Design and Fabrication of Epicyclic Internal Gear Pump

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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 where as 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.

IV. PROJECT SETUP

![Diagram of EPI-CYCLIC PUMP TRAIN]
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 an 1 stage reduction ;
  A) Motor pulley (\( \phi \) D1) = 20mm
  B) Main shaft pulley (\( \phi \) 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\( \beta \)) = 400
- Coefficient of Friction between belt and pulley = 0.25
- Allowable Tensile Stress = 8 N/mm²
Section of belt section
Ref: Manufacturers Catalogue

<table>
<thead>
<tr>
<th>C/S SYMBOL</th>
<th>USUAL LOAD (KW)</th>
<th>NOMINAL WIDTH (W mm)</th>
<th>NOMINAL THICKNESS T mm</th>
<th>WEIGHT DER METER KgF</th>
</tr>
</thead>
<tbody>
<tr>
<td>FZ</td>
<td>0.03 - 0.15</td>
<td>6</td>
<td>4</td>
<td>0.05</td>
</tr>
</tbody>
</table>

\[ \sin \alpha = \frac{O_2M}{O_1O_2} = \frac{R_2-R_1}{x} = \frac{D_2-D_1}{2x} = \frac{100-20}{2*210} \]

\[ \alpha = 10.980 \]

Angle of lap on smaller pulley i.e. motor pulley:
\[ \theta_o = 180 - 2\alpha \]
\[ = 180 - 2(10.98) \]
\[ \theta = 158.04 \]
\[ \Rightarrow \theta = 2.75^\circ \]

Now:
Mass of belt/meter length = 0.05 kgf
Centrifugal Tension \((T_c) = Mv^2 \)
\[ \Rightarrow T_c = 0.05 (26.67)^2 \]
\[ = 0.05 \times 700 \]
Max Tension in belt \((T) = f_{all} * Area \)
\[ = 8 * 20 \]
\[ = 160N/mm^2 \]

A) Tension in Tight side of belt = \(T_1 = T - T_c\)
\[ T_1 = 124.4 N \]

B) Tension in slack side of belt = \(T_2\)
\[ 2.3 \log \frac{T_1}{T_2} = \theta * \mu * \cosec \beta \]
\[ \Rightarrow \frac{T_1}{T_2} = 7.75 \]
\[ \Rightarrow T_2 = 16 N \]

POWER TRANSMITTING CAPACITY OF BELT:
P = \( (T_1 - T_2) v \)
\[ = (124.24 - 16) 26.67 \]
P = 3.13 KW

\( \Rightarrow \) 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

<table>
<thead>
<tr>
<th></th>
<th>BELT SELECTED</th>
<th>FZ 6 * 600</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Tight side Tension</td>
<td>(T_1 = 124.24 N)</td>
</tr>
<tr>
<td>2</td>
<td>Slack side Tension</td>
<td>(T_2 = 16 N)</td>
</tr>
<tr>
<td>3</td>
<td>Motor pulley did. (d D1)</td>
<td>(D1 = 20 MM)</td>
</tr>
<tr>
<td>4</td>
<td>Pulley (a) diameter (dD2)</td>
<td>(D2 = 100 MM)</td>
</tr>
</tbody>
</table>

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.
Assuming minimum section diameter on input shaft = 16 mm
=> d = 16 mm
\[ T_d = \frac{1}{16} \times (f_s)_{act} \times d^3 \]
\[ (f_s)_{act} = \frac{16T_d}{\pi d^3} \]
\[ (f_s)_{act} = 6.01 \text{ N/mm}^2 \]

As \((f_s)_{act} < (f_s)_{all}\)
=> I/P shaft is safe under torsional load

**DESIGN OF KEY**

Selecting parallel key from standard data book for given application.

<table>
<thead>
<tr>
<th>For Shaft Diameter</th>
<th>Above 17</th>
<th>Upto 22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Key cross section</td>
<td>Width 6</td>
<td>Height 6</td>
</tr>
</tbody>
</table>

Material of key ‘EN9’

- \(S_{ult} = 520 \text{ N/mm}^2\)
- \(S_{yield} = 340 \text{ N/mm}^2\)

\[ (f_s)_{all} = 85 \text{ N/mm}^2 \]
\[ (f_s)_{all} = 170 \text{ N/mm}^2 \]

Selecting parallel key:

6*6*30

Check for direct shear failure of key:

\[ T = L \times \frac{d}{2} \times \frac{t}{2} \times (f_s)_{act} \]
\[ 4.84 \times 10^3 = \frac{30 \times 6 \times 20 \times (f_s)_{act}}{30+6+20} \]
\[ (f_s)_{act} = 2.68 \text{ N. mm}^2 \]

As \((f_s)_{act} < (f_s)_{all}\)
=> Key is safe under shear load

Check for crushing failure of key

\[ T = L \times \frac{d}{2} \times \frac{t}{2} \times (f_s)_{act} \]
\[ 7.66 \times 10^3 = 30 \times 20/2 \times 6/2 \times (f_s)_{act} \]
\[ (f_s)_{act} = \frac{4.84 \times 10^3 \times 2 \times 2}{30+20+6} \]
\[ (f_s)_{act} = 5.367 \text{ N. mm}^2 \]

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}) \text{ pinion} = (S_{ult}) \text{ gear} = 600 N/mm^2
Service factor (C_s) = 1.5

Now; T = \frac{(P_t)}{2} \times \frac{d}{b}
\Rightarrow (P_t) = 201.6 N.

(P_{eff}) = \frac{P_t \times C_s}{1.5} = \frac{750 \times 1.5}{1.5} = 750 N.m

Now; C_v = \frac{3}{\left(\frac{4 + v}{10} \times \frac{3}{480}\right)} = 1.2
v = \pi DN = \frac{\pi \times 48 \times 10 - 3 \times 480}{60} = 1.2 m/sec
\Rightarrow C_v = 1.2

\Rightarrow P_{eff} = \frac{201.6 \times 1.5}{1.2} \times 1.5

P_{eff} = 378 \quad \text{(A)}

Lewis Strength equation
WT = S \times b \times y \times m
Where:
y = \frac{0.484 - 2.86}{32} = 0.394
\Rightarrow y_p = 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}) \times b \times m
= 236.4 \times 10 m \times m
WT = 2364 m^2 \quad \text{(B)}
Equation (A) \& (B)
2364 m^2 = 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
Dedendum diameter = 44.25 mm

DESIGN OF PLANET SPINDLES
MATERIAL SELECTION: - Ref : PSG (1.10 & 1.12) + (1.17)

<table>
<thead>
<tr>
<th>DESIGNATION</th>
<th>ULTIMATE TENSILE STRENGTH N/mm^2</th>
<th>YEILD STRENGTH N/mm^2</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

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.

\[
(f_s)_{max} = 0.18 \times 800
\]

\[
= 144 \text{ N/mm}^2
\]

OR

\[
(f_s)_{max} = 0.3 \times f_{yt}
\]

\[
= 0.3 \times 680 = 204 \text{ N/mm}^2
\]

Considering minimum of the above values:

\[
(f_s)_{max} = 144 \text{ N/mm}^2
\]

**DESIGN OF INPUT SHAFT.**

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

<table>
<thead>
<tr>
<th></th>
<th>ULTIMATE TENSILE STRENGTH N/mm²</th>
<th>YEILD STRENGTH N/mm²</th>
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\[
= 0.3 \times 680 = 204 \text{ N/mm}^2
\]

Considering minimum of the above values:

\[
(f_s)_{max} = 144 \text{ N/mm}^2
\]

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

\[
(f_s)_{max} = 108 \text{ N/mm}^2
\]

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

**TO CALCULATE INTERMEDIATE SHAFT TORQUE**

\[
\text{POWER} = \frac{2z\pi NT}{60}
\]

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

\[
\Rightarrow T = \frac{60 \times P}{2zN} = \frac{60 \times 50}{2 \times 1000}
\]

\[
\Rightarrow T = 0.48 \text{ N-m}
\]

\[
T_{design} = 0.48 \text{ N-m}
\]

**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
⇒ d = 16 mm
⇒ \( T_d = \frac{\Pi}{16} \ast (f_s)_{act} \ast d^3 \)

⇒ \( (f_s)_{act} = \frac{16 \ast T_d}{\Pi \ast d^2} \)

⇒ \( (f_s)_{act} = 0.6 \text{ N/mm}^2 \)

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.**

<table>
<thead>
<tr>
<th>ISI NO</th>
<th>Brg. (SKF) Series No.</th>
<th>d</th>
<th>D1</th>
<th>D</th>
<th>D2</th>
<th>B</th>
<th>Basic capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>20A C04</td>
<td>6004</td>
<td>20</td>
<td>23</td>
<td>42</td>
<td>36</td>
<td>12</td>
<td>C kgf</td>
</tr>
</tbody>
</table>

P = X \( F_r \) + Y \( F_a \).
Where;
P=Equivalent dynamic load, (N)
X=Radial load constant
\( F_r \)= Radial load (H)
\( F_a \)= Axial load contact
\( F_a \)= Axial load (N)

In our case;
Radial load \( F_r= T_1 + T_2 = 124.4 + 16 = 140.4 \text{ N} \)
Axial load \( F_a = 0 \)

P= 1 * 140.4 N
⇒ L= (C/p) p

Considering 4000 working hours
\( L = \frac{60 \pi L}{106} \) = 240 m rev
⇒ 240 = \( \frac{[C/140.4]^3}{106} \)
⇒ C = 872.5 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 |
As Per ASME Code; (FACTOR OF SAFETY=3)

\( \Rightarrow (f_s)_{max} = 166.6 \text{ N/mm}^2 \)

Check for torsional shear failure:

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

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

\( \Rightarrow (f_s)_{act} = 0.162 \text{ N/mm}^2 \)

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

\( \Rightarrow \) Hub is safe under torsional load

DESIGN OF SPUR GEAR PAIR FOR DRIVE FROM INPUT SHAFT TO PLANET SHAFT

Power = 01/15 HP = 50 watt

Speed = 1000 rpm

\( b = 10 \text{ m} \)

\( T_{design} = 0.48 \text{ N. m} \)

(Sult) pinion = (Sult) gear = 400 N/mm2

Service factor (Cs) = 1.5

\( d_p = 55.5 \)

Considering 1.5 module gear with 37 teeth

\( T = T_{design} = 0.48 \text{ N-m} \)

Now; \( T = P_t \times \frac{d_p}{2} \)

\( \Rightarrow P_t = 17.3N. \)

\[
P_{eff} = P_t \times \frac{C_S \times C_V}{C_V} = \frac{17.3 \times 1.5}{C_V}
\]

Neglecting effect of \( C_V \) as speed is very low

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

Lewis Strength equation

\( WT = S^*b*y*m \)

Where:

\[
y = \frac{0.484 - 2.86}{2}
\]

\[
\Rightarrow y_p = \frac{0.484 - 2.86}{37} = 0.4060
\]

\( \Rightarrow S_{yp} = 162.68 \)

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

\( S_{yp} = S_{yg} = 162.68 \)

\( WT = (S_{yp}) \times b \times m \)

\( = 162.68 \times 10m \times m \)

\( WT = 1626.8m^2 \) -----(B)

Equation (A) & (B)

\( 1626.8 m^2 = 26 \)

\( \Rightarrow m = 0.1 \)

Selecting standard module =1.5mm
VI. SELECTED MATERIALS

<table>
<thead>
<tr>
<th>SR NO.</th>
<th>PART CODE</th>
<th>DESCRIPTION</th>
<th>QTY</th>
<th>SPECIFICATION</th>
<th>MATERIAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>EGP-1</td>
<td>BASE PLATE</td>
<td>01</td>
<td></td>
<td>EN9</td>
</tr>
<tr>
<td>2</td>
<td>EGP-2</td>
<td>MOTOR</td>
<td>01</td>
<td>50watt, 0 to 6000rpm</td>
<td>EN9</td>
</tr>
<tr>
<td>3</td>
<td>EGP-3</td>
<td>PULLEY</td>
<td>04</td>
<td></td>
<td>STD</td>
</tr>
<tr>
<td>4</td>
<td>EGP-4</td>
<td>MAIN SHAFT</td>
<td>01</td>
<td>DIA. = 16mm</td>
<td>EN24</td>
</tr>
<tr>
<td>5</td>
<td>EGP-5</td>
<td>MAIN SHAFT HSG</td>
<td>01</td>
<td></td>
<td>EN9</td>
</tr>
<tr>
<td>6</td>
<td>EGP-6</td>
<td>PLANET GEARS</td>
<td>03</td>
<td>MODULE=1.5 No. of teeth=32</td>
<td>EN24</td>
</tr>
<tr>
<td>7</td>
<td>EGP-7</td>
<td>SUN GEAR</td>
<td>01</td>
<td>Module=1.5 No. of teeth=32</td>
<td>EN24</td>
</tr>
<tr>
<td>8</td>
<td>EGP-8</td>
<td>PUMP SHAFT</td>
<td>03</td>
<td></td>
<td>EN24</td>
</tr>
<tr>
<td>9</td>
<td>EGP-9</td>
<td>PUMP SHAFT HSG</td>
<td>03</td>
<td></td>
<td>EN9</td>
</tr>
<tr>
<td>10</td>
<td>EGP-10</td>
<td>PUMP BLOCKS</td>
<td>03</td>
<td></td>
<td>AL</td>
</tr>
<tr>
<td>11</td>
<td>EGP-11</td>
<td>PUMP</td>
<td>03</td>
<td>Crescent pumps</td>
<td>STD</td>
</tr>
<tr>
<td>12</td>
<td>EGP-12</td>
<td>BELT</td>
<td>01</td>
<td>FZ 6 * 600</td>
<td>MS</td>
</tr>
<tr>
<td>13</td>
<td>EGP-13</td>
<td>INLET MANIFOLD</td>
<td>01</td>
<td></td>
<td>AL</td>
</tr>
<tr>
<td>14</td>
<td>EGP-14</td>
<td>OUTLET MANIFOLD</td>
<td>01</td>
<td></td>
<td>AL</td>
</tr>
<tr>
<td>15</td>
<td>EGP-15</td>
<td>MOUNTING PLATE</td>
<td>01</td>
<td></td>
<td>STD</td>
</tr>
<tr>
<td>16</td>
<td>EGP-16</td>
<td>BOLTS</td>
<td>06</td>
<td></td>
<td>STD</td>
</tr>
</tbody>
</table>

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, 400 rpm and 500 rpm.

VIII. RESULTS

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

<table>
<thead>
<tr>
<th>SR. NO.</th>
<th>SPEED (RPM)</th>
<th>ACTUAL FLOW RATE (LPM)</th>
<th>THEORETICAL FLOW RATE</th>
<th>VOLUMETRIC EFFICIENCY</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>100</td>
<td>0.0317</td>
<td>0.036</td>
<td>86.35</td>
</tr>
<tr>
<td>02</td>
<td>200</td>
<td>0.063</td>
<td>0.073</td>
<td>86.82</td>
</tr>
<tr>
<td>03</td>
<td>300</td>
<td>0.092</td>
<td>0.11</td>
<td>83.7</td>
</tr>
<tr>
<td>04</td>
<td>400</td>
<td>0.122</td>
<td>0.147</td>
<td>83.2</td>
</tr>
<tr>
<td>05</td>
<td>500</td>
<td>0.15</td>
<td>0.183</td>
<td>81.65</td>
</tr>
</tbody>
</table>
IX. 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


