \text{ mm}$$

$$T_d = \Pi/16 * (f_s)_{act} * d^3$$

$$\Rightarrow (f_s)_{act} = \frac{16 * T_d}{\Pi * d^3}$$

$$= (16 * 0.48 * (10)^3)/(\Pi \times (16)^3)$$

$$\Rightarrow (f_s)_{act} = 0.6 \text{ N/mm2}$$

$$\text{As } (f_s)_{act} < (f_s)_{all}$$

⇒ I/P shaft is safe under torsional load.

DESIGN (SELECTION OF INPUT SHAFT BALL BRG

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. (SKF) Series No.	d	D1	D	D2	B	Basic capacity	
							C kgf	Co Kgf
20A C04	6004	20	23	42	36	12	4650	2850

$P = X F_r + Y F_a$.

Where;

P=Equivalent dynamic load, (N)

X=Radial load constant

F_r= Radial load (H)

Y = Axial load contact

F_a = Axial load (N)

In our case;

Radial load Fr = $T_1 + T_2$=124.4 +16 =140.4 N

Fa = 0

P= 1 * 140.4 N

⇒ L= (C/p) p

Considering 4000 working hours

$$L = \frac{60 \text{ n L h}}{106} = 240 \text{ m rev}$$

$$\Rightarrow 240 = [C/140.4]^3$$

$$\Rightarrow C = 872.5 \text{ N}$$

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

⇒ Bearing is safe.

DESIGN OF GEAR DRUM HUB: -

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

Material selection:

Designation	Ultimate Tensile strength N/mm2	Yield strength N/mm2
CI	500	380

As Per ASME Code; (FACTOR OF SAFETY=3)

⇒ $(f_s)_{max}$ = 166.6 N/mm2

Check for torsional shear failure: -

$$T = \frac{\pi * (f_s)_{act}}{16} * [\frac{Do^4 - Di^4}{Do}]$$

$$0.48 \times 103 = \frac{\pi * (f_s)_{act}}{16} * [\frac{56^4 - 35^4}{56}]$$

$$\Rightarrow (f_s)_{act} = 0.162 \text{ N/mm2}$$

$$\text{As; } (f_s)_{act} < (f_s)_{all}$$

⇒ Hub is safe under torsional load

DESIGN OF SPUR GEAR PAIR FOR DRIVE FROM INPUT SHAFT TO PLANET SHAFT
Power = 01/15 HP = 50 watts
Speed = 1000 rpm
b = 10 m
T_{design} = 0.48 N. m
(S_{ult}) pinion = (S_{ult}) gear = 400 N/mm2
Service factor (Cs) = 1.5 d_p = 55.5

Considering 1.5 module gear with 37 teeth
T = T_{design} = 0.48 N-m

Now; T = $P_t * \frac{d_p}{2}$
⇒ P_t = 17.3N.

$$P_{eff} = \frac{P_t x\ C_s}{C_v} = \frac{17.3*1.5}{C_v}$$

Neglecting effect of C_v as speed is very low

$\boxed{P_{eff} = 26N}$ --------(A)

Lewis Strength equation
WT = S*b*y*m
Where;

$$y = \frac{0.484 - 2.86}{Z}$$

$$\Rightarrow y_p = \frac{0.484 - 2..86}{37} = 0.4060$$

$$\Rightarrow S_{yp} = 162.68$$

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

$S_{yp} = S_{yg} = 162.68$
WT = (S_{yp}) * b * m
=162.68 * 10m *m
WT= $1626.8m^2$ ----------(B)

Equation (A) & (B)

$1626.8\ m^2 = 26$
⇒ m=0.1

Selecting standard module =1.5mm

VI. SELECTED MATERIALS

TABLE I Material Specification

SR NO.	PART CODE	DESCRIPTION	QTY	SPECIFICATION	MATERIAL
1	EGP-1	BASE PLATE	01		EN9
2	EGP-2	MOTOR	01	50watt,0to6000rpm	EN9
3	EGP-3	PULLEY	04		STD
4	EGP-4	MAIN SHAFT	01	DIA. = 16mm	EN24
5	EGP-5	MAIN SHAFT HSG	01		EN9
6	EGP-6	PLANET GEARS	03	MODULE=1.5 No. of teeth=32	EN24
7	EGP-7	SUN GEAR	01	Module=1.5 No. of teeth=32	EN24
8	EGP-8	PUMPSHAFT	03		EN24
9	EGP-9	PUMP SHAFT HSG	03		EN9
10	EGP–10	PUMP BLOCKS	03		AL
11	EGP–11	PUMP	03	Crescent pumps	STD
12	EGP–12	BELT	01	FZ 6 * 600	MS
13	EGP–13	INLET MANIFOLD	01		AL
14	EGP–14	OUTLET MANIFOLD	01		AL
15	EGP–15	MOUNTING PLATE	01		STD
16	EGP–16	BOLTS	06		STD

VII.TESTING PROCEDURE

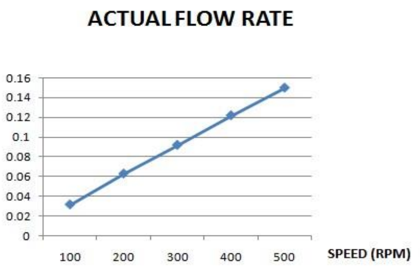
1. Maintain input speed at input =100 rpm.
2. Collect 100 ml of oil in measuring beaker.
3. Note time for collecting 100 ml of oil.
4. Change input speed to 200 rpm.
5. Repeat step 4 & 5.
6. Repeat procedure for 300 rpm, 400rpm and 500 rpm.

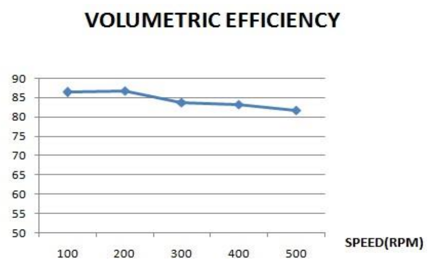
VIII. RESULTS

SR. NO.	SPEED (RPM)	VOLUME IN BEAKER (ml)	TIME (SECONDS)	FLOW RATE (LPM)
01	100	100	189	0.0317
02	200	100	94	0.063
03	300	100	65	0.092
04	400	100	49	0.122
05	500	100	40	0.15

SR. NO.	SPEED (RPM)	ACTUAL FLOW RATE (LPM)	THEORETICAL FLOW RATE	VOLUMETRIC EFFICIENCY
01	100	0.0317	0.036	86.35
02	200	0.063	0.073	86.82
03	300	0.092	0.11	83.7
04	400	0.122	0.147	83.2
05	500	0.15	0.183	81.65

IX. **GRAPHS**





ADVANTAGES

1. Minimum discharge available by use of one pump
2. Maximum discharge is available by use of three pumps in parallel
3. Common drive from single motor using epi-cyclic gear train; so less power required.
4. Low cost of manufacturing
5. Low cost of operation
6. Low maintenance cost
7. Easily available pump units in commercial market so easy to replace parts if they fail.

APPLICATIONS

1. Spring making machines
2. Sheet metal shearing machines
3. Sheet rolling machines
4. Sheet forming and bending machines
5. Conveyor plants

X. CONCLUSION

The pump must be of less cost as well as compact in size so that it can be use for many applications. By considering the requirements of industries we designed a pump which will fulfill their needs. It is approximately 70% cheaper in cost as compare to other conventional pumps available in market. As the parts of pump are easily available in market; they can be replaced when they get damaged.

XII. REFERENCES

[1].PLANETARY GEAR PUMP, Wilson A. Burtis,5011 Harvard Ave, Westminster, Calif, Filed Oct. 24, 1963, Ser.No. 318,670, 7 Claims. (Cl. (103–126).
[2].PISTON PUMP WITH PLANETARY GEAR DRIVE, Albert A. Zalis, Warren, Mass Appl.No.: 843, 955 , Field Oct 20 , 1977.
[3].OIL PUMP FOR CONTROLLING PLANETARY SYSTEM TORQUE , Perry Edward Phelan , Harsens Island , MI (U.S.); Joseph Palazzolo, Livonia, MI(U.S.), Appl. No.: 10/040,848, Filled: Dec. 28, 2001.
[4].Machine Design by R S Khurmi .
[5].Elements of Machine Design by V.B Bhandari.
[6].http://upload.wikimedia.org/wikipedia/commons/c/c4/Gear_pump.png
[7].http://upload.wikimedia.org/wikipedia/commons/7/78/Gear_pump_2.png .
[8].P.S.G. Data Book.