Kinematic Linkage Based Veriable Displacement Pump

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Abstract - In Variable displacement pump the displacement or amount of fluid pumped per revolution of pump"s input shaft can be varied as per requirement. This paper investigate the different improvements achieved for increase the efficiency of variable displacement pump .This paper also discussed about the alternative mechanisms for variable displacement pump to improve efficiency and cost reduction of 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 pumps. Pumps operate by some mechanism (typically reciprocating or rotary), and consume energy to perform mechanical work by moving the fluid. Pumps operate via many energy sources, including manual operation, electricity, engines, or wind power, come in many sizes, from microscopic for use in medical applications to large industrial pumps. Mechanical pumps serve in a wide range of applications such as pumping water from wells, aquarium filtering, pond filtering and aeration, in the car industry for water-cooling and fuel injection, in the energy industry for pumping oil and natural gas or for operating cooling towers. In the medical industry, pumps are used for biochemical processes in developing and manufacturing medicine, and as artificial replacements A positive displacement pump makes a fluid move by trapping a fixed amount and forcing (displacing) that trapped volume into the discharge pipe. Some positive displacement pumps use an expanding cavity on the suction side and a decreasing cavity on the discharge side. Liquid flows into the pump as the cavity on the suction side expands and the liquid flows out of the discharge as the cavity collapses. The volume is constant through each cycle of operation.

A. Objective

In variable displacement linkage pump we plan to use the conventional fixed discharge pump but discharge can be made continuously variable from zero to maximum by use of innovative kinematic linkage that uses a single control lever to vary displacement from to maximum.

1. Design and kinematic synthesis of a variable displacement linkage to give zero to maximum displacement , and point to point control of the displacement using manual linkage
2. Design and selection of an twin cylinder axial piston pump to which the variable displacement linkage will be applied to.
3. Testing of the twin cylinder axial piston pump to plot the following characteristics of pump: Discharge V/s Speed ,Pressure V/s Speed ,Volumetric efficiency V/s Speed
4. Comparative analysis of the results of discharge and cost in comparison to bent axis piston pump of analogous configuration.

B. Problem Statement

Axial piston pumps with constant pressure and variable flow have extraordinary possibilities for controlling the flow by change of pressure. But cost of the bent axis piston pump is extremely high over the radial piston pump. Hence, there is a need to develop a modification in the radial piston pump design that will offer a variable discharge configuration.

C. Scope

1.Power Press lubrication

2.Special purpose machine lubrication

3.Roll forming machine lubrication

4.Spring making machine lubrication

II. LITERATURE REVIEW

The purpose of the literature review is to go through design, analysis and experimental testing of Kinematic Linkage Based Veriable Displacement Pump. Variable displacement linkage that will enable to vary the stroke of a single cylinder axial piston pump, thereby offering to vary the discharge of the pump using manual control. The solution offered is in form of the linkage motion adjuster pump where in mechanism to convert rotary motion of crank element into oscillatory output of the output element.The angle of oscillation of the output is a function of the position of pivot element. The pivot element position can be varied as it is placed on a slide. Thus adjustment of the stroke can be done by varying the position of the pivot element.

III. METHODOLOGY

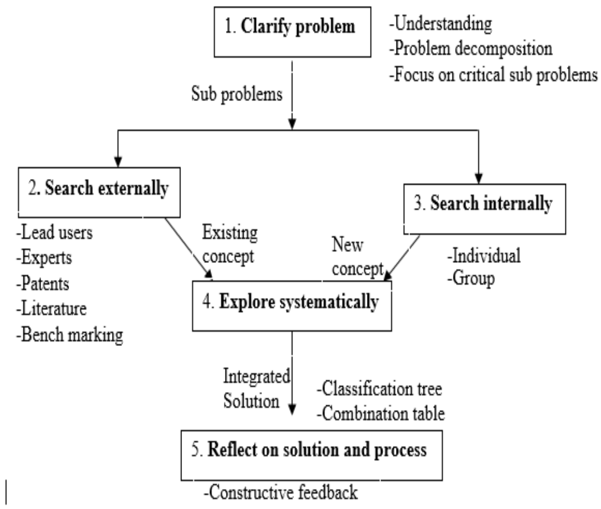

Fig. 2 Flowchart of Methodology

IV. DESIGN CALCULATION

1. Electric Motor

Power= 50 Watt
Speed = 0-9000 Rpm
Operating Speed = 4000 Rpm.
Now;

$$P = \frac{2\pi NT}{60}$$

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

$$T = \frac{60\times50}{2\pi\times4000}$$

T = 0.12 N.M
Belt Drive Between Motor And Pulley Have A Reduction Ratio Of 1:5
Hence Tdesign = Overload Factor × 5 ×Tmotor
Considering 100% Overlod
Tdesign = 2 × 5 ×.12
⇒ Tdesign = 1.2 N.M

2. .Design Of Input Crank Shaft:-

According to the maximum shear stress theory

F_{sy} =0.5

F_{yt} =300 N/mm^2

The permissible shear stress is given by

$$F_s = \frac{F_{yt}}{F.O.S}$$

$$= \frac{300}{2}$$

=150 N/mm^2

Section of the crank pin at xx is subjected to combined bending and torsional

$$\text{Crank force} = \frac{\mathrm{T_{design}}}{\text{eccentrycity}}$$

$$= \frac{1.2\times10^3}{25} \qquad = 48\ \text{N}$$

Moments,

M_t = 48 × 25 =1200 N-mm

M_b = 48 × 45 =2160 N-mm

$$f_b = \frac{M_b y}{I}$$

$$= \frac{1080 \times 32}{\pi d^3}$$

$$f_s = \frac{M_t r}{j}$$

$$= \frac{1200 \times 16}{\pi d^3}$$

$$F_{max} = \sqrt{(\frac{f_b}{2})^2 + {f_s}^2}$$

d = 3.5 mm

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.

3. Design Of Output Shaft :-

According To The Maximum Shear Stress Theory

F_{sy} =0.5

F_{yt} =300 N/Mm^2

The Permissible Shear Stress Is Given By

$F_s = \frac{F_{yt}}{F.O.S} = \frac{300}{2}$

$=150\ N/Mm^2$

Section Of The Crank Pin At Xx Is Subjected To Combined Bending And Torsional

$$\text{Crank Force} = \frac{T_{Design}}{\text{Eccentrycity}}$$

$= \frac{1.2\times10^3}{40}$ = 30 N

Moments

$M_t = 30\times40$ =1200N-Mm

$M_b = 30 \times 43$ =1290 N-Mm

$$f_b = \frac{M_b y}{I}$$

$$= \frac{1290 \times 32}{\pi d^3}$$

$$f_s = \frac{M_t r}{j}$$

$$= \frac{1200 \times 16}{\pi d^3}$$

$F_{max} = \sqrt{(\frac{f_b}{2})^2 + {f_s}^2}$ D = 4 Mm

But As Per Manufacturing Considerations We Have An H6h7 Fit Between The Dyno-Brake 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.

4. Design Of Connecting Pin-1

Connecting pin connects the connecting rod to connecting link.

We know that

T= force × radius

The eccentricity of the crank or eccentric =25 mm as per the mechanism design.

1200 = Force × 25

$$Force = \frac{1200}{25}$$

Force = 48 N

This force is transmitted by the connecting rod to the connecting link

Check for direct shear of connecting pin -1

$$Shear\ stress = \frac{shear\ force}{\text{Shear area}}$$

The lever pin-1 supports the connecting rod small end and is supported in the lever at other end hence will be subjected to a single shear failure

$$= \frac{48}{\frac{\pi}{4} \times d^2}$$

The connecting rod is an standard part with the small end pin of diameter 15 mm and minimum section on pin is 8mm for bearing mounting hence,

d=8mm

$$= \frac{48 \times 4}{\pi \times d^2}$$

Shear stress = 0.95 N/mm ^{2}

As $Fs_{act} < Fs_{all}$

Hence pin safe.

5. Design Of Connecting Pin-2

Connecting pin-2 connects the connecting link to output yoke.

We know that T= force × radius

The eccentricity of the crank or eccentric =25 mm as per the mechanism design.

1200 = force x 40

$$Force = \frac{1200}{40}$$

Force =30 N

1. Check for direct shear of connecting pin -2

$$Shear\ stress = \frac{shear\ force}{\text{Shear area}}$$

The lever pin-2 supports the output yoke end and is supported in the lever at other end hence will be subjected to a single shear failure

$$= \frac{30}{\frac{\pi}{4} \times d^2}$$

Connecting pin minimum section is 8mm for bearing mounting.

$$= \frac{48 \times 4}{\pi \times d^2}$$

shear stress = 0.14 N/mm ^{2}

As $Fs_{act} < Fs_{all}$

Hence pin safe.

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