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TRIPLE DISCHARGE PUMP WITH EPICYCLIC GEAR TRAIN- A DESIGN

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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- Pump, Epicyclic Gear, Maximum Discharge, Internal Gear Pump.

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. INTERNAL GEAR PUMP

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 favourable 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.

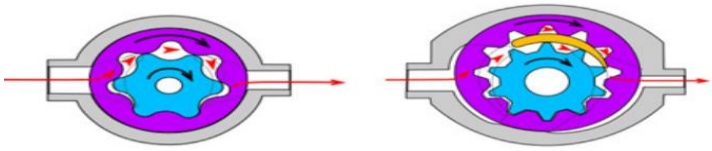

Fig.1.Internal gear pumps with and without a crescent-shaped partition resp.

Internal gear pumps are commercially available in product families with flows from 1 to 340 m3/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).

Internal Gear Pump Overview

Internal gear pumps are exceptionally versatile. While they are often used on thin liquids such as solvents and fuel oil, they excel at efficiently pumping thick liquids such as asphalt, chocolate, and adhesives. The useful viscosity range of an internal gear pump is from 1cPs to over 1,000,000cP.

In addition to their wide viscosity range, the pump has a wide temperature range as well, handling liquids up to 750F / 400C. This is due to the single point of end clearance (the distance between the ends of the rotor gear teeth and the head of the pump). This clearance is adjustable to accommodate high temperature, maximize efficiency for handling high viscosity liquids, and to accommodate for wear.The internal gear pump is non-pulsing, self-priming, and can run dry for short periods. They're also bi-rotational, meaning that the same pump can be used to load and unload vessels. Because internal gear pumps have only two moving parts, they are reliable, simple to operate, and easy to maintain.

III. WORKING

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 or without crescent shaped partition.When the gears disengage on the inlet side liquid comes into the pump and forces out discharge port by the meshing of the gears.

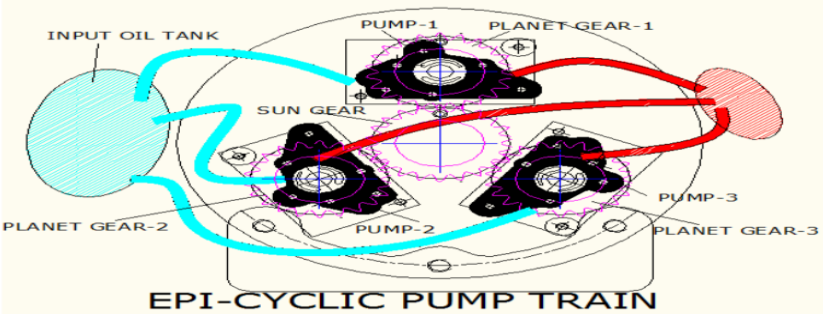

Fig.2.Detail information of each component of the pump

3.1 Liquid enters the suction port between the rotor (large exterior gear) and idler (small interior gear) teeth. The arrows indicate the direction of the pump and liquid.

3.2 Liquid travels through the pump between the teeth of the "gear-within-a-gear" principle.The crescent shape divides the liquid and acts as a seal between the suction and discharge ports.

Fig.3.Working of Internal Gear Pump

3.3 The pump head is now nearly flooded, just prior to forcing the liquid out of the discharge port.Intermeshing gears of the idler and rotor form locked pockets for the liquid which assures volume control.

3.4 Rotor and idler teeth mesh completely to form a seal equidistant from the discharge and suction ports. This seal forces the liquid out of the discharge port.

IV. LITERATURE REVIEW

E.A.P. Egbe (Mechanical Engineering Department, Federal University of Technology Minna, Nigeria) states in his Design Analysis and Testing of a Gear Pump that Nigeria depends heavily on importation of goods and machines. A shift from this trend requires the development of locally available technology. The design analysis of a gear pump that aimed at delivering 4.0913x10-4m3/s (24.55litres/min) of oil was carried out in this work.

Available technology was utilized in the design and fabrication of the external gear pump. The design considered relevant theories and principles which affect the performance of a pump. The parts of the pump were produced locally from available materials. The performance of the pump was characterized and the test results showed a volumetric efficiency of 81.47 per cent at a maximum delivery of 20litres/minute. The discharge dropped with increase in pressure head at a rate of - 0.344Litres/m.

Peng Dong1 , Yanfang Liu2 , Yang Liu2 and Xiangyang Xu2 gives in his paper a method of applying two-pump system in automatic transmissions for energy conservation In order to improve the hydraulic efficiency, modern automatic transmissions tend to apply electric oil pump in their hydraulic system. The electric oil pump can support the mechanical oil pump for cooling, lubrication, and maintaining the line pressure at low engine speeds. In addition, the start–stop function can be realized by means of the electric oil pump; thus, the fuel consumption can be further reduced.. The power loss transfers to heat which requires oil flow for cooling and lubrication. A leakage model is developed to calculate the leakage of the hydraulic system. In order to satisfy the flow requirement, a flow-based control strategy for the electric oil pump is developed.

V. DESIGN

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
- ✓ 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 ; weight of m/c from ground etc.

In Mechanical design the component in two categories.

- Design parts
- 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 tolerance on work 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.

System Design :

In system design we mainly concentrate on the following parameter

3.2.1) 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 care it is to be used in small scale industry So space is a major constrain .The system is to be very

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compact it can be adjusted to corner of a room. The mechanical design has direct norms with the system design hence the foremost job is to control the physical parameters so that the distinction obtained after mechanical design can be well fitted into that.

3.2.2) Arrangement of various component

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.

3.3.3) 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.

3.2.4) Man –m/c Interaction:- The friendliness of m/c with the operation is an important criterion of design. It is application of anatomical

Following are some e.g. of this section

- Design of machine height
- Energy expenditure in hand operation
- Lighting condition of m/c

3.2.5) 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 there over periodic maintenance is required to keep the m/c trouble free.

3.2.6) Servicing facility:- The layout of components should be such that easy servicing is possible especially those components which required frequent servicing can be easily disassembled.

3.2.7) Scope of future improvement:- Arrangement should be provided to expand the scope of work in future such as to convert the m/c motor operated this system can be easy configured to required one.

3.2.8) 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.

3.2.9) 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.

3.3 Mechanical Design

Mechanical design

Mechanical design phase is very important from the view of designer .as whole success of the project depends on the correct deign analysis of the problem.Many preliminary alternatives are eliminated during this phase. Designer should have adequate knowledge above physical properties of material, loads stresses, deformation, failure. Theories and wear analysis , He should identify the external and internal forces acting on the machine parts

These forces may be classified as ;

a) Dead weight forces
b) Friction forces
c) Inertia forces
d) Centrifugal forces
e) Forces generated during power transmission etc

Designer should estimate these forces very accurately by using design equations .If he does not have sufficient information to estimate them he should make certain practical assumptions based on similar conditions which will almost satisfy the functional needs. Assumptions must always be on the safer side.Selection of factors of safety to find working or design stress is another important step in design of working dimensions of machine elements. The correction in the theoretical stress values are to be made according in the kind of loads, shape of parts & service requirements.Selection of material should be made according to the condition of loading shapes of products environment conditions & desirable properties of material.Provision should be made to minimize nearly adopting proper lubrications methods.In ,mechanical design the components are listed down & stored on the basis of their procurement in two categories

- Design parts
- Parts to be purchased

For design parts a detailed design is done & designation thus obtain are compared to the next highest dimension which is ready available in market.

This simplification the assembly as well as post production service work. The various tolerance on the work are specified. The process charts are prepared & passed on to the work are specified.The parts to be purchased directly are selected from various catalogues & specification so that any body can purchased the same from the retail shop with the given specifications.

MOTOR SELECTION *:*

Thus selecting a motor of the following specifications

Single phase AC motor

Commutator motor

TEFC construction

Power = 1/15hp=50 watt

= 0.05kw

Speed =0-6000 rpm

DESIGN OF BELT DRIVE :

Selection of an open belt drive using V-belt;

Reduction ratio = 5

Planning a 1 stage reduction;

Motor pulley (f D1) =20mm

Main shaft pulley (f 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 b) = 400

Coefficient of friction = 0.25

Between belt and pulley

Allowable tensile strees = 8 N/mm2

SELECTION OF BELT *:*

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 α= 02M/0102 = R2-R1 / X = D2-D1/ 2X

=100 – 20 / 2 x 210

α= 10.980

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

Ø = 180 - 2α

=180 - 2(10.98) =158.04 = 2.75^0

Now;

Mass of belt /meter length = 0.05 kgf Centrifugal Tension (Tc) = Mv2

Tc = 0.05 (26.67) 2 Tc = 35.56 N

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Max Tension in belt (T) = f all x Area

= 8 x 20 =160N/mm2

Tension in Tight side of belt = T1 = T-Tc

= 160 - 35.56 T1 = 124.4 N

Tension in slack side of belt = T2

2.3 log T1 = q x μ x cos sec b

= 0.25 x 2.8 x cosec 20

log T_1/T_2 = 0.86

T_1 T_2 = 7.75

T_2=16N

POWER TRANSMITTING CAPACITY OF BELT:

P = $(T_1 – T_2)$ v

= (124.24-16)26.67

= 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 T1	=124.24 N
3.	Slack side Tension T2	= 16 N
4.	Motor pulley did.(D1)	=20 MM
5.	Pulley (a) diameter (D2)	=100MM

Figure4: Belt

DESIGN OF INPUT SHAFT :

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

	ULTIMATE TENSILE STRENGTH N/mm2	YEILD STRENGTHN/mm2
EN24	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/mm2

fs max = 0.3 fyt =0.3 x 680 =204 N/mm

Considering minimum of the above values fs max = 144 N/mm2

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

fs max = 108 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

POWER = $2\pi NT/60$

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

T = 60 x P/2 π x N

= 60 X 5 0/ 2 π X 1000

T = 0.48 N-m

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 = π/16 x fs act x d3

$$Fs\ act = \frac{16 \times Td}{\Pi \times d\ 3}$$

$$= \frac{16 \times 0.48 \times 10\ 3}{\Pi \times (16)\ 3}$$

fs act = 0.6 N/mm2

As fs act < fs all I/P shaft is safe under torsional load

VI. CONCLUSION

In this project we used the only one electric motor for running all the three simultaneously or individual pump due to which we save electricity and we gate the variable flow of liquid.By using planetary gear train and sun gear we can increase discharge also we can save electricity and time.Electricity can be reduced by using only one electric motor for running all the three simultaneously or individual pump and we get variable flow of liquid. which can be use to variable discharge of fluid.Due to compactness of the design of triple discharge pump the cost of the pump can be reduced.

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