Design of epi-cyclic internal gear pump for maximum discharge

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ABSTRACT
In many industrial 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 (which has high cost approx Rs. 90000/-) so it is not feasible 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. This paper deals with the design of such pump systems and its calculations.

Keywords: *Pump, Epicyclic gear, Maximum Discharge, Internal gear pump.*

INTRODUCTION
Internal gear
The internal gear pump is a rotary flow positive displacement pump design, which is well-suited for a wide range of applications due to its relatively low speed and inlet pressure requirements. These designs have only two moving parts and hence have proven reliable, simple to operate, and easy to maintain. They are often a more efficient alternative than a centrifugal pump, especially as viscosity increases. Internal gear pumps have one gear with internally cut gear teeth that mesh with the other gear that has externally cut gear teeth. Pumps of this type are made with or without a crescent-shaped partition. Either gear is capable of driving the other, or the design can be operated in either direction. Designs are available to provide the same direction of flow regardless of the direction of shaft rotation. As the gears come out of mesh on the inlet side, liquid is drawn into the pump. The gears have a fairly long time to come out of mesh allowing for favorable filling. The mechanical contacts between the gears form a part of the moving fluid seal between the inlet and outlet ports. The liquid is forced out the discharge port by the meshing of the gears.

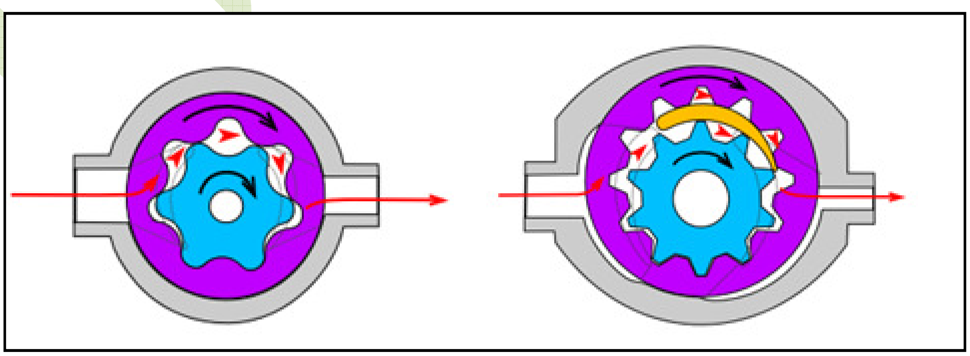

Fig. 1. Internal gear pumps with and without a crescent-shaped partition resp.

Internal gear pumps are commercially available in product families with flows from 1 to 340 m^3/h (5 to 1500 gpm) and discharge pressures to 16 bar (230 psi) for applications covering a viscosity range of 2 to 400,000 cSt (40 to 2,000,000 SSU). Internal gear pumps are made to close tolerances and typically contain at least one bushing in the fluid. They can be damaged when pumping large solids. They can handle small suspended solids in abrasive applications but will gradually wear and lose performance. Materials of construction are dictated by the application and include cast iron, ductile iron, bronze, cast steel, and stainless steel. Small internal gear pumps frequently operate at four-pole motor speeds (1800 rpm) and have operated at two-pole speeds (3600 rpm). As the pump capacity per revolution increases, speeds are reduced. Larger internal gear pumps typically operate below 500 rpm. Operating speeds and flow rates are reduced as the fluid viscosity increases. Pinion-drive internal gear pumps are a distinctive subclass with unique operating characteristics. They are typically direct-drive arrangements operating at two-, four-, and six-pole speeds for flows below 750 L/min (200 gpm) on clear to very light abrasion, low-viscosity, hydrocarbon-based fluids. They are available in single or multistage module designs capable of pressures to 265 bar (4000 psi).

BENEFITS OF GEAR PUMP

1. Operate at high speeds
2. Good efficiency
3. Non-pulsating flow
4. Reliable and easy to maintain
5. Handle higher viscosity fluids
6. Reduced speed for internal gear pumps will be able to pump higher viscosity liquids such as tar, molasses, and bitumen.
7. Suitable for high pressure
8. Internal can have smoother pumping for shear sensitive fluids

LIMITATIONS OF GEAR PUMPS

Pumping heavier viscosity fluids can sometimes build up within the pump and could make the gears rotate slower. Since the fluid is in contact with the gears, it can be extremely sheared as it is transferred to the discharge side of the pump. Internal gear pumps can have overhung loads on shaft bearings and cause pre-mature wear. If any gear pumps are not made to high standards and don’t have tight mechanical clearances between the internal components fluid could be able to leak backwards, which would decrease the pump efficiency. Shear sensitive liquids are not suitable for gear pumps.

APPLICATIONS OF GEAR PUMP

Gear pump provide continuous, non-pulsing flow making it ideal for certain metering applications. Further, these pumps can handle very high pressures ~3000 psi enabling them to be used in hydraulic application. Overall, the gear pumps have a wide variety of applications and these are just a few:

1. Oil pumps in vehicles
2. Used for hydraulic transmission system
3. Pump varies fuel oils and lube oils
4. Used for lubrication in machines
5. Handle corrosive liquids
6. Chemical metering
7. Metering molten plastics in forming synthetic fibers, filaments, films and pipes
8. Metering fuels and chemical additives
9. Internal gear pumps are greatly used in food industry for pumping things like chocolate, fillers and cacao butter

PROBLEM STATEMENT

As shown in fig. 2. There was a requirement of design a low cost pump system which is able to give maximum discharge. Also it should be able to provide flexibility in discharge. We studied the problem in detail and carried out the design of Epicyclic internal gear pump.

NEED FOR PROJECT

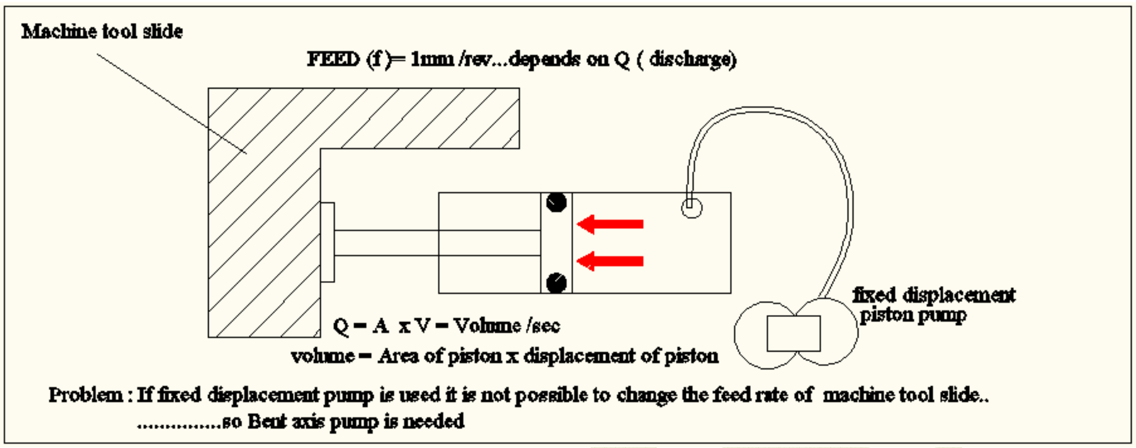

Fig. 2. Problem to be solved by designing a pump

SOLUTION

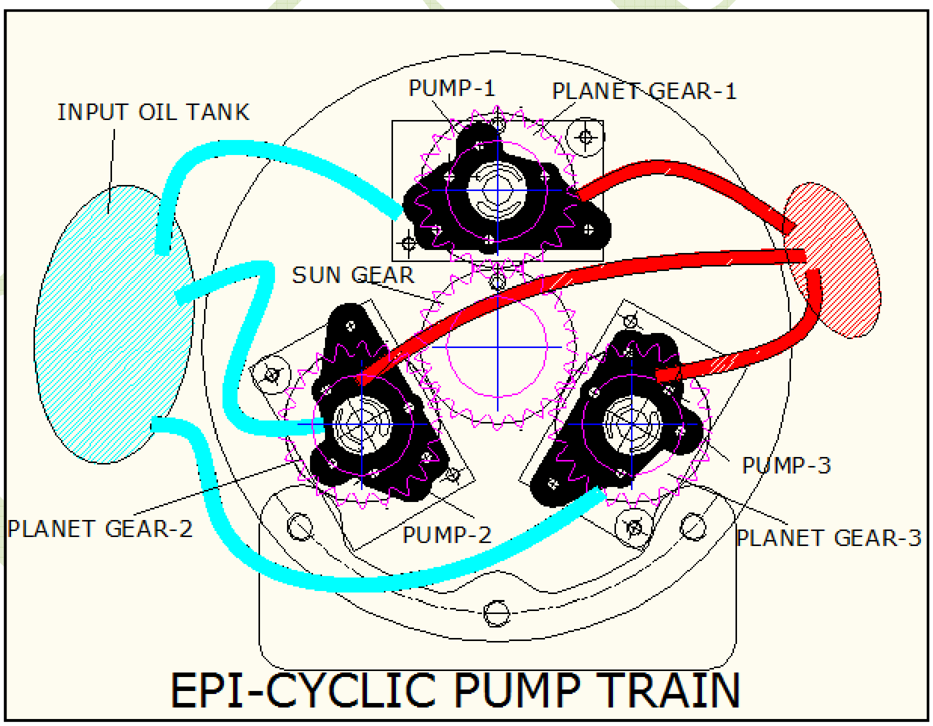

Fig. 3. CAD drawing of pump

DESIGN METHODOLOGY

In our attempt to design a special purpose machine we have adopted a very a very careful approach, the total design work has been divided into two parts mainly;

SYSTEM DESIGN

1. **Mechanical design**

System design mainly concerns with the various physical constraints and ergonomics, space requirements, arrangement of various components on the main frame of machine no of controls position of these controls ease of maintenance scope of further improvement; height of m/c from ground etc. In Mechanical design the components are categoriesed in two parts.

I. Design parts
II. Parts to be purchased.

For design parts detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work.
The various tolerances on work pieces are specified in the manufacturing drawings. The process charts are prepared & passed on to the manufacturing stage .The parts are to be purchased directly are specified &selected from standard catalogues.

2. **System Design**

In system design we mainly concentrate on the following parameter
System selection based on physical constraints:- While selecting any m/c it must be checked whether it is going to be used in large scale or small scale industry in our case it is to be used in small scale industry so space is a major constrain .The system is to be very compact. The mechanical design has direct norms with the system design hence the foremost job is to control the physical parameters.
Arrangement of various components:- Keeping into view the space restriction the components should be laid such that their easy removal or servicing is possible moreover every component should be easily seen & none should be hidden every possible space is utilized in component arrangement.
Components of system: - As already stated system should be compact enough so that it can be accommodated at a corner of a room. All the moving parts should be well closed & compact A compact system gives a better look & structure.
Following are some example of this section
Design of machine height
Energy expenditure in hand operation
Lighting condition of m/c
Chances of failure:- The losses incurred by owner in case of failure of a component are important criteria of design. Factor of safety while doing the mechanical design is kept high so that there are less chances of failure. Periodic maintenance is required to keep the m/c trouble free.
Servicing facility:- The layout of components should be such that easy servicing is possible especially those components which required frequent servicing can be easily dismantled.
Height of m/c from ground: - Fore ease and comfort of operator the height of m/c should be properly decided so that he may not get tired during operation .The m/c should be slightly higher than that the level also enough clearance be provided from ground for cleaning purpose.
Weight of machine: - The total wt of m/c depends upon the selection of material components as well as dimension of components. A higher weighted m/c is difficult for transportation & in case of major break down it becomes difficult to repair.

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 of an open belt drive using V-belt;
Reduction ratio = 5
Planning a 1 stage reduction;
Motor pulley (ϕ D1) =20mm
Main shaft pulley (ϕ D2) =100mm

Input data

Input power = 0.05kw
Input speed =1000 rpm
Center distance = 210 mm
Max belt speed = 1600 m/min = 26.67 m/sec
Groove angle (2 β) = 40^0
Coefficient of friction = 0.25
Between belt and pulley
Allowable tensile strees = 8 N/mm^2

SELECTION OF BELT SECTION

Ref Manufacturers Catalogue

C/s symbol	Usual load of drive (kw)	Nominal top width (wmm)	Nominal thickness t mm	Weight der meter kgf
FZ	0.03 - 0.15	6	4	0.05

$$\sin \propto = \frac{0_2M}{0_1 0_2} = \frac{R_2 - R_1}{x} = \frac{D_2 - D_1}{2x}$$

$$= \frac{100 - 20}{2 \times 210}$$

$$\Rightarrow \propto = 10.98^0$$

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

$$\theta^0 = 180 - 2\propto$$

$$= 180 - 2(10.98)$$

$$= 158.04$$

$$\Rightarrow \theta = 2.75c$$

Now;

Mass of belt /meter length = 0.05 kgf ⇒Centrifugal Tension (Tc) = Mv^2

$\therefore Tc = 0.05\,(26.67)^2$ Tc = 35.56 N

Max Tension in belt (T) = f_{all} x Area

= 8 x 20 =160N/mm^2

Tension in Tight side of belt = T_1 = T-Tc

= 160 - 35.56 $\boxed{T_1 = 124.4 \text{ N}}$

Tension in slack side of belt = T_2

$$2.3 \log \left(\frac{T_1}{T_2}\right) = \theta \times \mu \times \cos\sec \beta$$

= 0.25 x 2.8 x cosec 20

$$\log \frac{T_1}{T^2} = 0.86$$

$$\Rightarrow \frac{T_1}{T_2} = 7.75$$

T_2 = 16 N

POWER TRANSMITTING CAPACITY OF BELT;

P = ($T1$ - T_2) 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 x 600' from STD manufacturer's catalogue

MAKE: HELICORD

RESULT TABLE

1.	BELT SELECTED	FZ 6 x 600
2.	Tight side Tension	T_1 =124.24 N
3.	Slack side Tension	T_2 = 16 N
4.	Motor pulley did.(ϕ D_1)	D_1 =20 MM
5.	Pulley (a) diameter (ϕD_2)	D_2 =100MM

DESIGN OF INPUT SHAFT

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

	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTHN/mm^2
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 x 800 = 144 N/mm^2

OR

fs_{max} = 0.3 fyt =0.3 x 680 =204 N/mm

Considering minimum of the above values

$\Rightarrow$ fs_{max} = 144 N/mm^2

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

$\Rightarrow$ fs_{max} = 108 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

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

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

$$\Rightarrow T = \frac{60 \text{ x } P}{2 \text{ x } \Pi \text{ x } N}$$

$$= \frac{60 \text{ X } 50}{2 \text{ X } \Pi \text{ X } 1000}$$

T = 0.48 N-m $\Rightarrow$ T design = 0.48 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

$Td = \Pi/16 \text{ x } fs_{act} \text{ x } d^3$

$$\Rightarrow Fs_{act} = \frac{16 \text{ x } Td}{\text{X } d^3}$$

$$= \frac{16 \text{ x } 0.48 \text{ x } 10^3}{\text{x } (16)^3}$$

fs_{act} = 0.6 N/mm^2

As $fs_{act} < fs_{all}$ I/P shaft is safe under torsional load

DESIGN: SELECTION OF INPUT SHAFT BALL BEARINGS

In selection of ball bearing the main governing factor is the system design of the drive ie; 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

20A C04	6004	20	23	42	36	12	4650	2850

$$P = X\,Fr + Yfa.$$

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 FR= T1 + T2 =124.4 +16 =140.4 N

$F_a = 0$
$P = 1x\ 140.4$ N
$\Rightarrow L = (C/p)^{p}$
Considering 4000 working hours

$$L = \frac{60\,n\,L\,h}{10^6} = 240 \text{ mrev}$$

$$\Rightarrow 240 = \left(\frac{C}{140.4}\right)$$

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

AS; required dynamic of bearing is less than the rated dynamic capacity, hence bearing is safe.

DESIGN OF BRAKE DRUM HUB: -

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

Material selection

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
CI	500	380

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

$\Rightarrow fs_{max} = 166.6$ N/mm^2

Check for torsional shear failure:-

$$T = \left(\frac{\Pi\ x\ fs_{act}\ x}{16}\right)\frac{Do^4 - Di^4}{Do}$$

$$0.48 \times 10^3 = \left(\frac{\Pi\ x\ fs_{act}\ x}{16}\right)\frac{56^4 - 35^4}{56}$$

$fs_{act} = 0.162$ N/mm^2
As; $fs_{act} < fs_{all}$
Hub is safe under torsional load

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

Power = 01/15 HP = 50 watt
Speed = 1000 rpm
B = 10 m
Tdesign = 0.48 N.m
Sult pinion = Sult gear = 400 N/mm^2
Service factor (Cs) = 1.5
dp = 55.5
Considering 1.5 module gear with 37 teeth T = T design = 0.48 N-m

Now; $T = \dfrac{Pt\ x\ dp}{2}$

$\Rightarrow$Pt = 17.3N.

Peff = $\left(\dfrac{Pt\ x\ Cs}{Cv}\right) = \left(\dfrac{17.3\ x\ 1.5}{Cv}\right)$

Neglecting effect of Cv as speed is very low

$\boxed{\text{Peff} = 26\text{N}}$ -------- (A)

Lewis Strength equation
WT = Sbym
Where; Y = 0.484 – 2.86
Z

$\Rightarrow$ Syp $= 0.484 - \dfrac{2..86}{37}$ = 0.406

Syp = 162.68
Pinion and gear both are of same material and with same number of teeth hence
Syp = Syg = 162.68
W_T = (Syp) x b x m
=162.68 x 10m x m
W_T = 1626.8m^2---------- (B)
Equation (A) & (B)
1626.8. m^2 = 26
$\Rightarrow$m =0.1
Selecting standard module =1.5mm

This is done according to the geometry of the brake drum and roller i.e., the planet gears should remain in 70 % minimum mesh even when brake lever is operated…hence a larger module is selected which gives maximum tooth depth. Hence the planet & sum gear selected.

Conclusion

The problem stated to us was a fascinating one though we as an engineers had to solve it by using our expertise and at the end we succeeded. We were able to counter the problems raised at the site. Also we saved around **Rs. 33000/-** for a high discharge pump.

References

[1] Design Analysis and Testing of a Gear Pump Research Inventy: International Journal of Engineering and Science Vol.3, Issue 2 (May 2013), PP 01-07 ISSN (e): 2278-4721, ISSN (p):2319-6483.
[2] Proceedings 8th Modelica Conference, Dresden, Germany, March 20-22, 2011
[3] Kinematics of an Epicyclic Gear Pump C. K. Wojcik*J. Mech. Des*. 101(3), 449-454 (Jul 01, 1979) (6 pages) doi:10.1115/1.3454077History: Received June 01, 1978; Online October 21, 2010
[4] Reddy et al International Journal of Advanced Engineering Research and Studies E-ISSN2249–8974
[5] Shigley's Mechanical Engineering Design (McGraw-Hill Series in Mechanical Engineering
[6] Pump Characteristics and Applications, Third Edition (MECHANICAL ENGINEERING) By Michael Volk (Author)