DEPARTMENT **OF MECHANICAL ENGINEERING**
AHMADU BELLO UNIVERSITY, ZARIA

DEVELOPMENT OF A POSITIVE-DISPLACEMENT PUMP
PHASE II

BY

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A thesis submitted to the Postgraduate School, Ahmadu Bello University, Zaria, Nigeria, in partial fulfillment of the requirements for the award of the degree of M.Sc (Mechanical) in Engineering.

June, 1994

DEDICATION

I dedicate this thesis to my late grand father and the members of my family as a whole. I also dedicate it to my future wife and children.

DECLARATION

I hereby declare that this thesis has been written by myself and that it is a record of my research work. It has not been submitted or accepted in any previous application for a Higher Degree. All sources of information have been specifically acknowledged by means of references.

GYPam
G.Y. PAM

Date: 12.04.94

CERTIFICATION

This thesis, entitled "DEVELOPMENT OF A POSITIVE - DISPLACEMENT PUMP" by Pam, Gyang Yakubu, meets the regulations governing the award of the degree of Master of Science (MECH. ENG.) of Ahmadu Bello University, Zaria-Nigeria, and it is approved for its contribution to knowledge.

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ABSTRACT

A positive-displacement vane pump (pdvp) was designed, constructed and preliminary tests carried out on it in the phase I of this project. Also, the influence of clearances within the pump on the hydraulic efficiency of a pdvp was theoretically investigated in phase I, based on laminar flow within the clearances.

The work in this thesis is an extension of the work in phase I. In particular, the influence of the pump clearances on the hydraulic efficiency was similarly, theoretically investigated, this time, based on turbulent flow within clearances. A more ttailed experimental test was then carried out on the pdvp constructed in phase I and its performance characteristics established.

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NOMENCLATURE

Ar	-	aspect ratio
b	-	width of a slip passage (m)
C_d	-	coefficient of viscous drag dependent upon pump geometry.
C_f	-	coefficient of friction dependent upon pump geometry.
C_s	-	geometrical constant of the pump.
d	-	thickness of a slip passage (m)
D	-	pump displacement (m^3/rev)
E	-	efficiency
K	-	Coefficient dependent on pump geometry (m^3)
l	-	length of slip passage (m)
L	-	rotor length (m)
N	-	rotational speed (rpm)
P_i	-	input power (W)
P_o	-	output power (W)
Q	-	delivery (m^3/s)
Q_i	-	ideal delivery (m^3/s)
Q_r	-	loss in delivery due to vaporisation of liquid, or liberation of entrained gas at intake (m^3/s).
rpm	-	revolutions per minute
R	-	rotor radius (m)
T	-	Torque input (N-m)
T_c	-	Torque due to mechanical friction independent of pressure and is constant (N-m).
T_f	-	Mechanical friction torque which is proportional to pressure differential (N-m).
T_i	-	ideal torque (N-m)
T_v	-	Viscous Torque (N-m)

α - pump displacement (m^3/rad)

γ - coefficient dependent upon pump geometry (m^3)

δ - rotor -stator clearance (m)

Δp - pressure difference (N/m^2)

ρ - density (kg/m^3)

ω - angular speed of shaft (rad/s)

ψ - ratio of rotor stator clearance to rotor radius

μ - viscosity (Ns/m^2)

ϕ - $\rho\omega^2R^2/\Delta p$

min - minute

pdsvp - positive displacement sliding vane pump

μm - micrometer

CHAPTER ONE

INTRODUCTION, HISTORY, CLASSIFICATION AND APPLICATION OF PUMPS

1.1 Introduction

Development of a positive-displacement vane pump (pdvp) began in the phase I, of this project [1]. In the phase I, analysis of all slip flows within the clearances of the pump was assumed to take a laminar pattern. On the basis of this the influence of the sizes of the slip passages, the aspect ratio of the rotor, as well as the rotor-stator clearance on the hydraulic efficiency of a pdvp was theoretically investigated and established. In particular, the previous investigation showed that as the sizes of the slip passages increase, the hydraulic efficiency of a pdvp decreases, whereas as the aspect ratio of the rotor and the stator-rotor clearance increases, the hydraulic efficiency of the pdvp increases. Based on laminar flow considerations, the pump was then designed and constructed. However, no extensive experimental tests were carried on the pump.

As a follow-up and extension of the work carried out in the phase I, a similar theoretical analysis, however, based on turbulent flow considerations has been carried out to determine the influence of some geometric parameters of a pdvp on its hydraulic efficiency. The relevant flow parameter was $\rho R^2\omega^2/\Delta p$ in this case. The variation of the hydraulic efficiency with the flow parameter and two of the dimensionless geometrical parameters, ψ and Ar, were investigated theoretically based on turbulent flow considerations.

The pump constructed in the phase I was then extensively tested experimentally, so as to determine its performance characteristics. An analysis of the experimental data collected for the constructed pump [1] was carried out.

1.2 HISTORY

Pumps are among the oldest of man's aids. In this technological age, pumps rank next to "the most used of all machines," the electric motor.

Man's struggle from ancient times to his present civilization has been accompanied by an ever increasing use of water. Origin of wells dug by man is too ancient for any historical record. The Romans used aqueducts to carry water over long distances for supply of towns, but relied on the natural flow. Many-a-time water had to be lifted from a lower to a higher level and transported not only through short distances, but even over distances of several kilometers. This necessitated the development of pumping devices.

The earliest devices for lifting water are still in operation in India, Egypt, Nigeria e.t.c. A typical example is the scoop lowered into a river by means of a balanced beam or shadoof, or in later developments by a pulley, rope and a beast of burden.

Another example of the earliest pumps is variously known as the Persian wheels, water wheels, or norias depending on the locality. These devices were all undershot water wheels containing buckets which were filled with water when they were submerged in a stream and which automatically emptied into a collecting trough as they were carried to their highest point by the rotating wheel. Similar water wheels have continued in existence in the Orient even into the twentieth century.

Other developments includes the Archimedian screw for low-head applications where the liquid is frequently laden with trash.

The pump is the earliest form of machine which substituted natural energy for muscular effort in the fulfillment of man's needs.

1.3 Classification of Pumps

Pumps may be classified on the basis of the applications they serve, the materials from which they are constructed, the liquids they handle, type of flow in the pump, e.g. radial, axial or mixed flow pumps. Such classifications tend to overlap each other. A more reliable system of classification defines the principle by which energy is added to the fluid, goes on to identify the means by which this is carried out, and finally draws specific geometries commonly employed.

1.3.1 Dynamic Type

In this type, energy is continuously added to increase the fluid velocities within the machine to values in excess of those occurring at the discharge such that subsequent velocity reduction within or beyond the pump produces a pressure increase. Fig.1, presents in outline form a summary of the significant classification and sub-classfication within this category.

1.3.2 Displacement Type

In this type, energy is added periodically by application of force to one or more movable boundaries of any desired number of enclosed fluid-containing volumes, resulting in a direct increase in pressure up to values required to move the fluid through valves or ports into the discharge. Fig.2, presents a summary of classification and sub-classification within this category.

Positive-displacement is the transportation of fluid from an enclosure caused by decreasing the volume of the enclosure. A rotary positive-displacement vane pump is an arrangement of members so connected and constructed that they define an isolated space or spaces, that their relative motion causes these spaces alternately to increase and decrease in volume, and that these spaces are connected to an inlet port while expanding and to an outlet port while contracting.

In this thesis, the rotary positive-displacement sliding vane pump is the relevant pump to all discussions.

In general, rotary pumps have the following advantages over reciprocating pumps, they:-

i. deliver a continuous flow practically free from pulsations,

ii. avoid reciprocating complications,

iii. are simpler in construction,

iv. Occupy less space,

v. are much smaller in dimensions for a given capacity,

vi. and cost less to install and maintain.

1.4 APPLICATIONS

Rotary pumps are applicable to almost any liquid as long as it is free from grit and abrasive particles. If not specially designed and made of suitable materials, there will be excessive wear when liquids containing abrasive particles are handled.

Rotary pumps are particularly adopted to handling liquids of high viscosities, such as heavy fuel oils,

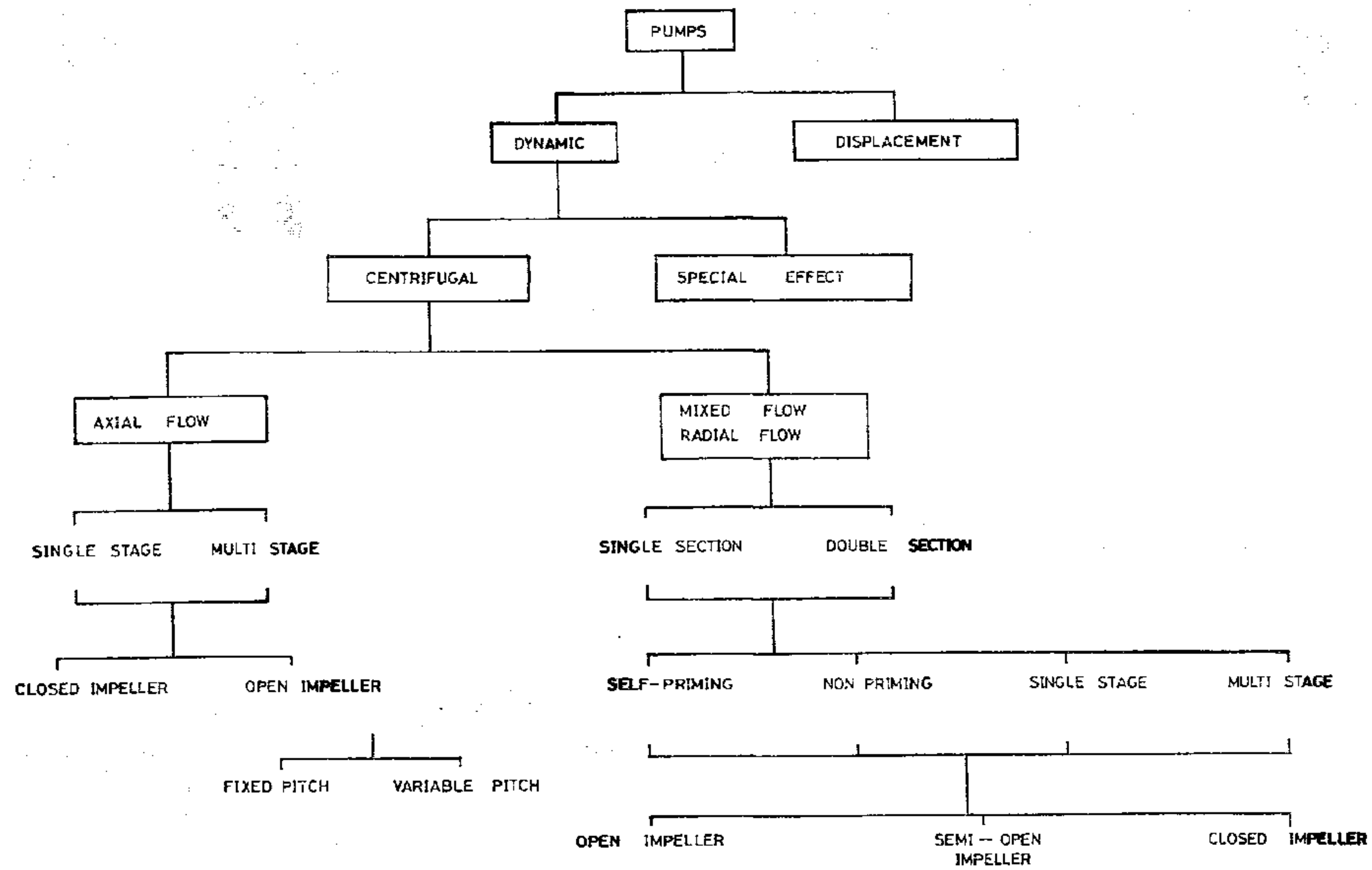

FIG. 1 : CLASSIFICATION OF DYNAMIC PUMPS .
SOURCE : Ref (2) Page 1 — 2 .

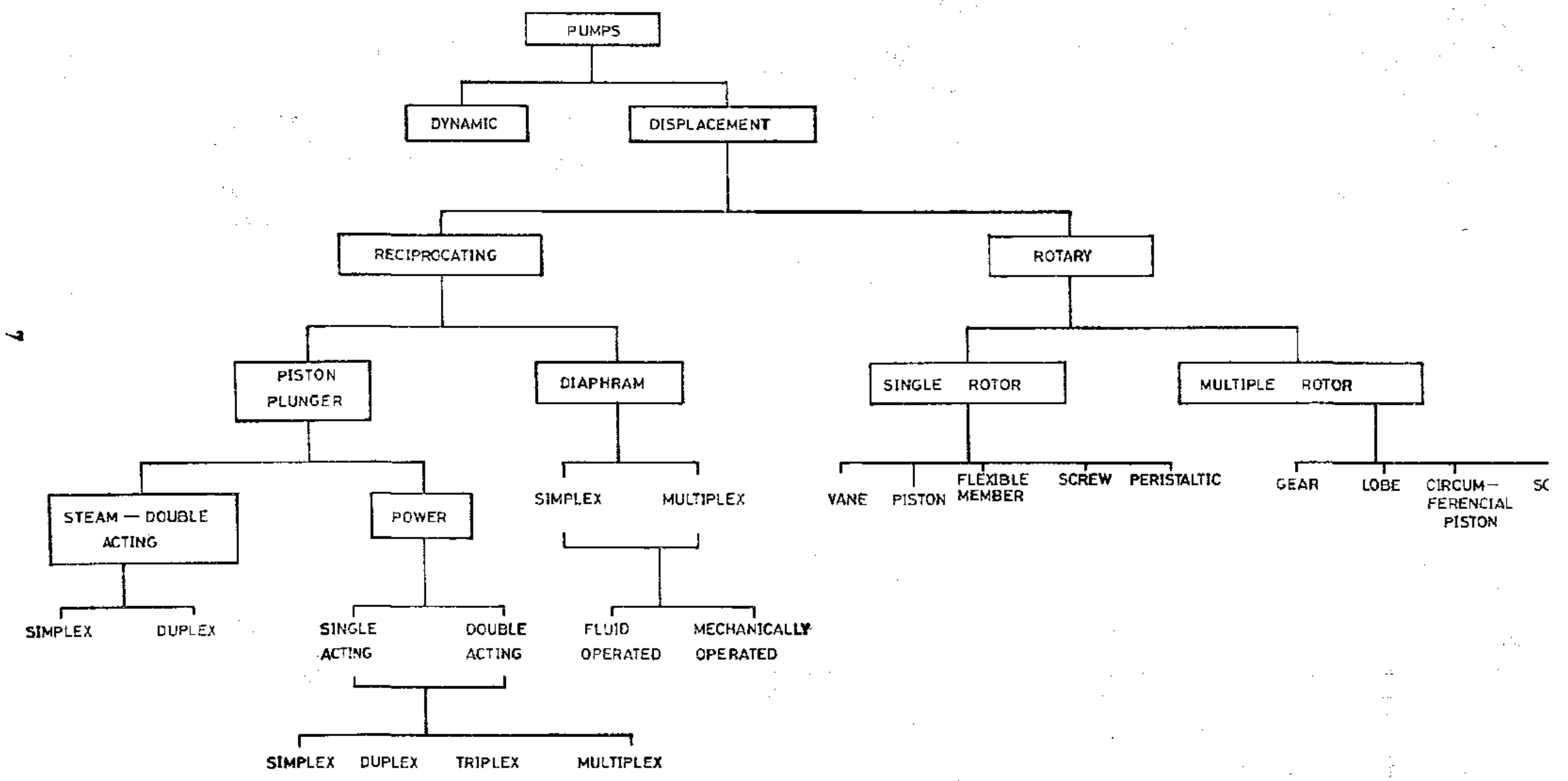

FIG. 2 : CLASSIFICATION OF DISPLACEMENT PUMPS .
SOURCE : Ref (2) Page 1—2 .

tars, pitch, asphalt and heavy lubricating oils.

They are not limited to handling of liquids of high viscosities, but have been used widely for pumping gas oil, gasoline, light fuel oils, benzene, etc.

Most industries use rotary pumps either directly or indirectly. They are used on hydraulic presses and dump trucks; in paint and chemical plants; for water pumping; in oil refineries, soap manufacturing, breweries, distillers; for handling volatile products such as gasoline.

CHAPTER TWO

THEORY OF A POSITIVE-DISPLACEMENT VANE PUMP (pdvp) EMPLOYING LAMINAR FLOW ANALYSIS

A positive-displacement sliding vane pump (pdsvp) is a rotary pump which operates, by transporting liquid confined between two consecutive vanes and the stator from inlet to discharge. At exit, the volume converges and hence the fluid is discharged at a higher pressure than at the inlet.

Figure 3, shows a schematic diagram of the essential features of a vane pump. The principal element is a rotor from which sliding vanes, actuated by mechanical or hydraulic means, project to make contact with the housing. A dividing seal separates inlet passage from the discharge passage. The confined liquid is transported bodily from intake to discharge. When a vane reaches the exit, the confined liquid is released and forced out of the discharge passage.

For an ideal unit, it is assumed that there is no mechanical or viscous friction, clearances are zero and hence no slip flow and the fluid handled is incompressible. This will lead to a hydraulic efficiency, of the unit, of unity [1]. In an actual unit, considerable deviation from ideal performance will be found due to slip cavitation and inlet restriction, frictional forces and torques.

2.1 Slip

It has been noted that there is a flow of liquid from the high pressure side, to the low pressure side of any positive-displacement pump, through the necessary clearances between solid surfaces [3]. Flow through the clearances and seals will cause an actual discharge differing from the actual discharge. These secondary flows are referred to as "slip". There are four slip paths in a vane pump depicted in fig. 4. These are between

i. rotor ends and end plates,

ii. rotor and dividing seal,

iii. Vane-tips and stator,

iv. Vane sides and endplates

2.2 Cavitation and Inlet Restriction

A loss in delivery of a pump, not properly designated as slip, is described. This loss is encountered when pumping liquids containing entrained gases, or liquids which vaporize readily.

The restriction to flow offered by the inlet passages of a pump leads to low pressures in this region. If a relatively volatile liquid is being pumped, low pressures will permit vaporization of the liquid. This will lead to the formation of vapour pockets which will displace liquid. When these pockets get into the high pressure zones, 416947

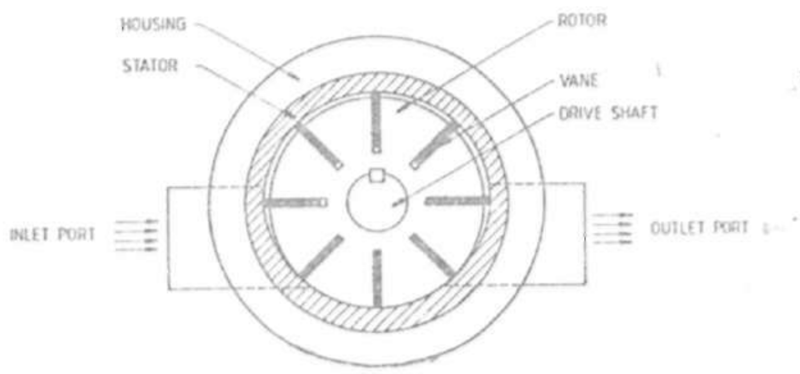

Fig. 3. Schematic representation of a vane pump.

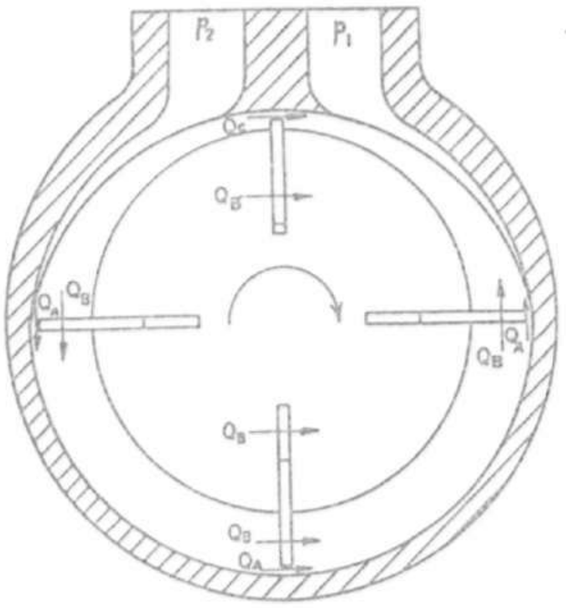

Fig. 4. Slip flow in a vane pump.

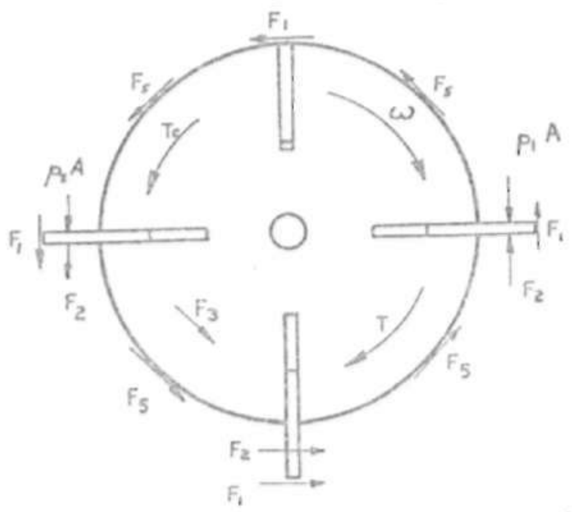

Fig. 5. Torques and forces acting on vane-pump rotor.

they will collapse. Considerable noise accompanies this process. The vapour occupies volume which otherwise, would have been filled by liquid and the net delivery of liquid is reduced by an amount equal to the difference between the volume of the vapour and the volume of the mass of matter in the liquid state. Frequently, a most destructive corrosive type of action on the metal parts of the pump interior is in evidence. This phenomenon is known as cavitation.

A similar reduction in delivery occurs if there is air entrained in the liquid since it will expand with reduction in pressure at intake, occupying space intended for liquid, exactly as in the case of the vaporization of the liquid.

2.3 FRICTIONAL FORCES AND TORQUES

Viscous bearing and sliding contact friction as well as other mechanical effects will tend to resist motion of the rotor. These affect the shaft torque by an amount which will be referred to as "torque loss". This influences the performance of the pump.

Three distinct types of forces are identified:

i. Friction due to viscous shearing of the liquid in the narrow passages between rotor and housing.
ii. Friction due to mechanical friction which is directly proportional to the pressure differential. It may originate in seals, if the sealing forces are proportional to pressure or in

bearings where the resistance is proportional to pressure.

iii. Constant friction that is both independent of pressure differential and speed.

Viscous forces arise at a large number of locations in all types of pumps. Figure 5 shows the principal locations in a vane pump. These are;

i. at the vane tips

ii. at the sides of vanes,

iii. at rotor ends,

iv. at seals,

v. On the cylindrical rotor surface; and

vi. in bearings.

The effect of each force is to produce a resisting torque.

The delivery of a pump is the fluid which is trapped due to the geometrical features of the pump and bodily transported from intake to discharge, minus that fluid which is returned from the discharge to the intake by the geometrical features of the pump, minus the slip and minus the volume of entrained gases or vapour which are expanded at the intake and compressed at the discharge side of the pump.

The torque required to drive a pump consists of the ideal torque due to pressure differential only, plus resisting torque due to viscous shearing of fluid, plus resisting torque due to mechanical friction which is directly proportional to the pressure differential, plus resisting torque due to mechanical friction which is constant.

In fig. 4 are shown the principal paths of leakage for a vane pump. The direction of flow indicated in the sketch is that which would exist in a vane pump. By assuming that the slip flow takes place in the laminar range, that the approximation of flow between parallel flat plates is valid, one can write a general expression for the rate of flow through a slip passage [3] thus:

$$\Delta Q_s = \frac{bd^3\Delta p}{12\mu l} \ldots\ldots\ldots (2\text{-}1)$$

The total slip flow, Q_s, is simply the sum of all the elementary slip flows and is written in the form [3] thus:

$$Q_s = C_s \frac{D\Delta p}{2\pi\mu} \ldots\ldots (2\text{-}2)$$

The actual discharge of the pump can be written as

$$Q = Q_i - C_s \frac{D\Delta p}{2\pi\mu} \ldots\ldots\ldots\ldots (2\text{-}3)$$

There will be a reduction in delivery of the pump due to inlet restriction, Q_r. The ideal delivery, Q_i = DN/60 [3]. The delivery equation becomes

$$Q = \frac{DN}{60} - C_s \frac{D\Delta p}{2\pi\mu} - Q_r \ldots\ldots\ldots\ldots (2\text{-}4)$$

The forces acting on the rotor of a vane pump shown in fig. 5, cause torques that oppose the motion of the rotor. The viscous torques may be expressed in general as follows [3]:-

$$T_v = C_d \frac{D\mu N}{60} \dots\dots\dots\dots (2\text{-}5)$$

The pressure dependent torque can be represented by [3]:-

$$T_f = C_f \frac{D\Delta p}{2\pi} \dots\dots\dots\dots (2\text{-}6)$$

A third torque, T_c, that opposes the motion, is independent of pressure and is constant, is also defined.

The total torque is expressed as

$$T = T_i + T_v + T_f + T_c \dots\dots\dots\dots (2\text{-}7)$$

Substituting equations (2-5), (2-6) and the expression for ideal torque, $T_i = \Delta pD/2\pi$ [3] into equation (2-7) we have

$$T = \frac{\Delta pD}{2\pi} + C_d \frac{D\mu N}{60} + C_f \frac{D\Delta p}{2\pi} + T_c \dots\dots\dots\dots (2\text{-}8)$$

Hydraulic Efficiency

The hydraulic efficiency, E, of a pump is defined as the ratio of the power output to the power input.

Power output, P_o, and power input, P_i, are given by the following basic relationships [3]

$$P_o = Q\Delta p \dots\dots\dots\dots (2\text{-}9)$$

$$P_i = \frac{T\pi N}{30} \dots\dots\dots\dots (2\text{-}10)$$

From the definition for hydraulic efficiency, it

$$P_o=\left(\frac{DN}{60}-C_s\frac{D\Delta p}{2\pi\mu}-Q_r\right)\Delta p \dots\dots\dots\dots (2-11)$$

$$P_I=\left(\frac{\Delta pD}{60}+C_d\frac{D\mu\pi N}{1800}+C_f\frac{D\Delta p}{60}+\frac{\pi T_c}{30}\right)N \dots\dots\dots\dots (2-12)$$

can be shown that

$$E=\frac{1-\frac{30C_s\Delta p}{\pi\mu N}-\frac{60Q_r}{DN}}{1+C_d\frac{\pi\mu N}{30\Delta p}+C_f+\frac{2\pi T_c}{\Delta pD}} \dots\dots\dots\dots (2-13)$$

The assumption that Q_r and T_c are zero coincides with the facts in many practical cases [3]. Under this assumption equation (2-13) becomes:

$$E=\frac{1-\frac{30C_s\Delta p}{\pi\mu N}}{1+\frac{C_d\pi\mu N}{30\Delta p}+C_f} \dots\dots\dots\dots (2-14)$$

The hydraulic efficiency is a function of C_s, C_f, C_d and the parameter $\mu N/\Delta p$. These quantities are significant factors descriptive of the performance characteristic of pumps. They may be used to compare the characteristics of various units effectively.

Large values of C_s indicate large slip; small values of C_f indicate small values of pressure - dependent frictional torques. Small values of these coefficients are necessary for high hydraulic efficiency as could be deduced from equation (2-13). A call for the redesign of a unit can be the result of excessively large values of T_c and C_f. This will indicate possible improvements to eliminate dry or thin film friction. Large values of Q_f indicate the need for better inlet design. The coefficients C_d and C_s are related and cannot both be made arbitrarily small [3]. There is an optimum value of each which gives the minimum value of the product C_sC_d, ensuring best hydraulic efficiency.

The value of the parameter $\mu N/\Delta p$ corresponding to maximum efficiency may be determined by differentiating equation (2-14) with respect to $\mu N/\Delta p$ and setting the result equal to zero. This operation leads to the following maximum efficiency [3].

$$E_{max}=\frac{1}{1+C_f+2C_sC_d[\sqrt{1+\frac{1+C_f}{C_sC_d}}]} \quad \ldots\ldots\ldots\ldots (2\text{-}15)$$

CHAPTER THREE

THEORY OF A POSITIVE-DISPLACEMENT VANE PUMP (pdvp) EMPLOYING TURBULENT FLOW ANALYSIS

The performance of a pump for which all slip flows are assumed to be in the turbulent range is the main concern of this chapter.

For a pump operating within the turbulent range, the fluid torque resistance is proportional to the square of the angular speed, and the slip is proportional to the square root of the pressure differential; slip is independent of the viscosity in this case [4]. The expressions for torque and delivery may be written as follows [4]:-

$$T = e\Delta\alpha + k\omega^2 R^2 + T_c \ldots\ldots\ldots\ldots (3\text{-}1)$$

$$Q = \alpha\omega - \frac{\gamma}{R}\sqrt{\frac{\Delta p}{\rho}} - Q_r \ldots\ldots\ldots\ldots (3\text{-}2)$$

e is a constant for any one pump and is given by the

following expression [4]:-

$$e = 1 + \frac{T_f}{\alpha\Delta p} \ldots\ldots\ldots\ldots (3\text{-}3)$$

But T_f is proportional to the pressure differential Δp, hence

$$T_f = m\Delta p \ldots\ldots\ldots\ldots (3\text{-}4)$$

Substituting equation (3-4) into equation (3-3), gives:-

$$e=1+\frac{m}{\alpha} \ldots\ldots\ldots (3\text{-}5)$$

The output power, P_o, of the pump is given by:-

$$P_o=Q\Delta p \ldots\ldots\ldots\ldots (3\text{-}6)$$

Substituting the expression for Q from equation (3-2) into equation (3-6), yields:-

$$P_o=\Delta p\alpha\omega\left[1-\frac{\gamma}{R}\frac{\sqrt{\Delta p/\rho}}{\alpha\omega}-\frac{Q_r}{\alpha\omega}\right] \ldots\ldots\ldots\ldots (3\text{-}7)$$

The input power, P_i, required to drive the pump is given by:

$$P_i=T\omega \ldots\ldots\ldots\ldots (3\text{-}8)$$

Substituting the expression for T from equation (3-1) into equation (3-8), yields:-

$$P_i=\Delta p\alpha\omega\left[e+\frac{k\rho\omega^2R^2}{\Delta p\alpha}+\frac{T_c}{\Delta p\alpha}\right] \ldots\ldots\ldots\ldots (3\text{-}9)$$

The hydraulic efficiency, E, is generally expressed as:-

$$E=\frac{P_o}{P_i} \ldots\ldots\ldots\ldots (3\text{-}10)$$

Substituting expressions for P_o and P_i from equations (3.7) and (3.9) respectively, into equation (3.10) gives:

$$E=\frac{1-\dfrac{\gamma}{\alpha\sqrt{\rho\omega^2R^2/\Delta p}}-\dfrac{Q_r}{\alpha\omega}}{e+\dfrac{k\rho\omega^2R^2}{\alpha\Delta p}+\dfrac{T_c}{\alpha\Delta p}} \ldots\ldots\ldots\ldots (3\text{-}11)$$

let

$$\phi=\rho\omega^2R^2/\Delta p$$

Hence, equation (3-11) may be rewritten as:-

$$E=\frac{1-\frac{\gamma}{\alpha\sqrt{\phi}}-\frac{Q_r}{\alpha\omega}}{e+\frac{k\phi}{\alpha}+\frac{T_c}{\Delta p\alpha}}\quad\text{..........(3-12)}$$

Assuming the quantities $T_c/\Delta p\alpha$ and $Q_r/\alpha\omega$ to be negligible within the range of performance, the efficiency equation (3-12) reduces to

$$E=\frac{1-\frac{\gamma}{\alpha\sqrt{\phi}}}{e+\frac{k\phi}{\alpha}}\quad\text{..........(3-13)}$$

The displacement, D, is given by [3]

$$D = 2\pi R\delta L \quad\text{........ (3-14)}$$

and

$$\alpha = D/2\pi = \delta LR \quad\text{...... (3-15)}$$

Substituting the expression of α from equation (3-15) into equation (3-13) gives:-

$$E=\frac{1-\frac{\gamma}{\delta LR\sqrt{\phi}}}{e+\frac{k\phi}{\delta LR}}\quad\text{..........(3-16)}$$

The following dimensionless parameters are defined:-

$$L/2R = Ar \quad\text{................. (3-17)}$$

Hence:-

$$L/R = 2Ar \quad\text{................ (3-18)}$$

and :-

$\delta/R = \psi$ .... (3-19)

Substituting equations (3-18) and (3-19) into equation

(3-16), yields:-

$$E=\frac{1-\dfrac{\gamma}{2\psi ArR^3\sqrt{\phi}}}{e+\dfrac{k\phi}{2\psi ArR^3}} \quad \text{..........} \quad (3\text{-}20)$$

Recall that:-

$\alpha = \delta RL$

This can be rewritten as:-

$$\alpha = 2\frac{\delta}{R}R^3\frac{L}{2R} = 2\psi ArR^3 \quad \text{..........} \quad (3\text{-}21)$$

It may be inferred from equation (3-21) that for a constant value of R, a change in either ψ or Ar results in a change in displacement,α. ψ and K are coefficients having dimensions, m^3, dependent on the clearances and geometrical dimensions of the pump.

A change in either clearances or displacement or both will alter the hydraulic efficiency of the pump. The effect of changes in displacement on the hydraulic efficiency of the pump will be investigated.

The influence of the dimensionless parameters,ψ and Ar, and hence the displacement on the hydraulic efficiency of the pump, was found out by varying each of the dimensionless parameters (ψ and Ar) while the other was kept constant. In this particular case, the flow parameter is $\rho\omega^2R^2/\Delta p$. The results for the turbulent analysis are shown graphically in Fig. 6 and Fig. 7.

Figures 6 and 7 show the effect of ψ, Ar and hence, α on the efficiency, E, of a positive-displacement vane pump. For both figures the efficiency starts to increase from zero as $\rho\omega^2R^2/\Delta p$ increases, attains a maximum at a certain value of $\rho\omega^2R^2/\Delta p$ and starts to drop, for the various curves.

The efficiency increases with an increase in clearance between rotor and stator signified by ψ. It also increases with increase in the aspect ratio , Ar. Since the displacement, α , is proportional to ψ and Ar, the efficiency increases with increase in the displacement.

A similar theoretical investigation based on laminar flow was carried out [1]. In it, the relevant flow parameter was $\mu N/\Delta p$; the efficiency increased with *increase in clearance between* rotor and stator and also, increased with increase in aspect ratio.

Though the flow parameters, $\rho\omega^2R^2/\Delta p$ and $\mu N/\Delta p$ relevant to turbulent and laminar analysis respectively are different, the curves for both followed the same basic trend.

The improvement on the hydraulic efficiency brought about by increases in both the rotor-stator clearance, ψ, and the aspect ratio, Ar, is not continuous as could be seen from fig. 6 and fig. 7. For both figures the curves tend to get more and more crowded as ψ, Λr approach 0.5 and 1.0 respectively. It may be inferred that a point will be reached where a further increase ψ,Λr will not significantly improve the performance.

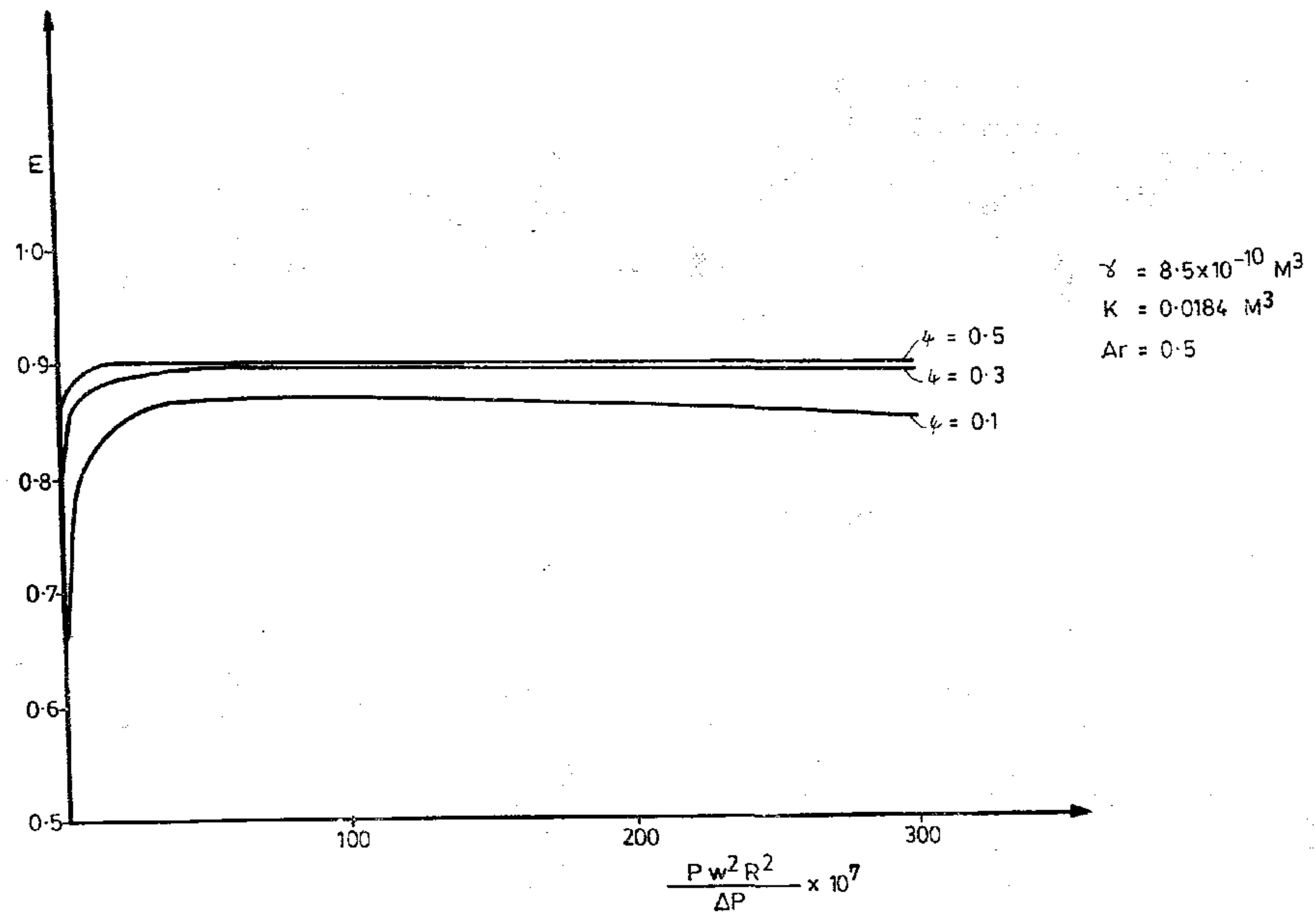

FIG. 6 : Influence of the clearance between Rotor and startor on hydraulic efficiency of a PDVP based on turbulent flow analysis .

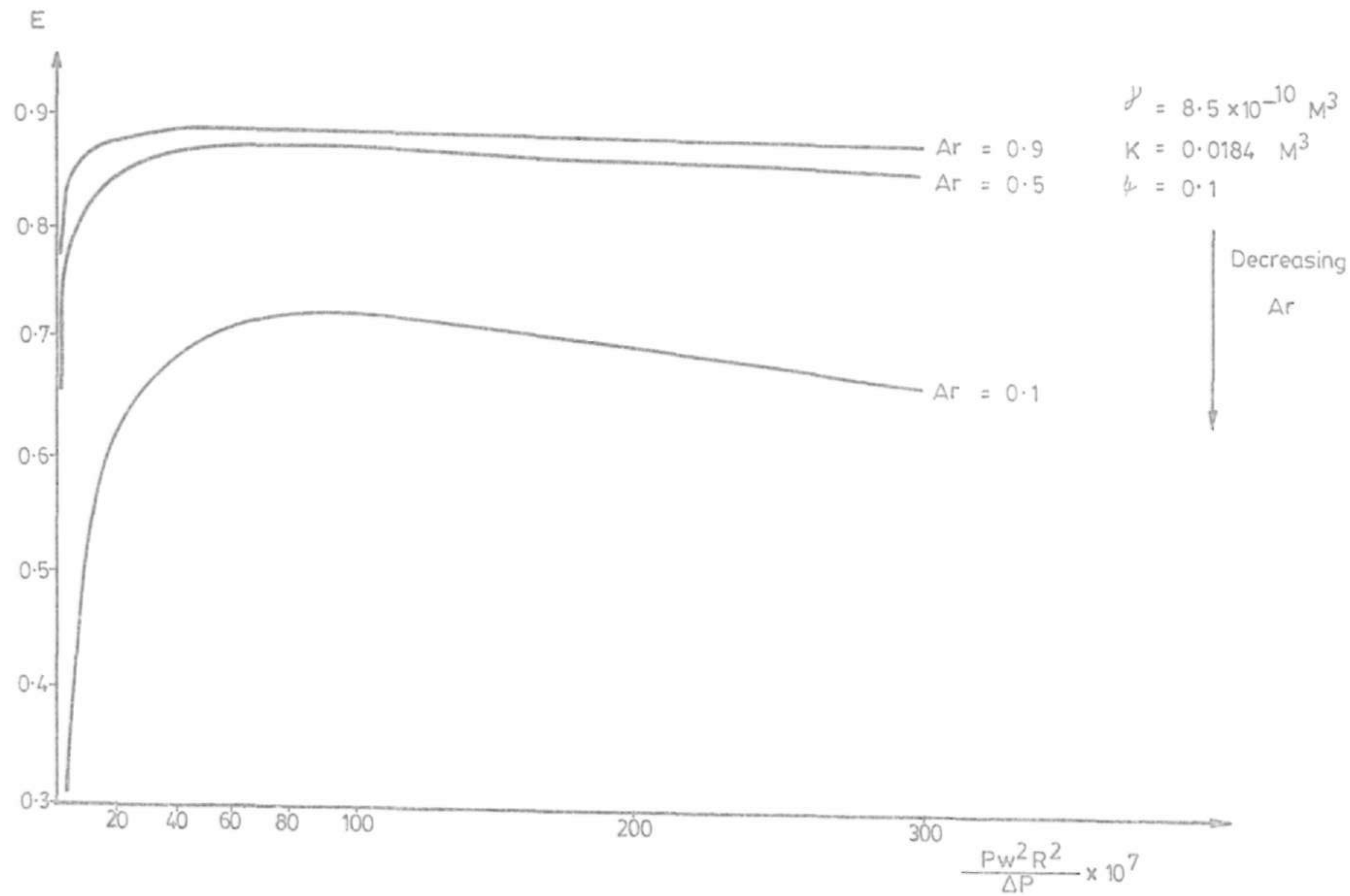

FIG. 7 : Influence of aspect ratio on the hydraulic efficiency of a PDVP based on turbulent flow analysis .

CHAPTER FOUR

EXPERIMENTAL STUDY OF THE pdvp CONSTRUCTED IN THE PHASE I

Rigorous tests on the sliding vane pump designed and constructed, in the phase I [1] of this thesis were carried out. The test observed the variation of the delivery with pressure, viscosity and speed, as well as the input torque and hence, power requirements. The object of testing the pump, is to determined by experimental means the performance characteristics of the pump. The data collected were subjected to detailed analysis.

4.1 The Test Rig

A flow diagram of the essential features of the test rig is shown in fig. 8.

The working fluid was admitted into the system from the reservoir. The two bourdon gauges, one at the discharge and the other at intake to the pump measured the pressure at these locations. A gate valve was placed some distance away from the discharge bourdon gange, for varying the discharge pressure. A heat exchanger, at the discharge of the pump was incorporated for cooling the working fluid to its temperature at intake. The working fluid was then stored in the reservoir for recirculation. The pump was driven by a 3 -phase, 3 horsepower induction motor, which was coupled to a dynamometer to measure the torque input to the pump.

4.2 Test Procedure and Results

With the fluid fully admitted into the pump which was driven by the motor, for a fixed rotational speed (which was measured by a tachometer and a stroboscope flash), the pump was ran at various discharge pressures. After a steady state has been reached for each rotational speed and discharge pressure, readings of the intake pressure, discharge pressure, time for fluid to fill a 2.2 litre container and reading on the dynamometer scale were taken. The temperature of the fluid in the reservoir was constantly measured. A sample of the fluid was taken and its viscosity measured using a Redwood viscometer. The test was repeated using a different working fluid. The working fluids used were groundnut oil and coolant solution. The processed results of the tests are tabulated in table 1 for groundnut oil and table 2 for coolant solution.

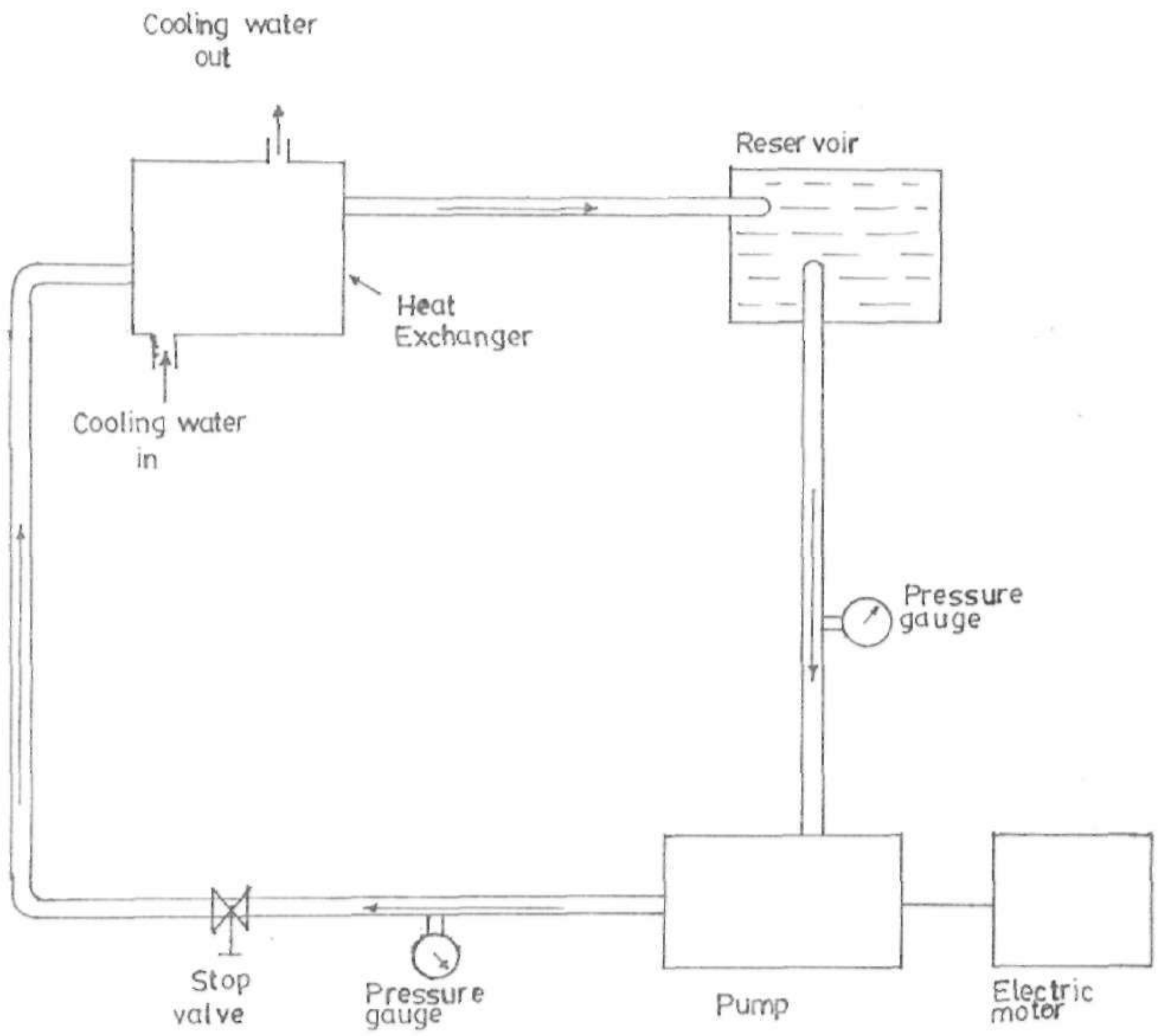

Fig. 8 : Schematic representation of the experimental test rig.

TABLE 1: GROUNDNUT OIL Viscosity = 0.047Nsm^{-2}

PUMP SPEED N (rpm)	PRESSURE DIFF. Δp (bars)	DELIVERY $QX10^6$ (m^3/S)	TORQUE T (N-m)
2010	0.26209	390	0.54
2310	0.26209	478	0.60
2550	0.26209	550	0.63
2690	0.26209	611	0.67
2010	0.5518	297	0.65
2310	0.5518	393	0.72
2550	0.5518	460	0.76
2690	0.5518	500	0.79
2010	0.9311	140	0.80
2310	0.9311	220	0.85
2550	0.9311	333	0.91
2690	0.9311	380	0.92
2010	1.2208	15	0.90
2310	1.2208	100	1.00
2550	1.2208	214	1.02
2690	1.2208	268	1.04

TABLE 2: COOLANT SOLUTION Viscosity = $0.00134 Nsm^{-2}$

PUMP SPEED N (rpm)	PRESSURE DIFF. Δp (Bars)	DELIVERY $Q X 10^6$ (m^3/s)	TORQUE T (N-m)
2020	0.9311	60	0.46
2290	0.9311	140	0.53
2530	9.9311	262	0.61
2680	0.9311	320	0.66

4.3 Data Analysis

The objective of the test is to determine by experimental means:-

1. the displacement D, of the pump,
2. the performance coefficients C_s, C_d, C_f and the torque T_c,
3. the performance of the pump with liquids of different viscosities.

4.3.1 Delivery

A plotting of the delivery versus speed for the various pressure differences was carried out, fig.9. From equation (3-3), the displacement of a pump is the slope of the delivery-speed curves. The inlet restriction is negligible in this operating range of speeds and pressure differences. The displacements found for the four curves are:

Δp (Bars) $D x 10^6 (m^3/rev)$

0.26209	19.11
0.5518	17.86
0.9311	21.73
1.2208	22.73

The average of the displacements is

$D = 2.036 \times 10^{-5}\ m^3/rev$

The intercept on the delivery axis is the slip Q_s, at the various pressure differences. They are:

Δp (bars)	$Q_s x10^6 (m^3/s)$
0.26209	254
0.5518	299
0.9311	597
1.2208	756

In order to determine the slip coefficient C_s, the vertical intercepts Q_s, of delivery-speed curves are plotted versus pressure differential Δp, as shown in fig. 10. From equation (2-2), the slope of this line was calculated to be $5.968 x10^{-11}\ N^{-1}\ m^5 s^{-1}$. The viscosity of the working fluid, groundnut oil, was found to be $0.047 Nsm^{-2}$

$$C_s = 2\pi\mu \times slope/D$$

$$C_s = \frac{44}{7} \times 0.047 \times \frac{5.968x10^{-11}}{2.036x10^{-5}} = 8.65x10^{-5}$$

4.3.2 Torque

The torque input to the pump was plotted versus the speed at constant viscosity and for each of the four pressure differences shown from fig. 11 to fig. 14. The slope of each of the torque-speed plotting is the quantity $C_d D\mu/60$. The vertical intercept to the torque axis will be denoted by T_o. It is the sum of the ideal torque $\Delta pD/2\pi$ and T_c as seen from equation (2-8).

$$T_o=\frac{\Delta pD}{2\pi}+C_f\frac{D\Delta p}{2\pi}+T_c$$

The slopes and T_o found for the four curves are:

Δp (bars)	Slope (Nmxmin.)	T_o (N-m)
0.26209	0.00018242	0.174
0.5518	0.000202703	0.246
0.9311	0.000186186	0.425
1.2208	0.000199699	0.513

The average of the slopes is

$$\text{Slope} = 1.9275475\text{x}10^{-4} \text{ (Nmxmin)}$$

$$C_d = \frac{\text{slope}}{\mu D} \times 60 = 60 \times \frac{1.9275475\text{x}10^{-4}}{0.045\text{x}2.036\text{x}10^{-5}} = 1.2086\text{x}10^{4}$$

The vertical intercepts were plotted against the corresponding pressure differentials to determine C_f

and T_c, as shown in fig. 15. The slope of this line is the quantity $(D/2\pi)$ $(1+C_f)$. The intercept on the torque axis is the friction torque T_c

$$\text{Slope} = 0.3696169 \times 10^{-5}$$

$$C_f = \frac{\text{Slope} \times 2\pi}{D} - 1$$

$$C_f = \frac{(0.3696169 \times 10^{-5} \times 44)}{2.036 \times 10^{-5} \times 7} - 1$$

$$C_f = 0.14$$

$$T_c = 0.065\text{N-m}$$

4.3.3 Comparison of Performance Between Groundnut oil and Coolant Solution:

The performance of the pump at 0.9311 bars and range of speeds with groundnut oil and coolant solution were compared. The efficiencies at 0.9311 bars for groundnut oil and coolant solution were calculated and tabulated below. A plotting of delivery versus speed and efficiiencies versus pump speed, for the groundnut oil and coolant solution were carried out. The plottings are shown in fig. 16 and fig. 17 respectively.

GROUNDNUT OIL

Pump Speed (rpm)	Pressure DIFF Δp (bars)	Efficiency (E)
2010	0.9311	0.08
2310	0.9311	0.10
2550	0.9311	0.13
2690	0.9311	0.14

COOLANT SOLUTION

Pump Speed (rpm)	Pressure DIFF Δp (bars)	Efficiency (E)
2020	0.9311	0.06
2290	0.9311	0.10
2530	0.9311	0.15
2680	0.9311	0.16

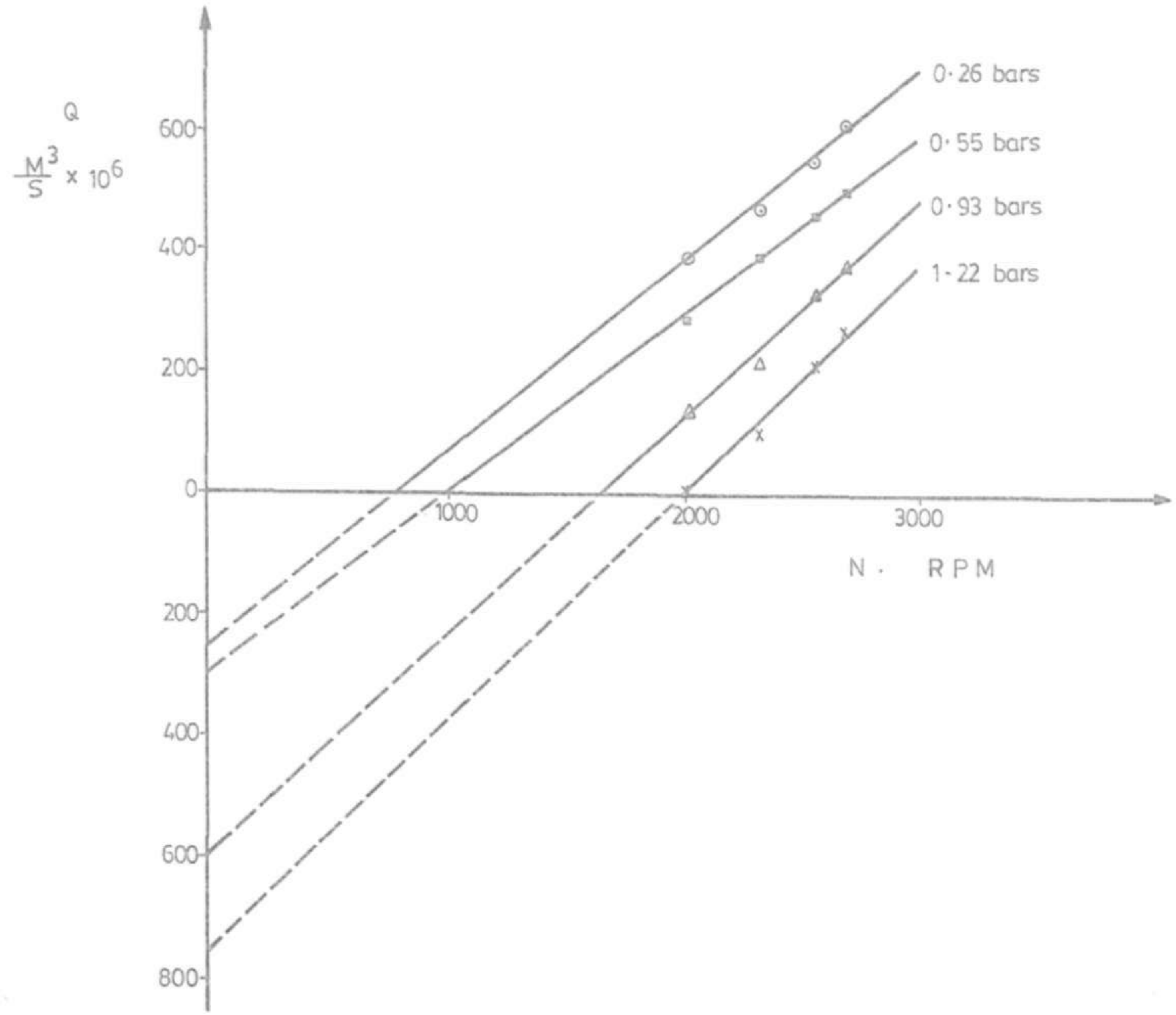

FIG. 9: Delivery — speed relation for the pump.

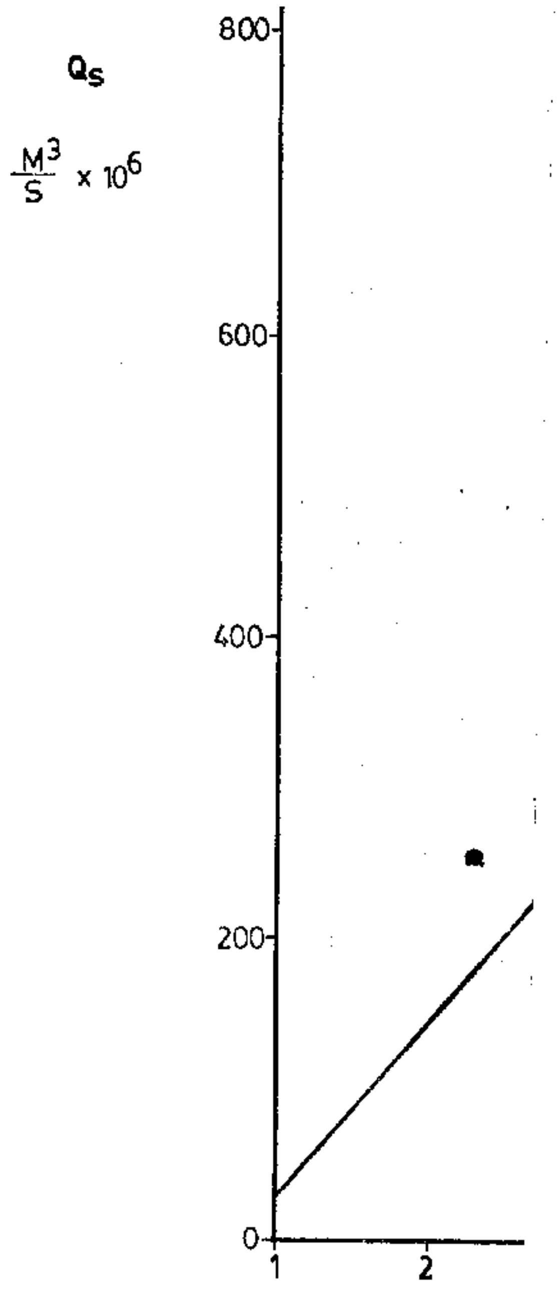

FIG. 10: Slip —

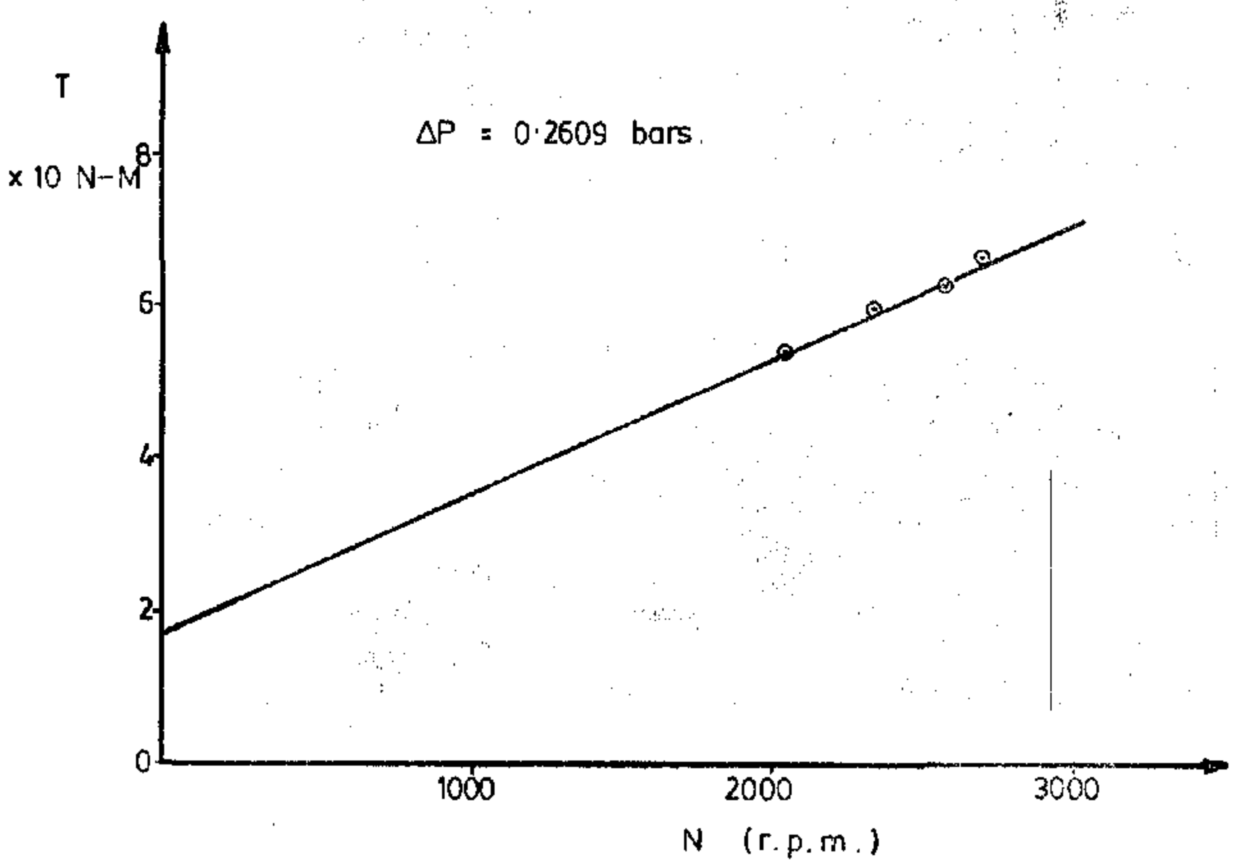

FIG. 11 : Torgue — speed relation for ΔP = 0·2609 bars.

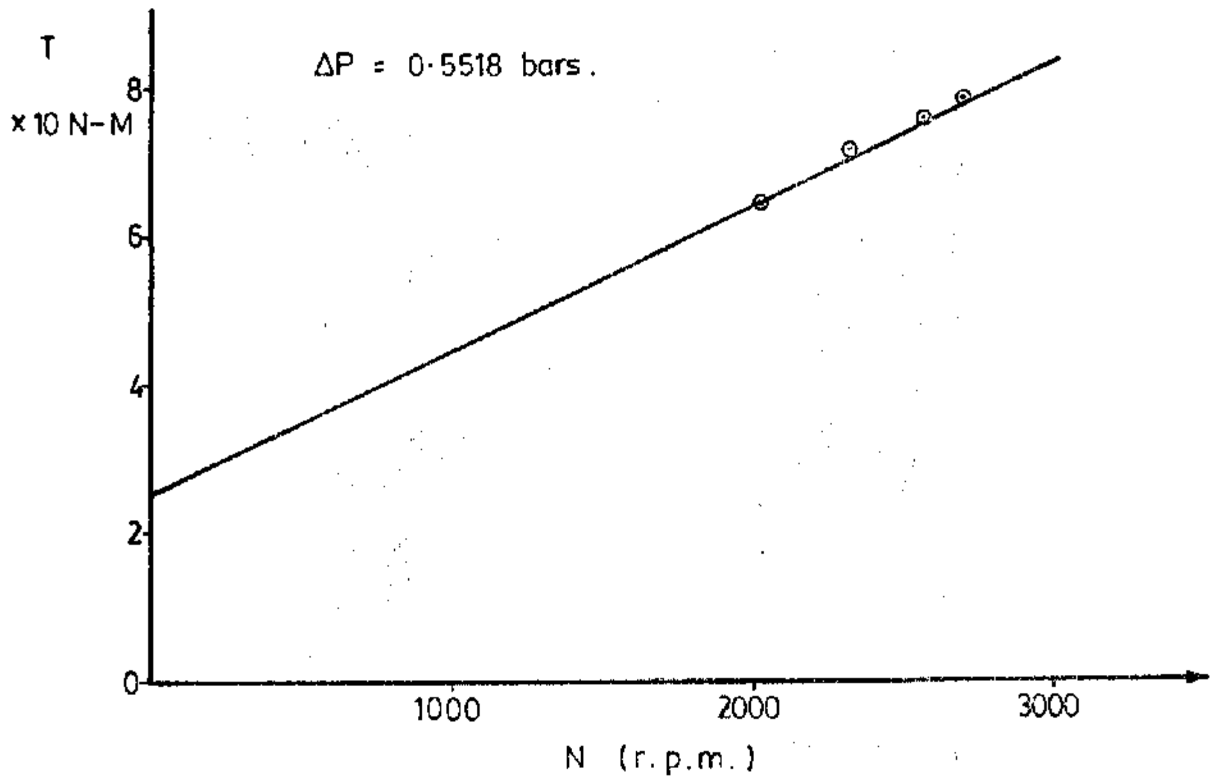

FIG. 12 : Torgue — speed relation for ΔP = 0·5518 bars.

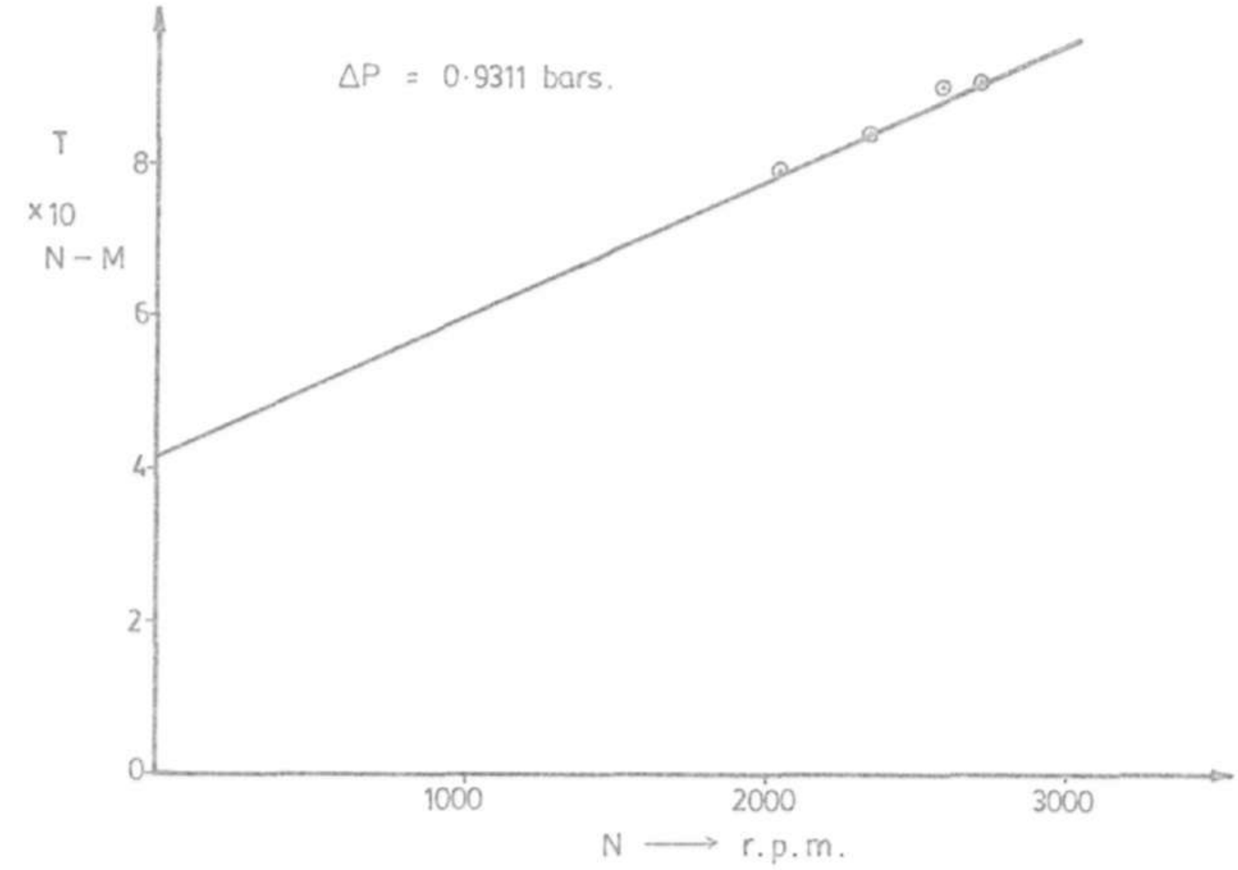

FIG. 13 : Torgue — speed relation for ΔP = 0·9311 bars.

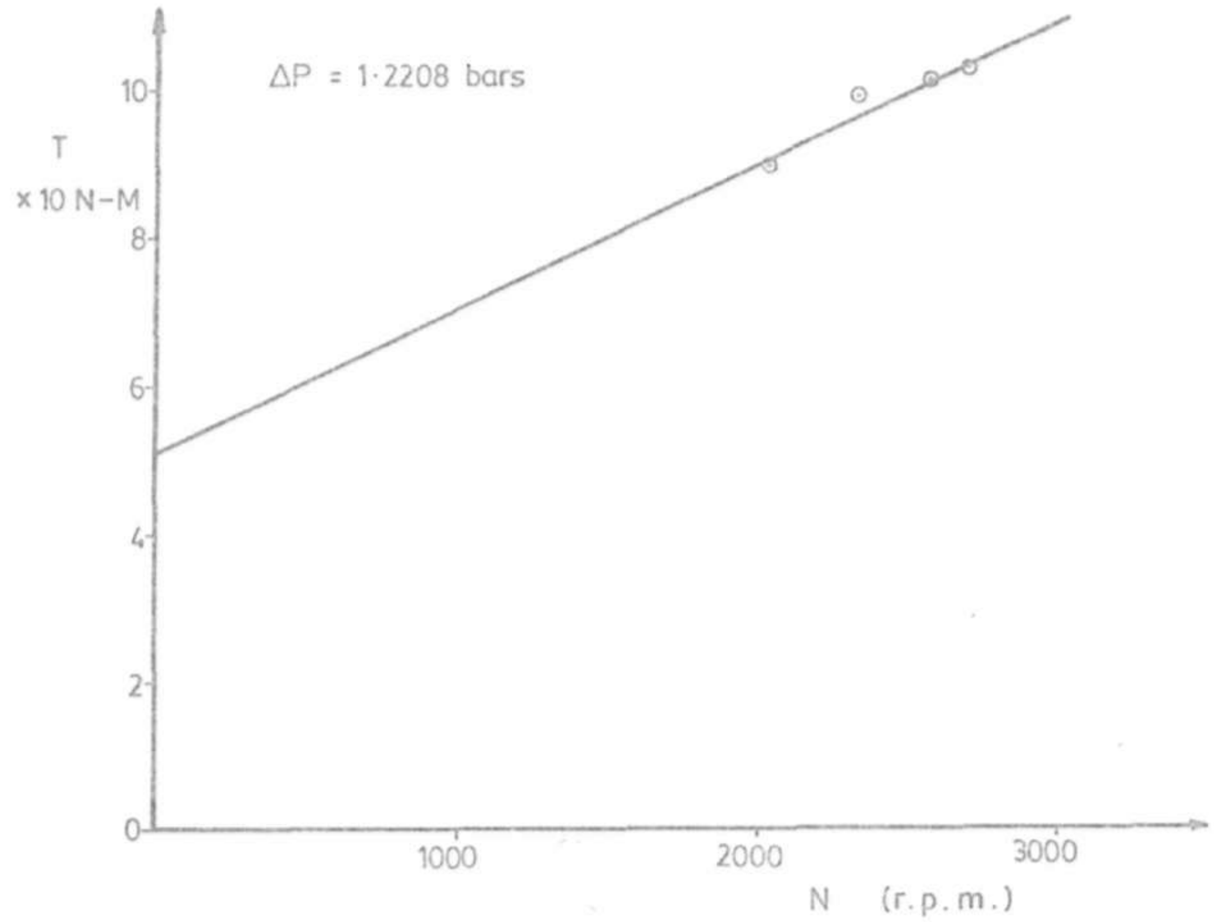

FIG. 14 : Torgue — speed relation for ΔP = 1·2208 bars.

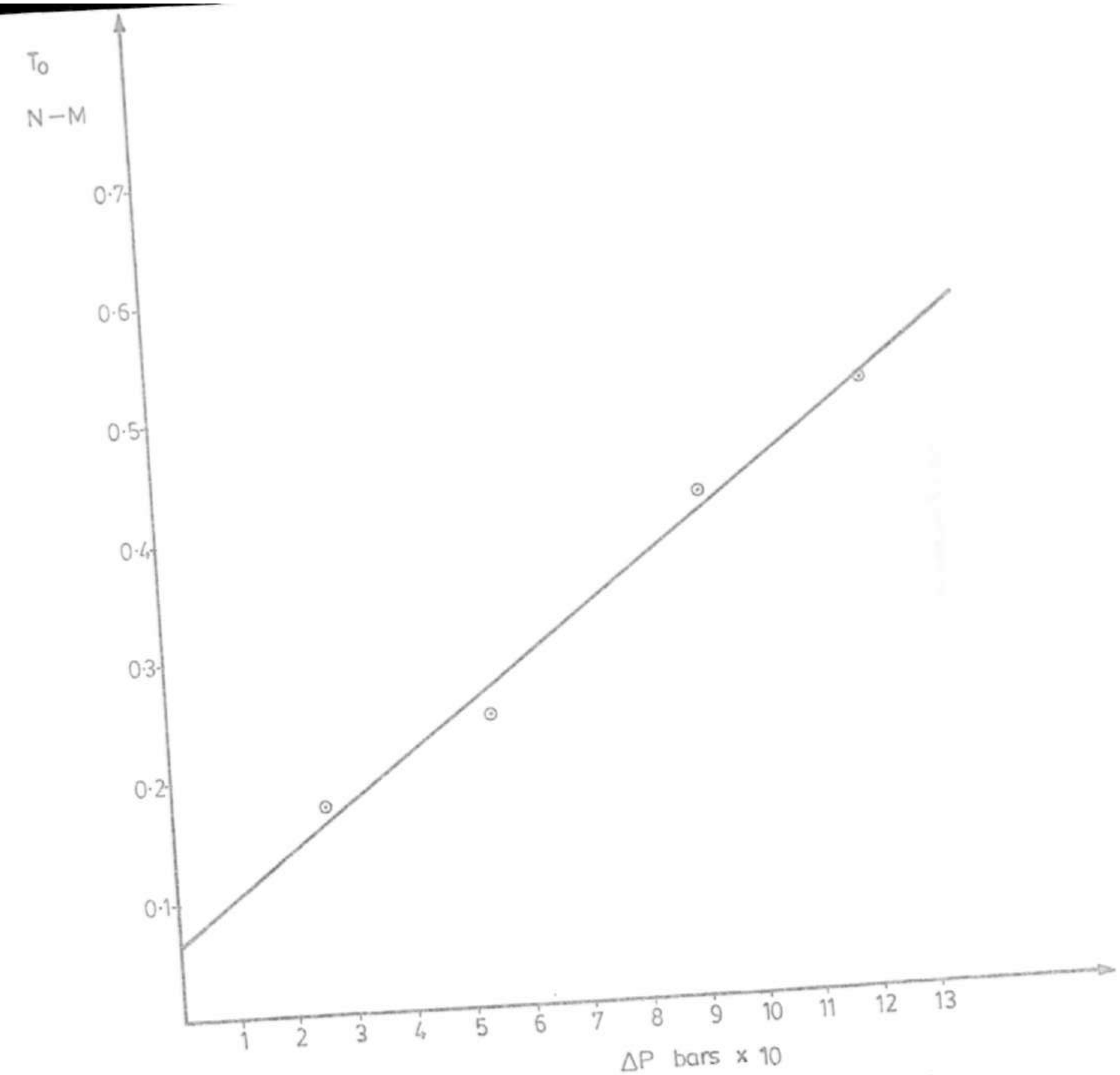

FIG. 15: Vertical intercept, To — pressure differential relation.

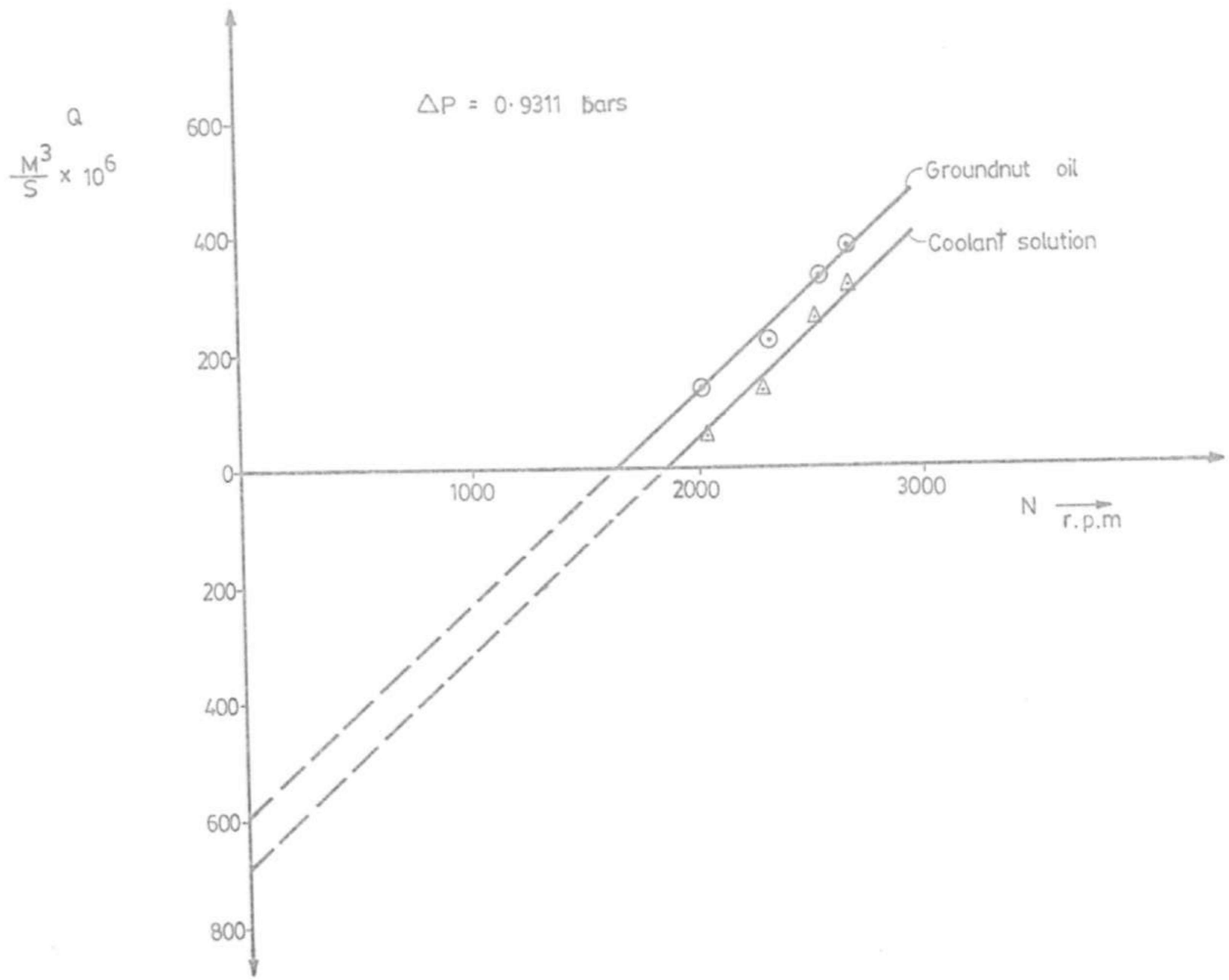

FIG. 16 : Delivery — speed relation using groundnut oil and coolant for Δp = 0·9311 bars.

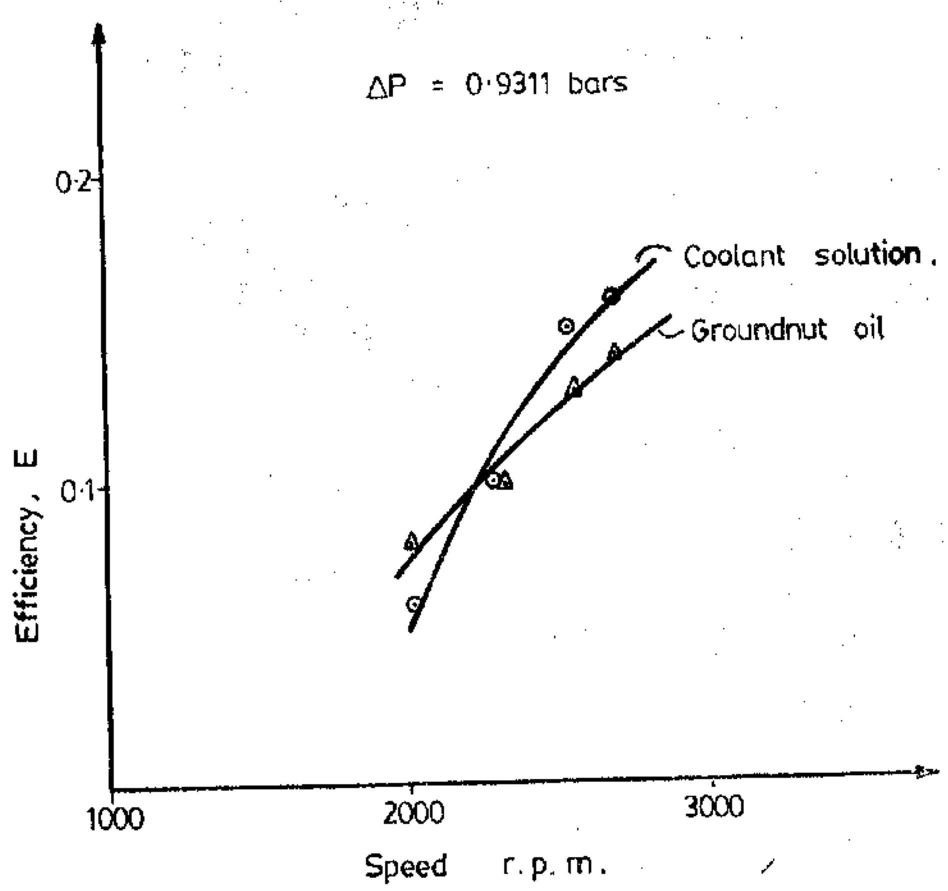

FIG. 17: Measured efficiency characteristics for the pump.

4.4 DISCUSSIONS AND RESULTS

Experimentally determined values:

$D = 2.036 \times 10^{-5}\ m^3/rev$

$C_s = 9.65 \times 10^{-5}$

$C_d = 1.2086 \times 10^{4}$

$C_f = 0.14$

$T_c = 0.065 N\text{-}m$

Substituting these experimentally determined performance coefficients into equation (2-15), to calculate the maximum efficiency that could be attained by the pump, we have:

$$E_{max} = \frac{1}{1+0.14+2x0.865x1.2086\left[\sqrt{1+\frac{1+0.14}{0.865x1.2086}}\right]}$$

$$E_{max} = 0.16$$

For the tested pump, because the displacement calculated from experimental data was lower than the design value and the general performance of the pump was poor, an investigation was embarked upon as to why.

1. The displacement, $2.036 \times 10^{-5} m^3/rev$ determined experimentally is smaller than the designed value of $5.339 \times 10^{-5} m^3 m^3/rev$ [1]. This indicates that the pump will not deliver the required amount of liquid at the required time. Measurements and observations revealed the following:-

(a) The rotor length was not machined to the design value of 48mm [1]. It is less by 1mm

(b) The rotor port in the stator was not machined to the actual dimensions marked out for it on the stator blank. The boundary of the rotor port fell below the actual marked out outline of the rotor port at some portions.

(c) The vanes occupied part of the volume swept out in one revolution of the rotor. The volume of liquid that should get into the pump per revolution of the rotor, is lessened by the volume occupied by the vanes. Consequently, the displacement got from experiment is bound to be less than the design value.

(d) The formula, $D = 2\pi R\delta L$, used in the design [1] assumes that the rotor-stator clearance, δ, is the same all round. Actually, it varies from zero at the dividing seal to a maximum value. For this pump it varies to a maximum of 6mm [1]. As a result, a smaller displacement is bound to be found experimentally.

2. The low maximum efficiency of 0.16 as calculated from equation (2-15) indicates that the performance coefficients of the pump are high which is undesirable. The pump was constructed in the department of mechanical engineering workshop, Ahmadu Bello University, Zaria - Nigeria. Discussions with the technologist, who

was involved in the construction of the pump revealed the problems that were encountered. These include the use of old machines, and unsatisfactory cutting tools which had to be ground very often. The grinding wheels used were old and cutting the correct rake angle, cutting edges, clearances, on the tool were not guaranteed. Mohammad, 1990,[1] concedes that the desired finish was not attained. The surface finish of a portion of the rotor port was measured and found to be $1.9\mu m$. This when compared with that of a Vickers double-lobe pump manufactured in the United Kingdom with a surface finish of $0.76\mu m$ indicated poor surface finish for the pump. The poor performance of the pump as indicated by the calculated maximum efficiency of 0.16, could be largely due to the poor construction of the pump parts.

3. The delivery versus speed plotting, fig. 16, shows more slip with the coolant solution than groundnut oil. This is due to the higher viscosity of groundnut oil over coolant solution. The groundnut oil has a viscosity of $0.047 NSM^{-2}$ at $33^{o}C$ and the coolant solution $0.00134 NSM^{-2}$ at $33^{o}C$.

4. The coolant solution gave a better performance as shown graphically in fig. 17, than groundnut oil. This was because the viscous torques with groundnut oil, were so high that they more than

off-set the improvement brought about by the greater lubrication of the pump.

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions:

The influence of rotor-stator clearance as well as the rotor aspect ratio, on the hydraulic efficiency of a pdvp was theoretically investigated, assuming turbulent flow within clearances. The theoretical analysis showed that as the rotor-stator clearance, ψ, increases the hydraulic efficiency also increases. Similarly, the hydraulic efficiency also increases with increase in the aspect ratio. However, the increase in the hydraulic efficiency of the pump becomes insignificant as ψ and Ar approach 0.5 and 1.0 respectively. Similar results were obtained in the phase I, where laminar flow was assumed to occur within clearances.

2. The pdvp constructed in Phase I, was experimentally tested using groundnut oil and coolant solution and its performance characteristics determined. The delivery-speed characteristic, torque-speed characteristic, as well as displacement and maximum efficiiency were established. In particular, the pump was found to have a displacement of about half the design value. A maximum efficiency of about 16% was also measured using groundnut oil as the working fluid.

5.2 Recommendations

The pump should be reconstructed, with the aim of meeting the dimensions of the pump as in the design, as precisely as possible. In particular, the rotor port surface, rotor ends, slots in the rotor for the sliding vanes should be machined to an acceptible surface finish comparable to that of the Vickers double-lobe pump measured. These therefore, call for the use of satisfactory machines and new tools ground to the correct rake angle, cutting edges for machining the pump parts.