Safety Hazards
Fluid Machinery Laboratory Room B-10

HAZARD: Rotating Equipment

Be aware of pinch points and possible entanglement

Personal Protective Equipment: Safety Goggles; Standing Shields, Sturdy Shoes
No: Loose clothing; Neck Ties/Scarves; Jewelry (remove); Long Hair (tie back)

HAZARD: Projectiles / Ejected Parts

Articles in motion may dislodge and become airborne

Personal Protective Equipment: Safety Goggles; Standing Shields

HAZARD: Electrical - Burn / Shock

Care with electrical connections, particularly with grounding, and not Using frayed electrical cords, can reduce hazard. Use GFCI receptacles near water.

HAZARD: High Pressure Air-Fluid / Gas Cylinders / Vacuum

Inspect system integrity before operating any pressure / vacuum equipment. Gas cylinders must be secured at all times.

Personal Protective Equipment: Safety Goggles

HAZARD: Water / Slip Hazard

Clean any spills immediately.

HAZARD: Laser / Eye - Cornea Damage

Do not look directly into laser

Personal Protective Equipment: Laser Specific Goggles

Experiment No. 2

GEAR PUMP PERFORMANCE

I. OBJECTIVE

The objective of this experiment is to determine the operating characteristics and efficiency of a gear pump at various speeds.

II. EQUIPMENT

A schematic drawing of the flow diagram used for this experiment is shown in figure 2-1.

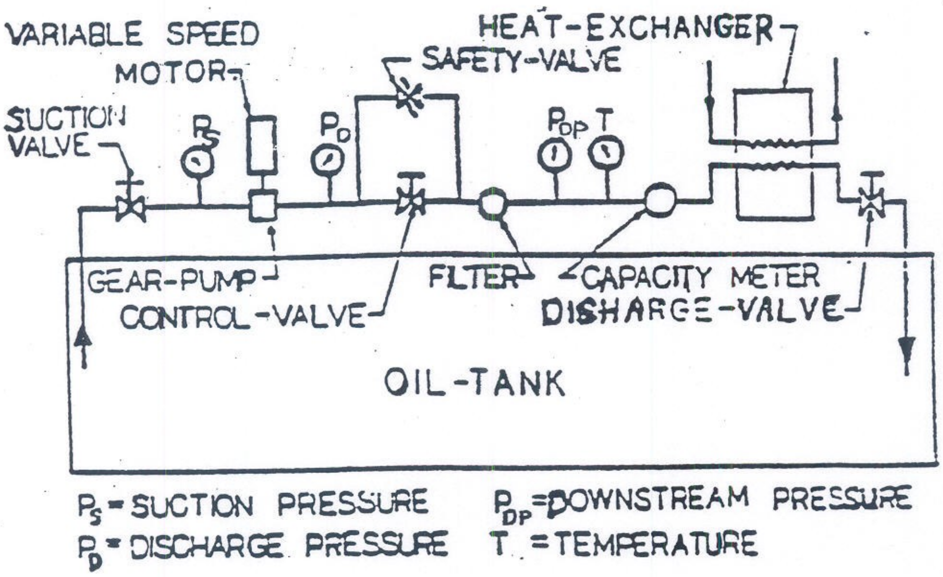

Figure 2-1. Schematic of Gear-Pump test-loop System

The oil is stored in a tank situated below the gear pump. The oil passes through the gear pump, control valve, filter, thermometer, heat exchanger and then return back to the tank. A motor with a variable speed transmission having maximum speed of 1,000 RPM drives the gear pump.

Gear pumps are widely used in lubrication systems such as of automotive engines, and for hydraulic power. A gear pump as well as piston pump is a positive displacement pump, i.e. the pump delivers, under ideal conditions, a fixed quantity of liquid per cycle, irrespective of the flow resistance (head losses in the system). However, it is possible to convert the discharge of constant flow-rate to discharge at constant pressure by installing a pressure relief valve that maintains constant pressure, and returns the surplus flow.

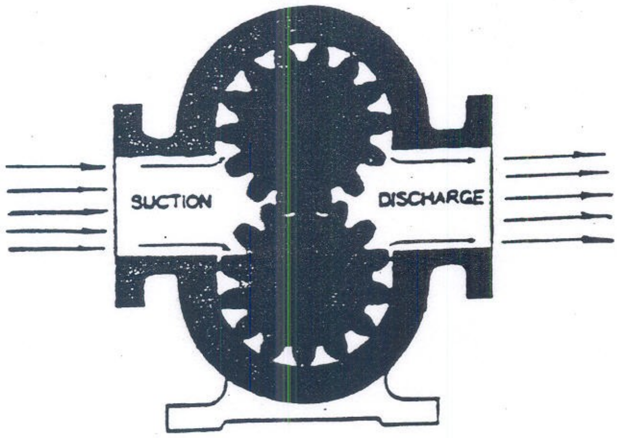

Fig. 2-2: Cross-section of a Gear Pump

A cross-section of a simple gear pump is shown in Fig. 2-2. A gear-pump consists of two spur gears (or helix gears) meshed inside a pump casing with one of the gears driven by constant speed, electrical motor. The liquid from the suction side is trapped between the gear teeth, forcing the liquid around the casing, and finally expelling it out through the discharge. The quantity of liquid discharged per revolution of the gear is known as displacement. Theoretically, it is equal to the sum of the volumes of all the spaces between the gear teeth and the casing. However, there are always tolerances and small clearance for a free fit between the gears and casing. The presence of clearance in pump construction makes it practically impossible to attain theoretical displacement.
The advantages of a gear pump, in comparison to other pumps, are as follows:

1. It is a simple and compact pump, which does not need inlet and outlet valve, such as in piston pump. However, gear pumps require close running clearances.
2. Continuous flow (unlike the positive displacement reciprocating pumps).
3. The gear pump can handle very high viscous fluid.
4. It can generate very high heads (or outlet pressure) in comparison to centrifugal pump.
5. It is self-priming (unlike centrifugal pump). It acts like a compressor and pumps out trapped air or vapors.
6. It has good efficiency at very high heads.
7. It has good efficiency over a wide speed range.
8. Gear pumps require relatively low suction heads.

The flow-rate of a gear pump is approximately constant irrespective of its head losses. If we close accidentally the discharge valve, the discharge pressure would rise, until the weakest part of the system would fail. To avoid this, a relief valve should be installed in parallel to the discharge valve.

A small amount of liquid escapes backward from the discharge side to suction side through the gear-pump clearances is referred to as "*slip*". The capacity (flow-rate) lost due to the slip, in the clearances is dramatically increasing with the clearance, h_o , between the housing and gears (proportional to h_o^3) and inversely proportional to the fluid viscosity. An idea about the amount of liquid lost in slippage can be obtained by the equation for a laminar flow between two parallel plates having a thin clearance, h_o, between them,

$$Q = \frac{l\, h_o^3}{12\mu\, b}\, \Delta p \qquad (2.1)$$

Here,

- Q – flow-rate of flow in the clearance (slip flow-rate),
- Δp - differential pressure (between discharge and suction),
- b - width of fluid path (normal to fluid path,
- h_o - clearance between two plates,
- μ - fluid viscosity,
- l - length along the fluid path.

This equation is helpful in the understanding the parameters affecting the magnitude of slip. It shows that the slip is mostly dependent on clearance since it is proportional to the cube of clearance. Also, slip is proportional to pressure differential Δp, and inversely proportional to the viscosity μ of the liquid. Gear-pumps are suitable for fluids of higher viscosity for minimizing slip, and are widely used for lubrication, since lubricants have relatively high viscosity (in comparison to water). Fluids with low viscosity, such as water or air, are not suitable for gear-pumps.

Gear-pump Characteristics

The constant theoretical displacement is a straight horizontal line. As the "slip" is proportional to the head of the pump (discharge head minus suction head), the actual capacity reduces with the head. When the head approaches zero, the capacity is equivalent to the theoretical displacement.

1. Capacity (flow-rate) versus head:

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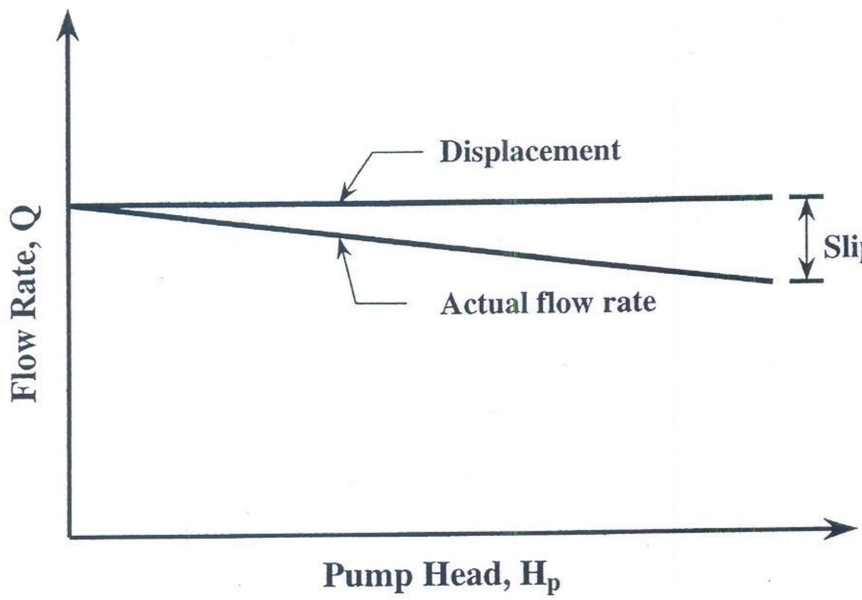

Figure 2-3: Gear pump Q-H characteristics.

2. Hydraulic power and pump efficiency:

The unit of Power $\dot{E}$ is *[Watt]* in SI units. Also, a widely used unit is the imperial unit; horse power [*HP*]. The brake-power, $\dot{E}_b$, (*BHP* in horsepower units) is the mechanical shaft power required to drive the pump by the electrical motor. In the pump this power is converted into two components: the useful hydraulic power, $\dot{E}_h$, and the frictional losses, $\dot{E}_f$. In the pump the friction losses are dissipated as heat. The friction losses result from friction in the bearings, stuffing box (or mechanical seal), and viscous shear of the fluid in the clearances.

In Fig. 2-3, the curves of the various power components, $\dot{E}$, versus pump head, H_p, are presented in horsepower [HP] units. This frictional horsepower, [FHP], does not vary appreciably with increased head, and it is horizontal line in Fig. 2-4. The other useful component is the Hydraulic Horsepower (HHP*)*. This power component is directly proportional to the pump head, and shown as a straight line with a positive slope. This component is added to constant FHP, resulting in the total brake horsepower [BHP].

The *BHP* curve in Fig. 2-4 is a straight line, and at zero head losses there is still a definite brake horsepower required due to friction in the pump. In a gear pump, the friction horsepower *FHP* is a function of the speed and the viscosity of the fluid, but not of the head of the pump. As the *FHP* is nearly constant versus the head, it is shown as a straight line in Fig. 2-4. On the other hand, *HHP* is nearly a linear function of H_p, see equation for hydraulic power. (This is an approximation, since Q is not constant because it is reduced by the slop of the slip). The sum of the friction and hydraulic power is the brake horsepower. The brake horsepower increases nearly linearly versus H_p, as shown in Fig. 2-

4. Since *FHP* is constant, the efficiency, η, is an increasing function versus H_p . It results that gear pumps have a better efficiency at high heads.

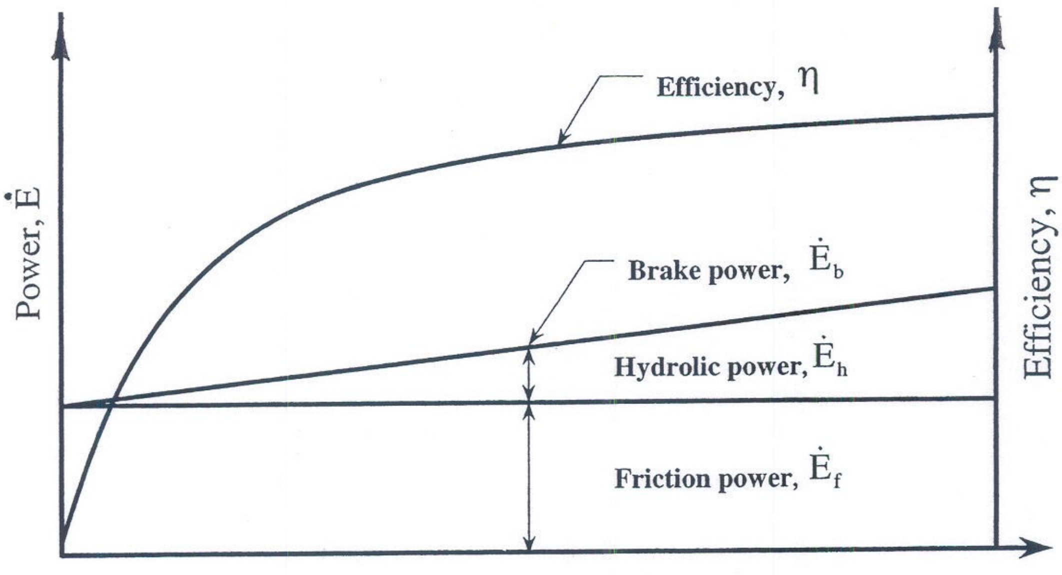

Fig. 2-4: Power and efficiency characteristics of gear-pump

The head of the pump, H_p, generated by the pump, is equal to the head losses in a closed-loop piping system such as in hydrostatic bearing system. If the fluid is transferred from one tank to another, at higher elevation, the head of the pump, is equal to the head losses in the piping system plus the height difference Δz.

The head of the pump *Hp*, is the difference of the heads between the two points of discharge and suction,

$$H_p = H_d - H_s \tag{2-2}$$

Pump head units are of length (*m, ft*). Head is calculated from Bernoulli's equation. The head of the pump is the difference in head at pump discharge and suction. The expression for discharge and suction head is:

$$H_d = \frac{p_d}{\gamma} + \frac{V_d^2}{2g} + Z_d \tag{2-3}$$

$$H_s = \frac{p_s}{\gamma} + \frac{V_s^2}{2g} + Z_s \tag{2-4}$$

where

H_d = head at the discharge side of pump (outlet),

H_s = head at the suction side of pump (inlet),
p_d = pressure measured at the discharge side of pump (outlet),
p_s = pressure measured at the suction side of pump (inlet),
γ = specific weight of fluid, *(for water $\gamma = 9.8 \times 10^3$ [N/m^3],*
V = fluid velocity,
g = gravitational acceleration,
Z = height

The pump head, H_d, (equal to the head loss in the system) is,

$$H_p = \frac{p_d - p_s}{\gamma} + \frac{V_d^2 - V_s^2}{2g} + (Z_d - Z_s) \tag{2-5}$$

The velocity of the fluid in the discharge and suction can be determined from the rate of flow and the inside diameter of the pipes. In most cases, the discharge velocity is equal to that of the suction, and there is no significant difference in height between the discharge and suction. In such cases, the last two terms can be omitted, and the pump head is determined only by the pressure difference:

$$H_p = \frac{p_d - p_s}{\gamma} \tag{2-6}$$

The hydraulic power of a pump is proportional to the pump head, H_d, according to the equations:

$$\dot{E}_h = Q \gamma H_p = Q(P_d - P_s) \tag{2-7}$$

where $\Delta p = (P_d - P_s)$.

In SI units, the units for Eq. (2-7) are:

Q [m^3/s]
γ [N/m^3]
H [m]
Δp [N/m^2 or Pascal]

In horsepower units, the HHP (hydraulic horsepower) is obtained from the equation:

$$HHP = \frac{1.341\, Q\, \Delta p}{1000} \tag{2-8}$$

In most cases, the inlet and outlet pipes are of the same diameters and the inlet and outlet velocities are equal.

In imperial units, the hydraulic horsepower (HHP) is given by,

$$HHP = \frac{Q\,\Delta p}{1714} \tag{2-9}$$

where the units are as follows:

$\Delta P\ [psi] = \gamma\,(H_p - H_s)$

and

$Q\ [GPM]$

The efficiency of the pump, η, is the ratio of the hydraulic and brake power,

$$\eta = \frac{HHP}{BHP} \tag{2-10}$$

The *BHP* can be measured by a motor dynamometer. If we are interested in the efficiency of the complete system of motor and pump, the input power is measured by the electrical power in watts, consumed by the electrical motor that drives the pump. The horsepower lost on friction in the pump, *FHP*, can not be measured, but can be determined by,

$$FHP = BHP - HHP \tag{2-11}$$

Net Positive Suction Head (NPSH)

Pumps operating on the suction side with low pressure that is close to the vapor pressure of the fluid (at the operating temperature) may result in cavitation. Cavitation is undesirable because it would slowly destroy the impeller or gear by metal erosion.

The Net Positive Suction Head (NPSH) is the sum of all forms of absolute hydraulic head, in relation to the vapor pressure head. This expression is important to predict if cavitation can take place inside the pump. In a gear pump, the cavitation reduces the flow-rate, particularly when the NPSH reduces below a certain critical value. The equation for the Net Positive Suction Head is:

$$\mathrm{NPSH} = \frac{P_s}{\gamma} + \frac{P_a}{\gamma} - \frac{P_v}{\gamma} + \frac{V_s^2}{2g} \tag{2-12}$$

Here,

P_s = gage pressure at the suction point (above the atmospheric pressure)
P_a = absolute atmospheric pressure
P_v = vapor pressure of liquid
V_s = flow velocity in the suction pipe

γ = specific weight of liquid

The pressure, P_s, is measured close to the suction side of the pump.

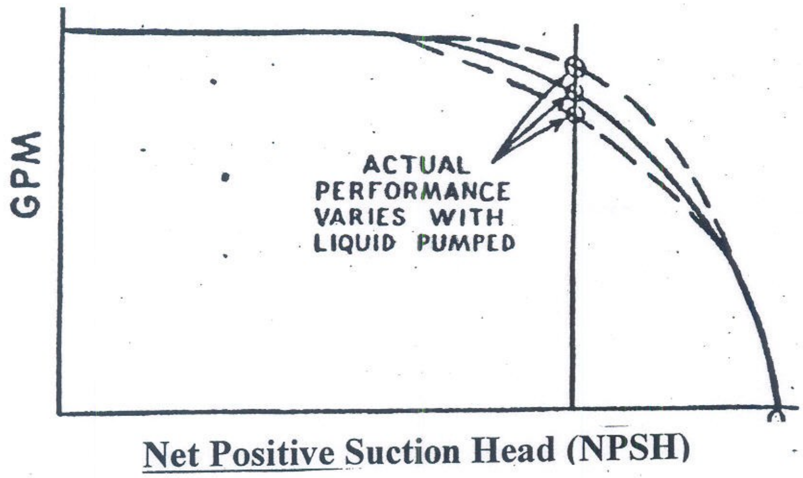

Fig. 2-5: Capacity versus NPSH in pumps

Fig. 2-5 shows the curve of the capacity Q versus the NPSH. The NPSH in this curve is reducing along the coordinate. If the value of the NPSH is below a certain critical value, cavitation initiates. Cavitation is highly undesirable because it can destroy the pump (particularly in the case of centrifugal pumps). Also, it results in a deterioration in the pump performance, in the form of a significant reduction in the flow-rate Q. For this reason it is important that the level of the NPSH would not be below a certain critical value that is determined from Fig. 2-5. One of the advantages of a gear-pump is that it can operate at relatively low levels of NPSH (in comparison to centrifugal pump).

VI. TEST PROCEDURE:

The experiment will be conducted in two parts:

i) Determining characteristic curves, and
ii) Capacity versus NPSH

i) Determining Characteristic Curves for 3 motor speeds,

1. Open the suction and discharge valves.
2. Start the motor and set it at a speed requested by the instructors.
3. Using the discharge valve vary the discharge pressure, by increments of 5 *psi*, from the minimum to maximum (90 *psi*).
4. For each step (i.e. each pressure) measure: (1) discharge flow-rate, (2) suction pressure [inches of water], (3) discharge pressure [*psi*], (4) motor power [Watts], and (5) fluid temperature.

5. Repeat the procedure by changing motor speed and noting results for at least 3 different motor speeds.
6. At the end of the experiment switch off the motor. After the motor has stopped completely close the suction and discharge valves.

ii) **Capacity versus NPSH**

1. Open the suction and discharge valves completely and leave the control valve fully open.
2. Start the motor and set it at a speed requested by the instructor.
3. Adjust the discharge valve to obtain a pressure of 40 *psi*.
4. Measure the suction pressure, discharge pressure, capacity and fluid temperature.
5. Reduce the suction pressure, by closing slowly the suction valve, (through increments of 2 inches of water on the suction pressure) and keeping the discharge pressure constant at 40 *psi*, until the flow rate is very low.
6. Repeat the procedure by changing motor speed and noting results for at least two different motor speeds.
7. At the end of the experiment switch off the motor. After the motor has stopped completely close the suction and discharge valves.
8. Measure the barometric (i.e. atmospheric) pressure.

VII GRAPHS REQUIRED IN THE REPORT

Use SI units for your results and graphs.

1. For each of the 3 speeds, plot capacity, Q, versus H_p and show the slip in each case.
2. For each of the 3 speeds, plot the hydraulic power, friction losses power and brake power versus pump-head, H_p . Plot the curves of all power components on the same coordinate system, similar to Fig. 3.
3. Plot the efficiency η versus H_p for each pump speed. To allow comparison, plot the 3 curves on the same coordinate system.
4. Plot the *volumetric efficiency* versus pump head, H_p, for each pump speed. To allow comparison, plot again the 3 curves on the same coordinate system.

$$Volumetric\ efficiency = \frac{Capacity}{Displacement} \qquad (2\text{-}13)$$

where the displacement is the theoretical capacity, without the effect of slip.

5. For each speed plot capacity, Q, versus NPSH. Use Eq. (2-12) to calculate the NPSH. Neglect the vapor pressure of oil.

6. **Preliminary report:** should include sample calculations and drawing of two graphs: capacity, Q, *vs.* - pump-head, H_p, and brake power *vs.* pump-head, H_p, curves, both for one speed (the highest speed) of the pump.