Combination Of Fixed And Variable Discharge Pump

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Abstract*- Objective of this project is defined to develop a variable displacement pump by varying the stroke of radial piston pump. This achievement is based on simple four bar linkage with one control link. Control link is joined at one end of connecting link and rotation of output yoke controlled by unidirectional clutch. By controlling the position of control link can vary the output without changing the input. Thus adjustment of the stroke can be done by varying the position of the pivot element. Aim of this project reduce cost of pump as compare to other pump.*

I. INTRODUCTION

A pump is a device that moves fluids (liquids or gases), or sometimes slurries, by mechanical action. Pumps can be classified into three major groups according to the method they use to move the fluid: direct lift, displacement, and gravity pump. Pumps operate by some mechanism (typically reciprocating or rotary), and consume energy to perform mechanical work by moving the fluid. An axial piston pump uses a variable angle swash plate; bent axis piston pump and vane pump. However the efficiency of variable displacement pump decrease at partial load. All of these architectures utilize planar joints that suffer from a trade-off between high mechanical friction and high leakage to maintain hydrodynamic bearings. Hence there is a need to develop a modification in the radial piston pump design that will offer a variable discharge configuration in addition to the advantages of high efficiency and maximum pressure.

1.1. Need for Project

Axial piston pumps with constant pressure and variable flow have extraordinary possibilities for controlling the flow by change of pressure. Owing to pressure feedback, volumetric control of the pump provides a wide application of these pumps in complex hydraulic systems, particularly in aeronautics and space engineering. The major obstacle in application of the bent axis piston pump is extremely high cost over that of the radial piston pump; it ranges in the range of 5 to 6 times the cost of radial piston pump. Hence there is a need to develop a modification in pump so that it will offer a variable discharge con figuration adjustment. Thus our aim is to develop an adjustable linkage that will enable us to vary the discharge and also keep it constant as per our requirement. Obtaining these above possible adjustment in minimum cost is also part our objective.

1.2. Objectives

1) Objective of our project is to develop a mechanism for pumping application.
2) The design so developed should be able to produce both the effect of fixed and variable type of pump.
3) The linkages designed should be able to provide various outputs at various operating position and condition of linkages.
4) Testing of pump to plot its performance curves:
    a) Flow rate vs. Speed.
    b) Efficiency vs. Speed.
5) Comparisons of flow rate vs. control angles at constant speed.
6) Comparative analysis of result of flow rate and cost required to produce the pump in comparison with bent axis or other type configuration pump.

1.3. Proposed Methodology

1.3.1. Theoretical Work:-

1) Literature review. Study of various configurations of hydraulic pumps, and variable displacement linkages using various Handbooks, United State Patent documents, Technical papers, etc.
2) Design and Development:-
    a) System design as to and theoretical derivation of dimensions of the variable displacement kinematic linkage pump parts using Auto –Cad software.
    b) System Design and theoretical derivation of dimensions of the pump for the above derived kinematic linkage system, selection of pump using standard manufacturer catalogue.
    c) System Design and theoretical derivations of linkage as for strength criterion for given pressure and flow specifications

d) Selection of pump and drive for circulation of oil through the hydraulic system to get desired flow rates.
e) Development of diagram for the circuit of hydraulic oil from the modules to the system and flow from the system back to the oil tank.

II. WORKING OF SETUP

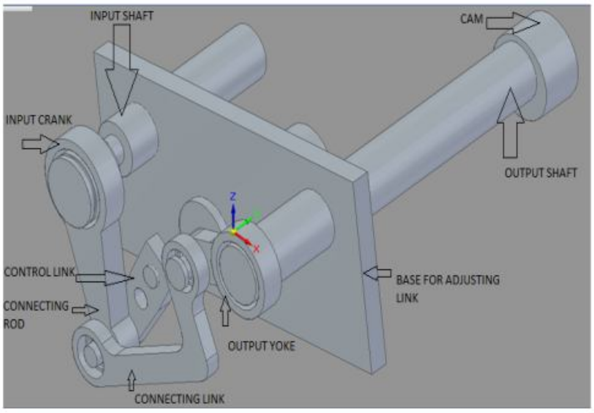


Fig. 1. Assembly

The speeds are instantly changed by turning the handle indicated by ‘G’. On the drive shaft A is mounted a series of eccentrics B. These eccentrics are connected to connecting links C by connecting rod D. As the drive shaft rotates , the eccentrics impart an oscillating movement to the left hand ends of the connecting links ‘C’ and as these are pivoted to the output yoke E they impart oscillatory movement to the roller clutches within yokes ‘E’ .Each reciprocating movement of clutch will cause the drive shaft to rotate a fraction of a revolution , and as the eccentrics are spaced uniformly about the drive shaft , the impulse given to the driven shaft will be successive and over lapping .In this way a uniform rotary movement of the driven shaft is obtained The oscillating movement of the right hand end of the link C determines the amount the driven shaft turns during each impulse, and this oscillating movement depends upon the position of joint M along the path determined by the control link end when the control shaft is rotated about hinge K by handle For example if joint M is moved towards the right by which reciprocating movement of clutch will be shorter , and a longer time will be required to rotate the driven shaft thereby reducing the speed of the output shaft..Obviously an entire range of speeds is covered smoothly, enabling the mechanism to glide from one speed to another.

III. DESIGN CALCULATIONS

3.1 Input Data:

3.1.1 Electric Motor Details

a) POWER= 50 WATT
b) SPEED = 0-9000 rpm
c) OPERATING SPEED = 4000 rpm.

NOW,

$$P = \frac{2\times\pi\times N\times T}{60}$$

$$50 = \frac{2\times\pi\times 1000\times T}{60}$$

$$T = \frac{60\times 50}{2\times\pi\times 1000}$$

$$\mathbf{T = 0.12\ N.m}$$

Belt drive between motor and pulley have a reduction ratio of 1:5

Hence T_{design} = Overload factor × 5 × Tmotor

Considering 100% overload

$T_{design} = 2 \times 5 \times 0.12$

$$\mathbf{T_{design} = 1.2\ N.m}$$

3.2 Design of Input Crank :-

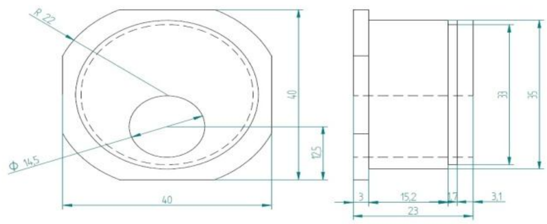


Fig. 2. Input Crank

As per the crank design we selected the bearing 6203 having standard outer diameter as per bearing catlogue is 40mm and inner diameter is 35mm. And this 35mm is our crank diameter.For parabolic action we select 7.5mm eccentricity with centre to centre distance.

$$\text{Force due to eccentricity} = \frac{\text{Tdesign}}{\text{Eccentricity}}$$

$$= \frac{1.2\times 10^3}{7.5}$$

$$= 160\ \text{N}$$

The permissible shear stress,

$$(Fs)all = \frac{F_{yt}}{FOS}$$

$$= \frac{380}{2}$$

$$= 190\ \text{N/mm}^2$$

Moment = 160 × 7.5 = 1200 N/mm

3.3 Design of Input Crank Shaft:-

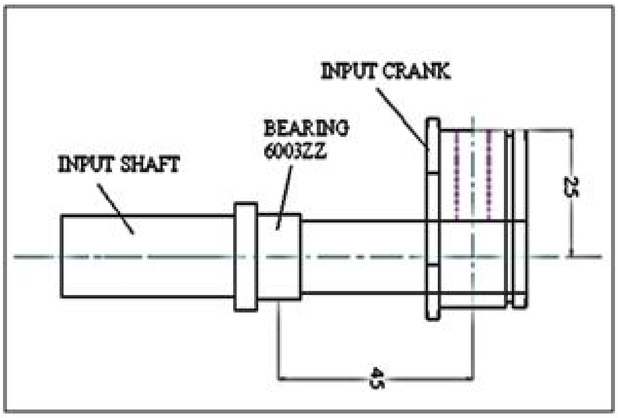


Fig.3. Input Crank Shaft

Diameter according to the maximum shear stress theory

$$D^3 = \left(\frac{32N}{\pi . F_y}\sqrt{T^2}\right)$$

Selection of material :- C40

Yield strength :- 380 N/mm^2

Tensile strength :- 580 N/mm^2

$$d^3 = \frac{32 \times 4000}{\pi \times 380}\left(\sqrt{(10^3 \times 1.2)^2}\right)$$

$$\mathbf{d = 15\ mm}$$

According to torsional equation,

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L}$$

$$\frac{T}{J} = \frac{\tau}{R}$$

$$\frac{1.2 \times 10^3}{4559.36} = \frac{63.16}{L}$$

$$\mathbf{L = 240\ mm}$$

3.4 Design of Connecting Rod

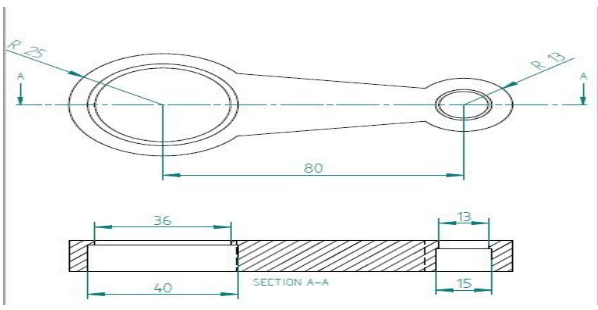


Fig. 4. Connecting Rod

While designing of connecting rod we have to consider the weaker side of the rod. Big end diameter is 40mm as per input crank bearing outside diameter value.

Now,

$$(F)_{\text{small end}} = \frac{\pi}{4} \times d^2 \times P_{max}$$

$$P_{max} = \frac{\text{Force}}{\text{Area}}$$

Our force for bigger end = 160 N

Now, area for bigger end is,

$$(A)_{\text{bigger end}} = \frac{\pi}{4} \times d^2$$

$$= \frac{\pi}{4} \times 40^2$$

$$\mathbf{(A)_{bigger\ end} = 1256.63\ mm^2}$$

$$P_{max} = \frac{160}{1256.63}$$

$$\mathbf{P_{max} = 0.1273}$$

$$(F)_{\text{small end}} = \frac{\pi}{4} \times d^2 \times P_{max}$$

$$160 = \frac{\pi}{4} \times d^2 \times 0.1273$$

$$\mathbf{d = 8\ mm}$$

Thickness,

$$t = 1.25 \times 8$$

$$\mathbf{t = 10\ mm}$$

3.5 Connecting Link

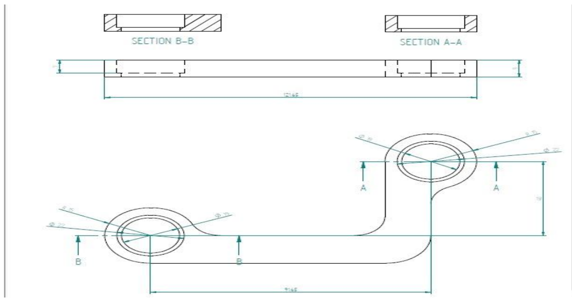


Fig. 5. Connecting Link

Smaller eye diameter = 8mm

Bigger eye diameter = 15mm

Thickness, t_1= 1.25 × d_1

= 1.25 × 8

= 10mm

t_2 =1.25 × 15

= 18mm

3.6 Output Yoke

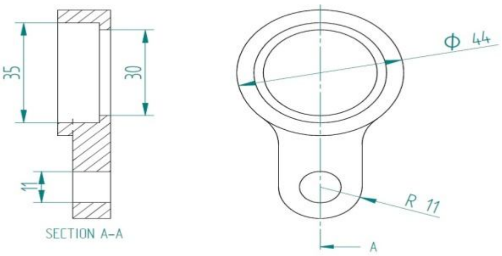


Fig.6. Output Yoke

The output yoke is a link that is subjected to direct tensile load in the form of pull = 66.66 N

MATERIAL DESIGNATION	TENSILE STRENGTH (N/mm^2)	YEILD STRENGTH (N/mm^2)
C40	600	380

From standard bearing 6001 whose small diameter is 12mm

Area $= \frac{\pi}{4} \times d^2$

$= \frac{\pi}{4} \times 12^2$

$\mathbf{A = 113.097\ mm^2}$

Check for failure of connecting rod under direct tensile load at the eye, this is the portion where the lever pin fits, the cross sectional area at this point is $113.097 mm^2$

$t = 0.75 \times d$

$= 0.75 \times 12$

t = 9 mm

3.7 Output Shaft

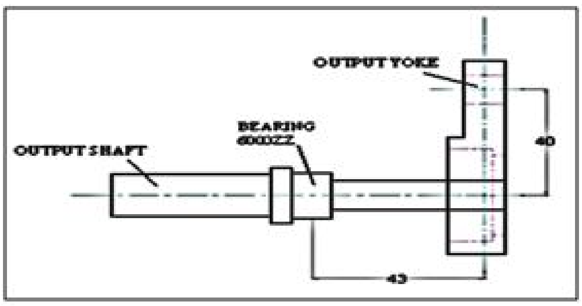


Fig.7. Output Shaft

Diameter according to the maximum shear stress theory

$$d^3 = \left(\frac{32N}{\pi . F_y} \sqrt{T^2}\right)$$

Selection of material :- C40

MATERIAL DESIGNATION		TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN24	800	600	

Yield strength :- 380 N/mm^2
Tensile strength :- 580 N/mm^2
Operating specific speed ;-4000 rpm

$$d^3 = \frac{32 \times 4000}{\pi \times 380} \left(\sqrt{(10^3 \times 1.2)^2}\right)$$

$$\mathbf{d = 15\ mm}$$

According to torsional equation,

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L}$$

$$\frac{T}{J} = \frac{G\theta}{L}$$

$$J = \frac{\pi}{32} \times d^4$$

$$= \frac{\pi}{32} \times 14.68^4$$

$$\mathbf{J = 4559.36\ mm^4}$$

$$\mathbf{G = 63.16\ MPa}$$

Here we have not cosidered the angle of twist(θ) because the output shaft is not fixed at any end of linkage.

$$\frac{1.2 \times 10^3}{4559.36} = \frac{63.16}{L}$$

$$\mathbf{L = 240\ mm}$$

3.8 Design of cam

Base diameter of cam selected 50mm.
Selection of cam diameter is not important.Design of eccentricity is important.
We select any diameter for cam but with selection of cam eccentricity taken is important.

3.9 Connecting pin1

This pin connects connecting rod to connecting link.
Now,

$$T = F \times Eccentricity$$

$$1.2 \times 10^3 = F \times 7.5$$

$$F = 160\ N$$

This force is transmitted by connecting rod to connecting link
Allowable shear stress,

$$F_{sy} = 0.5 \times F_{syt}$$

$$= 0.5 \times 600$$

$$= 300\ N/mm^2$$

By permissible shear theory,

$$F_{s\,all} = \frac{F_{sy}}{F_{os}}$$
$$= \frac{300}{2}$$
$$F_{s\,all} = 150 \text{ N/mm}^2$$

Checking pin-1 under direct shear stress,

$$F_{s\,act} = \frac{Force}{Area}$$
$$= \frac{160}{\frac{\pi}{4} \times 8^2}$$
$$F_{s\,act} = 3.18 \text{ N/mm}^2$$
$$F_{s\,all} > F_{s\,act}$$

Hence design of pin-1 is safe under shearing

Length of key

L = Connecting small eye thickness + Connecting rod thickness

L = 10 +10

L = 20mm

3.10 Connecting pin-2

MATERIAL DESIGNATION	TENSILE STRENGHT(N/mm²)	YEILD STRENGTH (N/mm²)
EN24	800	600

This link connects connecting link to output yoke

$$T = F \times \text{eccentricity}$$
$$1.2 \times 10^3 = F \times 18$$
$$F = 66.66 \text{ N}$$

This force is transmitted through output yoke to connecting link

Checking pin-2 under direct shear stress

$$F_{s\,act} = \frac{Force}{Area}$$
$$= \frac{66.66}{\frac{\pi}{4} \times 15^2}$$
$$= 0.37 \text{ N/mm}^2$$
$$F_{s\,all} > F_{s\,act}$$

Hence pin-2 is safe under direct shear stress

Length = Big eye thickness + output yoke thickness

= 15 + 9

= 24mm

3.11. Control Link

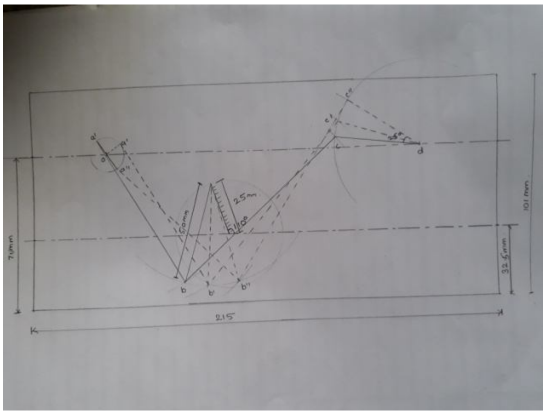


Fig. 8A. Synthesis

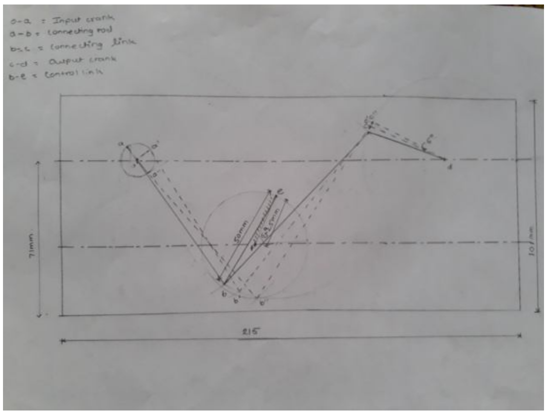
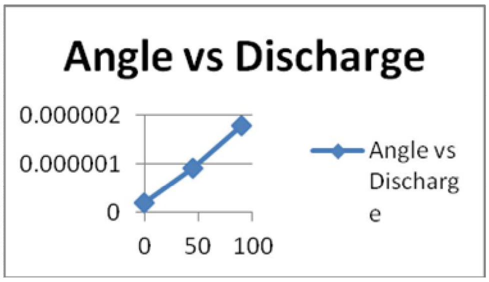


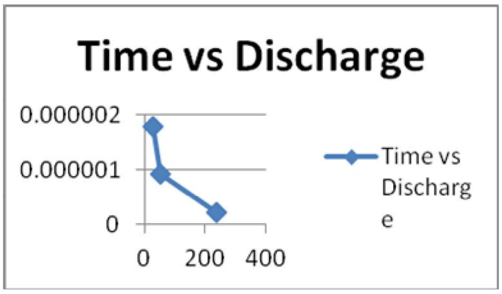
Fig. 8B. Synthesis

IV. RESULT

Volume = 50ml

sr. no.	Angle	Speed(RPM)	Time(sec)	Discharge(m^3/sec)
1	0	4000	2.37	0.00000021
2	45	4000	54.76	0.000000913
3	90	4000	27.8	0.000001798





V. CONCLUSION

A mechanism of four links in which one link acts as control link is developed to obtain the effect of fixed and variable displacement.It is seen that the discharge from the pump reduces at the control angle is changed from zero (0) degree to 120 degree. It is observed that flow rate is maximum when control link is set to 0 degree and flow rate is minimum when control link is set to 120 degree.From the graph of flow rate vs. speed the results obtained state that flow rate increases with increase in speed.From the graph of efficiency vs. speed we conclude that efficiency drops slightly as speed decreases. This is due to stiffness of spring used in pump and friction between piston and cylinder.As control angle increases increases flow rate decreases at constant speed. Cost from data obtained from various sources we conclude that our pump gives both functions in comparatively in minimum cost as compared to pumps available in market.

VI. ACKNOWLEDGEMENT

First of all I would like to give my sincere thanks to my guide Prof. Gauspira Makandar, who accepted us as his student & being mentor for us. He offered me so much advice, patiently supervising & always guiding in right direction. I have learnt a lot from him & he is truly a dedicated mentor. His encouragement & help made me confident to fulfill my desire & overcome every difficulty I encountered. I would also like to express my gratitude to Dr.M.Kate, H.O.D.Mechanical Engineering Department.I am also hugely obliged to Dr. Rajendra Kanphade, Principal, for giving me the opportunity to continue my education & enhance my knowledge. I would like to express my appreciation to all those colleagues & all other professors who have offered me their time & valuable advice for encouragement and suggestions during the partial fulfillment of the seminar.

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