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The reduction of gear pump pressure ripple by source flow modification

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THE REDUCTION OF GEAR PUMP PRESSURE RIPPLE BY SOURCE FLOW MODIFICATION

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SUMMARY

Pressure fluctuations in hydraulic systems may be substantially reduced by cancelling the flow ripple produced by the pump. This can be achieved by introducing an equal and opposite flow fluctuation from a secondary source mounted as close as possible to the pump outlet.

A secondary flow ripple generating device is described which, although specifically designed for external gear pumps and motors, may be applied to other types of positive displacement unit. The device, which consists primarily of a number of radial piston-followers controlled by a multi-lift cam driven directly by the pump driveshaft, is self-contained, very compact and capable of effecting considerable reduction in both inlet and outlet pressure ripples.

Tests performed over a range of speeds up to 3000 rev/min., and pressures up to 200 bar, have shown that up to four harmonics of pumping frequency can be virtually eliminated. The degree of reduction is dependent upon an accurate knowledge of the evaluation of the pump source flow characteristics. External gear pump inlet and outlet flow ripple characteristics have been evaluated, using specially developed test procedures designed to obtain pump and hydraulic system dependent parameters. These characteristics are compared over a wide range of mean pressure with characteristics derived from an analysis of involute gear geometry.

The reduction in gear pump pressure ripple has been shown to reduce overall airborne noise levels from a hydraulic system by as much as 10 dB, although the airborne noise radiated from the pump casing was not affected.

A method of utilizing the device as a means of determining gear unit flow ripple characteristics is proposed and time independent volume variations are discussed as a criterion for the comparison of pump fluidborne noise generating potential.

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Finally, many thanks to Mrs. J. Hughes for typing this record of the work.

NOTATION			Units
A	-	Internal area of pipe	m^2
B	-	Effective bulk modulus of the fluid	N/m^2
b	-	Gear width	m
C	-	Local speed of sound in fluid	m/s
f	-	Frequency	Hz
γ	-	Wave propagation constant	-
j	-	Complex operator	-
L	-	Characteristic length of pump discharge passageway	m
l	-	Length of line	m
m	-	Module of gear	m
Po	-	Pressure fluctuation at pump flange	N/m^2
Px	-	Pressure fluctuation at distance x from source	N/m^2
Qi	-	Instantaneous flow rate	m^3/s
Qm	-	Mean flow rate	m^3/s
Qs	-	Pump (source) flow fluctuation	m^3/s
R	-	Pressure drop/unit length/unit flow in pipe	Ns/m^6
ρ	-	Fluid density	Kg/m^3
ρ_S	-	Source reflection coefficient	-
ρ_T	-	Termination coefficient	-
t	-	Time	s
θ	-	Angle turned through by gear and cam	rad.
ϕ	-	Gear pressure angle	deg.
Vm	-	Volume variation produced by mechanism	m^3

NOTATION - continued

			Units
Vp	-	Internal volume of pump	m^3
Vs	-	Pump (source) volume variation	m^3
w	-	Frequency	rad.
x	-	Distance from source to point on line	m
Z_E	-	Entry impedance of hydraulic system	Ns/m^5
Zo	-	Line impedance	Ns/m^5
Zp	-	Characteristic impedance of pump discharge passageway	Ns/m^5
Zs	-	Source impedance	Ns/m^5
Z_T	-	Termination impedance	Ns/m^5
Z	-	Number of pumping elements	-
σ	-	Contact or Hertz stress	N/m^2
F	-	Force on piston-follower	N
fi	-	Pressure angle	degree
Cc	-	Curvature of cam	m
r	-	Piston-follower radius	m
cl	-	Contact length	m
Uc	-	Poisson's Ratio (cam material)	-
Uf	-	Poisson's Ratio (piston-follower material)	-
Ec	-	Young's Modulus (cam material)	N/m^2
Ef	-	Young's Modulus (piston-follower material)	N/m^2

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CHAPTER 1 - INTRODUCTION

1.1 - GENERAL BACKGROUND

The high specific power capabilities and the inherent flexibility and control of modern oil-hydraulic systemshas led to a growth in their use in a wide variety of environments. Today it is no more unusual to find hydraulic systems providing power for lifts in shops and hospitals or numerous vehicle applications, than it is for press tools and extrusion machines. Further more, with the advent of economic microprocessor control the demand for hydraulics is likely to increase, especially in the fields of machine tools and robotics.

The widespread use of hydraulic systems has led to their application in areas where the attendant noise from the hydraulic system is unacceptably high. Fluid power system noise is well known, particularly in mobile applications such as agricultural vehicles and fork lift trucks, where the operator is often very close to the noise source.

Prolonged exposure to noise at work increases the degree of hearing loss in a typical industrial population, ref. (1). In addition to the social handicap which deafness represents, levels of operator performance are also likely to be reduced. Legislation, in the form of the Health and Safety at Work Act 1974, has set maximum levels of noise to which an operator in an industrial environment may be subjected. This has promoted the need for the reduction of noise levels from many industrial sources, including hydraulic systems.

The noise radiated by a hydraulic system, excluding the effects of the prime mover, is caused predominantly by the discontinuous action of the hydraulic pump. In delivering oil at high pressures, cyclic loads are produced which act on the moving elements within the pump, causing vibrations or structureborne noise. This structural vibration may be transmitted through the pump casing, via the mounting stand or the connecting pipes, to the rest of the system causing sympathetic vibrations and consequent pressure fluctuations in the atmosphere known as airborne noise.

The structural vibrations due to load variations within the pump may be minimized by good pump design and manufacture. The transmission of structureborne noise to the rest of the system may be reduced by the use of flexible driveshaft couplings and mounting arrangements ref. (2) and flexible connecting pipes ref. (3).

The intake and delivery of fluid by most positive displacement pumps is unsteady and exhibits fluctuations dependent upon the mechanism of fluid transfer and the number of pumping elements. These periodic flow fluctuations produce pressure fluctuations in the fluid, termed fluidborne noise, which are transmitted via the fluid throughout the hydraulic system causing structural vibration and airborne noise.

Attempts to reduce the contribution made by the pump to overall system noise levels has shown that whereas a high degree of structural isolation may be obtained by flexible mounting of the pump, the isolation of pump pressure ripple is more difficult to achieve. A fluidborne noise silencer ref. (4), placed between the pump and system, can be effective over a limited frequency range, but reductions at low frequencies are generally limited by the size of the silencer and the acceptable pressure drop across it. Flexible hoses, which are known to have good structural vibration insulation characteristics, have also been shown to exhibit useful pressure ripple attenuation properties refs. (5, 6).

The reduction of pump pressure ripple by careful pump design has been the subject of a great deal of research, much of the effort being directed at piston pump port plate design refs. (7, 8). Although significant improvement can be achieved by attention to this area of design it is not practical to achieve the low levels of pressure ripple, characteristic of the internal gear pump. Unfortunately this machine is typical of low flow ripple pump designs in that its displacement at a given speed is constant. The use of variable displacement pumps, to improve overall system efficiency, often results in the necessary acceptance of higher levels of pump fluidborne noise. This exemplifies the need for a general means of pump pressure ripple reduction.

The considerable potential for active methods of pressure ripple reduction has been shown by the controlled interaction of pumping elements. A 75% reduction in gear pump flow ripple may be achieved by anti-phasing the flow ripple produced by the two halves of the gear ref. (9). Reductions may also be achieved by the interaction of gear and piston type pumping mechanisms ref. (10). However there are formidable problems in the application of such active methods. One of the difficulties in designing a dedicated pressure ripple reduction device is evaluating the variation in flow which causes the pressure ripple.

This problem has recently been solved by an evaluation technique which is the outcome of a program of work that first commenced in 1972 at the Fluid Power Centre, University of Bath. This work established that the airborne noise from an isolated pump was significantly less than the airborne noise produced by the pump and an associated system. This emphasized the dependence of hydraulic system airborne noise on the fluidborne noise produced by the pump.

The study of the generation and propagation of pressure waves in hydraulic systems, refs. (11, 12, 13, 14), led to the development of a technique for modelling pump and system fluidborne noise characteristics. This technique allowed a quantitative means of assessing the fluidborne noise generating potential of different pumps.

An important application of this modelling technique provided the means of experimentally evaluating pump flow fluctuation and source impedance. This has been essential to the work described in this thesis of reducing gear pump pressure ripple by source flow modification.

1.2 - OBJECTIVE OF WORK

The objective of this work was to determine a means of reducing the pressure ripple produced by positive displacement pumps, in particular an external gear unit, by modifying the inherent fluidborne noise characteristics.

The author has, in a previous work ref. (15), examined the effect of internal volume modification on gear pump pressure ripple. It was shown that, although a 20% reduction in pressure fluctuations could be achieved by doubling the pump internal volume, the method had insufficient potential to achieve significant reductions. This did however promote the realisation that modification of the flow fluctuation of the pump, by some active means, would be the most effective method of pressure ripple reduction. Of course it is important that the resulting pump modification should be a practical and commercially viable proposition.

The majority of work carried out on pump fluidborne noise has been associated with the transmission of pressure ripple from the pump outlet port through the connected system. Recent research refs. (16, 17) has however shown that under boosted conditions, the levels of pressure ripple produced by a pump in the inlet line are similar to those produced in the outlet. A similar situation occurs when a motor is operating with moderate back pressure. Consequently, airborne noise may be generated by both inlet and outlet pressure ripples. This was given due consideration in determining the method of pump modification and both sources were investigated.

1.3 - SCOPE OF THE THESIS

This thesis starts, chapter 2, with a description of the mechanism of wave propagation and its effects in hydraulic systems.

In chapter 3 the parameters influencing pump pressure ripple are discussed and fundamental expressions for the flow and volume variations produced by involute gear geometry are developed.

Chapter 4 reviews a number of active methods of reducing pump pressure ripple by modification of the source flow fluctuation. A basic design specification is also developed for a dedicated flow ripple cancellation device suitable for application to an external gear pump.

Chapter 5 discusses the feasibility of some design principles for a suitable flow ripple generating device.

Chapter 6 is a detailed account of the design of a multi-lift cam mechanism suitable for gear pump first harmonic flow ripple reduction. The performance of this MK I design is described in chapter 7, together with the test rig and methods of testing used to evaluate both pump and mechanism characteristics.

Chapter 8 describes the experimental tests performed on external gear pumps and motors with and without source flow modification by two improved multi-lift cam mechanism designs. The effects of single and multiple harmonic source flow reductions on pressure ripple, torque ripple and airborne noise are discussed.

Appendix 1 presents detailed documentation of the digital computer programs used to plot pump flow and volume variations and to design multi-lift cam profiles.

Appendix 2 presents a patent specification, relating to pumps and motors, which resulted from this work and describes an invention which is the property of the author and the sponsoring body:- Dowty Hydraulic Units Ltd., Cheltenham, Glos.

Appendix 3 presents details of hydraulic components and instrumentation used in this work.

CHAPTER 2 - FLUIDBORNE NOISE GENERATION AND ITS EFFECTS IN HYDRAULIC SYSTEMS

INTRODUCTION

The discharge of most positive displacement pumps is unsteady because the transfer of fluid, from inlet to outlet, is achieved by the action of a number of individual pumping elements (for example, pistons, vanes or gear teeth). Pump delivery therefore consists of a steady mean flow with a superimposed ripple or fluctuation which repeats at pumping element frequency. These flow fluctuations then produce pressure fluctuations (fluidborne noise) about a mean pressure level, which is determined by the resistance to mean flow.

Flow and pressure fluctuations often exhibit complex waveforms, but as they are periodic, it is possible to analyse them in terms of harmonic components using Fourier Analysis. Each flow or pressure signal can be represented by the sum of several sinusoidal waves, having frequencies of integer multiples of the fundamental pumping frequency. Each harmonic may then be considered in turn and the signal waveform constructed by summing the individual amplitudes of a number of harmonics at their respective phases. Ten harmonics is sufficient to describe the waveforms produced by most types of hydraulic pump or motor.

2.1 - PRESSURE WAVE PROPAGATION

The behaviour of pressure waves in a hydraulic system may be explained by considering one harmonic of flow ripple produced by a pump, discharging into a straight length of pipeline, terminated with a restrictor valve.

The motions of the pumping elements produce a flow disturbance which varies sinusoidally with time (a compression and expansion of the fluid). This flow variation, which originates at the surface of the moving elements, produces a corresponding pressure variation which propagates along the pipe at the speed of sound in the fluid (approximately 1400 m/s in oil). This pressure wave, known as the incident wave, is superimposed upon the mean pressure level generated by the resistance of the valve to the mean flow, and travels to the end of the line, where it is reflected back toward the pump by the valve.

The reflected wave is at the same frequency as the incident wave, but is of lower amplitude and lags the incident wave because of the energy lost during reflection. When the reflected wave reaches the pump it is again reflected and this process continues until all the wave energy is lost through reflections at the source and termination, and friction in the pipe.

The interaction of the incident wave and the subsequent reflections results in the formation of a "standing wave" at the incident wave frequency. The standing wave in a line shows the positions of high pressure fluctuations (anti-nodes) and low pressure fluctuations (nodes). The length of line required for a standing wave to have a complete period is called a wavelength and nodes are half a wavelength apart. Line lengths which produce large pressure fluctuations are referred to as resonant lengths for the system at a particular frequency. Resonant lengths are also half a wavelength apart.

In systems requiring low noise levels, resonant lengths should be avoided. This is not always possible as line lengths which generate low amplitudes at one frequency may correspond to resonant lengths at other harmonics. The problem is also more difficult with a varying frequency, as is often encountered with variable speed primemovers.

2.2 - PRESSURE FLUCTUATIONS IN A SIMPLE SYSTEM

Bowns and McCandlish ref. (11) used plane wave propagation theory to model the pressure fluctuations produced in a simple hydraulic system. The system, shown in fig. 2.1, consists of a positive displacement pump discharging into a straight rigid pipe, terminated with a restrictor valve.

The pump is characterized by its source flow fluctuation (Qs) and its source impedance characteristic (Zs). The pipeline, attached to the pump discharge flange is filled with fluid at some mean pressure and is represented by its "characteristic impedance" (Zo) which is a function of the internal pipe diameter, properties of the fluid and operating conditions. The attenuation and phase shift of the transmitted wave along the line are characterized by the "wave propagation constant" (γ) which is also dependent upon pipeline dimensions, fluid properties and operating conditions. The restrictor valve at the end of the line determines the mean pressure level and offers an impedance to the system known as the "termination impedance" (Z_T).

THE TRANSMISSION LINE EQUATION

The pressure fluctuation at a single frequency (w) measured at a point on the line a distance x from the source is given by:-

$$Px = \frac{Qs\ Zs\ Zo}{Zs + Zo} \cdot \frac{e^{-\gamma x} + \rho_T\ e^{-\gamma(2\ell - x)}}{1 - \rho_s \rho_T\ e^{-2\gamma\ell}} \qquad (2.1)$$

Equation 2.1 is derived in ref. (11), is known as the Transmission Line Equation and represents the pressure standing wave in an hydraulic line. All the variables, with the exception of x and ℓ , are complex and are defined as follows:-

Px - pressure fluctuation at distance x from source of frequency w

x - distance from source to measurement position

ℓ - length of line

Qs - source flow fluctuation at frequency w

Zs - source impedance at frequency w

ρ_s - source reflection coefficient at frequency w

ρ_T - termination reflection coefficient at frequency w

Zo - line impedance at frequency w

γ - wave propagation constant at frequency w

The line dependent parameters (Zo and γ) are determined by the fluid properties, pipe dimensions and operating conditions. The line characteristic impedance is given by:-

$$Zo = \left[\frac{R + \frac{\rho}{A}\ j\ w}{\frac{A}{B}\ j\ w} \right]^{\frac{1}{2}} \qquad (2.2)$$

where: R - pressure drop/unit length/unit flow in the pipe

A - internal area of pipe

B - effective bulk modulus of the fluid

ρ - fluid density

j - complex operator

w - frequency of harmonic component considered.

For hydraulic lines which have good internal surface finishes and are of only a few metres in length, losses due to friction can be considered negligible. In which case, for a lossless line with R = o equation 2.2 becomes:-

$$Zo = \left(\frac{\rho \ B}{A}\right)^{\frac{1}{2}} \qquad (2.3)$$

In this simplified lossless case the line characteristic impedance is now a real value and not a complex one.

The other line dependent parameter, the wave propagation constant is given by:-

$$\gamma = \left[-\frac{\rho}{B} w^2 + \frac{j \ R \ A \ w}{B}\right]^{\frac{1}{2}} \qquad (2.4)$$

where A, B, ρ, R, j and w are as in equation 2.2. With R = o, for a lossless line

$$\gamma = j \ w \left[\frac{\rho}{B}\right]^{\frac{1}{2}} \qquad (2.5)$$

as the local speed of sound in the fluid $C = (B/\rho)^{\frac{1}{2}}$ equation 2.5 becomes:-

$$\gamma = j\frac{w}{c} \qquad (2.6)$$

For normal lossless lines the attenuation of the wave is negligible and the imaginary value of γ determines the change in phase or delay along the line.

The reflection coefficients ρ_s and ρ_T determine the relationship between the incident and reflected waves at the source and termination respectively. These are both functions of the line and end impedances as follows:-

$$\rho_s = \frac{Zs - Zo}{Zs + Zo} \quad (2.7) \quad \text{and} \quad \rho_T = \frac{Z_T - Zo}{Z_T + Zo} \qquad (2.8)$$

As a reflected wave cannot have an amplitude greater than the incident wave which caused it, reflection coefficients have amplitudes which are less than or equal to unity.

Equation 2.1 consists of two distinct groups of terms. The first, $\frac{Qs\ Zs\ Zo}{Zs + Zo}$, does not vary with the line length and represents a mean level of pressure fluctuation which is modified by the remaining group of terms to characterize the standing wave in the line. The mean pressure multiplied by $e^{-\gamma x}$ represents the incident wave at a distance x from the source. Similarly $\rho_T\ e^{-\gamma(2\ell - x)}$ represents the incident wave after it has been reflected at the termination when it is again a distance x from the source. The factor $\left[1 - \rho_s \rho_T\ e^{-2\gamma\ell}\right]^{-1}$ when multiplied by the sum of the incident and reflected waves represents all subsequent reflections which occur at the source and termination.

2.3 - PARAMETERS INFLUENCING PUMP GENERATED PRESSURE RIPPLE

From equation 2.1 it can be seen that the pressure fluctuation at a distance x along the pipe, Px, is dependent upon both pump and system parameters. If it is remembered that each of these parameters is a phasor quantity, having amplitude and phase components at the particular frequency under consideration, it will be realised that the pressure ripple, Px, generated by the pump is also a phasor quantity which is both pump and system dependent.

A detailed computer study of pressure standing waves and their sensitivity to changes in both pump and system parameters has been carried out by Freitas ref. (17), which shows Px to be strongly influenced by some system parameters. As the same pump may be used in many different systems of unknown impedance characteristics it is important to have a means of evaluating the potential fluidborne noise generating capability of a pump independent of the effect of system influences. This type of evaluation would then provide a means of comparing one pump with another on a fluidborne noise basis.

PUMP FLOW RIPPLE AND IMPEDANCE

An evaluation of the fluidborne noise generating potential of a pump, without the influence of system parameters, may be achieved by considering the pressure fluctuations at the pump outlet flange, ref. (18). Taking x = o, then

$$Po = \frac{Qs\ Zs\ Zo}{Zs + Zo}\left(\frac{1 + \rho_T\ e^{-2\gamma l}}{1 - \rho_T \rho_s\ e^{-2\gamma l}}\right) \qquad (2.9)$$

If the entry impedance of the system at the pump flange is given by

$$ZE = Zo\left(\frac{1 + \rho_T\ e^{-2\gamma l}}{1 - \rho_T\ e^{-2\gamma l}}\right)$$

then $$Po = \frac{Qs\ Zs\ ZE}{Zs + ZE}$$

or $$Po = \frac{Qs\ Zs}{1 + \frac{Zs}{ZE}} \qquad (2.10)$$

The pressure fluctuation at the pump flange is now expressed in terms of two pump dependent parameters, the source flow fluctuation Qs and the source impedance Zs, and a single system parameter, the system entry impedance ZE. In this form, it is now easier to see the effect of the various parameters on the fluidborne noise generating potential of the pump. It is still important to note that each parameter is a phasor quantity, evaluated for a single frequency at the pump outlet flange. The full characteristic would be obtained by summing all significant harmonic components of source flow.

Considering equation 2.10 in detail, the following points become apparent:-

a) Po is directly proportional to Qs. Any reduction in Qs results in a direct reduction in Po. This is a prime area for further consideration and is the basis used in this work to reduce pump pressure ripple.

b) If $|Zs| \ll |ZE|$ then $Po = Qs.Zs$. The pressure ripple at the pump flange is only dependent upon pump Qs and Zs and can be used as a means of evaluating pump fluidborne noise generating potential ref. (19). Reducing Zs is a useful means of reducing pressure ripple. This approach has been examined in some detail by the Author and is reported in ref. (15).

c) If $Zs = ZE$ then $Po = \frac{Qs\ Zs}{2}$. In this event the pressure ripple at the pump flange is half the fluidborne noise rating given by b).

d) As $Zs \to -ZE$ then Po becomes large and sensitive to Zs. This shows that if the pump source impedance is or tends to 180 degrees out of phase with the system entry impedance i.e. mismatched, then large amplitudes of Po are possible especially with pumps of compact design having high source impedances.

e) If $|Zs| \gg |ZE|$ then $Po = Qs.ZE$, independent of Zs. In this situation Po is determined largely by the system entry impedance, which in most circumstances is unknown. A reduction in Po is still possible by a reduction in Qs.

f) As $ZE \to 0$ then Po becomes small. Hence the pump parameters have little effect on Po and amplitudes would be small. Unfortunately the practical requirements for low system entry impedance are large pipe diameters, and low mean pressures.

This is because Zo is inversely proportional to the pipe cross-sectional area and valve termination impedance is proportional to the mean pressure level. These requirements are directly opposed to the principal advantage of modern hydraulic systems over other forms of power transmission, that of high power density.

Summarising these points, which include the extremes of pump and system impedance matching and mismatching, it can be seen that under certain circumstances the fluidborne noise generating potential of a pump and system is greatly influenced by the relative amplitudes and phases of Zs and ZE. Unfortunately these conditions are not readily identifiable as Zs and ZE cannot, in general, be quantified. The only parameter which appears under all circumstances and offers the greatest scope for the reduction of system pressure ripple is the source flow fluctuation Qs. This fact is supported by the now popular practice of using pumps with inherently low values of Qs (for example internal gear pumps) when low fluidborne and airborne noise levels are required.

2.4 - THE ADVERSE EFFECTS OF FLUIDBORNE NOISE

2.4.1 - AIRBORNE NOISE RADIATED FROM HYDRAULIC SYSTEMS

Structural excitation and airborne noise can be a serious problem if the hydraulic system is sited in close proximity to a machine operator. It is this sort of problem and the high cost or design inconvenience of isolating the operator from high levels of airborne noise that has promoted research into techniques to reduce airborne noise, at source, by the reduction of fluidborne noise. Much of this work has been directed at two main objectives. Firstly to reduce the fluidborne noise generated by the pump. For example the improving of axial piston pump port plate design ref. (7), changes in the flow fluctuation ref. (20) and impedance of gear pumps ref. (15). Secondly to isolate the system from the fluidborne noise produced by the pump. This may be achieved by the use of fluidborne noise silencers ref. (21) which either absorb the pressure fluctuations or reflect them back toward the pump.

This concept of considering the pump as a generator and the system as an amplifier of both fluidborne and airborne noise is fundamental to the design of quiet hydraulic systems.

McCandlish and Petrusawicz ref. (22) showed that the airborne noise levels produced by isolated pumps of similar capacity and installational volume, but of fundamentally different pumping action, varied by only 10 dBA. For similar running conditions of pressure and speed, the noise levels varied from 65 dBA for the best pump to 75 dBA for the worst. These airborne noise levels are not excessive and are in fact very much below the limit of 90 dBA Leq (8h) recommended by the Health and Safety Commission ref. (23). This limit corresponds to an average noise level of 90 dBA over an eight-hour working day. However different pump designs, internal and external gear, vane and axial piston, can have a considerably different airborne noise generating effect when used in the same system.

This may be explained by considering the fluidborne noise rating for these designs ref. (19). Edge and McCandlish ref. (24) show these ratings as 0.8, 6.5 and 12 bar rms for internal and external gear pumps and axial piston pump respectively, a difference in fluidborne noise generating potential between an internal gear and axial piston unit of 15 times. This probably accounts for the increasing use of internal gear pumps for low noise systems.

In conclusion, the evidence suggests that the airborne noise radiated from the casing of the majority of hydraulic pumps is not a serious problem, especially when compared to the airborne noise radiated by some primemovers (for example diesel engines and electric motors). The serious problem of high airborne noise levels occurs through the high levels of structure and fluidborne noise, which are generated by some types of pump, being propagated and amplified by the hydraulic system itself. The obvious solution to system noise which originates from the pump, is to reduce the structureborne noise transmitted to the system, by isolating the pump with suitable flexible mountings and couplings, and to reduce the level of fluidborne noise in the system.

It is important to remember that pump structural vibration and pressure ripple may both be generated at pumping frequency and harmonics thereof, so that it may be difficult to evaluate the effect of each source independently. Effective system noise reduction will therefore rely on the minimization of both sources and their restriction to the smallest practicable system surface area which can radiate airborne noise.

2.4.2 - THE REDUCTION OF COMPONENT FATIGUE LIFE

High levels of fluidborne noise in hydraulic systems can also cause a significant reduction in the actual or service life of individual hydraulic components. The need to snub Bourdon type pressure gauges is well known and often is a direct result of pump produced pressure ripple. Many component manufacturers have fatigue test procedures which basically involve subjecting the component to a number of cyclic variations in mean pressure which is often of an on-off or square form nature. Typically this may be 10^6 cycles of 0 to 250 bar. The length of time during which the pressure is on or off may be of the order of one second and the rates of pressure rise and decay may be specified.

Such tests have been found necessary to evaluate the likely length of service a particular pump or flexible hose will give in a system with a certain duty cycle.

The effect on fatigue life of additional pressure fluctuations due to the presence of fluidborne noise are rarely considered, but under certain conditions these can be very important. King and O'Neal ref. (25) have predicted the reduction of actual pump life by a factor of 3 due to the additional effects of fluidborne noise. This conclusion was reached by consideration of the fatigue life of a 12 tooth gear pump body in 2024-T3 aluminium alloy. The first harmonic (200 Hz) of fluidborne noise at the pump was predicted for non-resonant and resonant load system line lengths. The values obtained were 1.4 bar and 50 bar zero to peak, respectively. The assumption was made that the pump body was designed to last for 10^6 cycles of zero to 172 bar with an equal on-off duration of 0.1 second. The pressure fluctuation the pump would actually experience under the resonant test condition is the sum of the duty cycle alternating pressure and the fluidborne noise, i.e. an alternating square wave of zero to 86 bar with a period of 0.1 second about a mean of 86 bar and a sinusoidal wave of zero to 50 bar and period 0.005 second. The overall effect of the two cycles on pump body life was evaluated using Minor's Law of Linear-cumulative-damage.

The calculations revealed the pump body would fail after 333,000 design duty cycles, corresponding to 6.67×10^6 cycles of pressure ripple at fundamental pumping frequency. This is only one third of the 10^6 cycle design duty life assumed for the pump without considering the effect of fluidborne noise. In addition, it is important to note that only the largest fundamental component was considered, and other, smaller amplitude harmonic components were ignored. This example shows the possible effect of high levels of fluidborne noise on component fatigue life.

Pressure fluctuations can cause wear of elastomeric seals, such as "O" rings, if they are installed in such a manner that repeated distortion by pressure fluctuations can occur.

The small relative movement between the seal and adjacent metallic surfaces, caused by this distortion, can produce seal wear and in the case of "O" sectioned rings reduce the effective section diameter to the extent where there is insufficient compression to produce an effective seal.

On the inlet or suction side of a pump, pressure fluctuations acting on incorrectly designed "O" sectioned sealing ring installations can cause the seal to oscillate and pump air into the system.

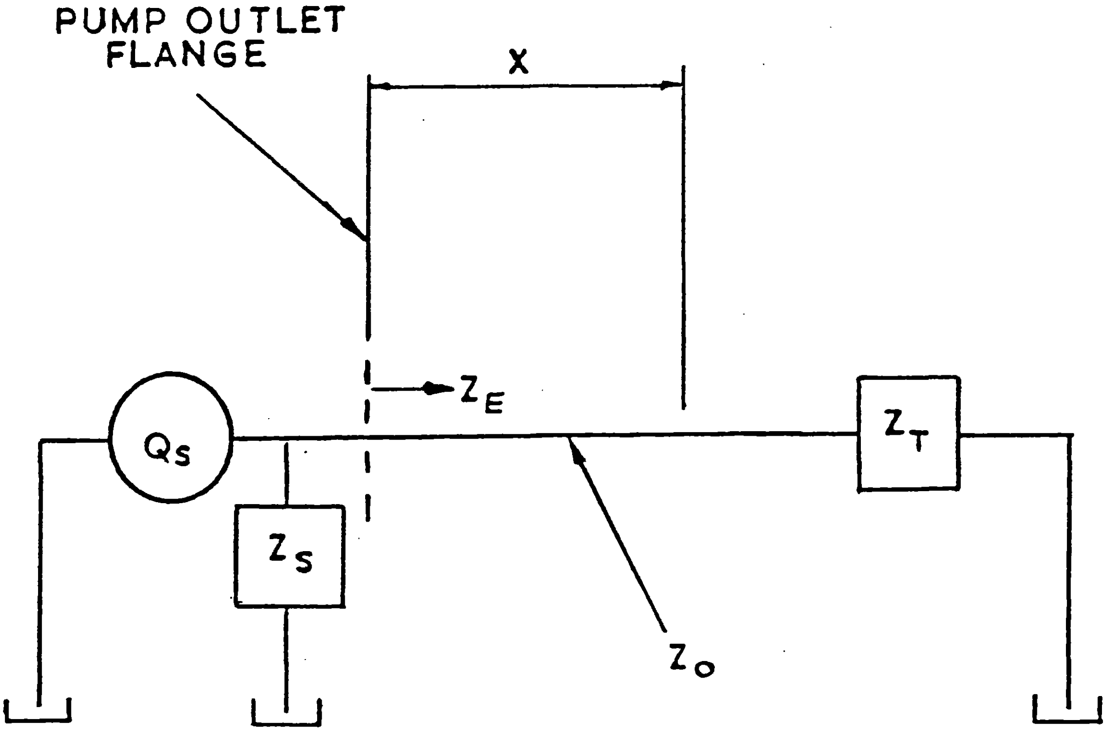


FIG. 2.1 MODEL REPRESENTATION OF A SIMPLE SYSTEM

CHAPTER 3 - THE FLUIDBORNE NOISE GENERATING PARAMETERS OF POSITIVE DISPLACEMENT PUMPS

The fluidborne noise produced by a positive displacement pump was shown in chapter 2, to be affected by only two pump dependent parameters, the source flow fluctuation Qs and the source impedance Zs. It follows therefore that any method of reducing pump generated pressure ripple by a pump modification, must involve the reduction of one or both of these parameters. To achieve this end it is necessary to determine what factors affect the source impedance and the source flow fluctuation of a pump.

3.1 - SOURCE IMPEDANCE

The source impedance of a pump is primarily dependent upon its internal fluid volume. The often irregular form of this volume, which is determined by the pumping mechanism geometry and the porting arrangement, has prevented the accurate prediction of pump impedance.

However, pump impedance spectra may be experimentally determined if the source flow fluctuation Qs and the isolated pressure fluctuation Ps are known.

Then the pump impedance, $Zs = \frac{Ps}{Qs}$, if Ps is the pressure fluctuation resulting from the interaction of Qs with the pump impedance only. This may be achieved by a special measurement technique such as the High Impedance Pipe method ref. (19).

The ability to determine experimentally the impedance of a pump has led to a number of researchers to propose theoretical models for the impedance characteristics of pumps. Edge ref. (18) compares several proposals, including resistive, capacitive and capacitive plus inductive models, with experimental results and draws attention to their inability to represent known pump impedance spectra.

A capacitive model representing the compliance of the oil in the pump casing appears to be viable at low frequencies but is completely inadequate for high frequencies. Edge goes on to propose a distributed parameter model which assumes the pump internal, high pressure, volume may be considered as a pipe of constant cross-sectional area A and length L. The leakage losses are assumed to be lumped at the end of this pipe and the leakage impedance is assumed to be much greater than the characteristic impedance of the pipe Zp. Ignoring pipe friction effects pump impedance is given by:-

$$Zs = \frac{Zp}{\tan\left(\frac{2\pi fL}{C}\right)} \qquad \underline{/-90^{\circ}\text{sgn } \tan\left(\frac{2\pi fL}{C}\right)}$$

where $Zp = \frac{B}{AC}$, the characteristic impedance of the "pipe"

This model gives a good representation of the form of pump impedance spectra but requires an effective volume approximately twice that of the actual pump.

The author ref. (15) found that with empirical adjustments to the model precise agreement with experimental results, for an external gear pump, could be achieved (see fig. 3.1).

Although the necessary adjustments could not be determined from examination of the pump internal volume, it was found that increasing this volume caused a reduction in pump impedance which under certain conditions resulted in a reduction in pressure ripple.

The potential of modifying pump impedance as a means of reducing pump pressure ripple is however limited. Significantly greater potential exists in the modification of the other pump parameter, source flow fluctuation.

3.2 - SOURCE FLOW FLUCTUATION

The source flow fluctuation Qs, previously referred to in section 2.2, when related to a positive displacement pump, is directly dependent upon the fundamental geometry of the pumping mechanism. Different pumping actions can produce significantly different variations in delivery. Screw and piston type machines are representative of the extremes of flow variation for the range of positive displacement pumps generally used in hydraulic systems. They also demonstrate two basic effects which are the cause of pump flow ripple.

Firstly, the methods of fluid transfer, from inlet to outlet, are inherently dissimilar. A screw pump has a smooth continuous action which produces a very small or insignificant amount of variation in mean flow. A piston pump produces a cyclic variation caused by the summation of the overlapping deliveries of a relatively small number of pistons, usually seven or nine.

Secondly, the screw pump raises the pressure of the fluid being transferred gradually and continuously over the whole delivery process which makes it more prone to constant leakage and less prone to cyclic or dynamic leakage. The axial piston pump is very susceptible to back flow from the system into the delivering cylinder because of the timing of the piston compression stroke and the opening of the delivering cylinder to the outlet port. This back flow effect may be many times greater than the flow ripple effect caused by the summation of the individual piston deliveries, ref. (26), complicates the problem by varying with speed of rotation, operating pressure and swash plate position. Thus the flow ripple produced by a piston pump is greatly affected by port plate timing and may vary considerably from one manufacturer's design to another.

A measurement technique has been established, ref. (27), which makes it possible to analyse the flow fluctuation of a pump in terms of amplitude and phase of the fundamental pumping frequency and its harmonics. Further details of this technique will be given in chapter 7. Reconstruction of these harmonic components then enables the source flow fluctuation to be presented in graphical form. Fig. 3.3 shows the evaluated instantaneous delivery (the source flow component combined with the mean flow component) for two different external gear pumps.

To enable a better understanding of the source flow generation of a pump it is necessary to analyse in detail the pumping geometry of its design and determine the parameters which influence the amplitude and frequency of the source flow fluctuation it produces.

At this point it is pertinent to mention that the methods of measurement and analysis used in this work are generally applicable to all types of positive displacement pump. However the purpose of this work was to find a means of modifying the source flow of a particular type of pump with the view that the principles of the solution may be applicable to other pump types. In this case the pump design in question was of the external involute gear type, details of which are shown in appendix A.3.1. The reason for this was to provide continuity with previous work on source impedance ref. (15) and maintain the co-operative effort of the sponsoring body.

To carry out this work, the derivation of a suitable theoretical model for an external involute gear pump was undertaken. In addition, in order that flow ripple comparisons could be made with equivalent displacement internal gear pumps, internal involute gearing was also investigated.

3.3 - THE SOURCE FLOW AND VOLUME VARIATIONS PRODUCED BY GEAR PUMPS

The output flow of a pump may be defined by the following terms:-

i) The instantaneous output flow, Qi - the output flow rate at any instant in time or position of driveshaft.

ii) The average or mean output flow, Qm - the result of averaging the instantaneous flow rate over a complete number of pumping cycles.

iii) The fluctuation in output flow, Qs - the variation in instantaneous output flow about the mean output flow, over a period of time, or equivalent rotational movement of the driveshaft, equal to the fundamental pumping period.

$$\text{hence} \quad Qs = Qi - Qm$$

As Qs represents the variation in output flow rate it is inherently time dependent and hence will vary, for a given positive displacement pumping geometry, with the speed of rotation of the pumping elements. It is therefore convenient to express Qs in a manner which is independent of the speed of rotation of the pump and which may be directly related to the movement of the pumping elements. This may be achieved by summing, over one complete pumping cycle, the instantaneous values of Qs. The result is the instantaneous variation in output volume Vs. Where:-

$$Vs = \int_0^t Qs \, dt$$

where t = period of pumping frequency.

This method of interpreting Qs has proved useful in understanding and relating the volume changes produced by pumping mechanisms and is fundamental to the conception of an active pressure ripple reduction device.

3.3.1 - EXTERNAL INVOLUTE GEAR PUMP ANALYSIS

A theoretical source flow model for external involute gearing will now be developed by a geometrical analysis of the areas bounded by two meshing gear teeth. The result is a simple expression relating changes in pump output with the angle of shaft rotation for an ideal pumping geometry and an ideal fluid. The ability of the model to accurately predict the flow fluctuations of a real pump working on a real fluid therefore depends on the validity of the following assumptions:-

a) The involute gear profiles are unmodified and have a contact ratio of unity.

b) There is no "carry-over" volume from outlet to inlet.

c) There are ideal "fluid-trapping" relief grooves.

d) The gears are constrained, by bearings to run smoothly with a constant distance between their centres, equal to the pitch circle diameter.

e) There are no leakage or compressibility losses.

Consider the driver and driven gears in mesh at the pitch point as shown in fig. (a) below where the notation refers to the following parameters:-

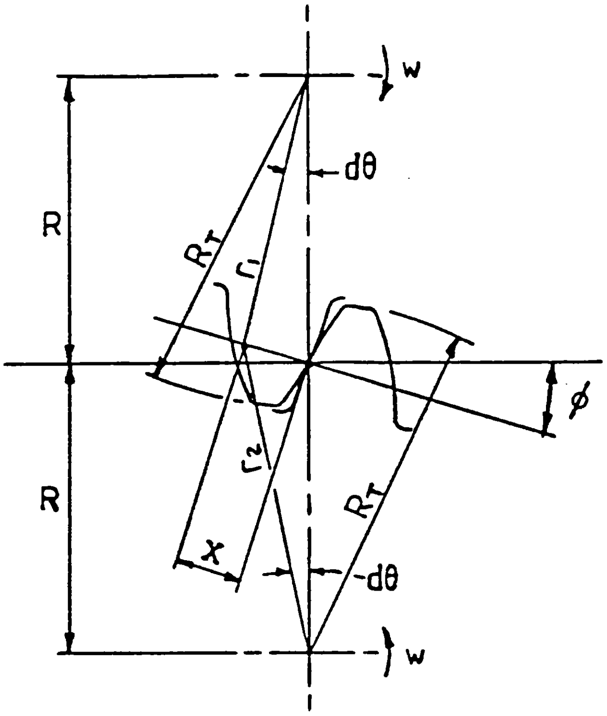


FIG.(a)

where:-

suffix 1 - driver gear

suffix 2 - driven gear

Z - number of teeth

$d\theta$ - angular increment

R - pitch circle radius

R_T - tip circle radius

r - distance from centre to point of mesh

x - distance along line of contact from pitch point to point of mesh

ϕ - pressure angle

b - gear width

w - rotational velocity

m - gear module

For the normal configuration, where driver and driven gears are identical, from ref. (28), a small change in driver/shaft angle $d\theta$ produces the following change in output volume:-

$$dq = b\left(\tfrac{1}{2} R_T^2\, d\theta - \tfrac{1}{2} r_1^2\, d\theta + \tfrac{1}{2} R_T^2\, d\theta - \tfrac{1}{2} r_2^2\, d\theta\right)$$

$$dq = \frac{b}{2}\, d\theta \;(2 R_T^2 - r_1^2 - r_2^2)$$

by Pythagoras from fig. (a) above

$$r_1^2 = (R - x \sin\phi)^2 + x^2 \cos^2\phi$$

and $r_2^2 = (R + x \sin\phi)^2 + x^2 \cos^2\phi$

$$r_1^2 + r_2^2 = 2(R^2 + x^2)$$

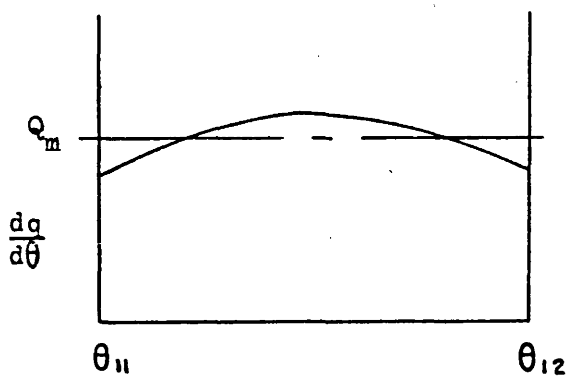
hence $\dfrac{dq}{d\theta} = b\,(R^2 - R^2 - x^2)$

from involute geometry $x = R \cos\phi \,.\, \theta$. Substituting for x the instantaneous output variation is:-

$$Qi = \frac{dq}{d\theta} = b\,(R_T^2 - R^2 - R^2 \cos^2\phi \,.\, \theta^2)$$

this is valid in the limit $-\dfrac{\pi}{Z} \leqslant \theta < \dfrac{\pi}{Z}$

The mean output volume/rad. for one tooth cycle is the area under the $dq/d\theta$ curve divided by the angular rotation of the gear between θ_{11} and θ_{12}



$$Qm = \frac{1}{\theta_{12} - \theta_{11}} \int_{-\frac{\pi}{Z}}^{\frac{\pi}{Z}} \frac{dq}{d\theta} \cdot d\theta$$

$$Qm = \frac{b}{\frac{\pi}{Z} - \left(-\frac{\pi}{Z}\right)} \int_{-\frac{\pi}{Z}}^{\frac{\pi}{Z}} (R_T^2 - R^2 - R^2 . \cos^2\phi . \theta^2)\, d\theta$$

integrating w.r.t θ , substituting limits and collecting terms gives the mean output as:-

$$Qm = b\left(R_T^2 - R^2 - R^2 \cos^2\phi \frac{\pi^2}{3 Z^2}\right)$$

The variation in output is now:-

$$Qs = Qi - Qm$$

$$= b(R_T^2 - R^2 - R^2 . \cos^2\phi \; . \theta^2) - b\left(R_T^2 - R^2 - R^2 . \cos^2\phi . \frac{\pi^2}{3 Z^2}\right)$$

$$Qs = b\,R^2 \cos^2\phi \left(\frac{\pi^2}{3\,Z^2} - \theta^2\right)$$

This shows Qs to be of parabolic form where Qs = 0 when

$$\theta = \pm \frac{\pi}{3\,Z}$$

At this point Qs is the change in output volume/radian. To convert Qs to the more usual volume/sec. units it is necessary to multiply by the gear rotational velocity w. Also the pitch circle radius, R, may be substituted in terms of gear module, m, and number of teeth. $R = mZ/2$

then $$Qs = wb\,\frac{m^2 Z^2}{4} \cos^2\phi \left(\frac{\pi^2}{3\,Z^2} - \theta^2\right)$$

which reduces to the form quoted by Duke and Dransfield ref. (29):-

$$Qs = \frac{wb\,m^2 \cos^2\phi}{12} \left(\pi^2 - 3\,Z^2\theta^2\right) \tag{3.1}$$

This is valid in the limit $-\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$

Qs shows how the gear outlet flow rate changes with rotation of the driver gear (or driveshaft). Integrating Qs w.r.t time gives the variation in outlet volume with gear rotational angle

hence $$Vs = \int Qs.dt$$

$$Vs = \int \frac{wb\ m^2\ \cos^2\phi}{12}\ (\pi^2 - 3\ Z^2 .\ \theta^2)\ dt$$

as $w = \frac{d\theta}{dt}$

$$Vs = \int \frac{b\ m^2\ \cos^2\phi}{12}\ (\pi^2 - 3\ Z^2 .\ \theta^2)\ d\theta$$

integrating and substituting values gives

$$Vs = \frac{b\ m^2\ \cos^2\phi}{12}\ (\pi^2\theta - Z^2 .\ \theta^3) \qquad (3.2)$$

which is valid in the limit $-\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$

GRAPHICAL REPRESENTATION

Using equations 3.1 and 3.2, the flow and volume variations for an external gear pump over one tooth cycle have been evaluated. The results, which relate to pump A (appendix 3) at 1500 rev/min. are shown in fig. 3.2. The flow variation, is shown superimposed on the mean flow as a function of time. The fundamental time period of 5 ms for an eight tooth unit at 1500 rev/min. is equivalent to 45 degrees of rotation of the driveshaft.

It is important to note that because the flow variation is the rate of volume change, doubling the rotational speed of the unit would not only halve the time for one cycle to 2.5 ms but would also double the peak to peak variation.

The equivalent volume variation shown in fig. 3.2 is plotted against the angle of driveshaft rotation and the peak-to-peak amplitude is now independent of time. Comparison of the two curves shows that the volume variation is equal to the area between the curves representing the mean and instantaneous pump deliveries.

Fig. 3.3 shows the degree of agreement between predicted flow fluctuations and evaluated results for two external gear pumps, A and E (A.3.1) at a rotational speed of 1500 rev/min. and various mean pressures. A detailed discussion of the differences between predicted and evaluated results is given in chapter 7.

HARMONIC REPRESENTATION

The flow and volume variations, due to external involute gear geometry, may also be represented by a Fourier series consisting of sinusoidal components at integer multiples of fundamental pumping frequency.

The amplitude of the Fourier components of external gear pump flow ripple may be determined by the following equation from ref. (30):-

$$Af = -\frac{4b\ Rg^2\ w\ (-1)^M}{M^2\ Z^3} \tag{3.3}$$

From this it can be shown, by integration in the time domain that the amplitude of the Fourier components of volume ripple are given by:-

$$Av = -\frac{4b\ Rg^2\ (-1)^M}{M^3\ Z^3} \tag{3.4}$$

where Af = peak amplitude of harmonic component of flow ripple

Av = peak amplitude of harmonic component of volume ripple

M = number of harmonic

b = gear width

Rg = base circle radius

w = fundamental angular frequency

Z = number of teeth on driver

3.3.2 - INTERNAL INVOLUTE GEAR PUMP ANALYSIS

Using a similar approach to that used in section 3.3.1, a theoretical source flow model may also be developed for internal involute gearing as follows:-

Consider an internal gear (wheel) in mesh with an external gear (pinion) as shown in fig. (b) below, where the notation refers to the following parameters:-

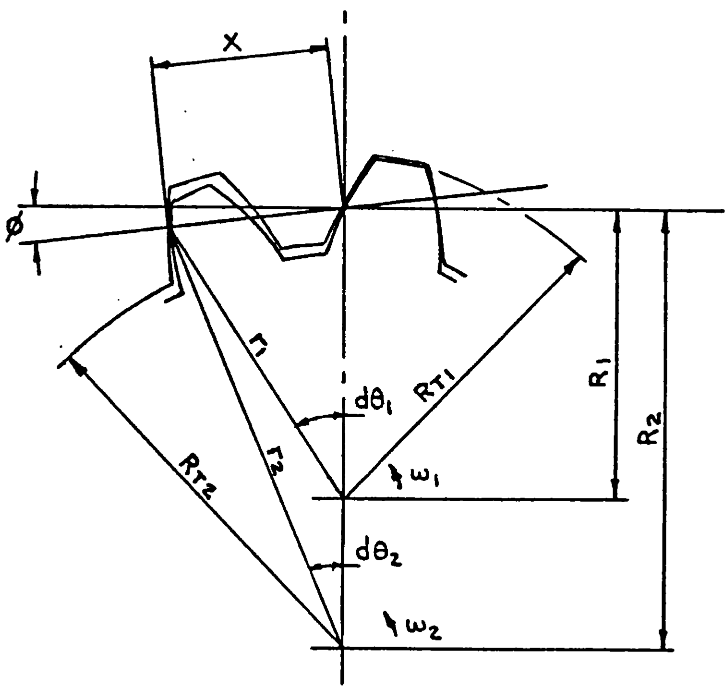


FIG.(b)

suffix 1 - pinion

suffix 2 - wheel

Z - number of teeth, $n = Z_1/Z_2$

$d\theta$ - angular increment

R - pitch circle radius

m - gear module

R - tip circle radius

r - distance from centre to point of mesh

x - distance along line of contact from pitch point to point of mesh

ϕ - pressure angle

b - gear width

w - rotational velocity

Practical internal gearing configurations require that Z_2 must be greater than Z_1 and $Z_2 = Z_1 + 3$ is considered a minimum.

For a small change in pinion shaft angle $d\theta_1$, the change in output volume is given by:-

$$dq = b\left(\tfrac{1}{2} R_{T_1}^2 . d\theta_1 - \tfrac{1}{2} r_1^2 . d\theta_1 + \tfrac{1}{2} r_2^2 . d\theta_2 - \tfrac{1}{2} R_{T_2}^2 \, d\theta_2\right)$$

as $n = Z_1/Z_2$ then $d\theta_2 = n \, d\theta_1$ and:-

$$dq = \frac{b}{2} d\theta_1 \left(R_{T_1}^2 - n R_{T_2}^2 - r_1^2 + n r_2^2\right)$$

from fig. (b) by Pythagoras

$$r_1^2 = (R_1 - x \sin\phi)^2 + (x \cos\phi)^2$$

and $r_2^2 = (R_2 - x \sin\phi)^2 + (x \cos\phi)^2$

substituting for $r_1^{\ 2}$ and $r_2^{\ 2}$, expanding bracketed terms, using $\cos^2\phi + \sin^2\phi = 1$ and $R_1 = n\ R_2$, dq can be expressed as follows:-

$$dq = \frac{b}{2}\, d\theta_1 \left(R_{T_1}^2 - n\ R_{T_2}^2 - R_1^{\ 2} + n\ R_2^{\ 2} + (n-1)\ x^2 \right)$$

from involute geometry $x = R_1 \cos\phi \cdot \theta_1$ and the instantaneous output volume with angle of rotation Qi is given by:-

$$Qi = \frac{dq}{d\theta_1} = \frac{b}{2} \left[R_{T_1}^2 - R_1^{\ 2} + n\ (R_2^{\ 2} - R_{T_2}^2\ ) + (n-1)\ (R_1^{\ 2} \cos^2\phi\ )\, \theta_1^{\ 2} \right]$$

which is valid in the limit $-\frac{\pi}{Z_1} \leqslant \theta_1 \leqslant \frac{\pi}{Z_1}$

The mean output volume/rad. for one tooth cycle is the area under the $dq/d\theta_1$ curve divided by the angular rotation of the pinion between θ_{11} and θ_{12}

$$Qm = \frac{1}{\theta_{12} - \theta_{11}} \cdot \int_{-\frac{\pi}{Z_1}}^{\frac{\pi}{Z_1}} \frac{dq}{d\theta_1} \cdot d\theta_1$$

where $\theta_{11} = -\pi/Z_1$ and $\theta_{12} = \pi/Z_1$

integrating w.r.t. θ_1, substituting limits and collecting terms gives the mean output as:-

$$Qm = \frac{b}{2} \left[R_{T_1}^2 - R_1^{\ 2} + n\ (R_2^{\ 2} - R_{T_2}^2) + (n-1)(R_1^{\ 2} \cos^2\phi\ ) \frac{\pi^2}{3\ Z_1^{\ 2}} \right]$$

the variation in output is now given by:-

$$Qs = Qi - Qm$$

after substitution and re-arranging.

$$Qs = \frac{b}{2}\left[(1-n)\,(R_1^2 \cos^2\phi)\left(\frac{\pi^2}{3Z_1^2} - \theta_1^2\right)\right]$$

This is again of parabolic form where Qs = O when

$\theta_1 = \pm \frac{\pi}{\sqrt{3}\,Z_1}$ similar to that obtained for an external gear pump. At this point Qs is the change in output volume/radian. To convert Qs to the more usual volume/sec. units it is necessary to multiply by the pinion gear rotational velocity w_1. Also the pitch circle radius, R_1, may be substituted in terms of gear module m_1, and number of teeth $R_1 = m_1 Z_1/2$

then $$Qs = \frac{b.w_1\, m_1^2 \cos^2\phi}{12} \cdot \frac{(1-n)}{2}\,(\pi^2 - 3\,Z_1^2\,\theta_1^2) \qquad (3.5)$$

valid in the limit $-\frac{\pi}{Z_1} \leqslant \theta \leqslant \frac{\pi}{Z_1}$

Comparison of equations 3.5 and 3.1 shows that the flow fluctuation of an internal involute gear pump is the same as for an external gear pump multiplied by the factor
$(1 - Z_1/Z_2)/2$

Qs shows how the gear outlet flow rate changes with rotation of the pinion gear (or driveshaft). Integrating Qs w.r.t. time gives the variation in outlet volume with gear rotational angle.

hence

$$Vs = \int Qs\ dt$$

$$V_s = \int \frac{b\, w_1\, m_1^{\,2} \cos^2\phi}{12} \cdot \left(\frac{1-n}{2}\right)(\pi^2 - 3Z_1^{\,2}\,\theta_1^{\,2})\, dt$$

as $w_1 = \frac{d\theta_1}{dt}$

$$V_s = \int \frac{b\, m_1^{\,2} \cos^2\phi}{12} \left(\frac{1-n}{2}\right)(\pi^2 - 3\, Z_1^{\,2}\, \theta_1^{\,2})\, d\theta_1$$

integrating and substituting values gives

$$V_s = \frac{b\, m_1^{\,2} \cos^2\phi}{12} \left(\frac{1-n}{2}\right)(\pi^2\theta_1 - Z_1^{\,2}\, \theta_1^{\,3}) \qquad (3.6)$$

which is valid in the limit $-\frac{\pi}{Z_1} \leqslant \theta_1 \leqslant \frac{\pi}{Z_1}$

Comparison of equations 3.6 and 3.2 show that internal and external gear pump volume variations are related in the same manner as their flow variations, by the factor $(1 - Z_1/Z_2)/2$.

GRAPHICAL PRESENTATION

The theoretical flow and volume variations for an internal gear pump (pump G in A.3.1) are shown in fig. 3.4. Again the figure relates to one tooth cycle at 1500 rev/min. This unit has a mean delivery of o.437 l/s and 13 teeth on the pinion gear. The axes of the graphs are the same as in fig. 3.2, for the external gear pump of similar delivery (0.45 l/s), so that the significant differences in flow and volume variations due to the pumping mechanism geometries of the two units can clearly be seen.

Over one pumping cycle the external gear pump produces a peak to peak flow variation of 25% of the mean, compared with 3% for the internal unit. This considerable difference however is not entirely representative of the likely pressure ripple each would produce because of the difference in fundamental pumping frequency. The pressure ripple generating potential of each unit may be compared directly, at the same fundamental frequency by comparing volume variations. In this case, if the flow ripple from both units were discharged into systems having the same effective impedance, then at the same fundamental pumping frequencies the external gear pump would produce a pressure ripple 15 times greater than the internal unit.

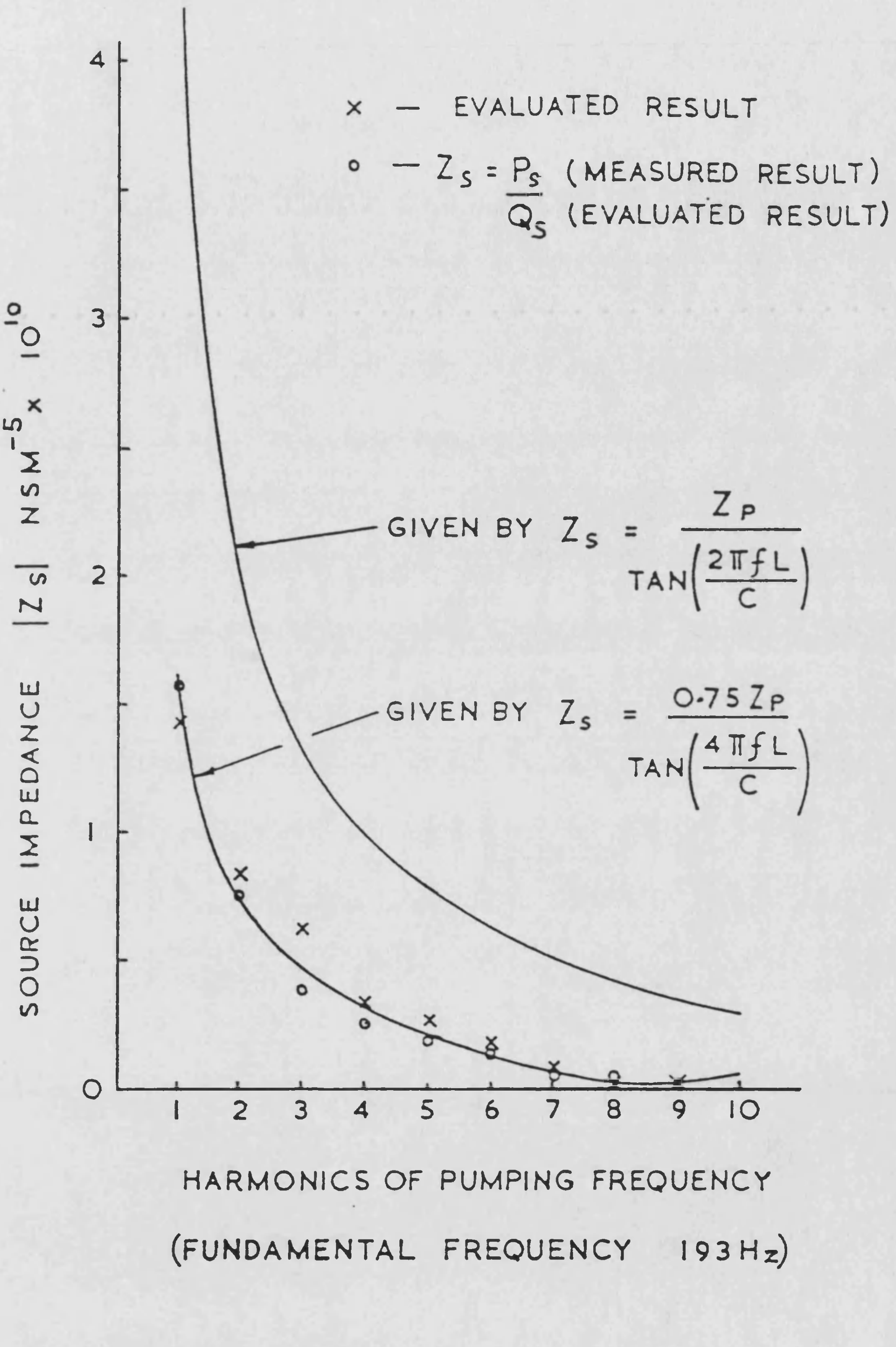


FIG. 3.1 COMPARISON OF EXPERIMENTAL AND PREDICTED SOURCE IMPEDANCE RESULTS FOR AN EXTERNAL GEAR PUMP

REPRODUCED FROM REF. (15)

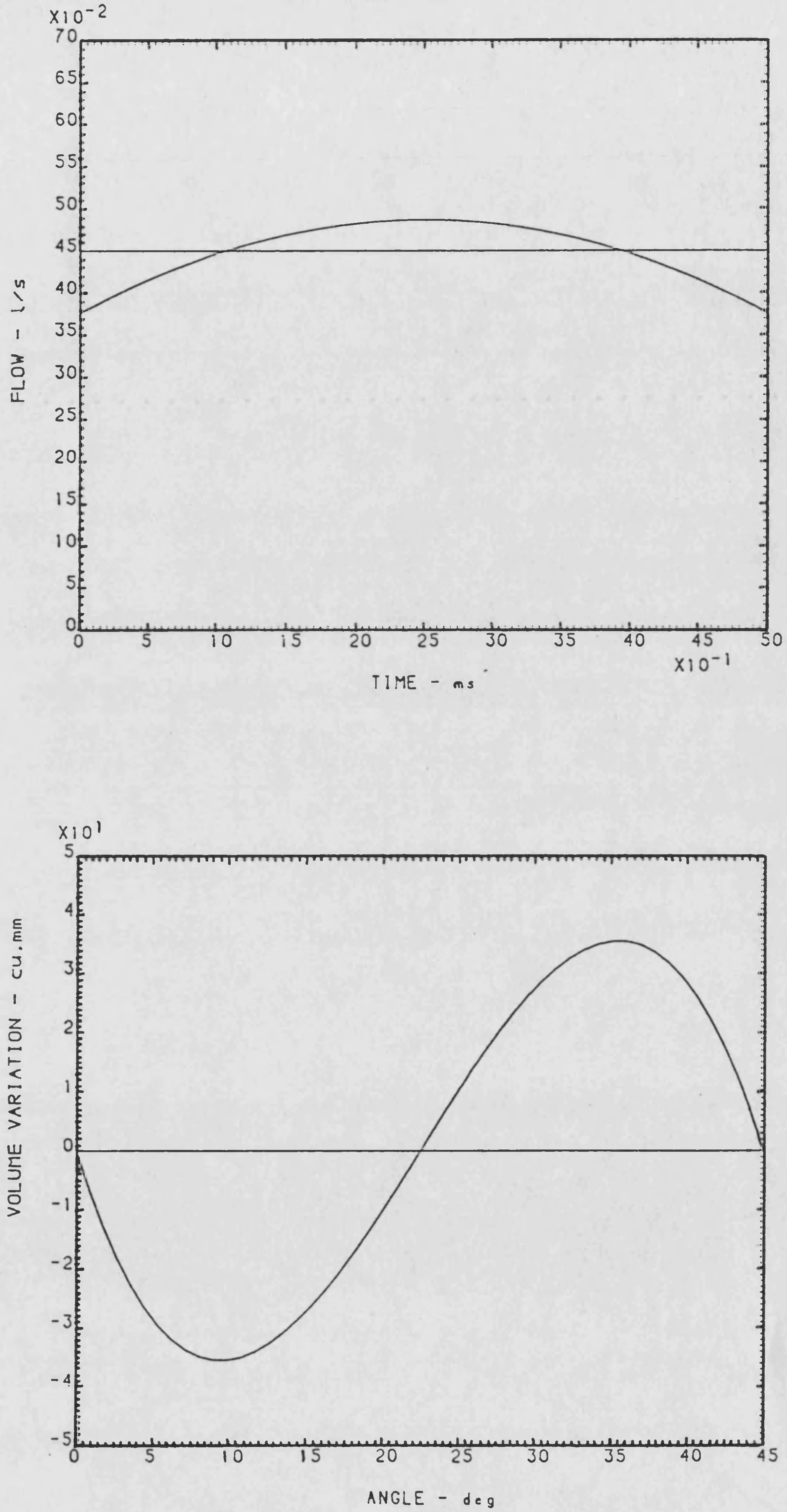


FIG. 3.2 THEORETICAL EXTERNAL GEAR PUMP FLOW AND VOLUME VARIATIONS

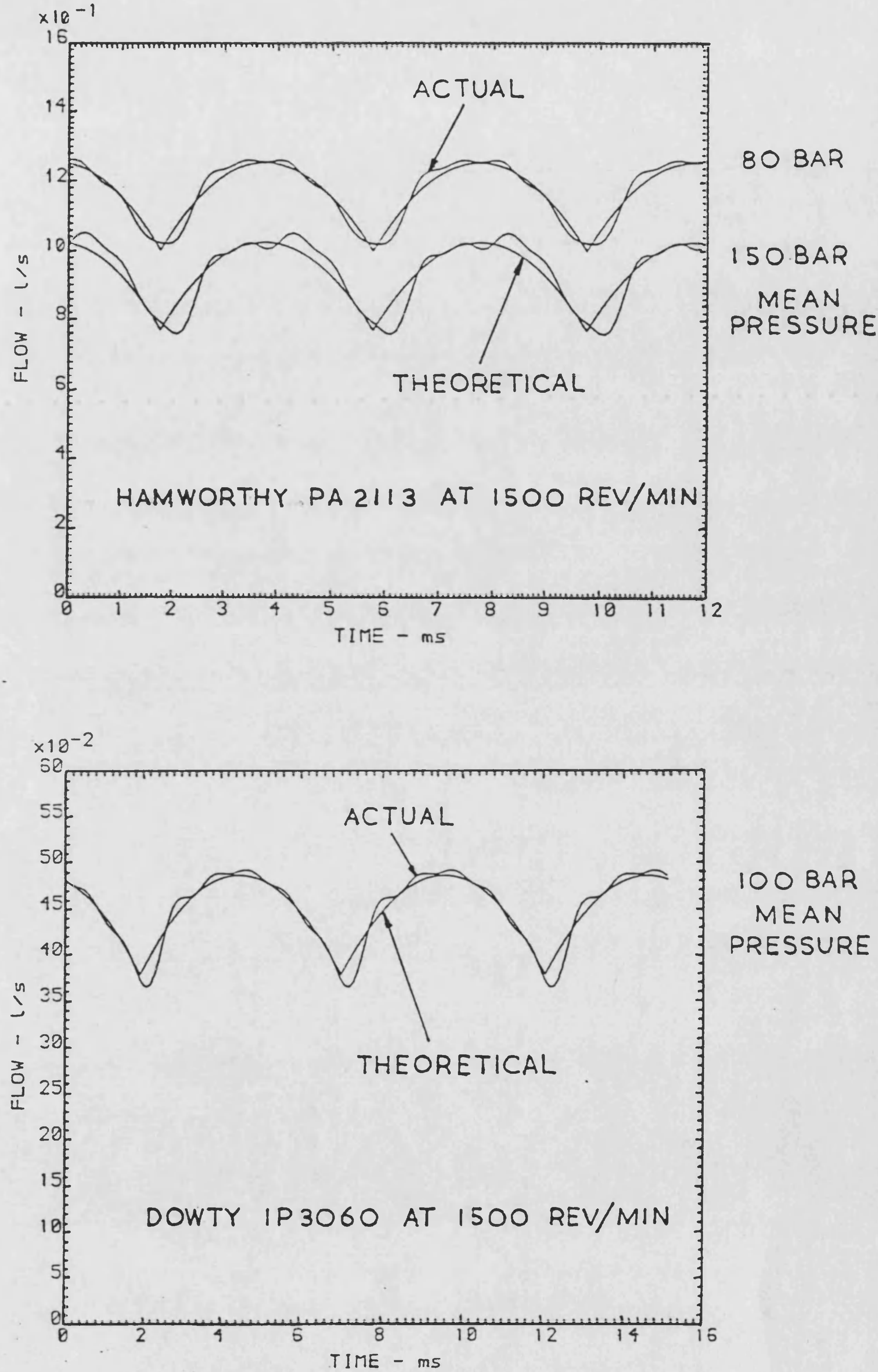


FIG. 3.3 COMPARISON OF THEORETICAL AND ACTUAL EXTERNAL GEAR PUMP FLOW VARIATIONS

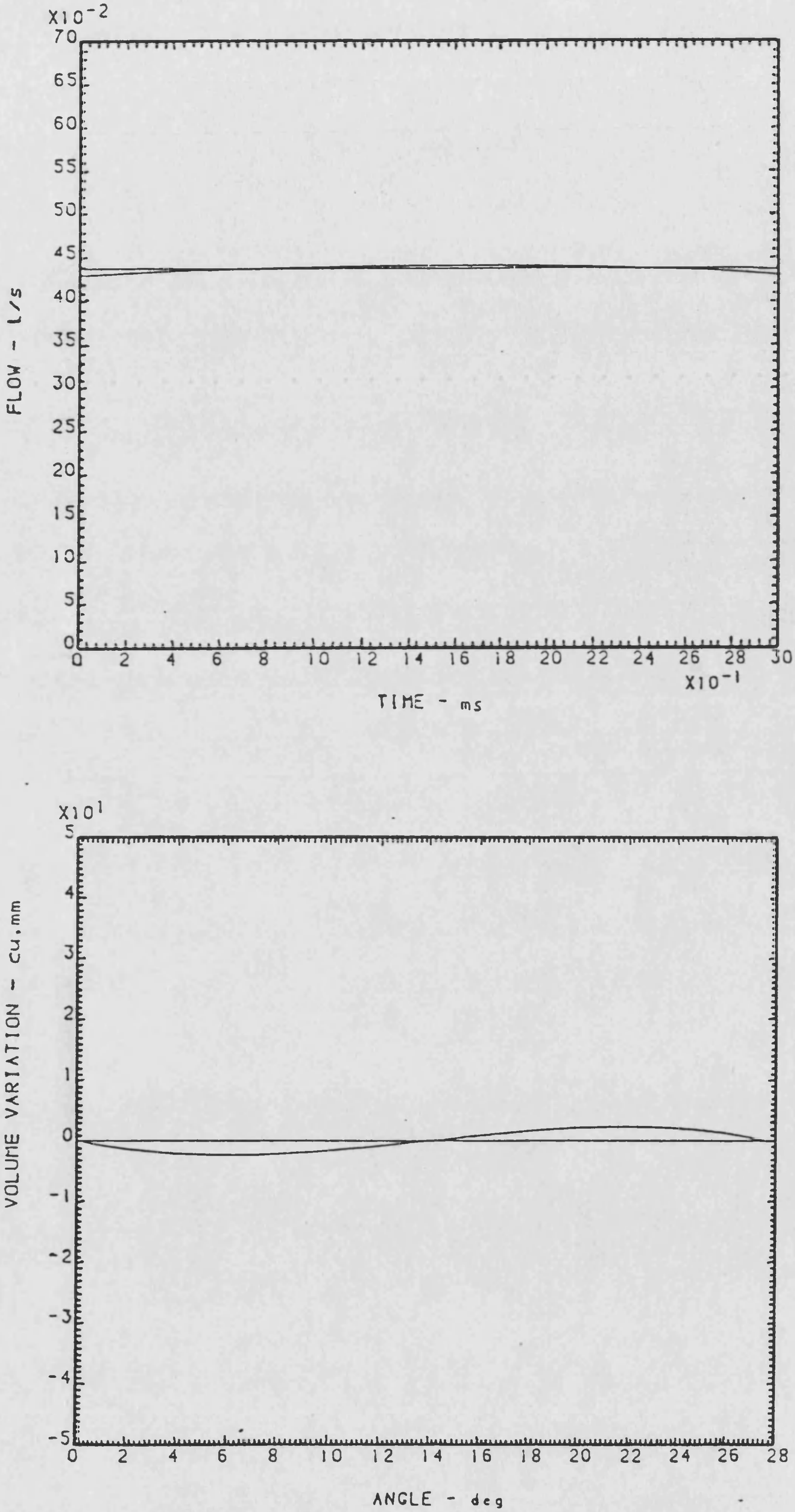


FIG.3.4 THEORETICAL INTERNAL GEAR PUMP FLOW AND VOLUME VARIATIONS

CHAPTER 4 - ACTIVE METHODS OF FLUIDBORNE NOISE REDUCTION

The principal methods of reducing structureborne,fluidborne and airborne noise levels may be divided into two main categories; passive and active. Traditionally passive methods have had a wider field of application, but research into techniques for active noise suppression has revealed a potential for considerable improvement in performance over passive methods.

It is the application of these techniques to fluidborne noise that forms the basis of this work. An idea of the extent of this potential may be given by a brief description of the two aforementioned categories.

Passive methods utilize particular inherent properties of materials and geometric shapes to effect changes in energy state for example, some silencers use soft or porous materials ref. (31) to dissipate or absorb dynamic pressure energy as heat energy. This principle is very effective for gaseous media, but suitable materials with the required acoustical and mechanical properties have yet to be developed for high pressure hydraulic systems. The working media itself may be used to effect pressure ripple reductions as in air receivers or hydraulic accumulators. In general large volumes of the working media offer a low impedance to flow fluctuations and hence produce low levels of pressure ripple.

The other type of passive silencer, often referred to as reactive or reflective, relies on the ability of a discontinuity in the shape of the noise path to partially reflect pressure waves. Careful design can optimise the reflective effect of oil volumes interconnected with tubes over a particular frequency range. However there are practical limitations. A large size device is required for low frequencies and high strength is required for high mean pressures. Some success in reducing airborne noise levels has been achieved with these devices by reducing the fluid-borne noise entering a hydraulic system. A fluidborne noise silencer between the pump and system reflects a proportion of the fluidborne noise back towards the pump and reduces the effective surface area over which fluidborne noise can excite structureborne and therefore airborne noise.

Active methods of "noise" reduction involve the application of a second, often external, source of anti-phase noise which when combined with the original noise source produces destructive interference - a cancelling or neutralising effect. This principle has been used by Chaplin (ref.32) to quieten the exhaust noise emitted by a diesel engine. A microphone monitors the ABN from the exhaust and feeds a proportional signal to a microprocessor which synchronises the signal with engine rotation, and controls a wave form generator to produce an inverse signal. A loudspeaker converts this inverse signal to a pressure fluctuation which is directed at the exhaust outlet. The anti-noise combines with the exhaust noise resulting in a 20 dB reduction in airborne noise level in the frequency range 50-150 Hz.

Another application where anti-noise has been successful is in the reduction of airborne levels in military aircraft. Here the inverse signal is again created electronically but the spacial field of application of the anti-noise is limited to the crewman's helmet. The acoustic power levels are therefore low and easily produced electrically. When applying the anti-noise principle to fluidborne noise in hydraulic systems a major difficulty is posed by the need to create the inverse pressure variation about a mean pressure which may vary between zero and 300 bar. Another difficulty is ensuring a consistent phase relationship between the noise and anti-noise such that the interference is destructive rather than constructive, when the amplitude and frequency of the noise source may be varying considerably.

A number of researchers have developed different active methods of reducing fluidborne noise in hydraulic systems, and in particular the pressure ripple produced by positive displacement pumps and motors. Examination of these methods shows the effectiveness of flow ripple reduction and the difficulties involved in its achievement.

4.1 - SERVO VALVE CONTROL OF NEUTRALISING-PRESSURE-PULSE

As pressure variations in hydraulic systems are caused directly by volume changes within the fluid, it has been reasoned that if fluid is removed at the point where a pressure-peak exists and re-introduced at a pressure-trough, then a considerable reduction in the pressure variation could be achieved.

A practical embodiment of such a method requires some means of dynamic pressure detection; a means of reference, control and generation of an inverse signal; and a means of application of the signal as a fluid input to or output from the hydraulic system. Pressure ripple may be easily represented by the electrical signal produced by a suitable transducer. After conditioning, this signal may then be used to control an electro-hydraulic servo valve to effect fluid input to or output from the hydraulic system.

Active fluidborne noise suppression, using this technique, has been investigated by Rébel ref. (33). In his paper he describes possible hydraulic circuits, fig. 4.1 for the extraction and insertion of fluid using a servo valve. Fig. 4.1a shows a circuit for the extraction or bleeding-off of fluid from the hydraulic system. As this is limited to reducing those flow variations which are above the mean flow, and because of the reduction in system volumetric efficiency, it is also desirable to be able to insert flow into the system. This may be achieved by using an additional hydraulic pump as shown in fig. 4.1b. This circuit utilizes the closed centre servo valve so that fluid may be subtracted from or added to the system by electrical actuation of the valve in either direction.

The arrangements, used by Rébel, for pressure measurement and servo valve control are shown in fig. 4.2. Open loop control appears to permit single sinusoidal control of the valve. The closed loop arrangement allowed the measured pressure signal, containing numerous harmonics, to be applied to the valve.

Based on tests carried out on a radial 3 piston pump of

15 cm^3/rev. displacement, running at a maximum speed of 2000 rev/min. and 75 bar mean pressure, Rébel makes the following conclusions:-

a) A reduction in pump fluidborne noise level of 9 dB can be achieved by active suppression at the fundamental pumping frequency. This may be increased to 17 dB when the servo valve is regulated by the measured pressure signal, which allows the fundamental and higher harmonics to be suppressed.

b) There are difficulties in achieving suppression of pressure wave-forms, other than sinusoidal at frequencies above 50 Hz. The cancellation of higher frequency components is limited by the frequency response of the servo valve.

c) Improvements may be achieved by bleeding-off at "pressure peaks" only, but with the consequent loss of system volumetric efficiency. Bleeding-off and adding fluid restores system volumetric efficiency at the cost of an additional hydraulic pump to provide an independent servo valve supply.

d) Active fluidborne noise suppression would be applied most practicably in conjunction with traditional passive methods which are more suited to the economic suppression of high frequency pressure fluctuations.

In addition to these conclusions it must also be pointed out that there are problems other than those related to frequency and phase. Although pressure fluctuations are not necessarily dependent upon mean pressure levels, the delivery of a servo valve, for a given input signal, will vary with the mean pressure difference between its supply and delivery ports. To accurately control the input and output of fluid to and from a system, the servo valve input signal must have mean system pressure compensation. The addition of a servo supply pump to the system will inevitably cause problems because of its own pressure ripple generating characteristics affecting the servo valve operation or interacting with the pressure fluctuations from the pump which are to be suppressed.

Assuming that the servo valve method could be made effective in suppression of pump generated fluidborne noise, the likely cost of the necessary components would probably be many times the cost of a typical positive displacement pump used in the mobile hydraulic industry. Although this additional expenditure may be justifiable in particular installations, in general it is not consistant with low cost hydraulic systems. It is therefore desirable to achieve a similar end by alternative, less expensive means.

4.2 - ANTI-FLOW RIPPLE GROOVES

The use of carefully designed flow relieving grooves to reduce the occurance of dynamic pressure pulsation within positive displacement hydraulic pumps is well known. It is a design feature, in one form or another, of the majority of high performance piston, vane and gear machines.

4.2.1 - PRESSURE RELIEVING GROOVES

Consider a positive displacement pump of the piston type; movement of the piston compresses and therefore increases the pressure of the fluid in the cylinder (a 1% change in volume produces an approximate change in pressure of 150 bar for a normal mineral oil) until the cylinder connects to the outlet port and the piston delivers its swept volume. Now depending at what point in the piston compression stroke the cylinder is connected to the outlet port, will determine the pressure of the fluid in the cylinder at the start of delivery. If the pressure in the outlet port is higher than the cylinder pressure at the instant of connection then fluid will flow back from the outlet into the cylinder to equalise the pressure. To avoid this sudden change in flow the connection of the cylinder and the outlet port must be made when the pressures in each are identical. For a given pumping geometry this can only be achieved for one particular value of outlet pressure.

Thus the back flow changes in magnitude with the pump mean displacement and outlet pressure and adds considerably to the flow fluctuation produced by the summation of the individual piston deliveries. A typical result is shown in fig. 4.3.

The addition of pressure relieving grooves - restricted pathways machined into the port plate at the beginning of the traditional kidney port - have been found to be very effective in making the change in flow more gradual and consequently smoothing the pressure fluctuation produced.

Considerable work has been carried out in this field to optimise piston pump port plate design; ref. (7), but even if perfect port plate timing could be achieved it would still be only a partial solution. The flow fluctuation inherent in the geometry of the pumping action would still be present and could therefore create fluidborne noise.

In section 4.2.2 it will be shown that by using specially designed grooves to control dynamic leakage even the flow ripple, inherent to the pumping geometry, may be reduced with a consequent reduction in pump fluidborne noise.

The design of modern gear pumps also includes a means of relieving undesirable pressure pulses, caused by fluid becoming trapped between intermeshing teeth. If these volumes are not vented to a point outside the envelope swept by the path of tooth profile contact then very high instantaneous pressures are developed in the fluid trapped between the meshing teeth. These pressure pulses are generated at pumping frequency and are large enough to cause cyclic movements of the rotating gears and bushings or wear plates. This generally results in a reduction of the pump life and volumetric efficiency and an increase in the fluid and structure borne noise produced by the pump.

The pressure trapping relief grooves are usually designed such that the majority of the trapped fluid volume is directed toward the outlet or high pressure side of the pump, thus maintaining low leakage levels to preserve volumetric efficiency. Correctly designed grooves should allow all the fluid carried in the tooth spaces to be delivered to the outlet port and so the outlet flow fluctuation will be determined by the geometry of the pumping action. If however the trapped volume flow is directed to the inlet side of the pump then the outlet flow ripple would not be entirely determined by pumping geometry.

4.2.2 - FLOW RELIEVING GROOVES

Duke and Dransfield ref. (29) reasoned that if a flow fluctuation, equal and opposite to the outlet flow ripple, could be induced by controlled dynamic leakage from the pump outlet to inlet, then a pump with a ripple free delivery could be produced. Their application of this reasoning to an external gear pump showed that the potential for delivery ripple reduction was of course inherently dependent upon a loss in volumetric efficiency.

Fig. 4.4 shows the theoretical outlet flow ripple for such a pump and the ideal induced leakage flow required for cancellation. The control of the induced leakage flow was achieved by redesigning the original pressure trapping relief grooves into a single gear-side-plate and incorporating a flow relieving groove in the other side-plate as shown in fig. 4.5. The flow relieving groove was positioned so that the passage of the gear teeth over the groove controlled the leakage flow from outlet to inlet in a cyclic manner at gear tooth frequency.

The reduction of pump outlet pressure ripple when tested with a short discharge line and variable restrictor valve termination was reported to be 80%.

A frequency analysis of the pressure ripple was not given but from inspection of the pressure-time traces shown it is probable that the majority of this reduction was at the fundamental pumping frequency. Because the method deliberately bled off outlet flow a reduction in volumetric efficiency would be expected and this was reported to be approximately 10% at a mean pressure of 140 bar. This is an unacceptably high price to pay for the corresponding reduction in pressure ripples, especially as the groove design must inherently be pressure and viscosity dependent and many gear pumps have an upper limit of working pressure in the range 200-300 bar.

4.3 - VARIATION IN PUMPING MECHANISM GEOMETRY

A fundamental means of reducing the flow fluctuation of any pump design is to reduce the proportion of mean flow delivered by each pumping element per revolution of the pump drive shaft. This is simply achieved, by increasing the number of pumping elements. As this would affect the mean flow rate of a particular design as well as the peak to peak fluctuation in flow, it is necessary to express the fluctuation as a proportion of the mean to provide a valid criterion for comparison, i.e.

$$\text{flow fluctuation/unit mean flow} = \frac{Qs\ max - Qs\ min}{Qm}$$

If it is assumed that the flow fluctuation produced by any pump design may be evaluated by consideration of the kinematics of the pumping elements, then the effect the number of elements has on flow fluctuation may be compared for various pump designs. This assumption infers that piston and vane-type machines have "perfect" port plate timing and that gear units have "ideal" pressure trapping relief grooves.

4.3.1 - AXIAL PISTON PUMPS

The flow fluctuation/unit mean flow produced by the combined sinusoidal deliveries of a number of pistons (whether disposed axially or radially) is given by ref. (34) as follows:-

$$\frac{Qs}{Qm} = \frac{\pi}{Z} \sum_{i=1}^{i=Z} \sin\left(wt + \frac{2\pi i}{Z} \right)$$

and is only valid for positive values of $\sin(wt + 2\pi i/Z)$.

Units with an even number of pistons produce a flow variation which repeats at piston frequency. Maximum and minimum values of Qs occur when wt has the value π/Z and $2\pi/Z$ respectively.

Units with an odd number of pistons produce a flow variation which repeats at twice piston frequency. Hence, maximum and minimum values of Qs occur when wt has the value $\pi/2Z$ and π/Z respectively.

Fig. 4.6 shows (Qs max - Qs min)/Qm for pumps having 5 to 20 pistons. For pumps with an odd number of pistons the magnitude of the flow variation decreases from 5% with 5 pistons to 0.34% with 19. A 9 piston arrangement is commonly used in piston pump designs and this produces a flow variation of 1.52%.

The graph shows that for even numbers of pistons the change is from 13.4% with 6 to 1.2% with 20 pistons. This would suggest a significant advantage, on a flow ripple and hence fluidborne noise basis, to have an odd number of pumping elements in an axial piston design. Unfortunately the actual flow ripple produced by a piston unit is very much higher than these theoretical evaluations. For example, a 7 piston unit with a 2.5% theoretical variation in mean flow can have as much as 50% variation when working at a pressure of 200 bar.

The magnitude of this actual variation, depends on the effectiveness of the port-plate timing. This effect has also been revealed by the comparison of piston pump fluidborne noise ratings with those of internal and external gear pumps as reported by Edge and McCandlish ref. (24). For nominally similar mean flows and working pressures the fluidborne noise ratings for the three designs were shown to be of the following order:-

axial piston	14 bar
external gear	7 bar
internal gear	1 bar

The ratings for the two types of gear unit were shown to be largely independent of mean pressure whereas the piston pump varied in rating from 7 to 14 bar for mean pressures of 100 and 200 bar respectively.

4.3.2 - VANE PUMPS

Vane pumps, of the simple eccentric rotor type with a number of vanes running in a circular track, have very similar flow variation characteristics to axial piston machines. The variation in theoretical flow ripple with respect to mean flow as derived by Wüsthoff and Willekins ref. (35) is shown in Fig. 4.6.

The change with increasing number of vanes (for pumps with an even number of vanes) is identical with that of axial piston pumps. However the same does not apply for units with odd numbers of vanes and pistons, over the whole range of rotor eccentricity to vane track radius ratio that are possible with vane units. For eccentricity to track radius ratios approaching zero vane and piston units with an odd number of elements are similar. As the ratio approaches 0.25, vane flow ripple values are considerately higher than piston values. The increase is a constant factor of approximately 2.5 times, making the flow fluctuation of a 5 vane pump with an eccentricity/track ratio of 0.25,12% of the mean. An axial five piston unit would have a 5% variation.

In general vane and axial piston pumps have similar porting arrangements and this would imply similar back-flow characteristics. However vane units usually have between 10 and 15 pumping elements and some designs, which are pressure balanced, have two pumping strokes per revolution. Consequently there is considerable scope for designs with low flow fluctuation characteristics.

4.3.3 - GEAR PUMPS

Gear pumps utilizing external and internal involute spur gear geometry produce considerably different levels of flow fluctuation. The expressions derived for external and internal gearing in sections 3.3.1 and 3.3.2 respectively, show the cyclic flow ripples of both types to be of parabolic form, thus having similar relative harmonic content, but of significantly different amplitude. The peak to peak flow variation, given by these expressions, expressed as a percentage of mean flow is shown in Fig. 4.6 for various numbers of pumping elements.

The flow fluctuation for an external gear machine having equal numbers of teeth on driver and driven gears varies from

25% for 7 teeth to 11% for 20 teeth. Practical considerations have been taken into account in these results. The minimum number of teeth which may be produced without interference is 7 and the results include a variation in the pressure angle of the gears, over the range of number of teeth, to achieve profiles of practical form. (i.e. the pressure angle for 7 teeth is 30° and for 20 teeth is 15°).

Evaluation of the flow fluctuation produced by external gear pumps tends to indicate that the pumping mechanism exhibits characteristics closely related to those predicted by a simple geometric model and are largely independent of mean working pressure. This is also likely to be true for internal gear units since with the exception of the differences in involute geometry, the pumping actions of the two types are identical.

Internal gearing has an additional factor to be considered; the difference in the number of teeth between pinion and ring or internal gear. The minimum difference for any form of internal gearing is three (expressed $Z_2 = -(Z_1+3)$ the -ve sign denoting internal gear). The practical minimum for internal involute gearing is $Z_2 = -(Z_1+7)$ as used in the well known Eckerle-Voith design ref. (36). The flow fluctuation with variation in number of pinion teeth (Z_1) for both these conditions is also shown in Fig. 4.6. Increasing the difference between Z_1 and Z_2, increases the flow fluctuation by up to a factor of two throughout the range for the particular conditions shown. For the case where $Z_2 = -(Z_1+7)$ compared with the external case of $Z_1 = Z_2$ the flow fluctuation is less, by a factor which varies from approximately 3 to 10 over the range of pumping elements considered.

Consider a typical example. An internal gear pump with $Z_1 = 13$ and $Z_2 = 20$ (Voith design) has a flow ripple which is 3% of the mean flow (this is in fact what is claimed by the manufacturer ref. (36)). An external gear pump, $Z_1 = Z_2 = 13$ has a flow ripple 15% of the mean flow, 5 times that of the internal gear unit. By inappropriate selection of pumpi ng elements an external gear pump may have flow ripples up to 20 times that of an internal unit.

4.4 - DUPLICATION OF PUMPING ELEMENTS

A particular method of active flow ripple cancellation, which involves duplication of the pumping elements of the machine is of special interest because of its utilization of two basic principles:-

a) Generation of an anti-flow ripple is achieved inside the pump without significant loss of volumetric or mechanical efficiency.

b) The degree of cancellation is predictable and the matching of amplitude and mismatching of the phase of the anti-flow ripple is achieved by a simple mechanical means.

The means however relies on a precise understanding of the flow ripple generation characteristics of the pumping geometry.

Both of these basic concepts are fundamental to the design of a practical, cost effective active method of pump pressure ripple reduction by source flow modification.

This method of reducing pump flow ripple by duplication of the pumping elements is utilized in an external gear design, known as a "Duo-Pump" ref. (9). This is currently marketed as a low "noise" pump by the hydraulic division of the German based company Bosch. The benefits of the "Duo" or dual-gear method were reported by Yudin ref. (28) at least as early as 1964 but it has necessitated market demands for quieter hydraulic systems to encourage its development as a practical alternative to internal gear designs.

The flow ripple produced by an external gear pump is proportional to the width of the meshing gears. Therefore by dividing the gear width into two halves, each half produces a ripple which is equal to that produced by the other and which when combined in phase with the other will produce a ripple equal to that produced by the whole gear width. If however the ripples produced by each half gear width are combined in anti-phase, interference occurs.

Anti-phasing of the two ripples is achieved by displacing the two half-gear widths by a half tooth pitch (180° phase shift in the fundamental tooth frequency components).

The effect, on flow ripple, of this duplication of pumping elements is best shown with an example. Table 4.1 shows the first ten harmonics of the theoretical flow ripple (as given by equation 3.1) for a Dowty 1P3060 external involute gear pump in standard and dual-gear forms. The amplitudes of the components produced by each half of the dual-gear are half of those produced by the standard design. A 180° phase shift in the fundamental, and consequent changes in phase of other harmonics, has been applied between the 1st and 2nd gear width halves. The combination of the two ripples results in the cancellation of all odd harmonics and the reinforcement of even harmonics. The resultant levels of the even harmonics is of course equal to those produced by the original unit in its standard form.

The dual-gear method therefore effects a cancellation of all odd harmonics of the original unit, leaving a ripple consisting of even harmonics only. This remaining flow ripple is at a fundamental frequency twice that of the original pump and approximately 25% of the original peak to peak amplitude. A comparison of the standard and dual-gear 1P3060 flow and volume ripple is shown in fig. 4.7.

The variation of output flow with time clearly shows the reduction in amplitude and doubling of frequency.

It is interesting to note that the dual-gear design reduces the peak to peak flow ripple from 24% to 6% of the mean flow. There is however still double the 3% variation which is produced by an equivalent displacement internal gear pump.

The variation in output volume with shaft rotational angle also shown in fig. 4.7 for the standard and dual-gear designs again shows the doubling of frequency effect. Note the amplitude reduces from a zero to peak value of 36 mm^3 to

4.5 mm^3, a reduction by a factor of 8 compared to a flow ripple reduction factor of 4. These are in fact equivalent reductions, the factor of 2 difference between flow and volume variation reductions being inherent in the integration process to interpret flow variations as variations in volume.

The reduction in flow ripple caused by the dual-gear design is not achieved without penalty. Division of the gear produces twice the number of gear face leakage paths and must therefore adversely affect volumetric efficiency. Separating the gear halves on common shafts (as in the Bosch design) reduces shaft stiffness and therefore the operating pressure. Most importantly the dual-gear design is inherently an estimated 25 to 30% more expensive to manufacture than the standard unit.

4.5 - A DEDICATED FLOW RIPPLE CANCELLATION DEVICE

The various methods of reducing pump flow ripple previously described in this section, all show considerable potential in principle but suffer from a number of practical and economic limitations. These detract from the effectiveness of each method and as a result no single method has the potential for a practical, cost effective solution to the pump flow ripple reduction problem. However consideration of the advantages of these methods can provide an insight into the design specification of a method or device dedicated to the production of a precisely controlled anti-flow ripple.

To enable a detailed design specification to be drawn up it is convenient to consider the problem as related to a particular situation. This design specification is generally applicable to the Dowty P3000 range of external gear pumps and in particular the 1P3060 gear pump unit.

4.5.1 - DESIGN SPECIFICATION OF FLOW RIPPLE CANCELLATION DEVICE FOR 1P3060 GEAR PUMP

The flow ripple cancellation device should:-

i) allow the production of flow fluctuations at variable pumping element frequencies and their harmonics. The relative amplitude and phase of constituent harmonic components must be capable of initial adjustment.

ii) Produce a fundamental sinusoidal volume variation component of the order of 72 mm^2 peak to peak at pumping element frequency.

iii) Be able to produce this volume variation at frequencies varying between 0-400 Hz, while maintaining correct synchronization with pumping elements. If possible synchronization should be accomplished mechanically.

iv) Be able to produce the volume variation in a fluid (of type SAE 10-20 at temperatures up to 80°C) at continuous or changing mean pressures between zero and 210 bar.

v) Not adversely affect the steady state performance of the pump. In particular the volumetric efficiency should not be less than 92% at the maximum rated pressure.

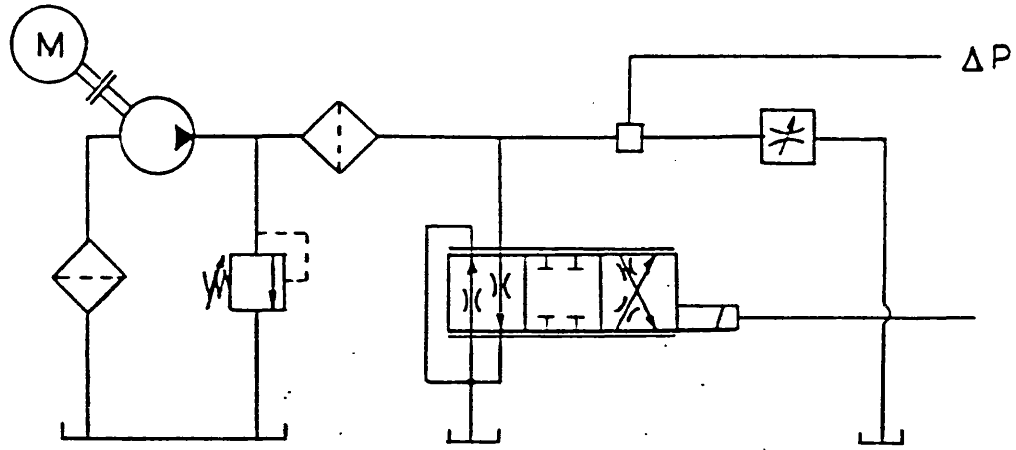
vi) Not significantly increase the installation volume or cause installation restrictions on standard mounting or porting options.

vii) Not increase the works cost of the pump by more than 25% unless the potential performance is significantly better than competitive units. To this end standard components should be incorporated where possible.

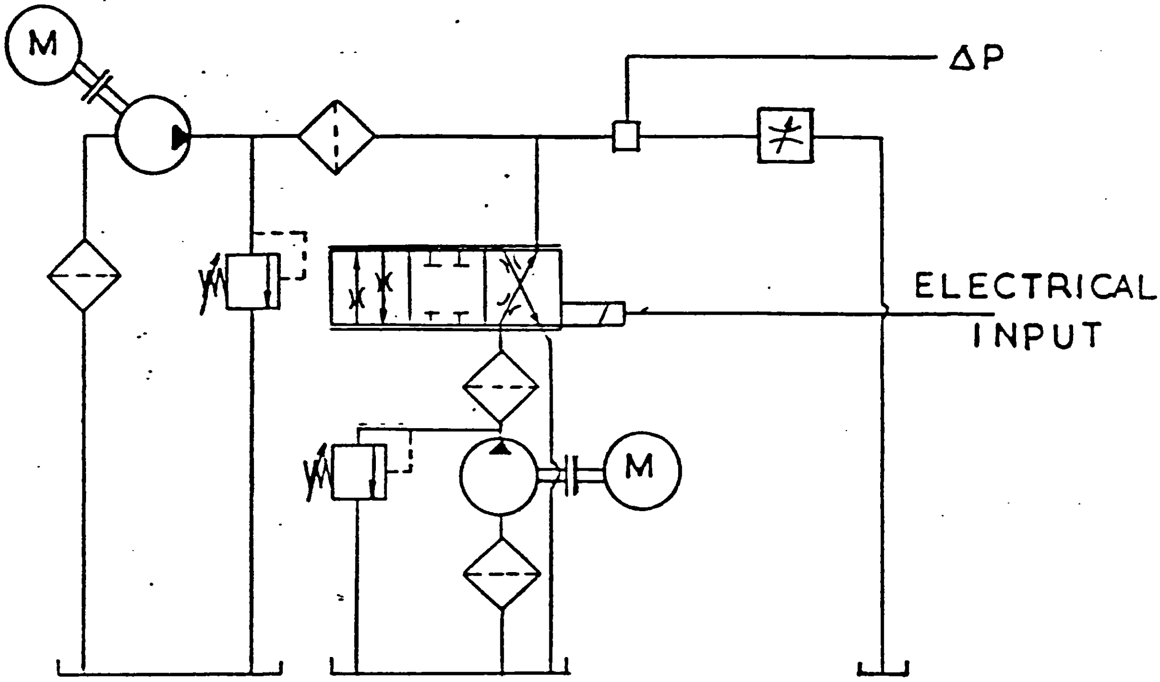
viii) Be economically applicable to a range of pump or motor displacements.

HARMONIC OF OF 200 Hz	STANDARD IP3060		DUAL-GEAR IP3060					
			1 ST HALF		2 ND HALF		RESULTANT	
	AMPLITUDE $\times 10^{-3}$ M^3/S	PHASE DEGREE	AMPLITUDE $\times 10^{-3}$ M^3/S	PHASE DEGREE	AMPLITUDE $\times 10^{-3}$ M^3/S	PHASE DEGREE	AMPLITUDE $\times 10^{-3}$ M^3/S	PHASE DEGREE
1	·04502	0	·02251	0	·02251	180	0	0
2	·0113	90	·00565	90	·00565	90	·0113	90
3	·00505	-180	·00252	-180	·00252	0	0	0
4	·00287	-90	·001435	-90	·001435	-90	·00287	-90
5	·00184	0	·00092	0	·00092	180	0	0
6	·00131	90	·00065	90	·00065	90	·00131	90
7	·00098	-180	·00049	-180	·00049	0	0	0
8	·00076	-90	·00038	-90	·00038	-90	·00076	-90
9	·00062	0	·00031	0	·00031	180	0	0
10	·00045	90	·000225	90	·000225	90	·00045	90

TABLE.4.1 THEORETICAL HARMONIC COMPONENTS OF FLOW RIPPLE FOR STANDARD AND DUAL-GEAR IP3060 EXTERNAL GEAR PUMPS



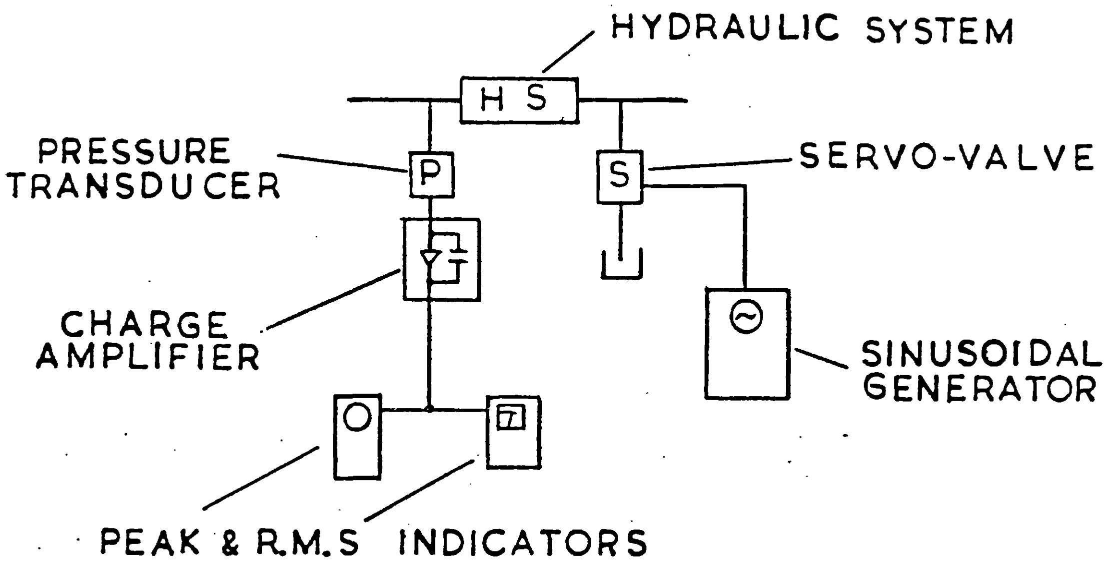
4.1.a FLOW BLEED-OFF



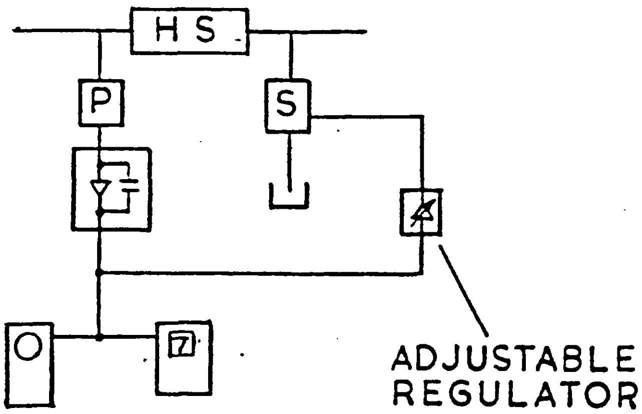
4.1.b FLOW BLEED-OFF AND ADDITION

FIG. 4.1 HYDRAULIC CIRCUITS FOR PRODUCING COUNTER-PRESSURE PULSES WITH A SERVO-VALVE

REPRODUCED FROM REF (33)



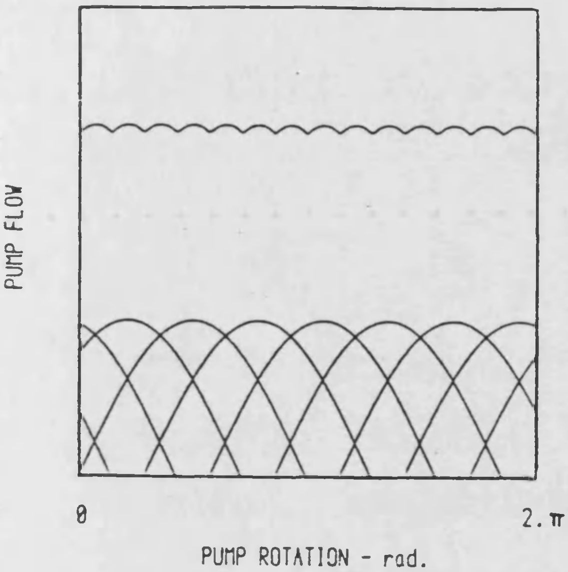
OPEN-LOOP CONTROL



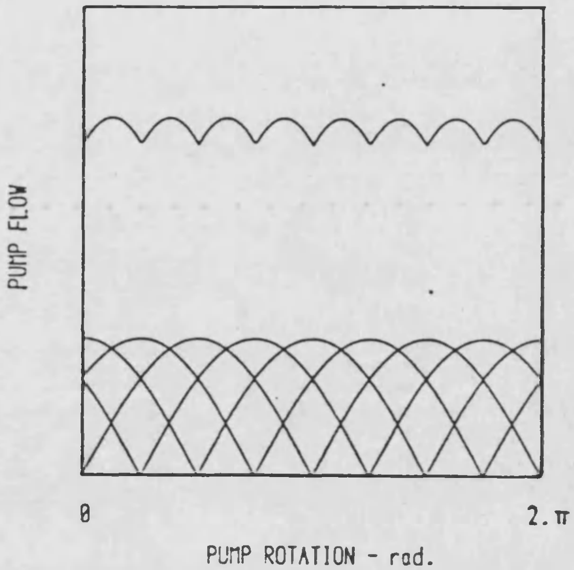
CLOSED-LOOP CONTROL

FIG. 4.2 ELECTRICAL CIRCUITS FOR CONTROL OF A COUNTER-PULSE PRODUCING SERVO-VALVE

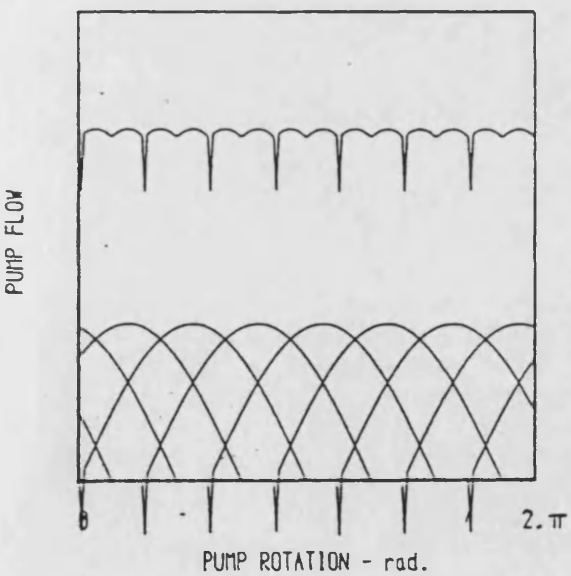
REPRODUCED FROM REF. (33)



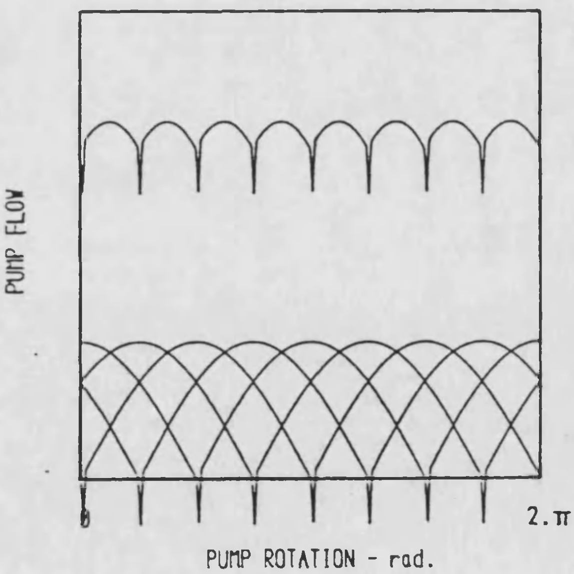
(a) 7 PISTON PUMP
UNLOADED



(b) 8 PISTON PUMP
UNLOADED

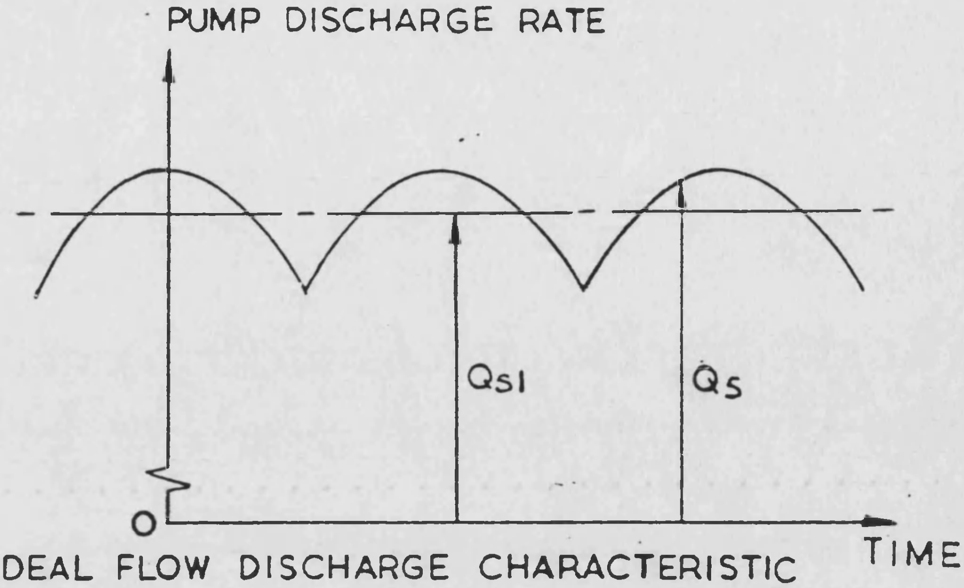


(c) 7 PISTON PUMP
AT 100 BAR

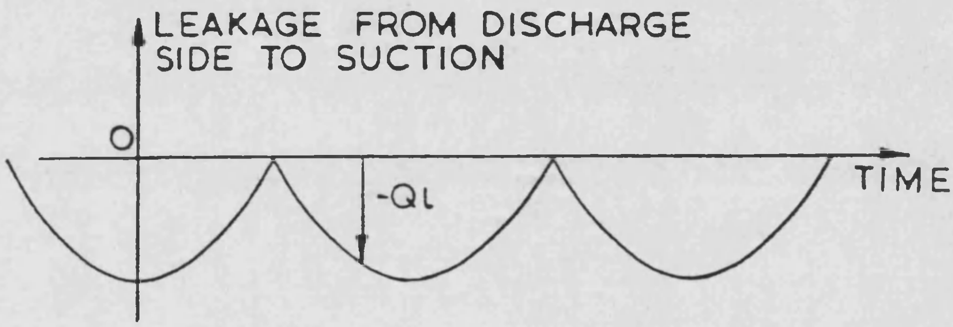


(d) 8 PISTON PUMP
AT 100 BAR

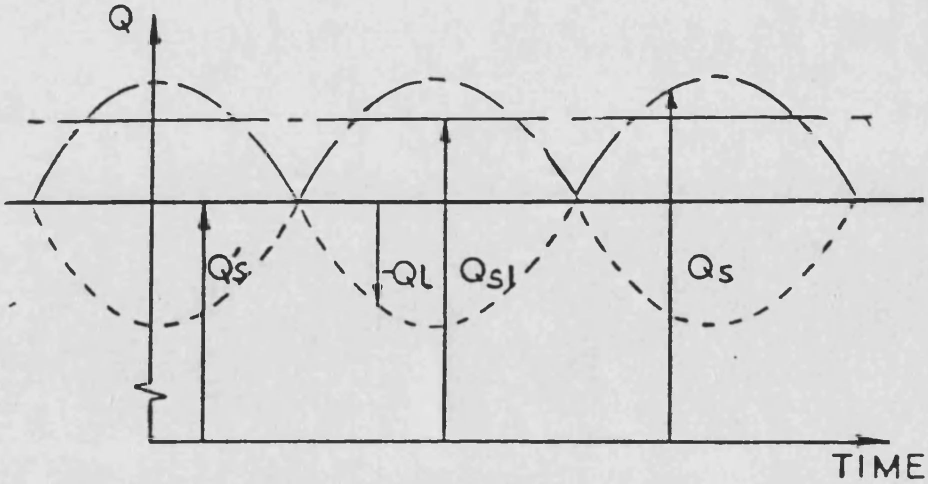
FIG. 4.3 TYPICAL THEORETICAL AND ACTUAL FLOW FLUCTUATIONS PRODUCED BY PISTON PUMPS



(a) IDEAL FLOW DISCHARGE CHARACTERISTIC FOR GEAR PUMP WITHOUT LEAKAGE GROOVE

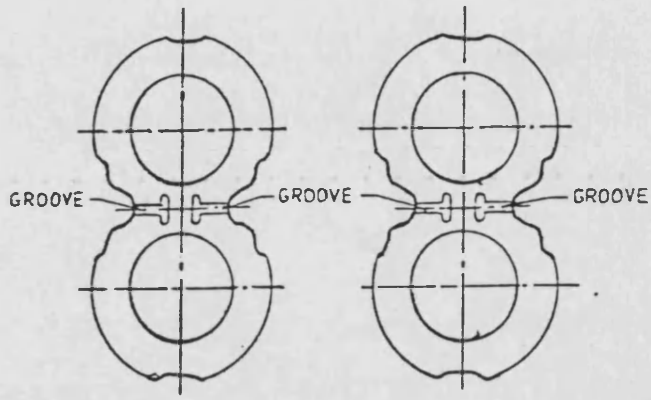


(b) DESIRED CONTROLLED LEAKAGE CHARACTERISTIC

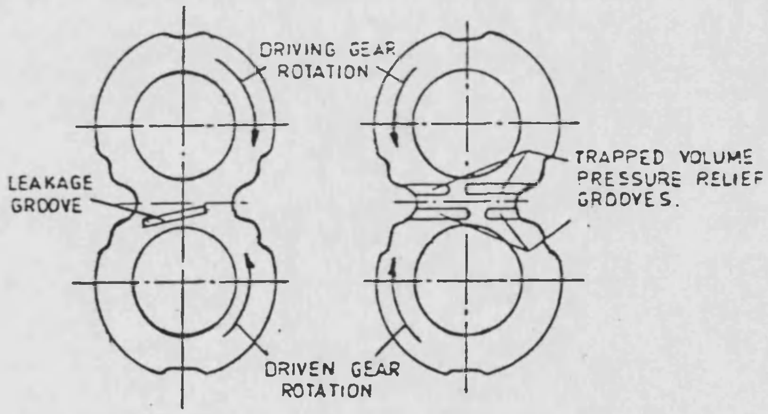


(c) USE OF CONTROLLED LEAKAGE (Ql) TO INDUCE CONSTANT DISCHARGE FLOWRATE (Q's)

FIG. 4.4 GEAR PUMP CONTROLLED LEAKAGE TO INDUCE CONSTANT DISCHARGE FLOWRATE REPRODUCED FROM REF.(29)



(a) ORIGINAL PRESSURE RELIEF GROOVES IN SIDE PLATES OF GEAR PUMP.



(b) PROPOSED RELIEF GROOVE (right side) PLUS LEAKAGE GROOVE (left side) SYSTEM

FIG. 4.5 GEAR PUMP SIDE PLATE GROOVE SYSTEMS FOR PRESSURE TRAPPING AND DYNAMIC LEAKAGE CONTROL

REPRODUCED FROM REF. (29)

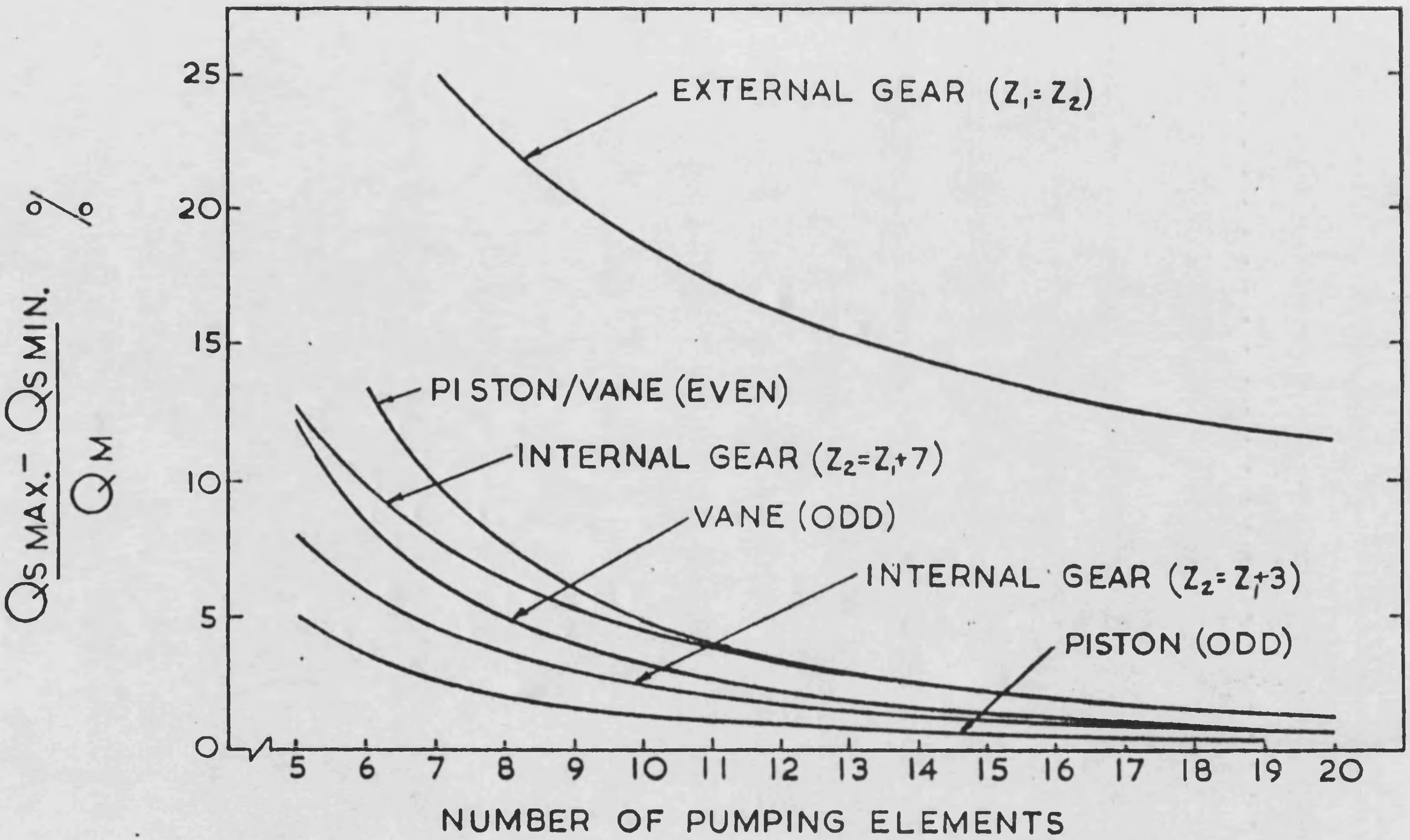


FIG. 4.6 THE RATIO OF THEORETICAL FLOW VARIATION TO MEAN FLOW WITH VARIATION IN NUMBER OF PUMPING ELEMENTS FOR DIFFERENT PUMP DESIGNS

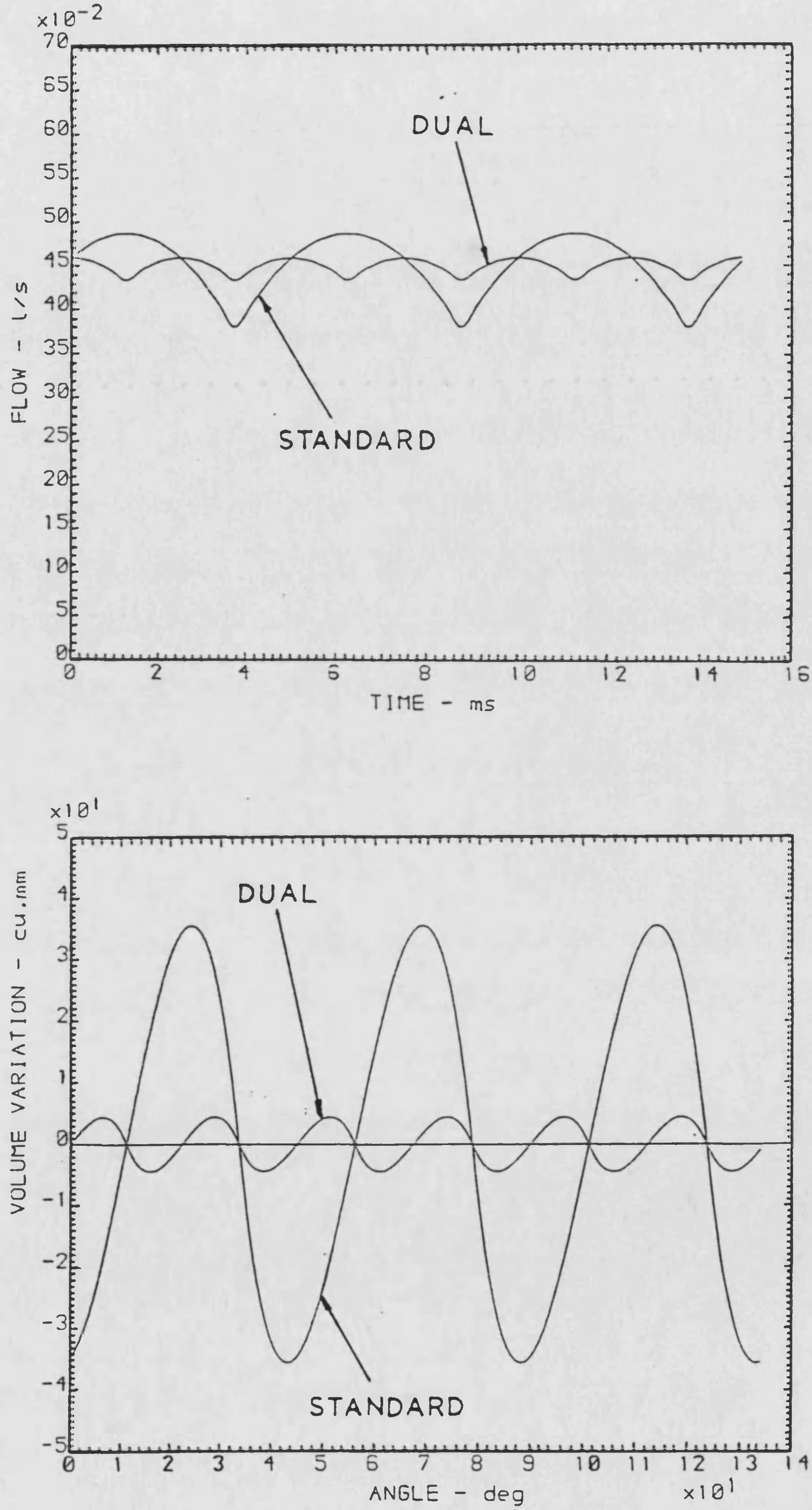


FIG. 4.7 THEORETICAL FLOW AND VOLUME VARIATIONS FOR STANDARD AND DUAL IP3060 GEAR PUMPS

CHAPTER 5 - FEASIBILITY OF FLOW RIPPLE CANCELLATION DEVICE

The purpose of this chapter is to discuss some possible solutions to the design specification described in section 4.5.1. All the ideas presented are based on concepts utilised in the methods described in section 4.1 to 4.4.

Concept Summary:-

i) Flow ripple cancellation to be achieved by the addition of an anti-flow variation, actively produced by some controllable mechanism.

ii) Energy input to the mechanism to be derived from the source producing the initial flow fluctuation. Close coupling of the source and mechanism is preferable to produce a modified pump/motor unit of compact design.

iii) Mechanism control to be mechanical. Electrical control methods are inherently too expensive and not all hydraulic systems have an electric power source.

Effecting Fluid Displacements

A cylindrical piston sliding in a close fitting bore is an accepted and well proven method of economically displacing fluid. Close geometric tolerances may be held easily by conventional machining techniques so that fluid displacement, with minimal cross-piston leakage, can be achieved even at high fluid working-pressures.

The volume of fluid displaced can be easily controlled by manipulation of the stroke and diameter of the piston. It was therefore decided that a cylindrical piston should form the interface, between the pump high pressure outlet volume and the inlet volume (or atmosphere), through which a cyclic fluid-volume variation could be introduced. The problem then is achieving a variable yet controllable oscillatory piston motion from the rotating source available (the pump driveshaft).

5.1 - GEAR TRAIN WITH ECCENTRIC CAM AND PISTON FOLLOWER

A simple eccentric cam was considered as a means of providing a piston with a controllable oscillatory motion. One solution is shown schematically in Fig. 5.1. An eccentric cam shaft is driven via a simple gear train from the gear pump drive shaft. The gear ratio required is equal to the number of pumping elements (gear teeth) on the pump drive gear.

The piston displacement is dependent upon the profile of the eccentric cam so that additional harmonic components of the fundamental frequency could be incorporated to produce a complex volume ripple by adjustmentof the cam profile.

In principle the basic requirements of the specification appear to be satisfied. However, consideration of the rotational speed of the eccentric cam shaft over the range of pump driveshaft speeds reveals serious practical limitations. For the Dowty 1P3060 pump with 8 teeth, the maximum driveshaft speed is 3000 rev/min. which means the cam shaft would have to rotate at 24000 rev/min. Such high speeds would create particular problems with mechanical vibration and airborne noise from the meshing teeth of the gear train. The eccentric cam would require balancing and the very high surface velocities between the cam and the piston follower would seriously limit the load which the piston follower could carry. This prompted a search foran alternative method of driving the piston without incurring the problems of excessive rotational velocities.

5.2 - ROCKING PAWL WITH CONNECTING ROD AND PISTON

The requirement for a fundamental piston oscillation at tooth frequency led to the investigation of methods of using the pump gear teeth as a means of converting rotational to oscillatory motion. As modification of the existing involute tooth form would be impractical the prospects were not encouraging. However one arrangement, often used in clock mechanisms and known as the "rocking pawl" was considered.

A schematic representation of how oscillatory motion may be extracted directly from the pump gear teeth is shown in Fig. 5.2. A horse-shoe shaped component,known as the pawl, is positioned so that it may rock back and forth about a pivot as dictated by the contact between the tooth profile and the extremities of the pawl. The three positions shown are intended to illustrate the motion of the pawl with rotation of the gear teeth.

In position a, anti-clockwise rotation of the tooth flank contacting the pawl produces a clockwise movement of the pawl. Attachment of a piston via a connecting rod to a point on the pawl, as shown, produces an upward movement of the piston. At position b, the pawl is still rotating clockwise, the piston has reached the top of its stroke, and the right hand side of the pawl is encroaching into a tooth space. In position c, continued anti-clockwise rotation of the gear transfers contact from the left to the right of the pawl and reverses the direction of the pawl rotation and the cycle repeats. It will be noticed however that for a gear movement of half a tooth pitch the piston has travelled one cycle, up and down. This arrangement actually produces a piston oscillation at twice tooth frequency.

If instead of contacting the actual pump gear teeth with the pawl, a special gear or cam is added to the pump driveshaft, the number of teeth or lobes may be halved to achieve the correct frequency. This is obviously only applicable to gear pumps with an even number of gear teeth, but the additional special cam may now be profiled to produce the required piston displacement profile. The offset pivot and con-rod arrangement also provides a velocity ratio and mechanicaladvantage adjustment between pawl and piston movements, a useful facility when the piston has to work at high velocities against high pressures.

Unfortunately this method has two drawbacks which make it unsuitable as a solution. Firstly, the piston displacement is only controlled during an upward piston movement. The return, whether by spring force or fluid pressure on the piston, is uncontrolled because neither end of the pawl is in contact with the gear profile. Secondly, because of the thickness of the pawl ends and the gear tooth tip width, there is a dwell period where no pawl and therefore no piston movement is possible. This is also unacceptable as gear pump volume ripple is of a smooth continuous form.

At this point it was apparent that a satisfactory oscillatory motion could be achieved by a modified version of this mechanism. A common method of accomplishing continuous control of cam-follower displacement is to use two cams, of similar profile, each of which controls motion in a single direction. In this case one cam profile would control the upward piston movement and a conjugate cam profile would control the movement downward. The following section describes a mechanism incorporating the over-centre or toggle device with a conjugate cam arrangement.

5.3 - TOGGLE MECHANISM WITH CONJUGATE CAMS AND PISTON

A variation of the rocking pawl method, previously described. is shown in Fig. 5.3. The piston is now at the free end of a toggle or over-centre arrangement and moves through a single displacement cycle each time the central pivot moves from one side of the centre line to the other.

Two contra-rotating cams, which are 180° out of phase with each other, dictate the motion of a roller follower which is the centre pivot of the toggle mechanism. Hence rotational cam motion is transformed into linear piston motion depending on the geometry of the toggle and the cam profile.

Fig. 5.3.a shows the piston at the start of the stroke with the roller-follower at the centre pivot at its maximum distance from the centre-line. The roller-follower is displaced toward the centre by the clockwise rotation of the right-hand cam. The trailing profile of the synchronous left-hand cam is the conjugate of the active profile of the right-hand cam. At

position b, the roller-follower is in the centre position and contact is about to be transferred to the left-hand cam. The piston return force F, due to the resultant of fluid pressure and piston acceleration force, biases the roller-follower contact to left or right depending on whether the roller-follower is to the left or right of the centre line. Piston displacement is now controlled by the left-hand cam until the end of stroke shown in position C.

This arrangement was considered to be the first to embody in a practical manner all the variable quantities needed to satisfy the specification. It was therefore decided to scheme the device into a pump, as shown in Fig. 5.4, in order to assess the viability of the arrangement when installed in a pump.

Section B-B of Fig. 5.4 shows how well the two conjugate cam arrangement lends itself to the external gear pump. Each cam may be supported, driven and phased by simply adding to the existing driver and driven gear shafts.

Section A-A illustrates the principle of the device and shows how the toggle mechanism naturally extends across the pump axis between the pitch centres of the gears.

Section C-C shows how fluid displaced by the spooled piston is easily transferred, by a short connecting passageway, to the pump outlet port.

Allowing the displaced flow to travel through the side of the piston permits this arrangement and also enables the piston mass to be reduced to minimise inertia effects. It is also possible, for testing purposes, to isolate the pump from the effect of the mechanism by substituting the blanking plug for the isolating plug shown to the right of section C-C.

Section C-C also shows the toggle mechanism, comprising two yoke shaped components, which allow the centre pivot to be accessed by the two cams. A thin tubular roller around the centre pivot pin is intended to provide rolling, instead of sliding, contact between the pin and the cam profile.

5.3.1 - DISPLACEMENT AND FORCE ADVANTAGES PRODUCED BY TOGGLES

The cam profiles shown in Fig. 5.4 were derived by graphical means and are intended to be representative of the shape required to produce a fundamental peak to peak volume variation of approximately 72 mm^3. With a 20mm diameter piston this is achieved by a piston stroke of 0.23 mm. Equal yoke connecting rod lengths of 20 mm mean that this stroke is produced by a 2.14 mm lateral displacement of the centre pivot from the centre position.

The piston force due to a fluid pressure of 210 bar is approximately 6,600N. This translates to a maximum lateral force of 1,400N to be provided by the cam, a force advantage of 4.7.

The variable parameters of this design (piston diameter, cam throw and toggle dimensions) offer considerable scope for functional development and application to a range of different pump displacements. However, from a production point of view, some aspects of the scheme are less than ideal:-

i) The number of additional components incorporated in the modification is 7. This is high compared to a standard pump which contains 9 major items excluding seals and through bolts. This combined with an increase of up to 35% in installational length, which may be approximated to a proportional raw material on cost, means that the target of 25% work cost increase cannot be achieved. Additional assembly time because of the increased complexity of the modified unit would add further cost.

ii) The transfer of roller-follower contact from one cam to the other at mid-piston stroke may be a source of mechanical noise. Close tolerancing of the conjugate cam profile and support bearings would minimise this but attention would have to be given to the back-lash between the meshing gear pump teeth, as cam drive must transfer through this mesh.

iii) Utilizing changes in the piston diameter, cam profile or toggle dimensions to make the mechanism suit different pump displacements would introduce a number of component variants for each displacement. The number of component variants necessary should be kept to a minimum as component costs generally increase with the variety of components produced.

Consideration of these aspects of the design promoted changes which would considerably simplify and improve the mechanism and its adaptability. Using only a single cam and direct acting piston-followers of significantly reduced diameter, achieved a reduction, in the number of additional components, the installational volume and the potential cost. The conception of this improved design is explained in the description of the multi-lift cam mechanism.

5.4 - MULTI-LIFT CAM WITH PISTON-FOLLOWERS

The required oscillatory control of a piston operating against high fluid pressures and at high frequencies may be achieved, in a direct manner, by a simple rotating cam. A cam which, for each revolution, imparts eight outward and return cycles to a piston-follower is the most important factor in producing a simple and compact mechanism design.

Another important factor, which results from the symmetry of the multi-lift cam, is that a number of piston-followers can each be controlled by a single cam. For a particular displacement volume, selecting a cam lift determines an effective piston area. If this area is divided between two separate pistons, acting at diagonally opposed points on the cam, the individual piston displacements are identical in amplitude and phase, and the net side load on the cam is zero.

This provides the means of producing a required volume ripple by distributing the area between a number of pistons with the same displacement characteristics. Reducing the area of the individual pistons, proportionally reduces the piston load, for a given working pressure, and the piston mass. The latter, reduces the inertia forces acting on the pistons at high frequencies and obviates the need for bi-directional piston drive provided by conjugate cams. Under conditions of low working fluid pressure and hence low piston returning force, control of the piston displacement during the return stroke is determined by the cam profile as long as piston-cam contact is maintained. This may be ensured by the provision of a returning force, greater than the acceleration and friction forces acting on the piston, from a return spring or a minimum fluid pressure.

A schematic representation at the multi-lift cam arrangement is shown in Fig. 5.5. The cam profile shown is of the simplest, straight-sided, form to illustrate the principle. View a shows the pistons in the minimum displacement condition. View b shows the pistons at the point of maximum displacement after a cam rotation of 22.5 degrees - equal to one half of a tooth cycle (one tooth cycle for 8 teeth corresponds to a rotation of 45 degrees). View c shows the piston in a mid-stroke position after a further 11.25 degrees of cam rotation. The movement of the point of contact, between the cam and the piston-follower, to the right of the centre-line should be noted. The profile of the piston-follower obviously affects the piston displacement for any particular cam profile and a constant curvature piston-follower profile has a number of theoretical and practical advantages which will be explained later. A further 11.25 degrees of cam rotation completes one tooth cycle and returns the pistons to the minimum displacement position (as shown in a) after one complete cycle of piston movement.

In principle, the multi-lift cam design offers the most promising solution to the problem of flow ripple generation. Consequently the following chapters are devoted to the discussion of the design, development and experimentation carried out in pursuing the reduction of gear pump flow ripple, using multi-lift cam type devices.

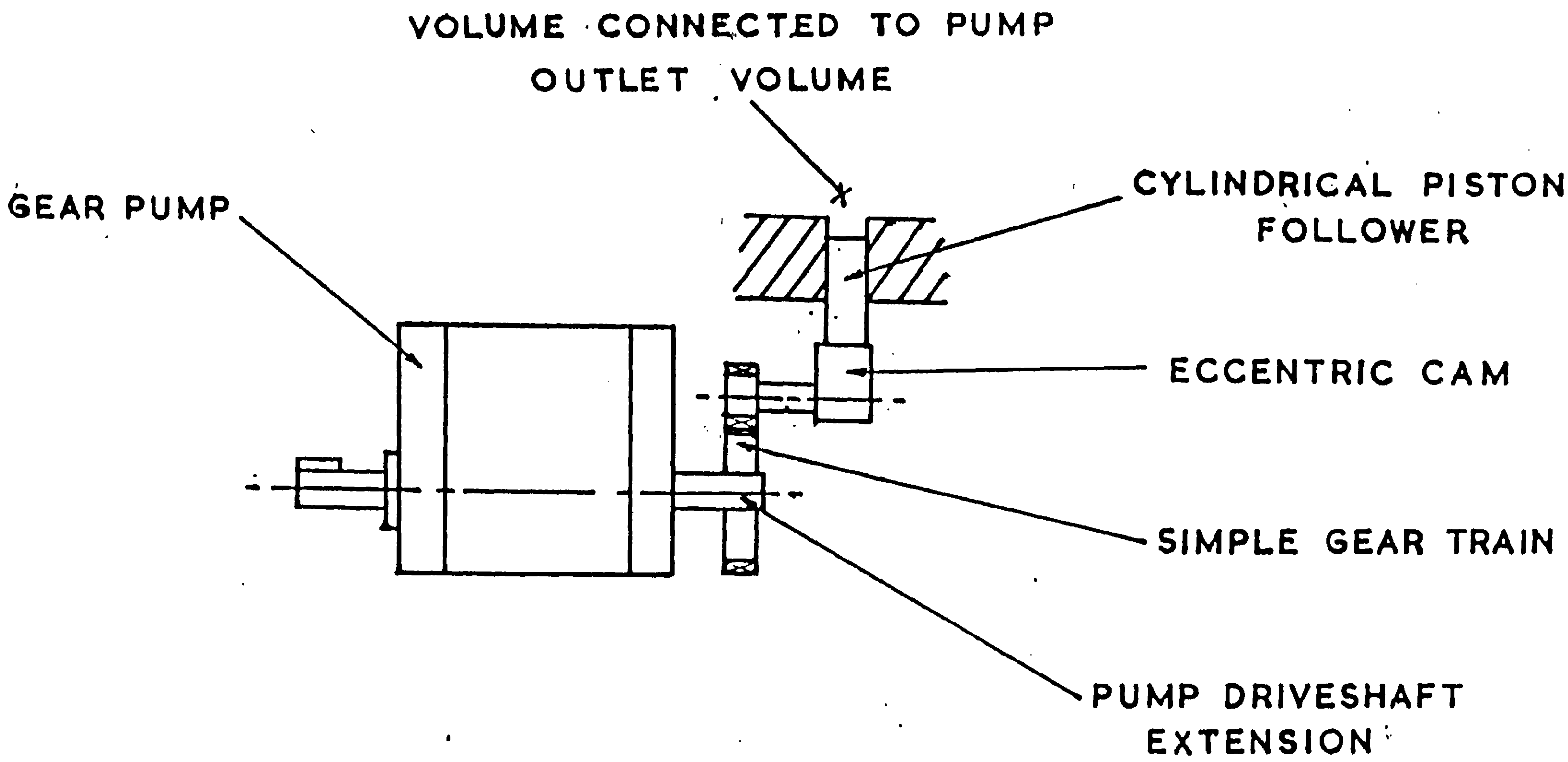


FIG. 5.1 GEAR TRAIN WITH ECCENTRIC CAM AND PISTON-FOLLOWER

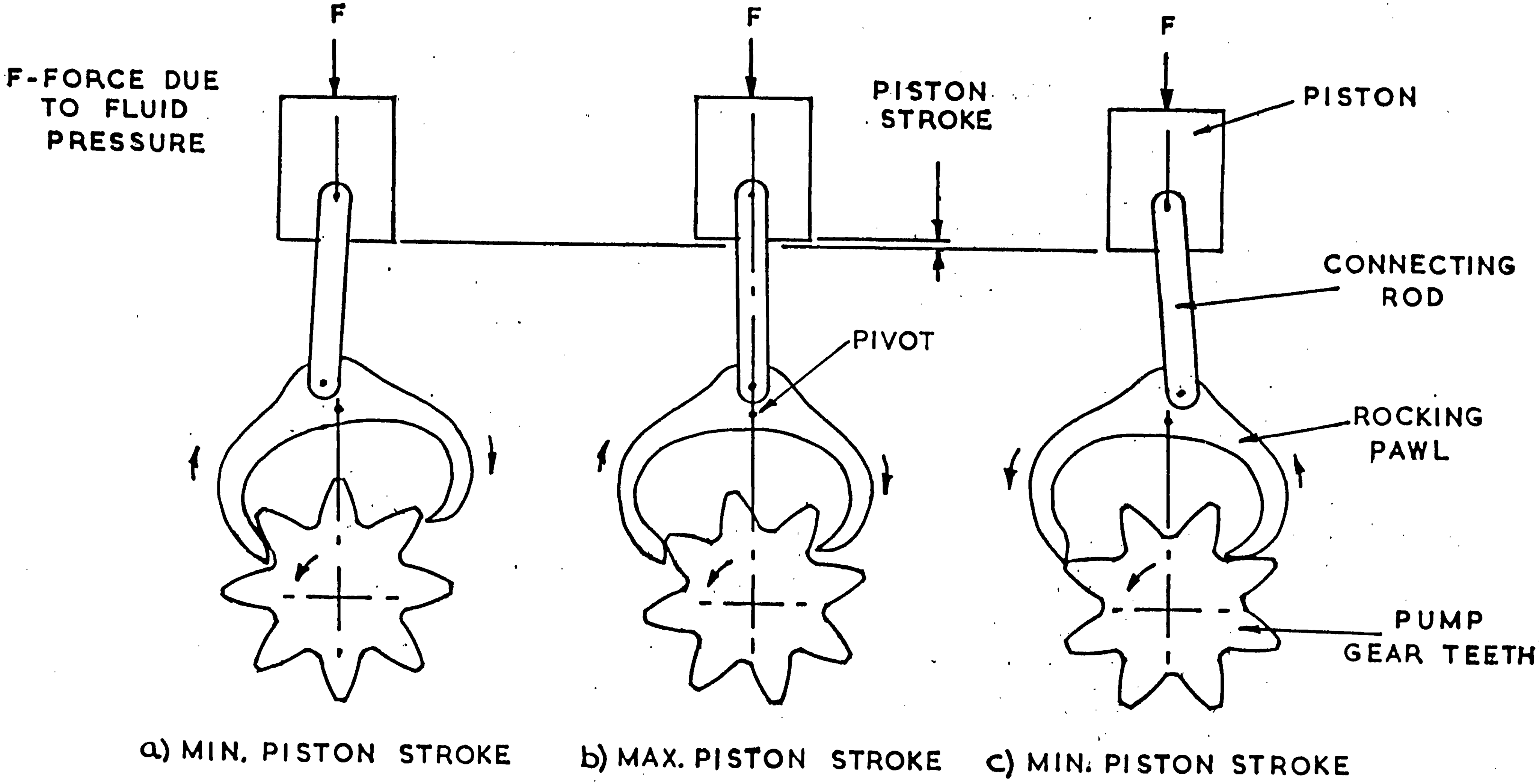


FIG. 5.2 SCHEMATIC OF ROCKING PAWL MECHANISM

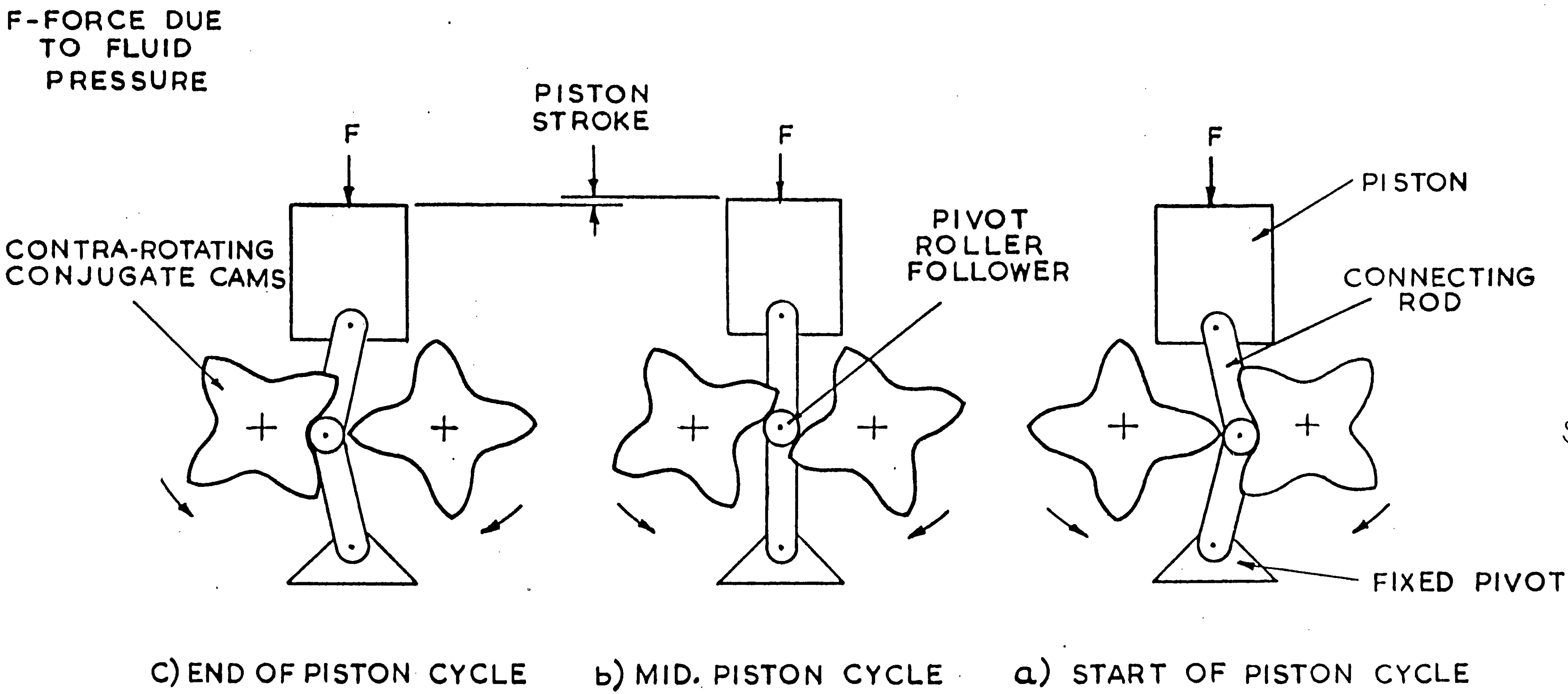


FIG. 5.3 TOGGLE MECHANISM WITH CONJUGATE CAMS AND PISTON

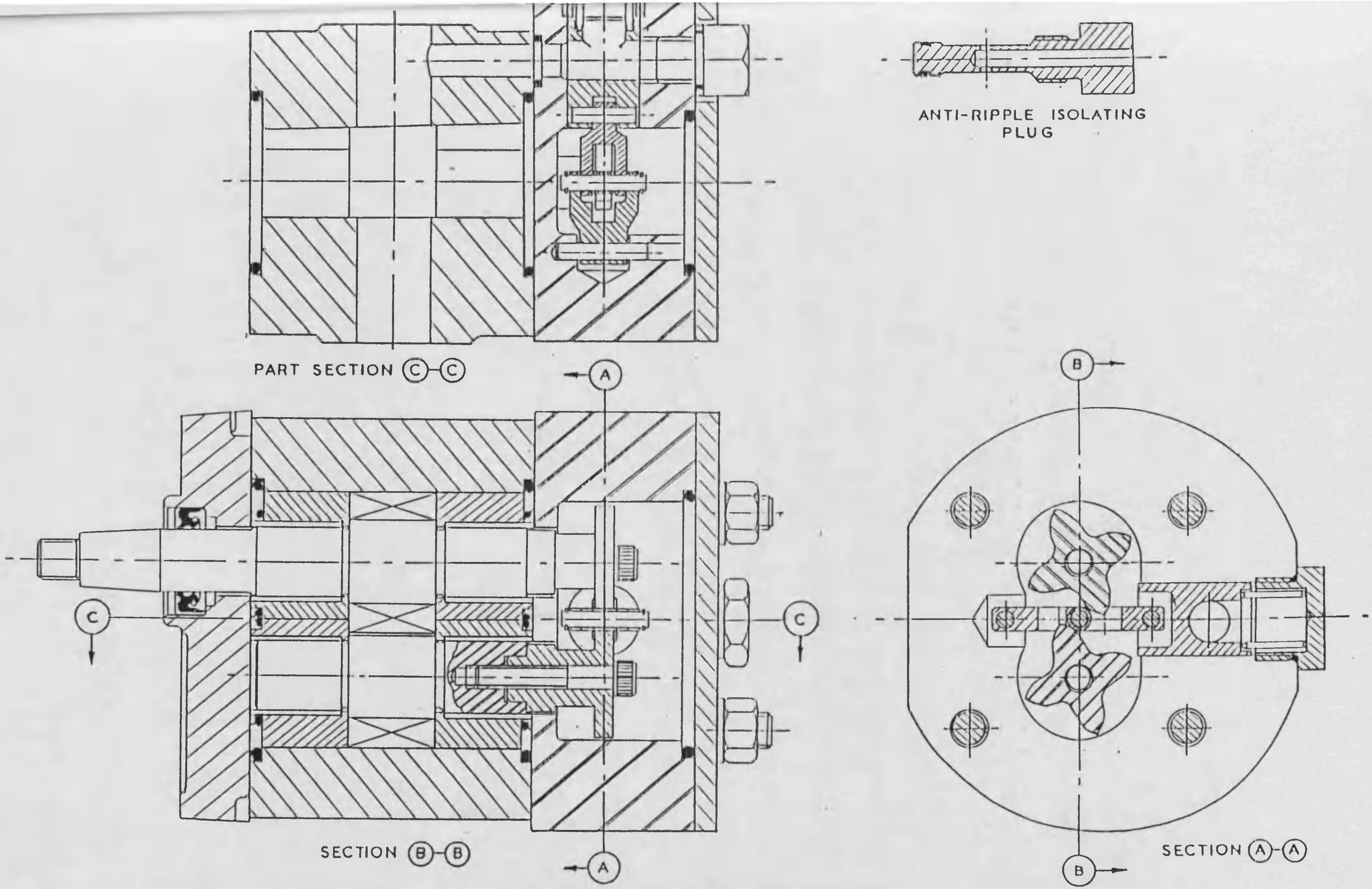
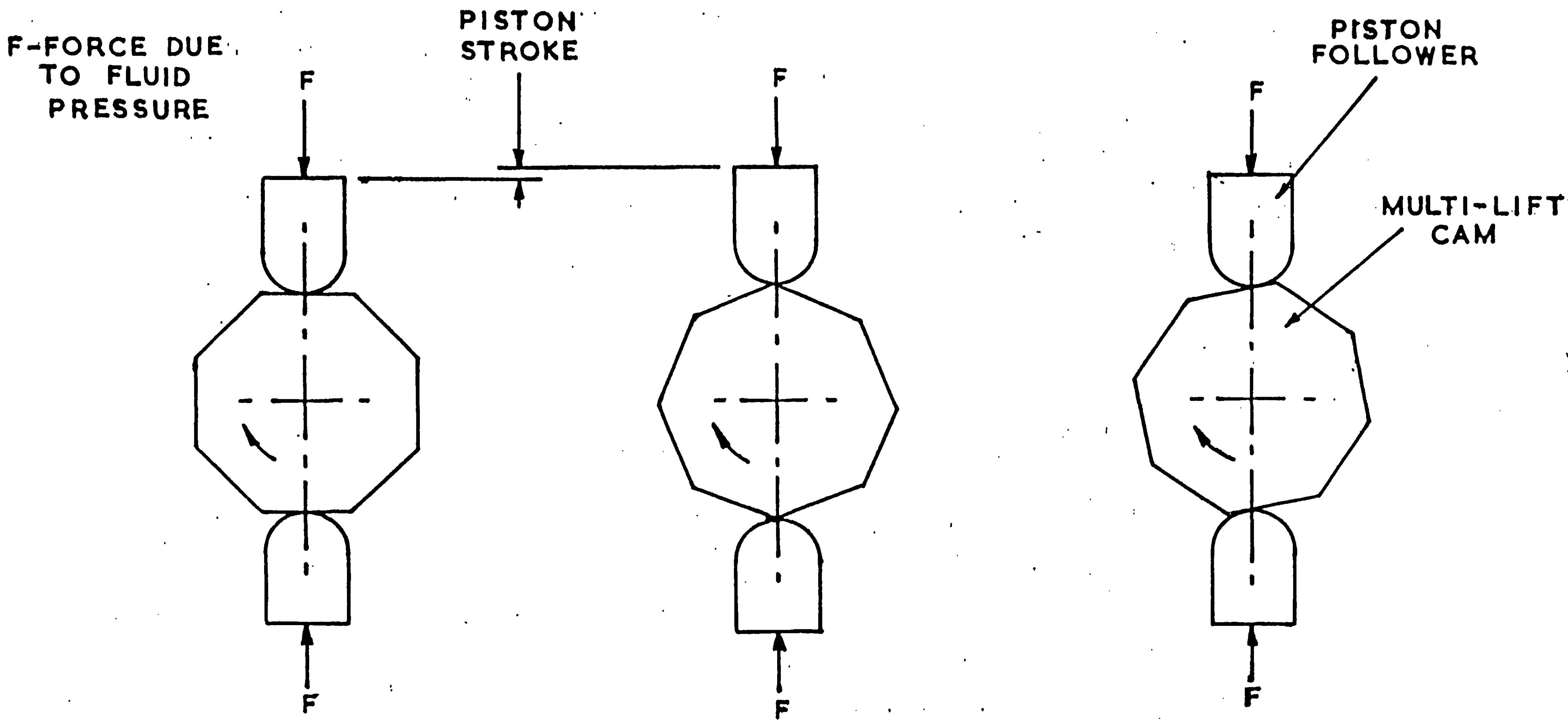


FIG. 5.4 SCHEME OF IP3060 WITH CONJUGATE CAM ARRANGEMENT



a) MIN. PISTON STROKE b) MAX, PISTON STROKE c) MID PISTON STROKE

FIG.5.5 SCHEMATIC OF MULTI-LIFT CAM MECHANISM

CHAPTER 6 - FEASIBILITY STUDY OF MULTI-LIFT CAM MECHANISM

6.1 - INTRODUCTION

This chapter presents the design considerations for a multi-lift cam mechanism proposed as a solution to the design specification given in 4.5.1. The mechanism design progressed through three stages of development and these will be referred to as Mk I, Mk II and Mk III. The performance of the mechanism was evaluated at each stage using hardware specially designed for adapting standard gear pumps and motors. The manufacture of this hardware necessitated the development of a cam profile analysis technique which enabled complex cam profiles to be designed by digital computer.

6.2 - MK I MULTI-LIFT CAM MECHANISM DESIGN

The aims of this initial design, as shown in fig. 6.1, were threefold:-

i) To prove the principle of pressure ripple reduction by anti-flow generation without undue waste of time or resources. (If this particular avenue of research were to prove unsuccessful then it was important to have sufficient time to explore others.)

ii) To evaluate the likely increase in installational volume of a modified pump compared with a standard unit.

iii) To evaluate certain design criteria related to the basic mechanics of a cam and piston-followers, such as the limit of acceptable contact stress and the possibility of the production of any additional mechanical vibration.

A harmonic analysis of the volume ripple produced by an external gear pump shows that it consists largely of a component at pumping frequency together with two or three other strong low-frequency harmonics. In order to cancel the first four harmonics of pump volume ripple a multi-lift cam profile of complex form is required. At this stage a technique for evaluating the cam profile had not been developed. Even if it had, the production of such a cam would have required the utilization of specialized numerically controlled grinding facilities, the expense of which at this stage could not be justified.

It was therefore decided to produce a cam of much simpler design, which would attempt to cancel the dominant fundamental component of volume ripple and which could be produced by more traditional machining techniques. The displacement properties of a straight-sided cam with blending radii between the sides were shown, by graphical construction, to exhibit a strong fundamental harmonic component with a frequency which is directly related to the number of cam sides. This made the straight-sided cam particularly suitable for cancellation of the first harmonic of pump volume ripple and led to its incorporation into the MK I mechanism design.

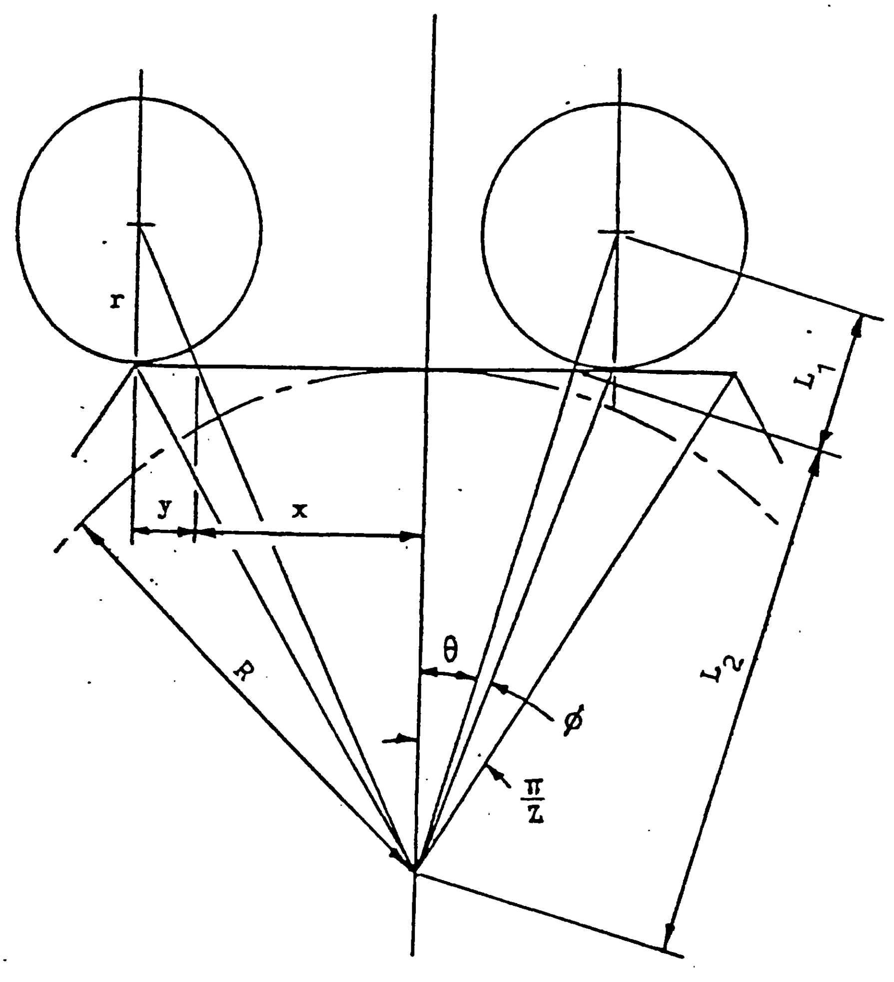
6.2.1 - ANALYSIS OF THE STRAIGHT-SIDED CAM

The purpose of the multi-lift cam is to control the displacement or lift of single or multiple pistons, such that a volume variation may be produced to cancel the volume variation produced by another source.

The fundamental harmonic component of pump volume ripple is at pumping element frequency, a period of $\frac{2\pi}{Z}$ expressed as a proportion of shaft rotational angle, where Z is the number of pumping elements. Any volume compensating pistons must also have this same fundamental period of extend and retract in $\frac{2\pi}{Z}$ of cam rotational angle. Therefore the number of sides on a straight-sided cam is the same as the number of pumping elements.

Multi-Straight-Sided Cam

A cylindrical faced piston-follower driven by a multi-straight-sided cam may be represented as follows:-



Where:-

θ - angle between perpendicular to cam face and centre of follower radius

ϕ - angle between centre of follower radius and point of contact

R - minimum radius of cam

r - radius of piston-follower

Consider the piston-follower in contact with the cam at some intermediate point between the extremes of piston displacement: minimum displacement where $\theta = 0$ and maximum displacement where $\theta = \pm \frac{\pi}{Z}$. (As shown in the right hand side of the above diagram). Then piston displacement is given by:-

$$d = L_1 + L_2 - (R+r)$$

as $R = L_2.\cos\theta$ and $r = L_1.\cos\theta$

$$d = \frac{r}{\cos\theta} + \frac{R}{\cos\theta} - (R+r)$$

or $$d = (R+r)(\sec\theta - 1) \qquad (6.1)$$

Equation 6.1 applies for values of θ in the limit

$-\frac{\pi}{Z} + \phi \leq \theta \leq +\frac{\pi}{Z} - \phi$. As $\theta \rightarrow \pm\frac{\pi}{Z}$ the piston-follower-radius pivots around the point of intersection of the adjacent sides of the cam and then the relationship is no longer valid.

Validity Limitations of Equation 6.1

Consider the point of contact to be just coincident with the intersection of the cam sides then:-

$$\theta + \phi = \pm \frac{\pi}{Z} \quad , \; x = R \tan\theta \quad , \; y = r \tan\theta$$

and $\quad x + y = R \tan (\theta + \phi) = R \tan \left(\frac{\pi}{Z}\right)$

therefore $\quad R \tan\theta + r \tan\theta \quad = R \tan \left(\frac{\pi}{Z}\right)$

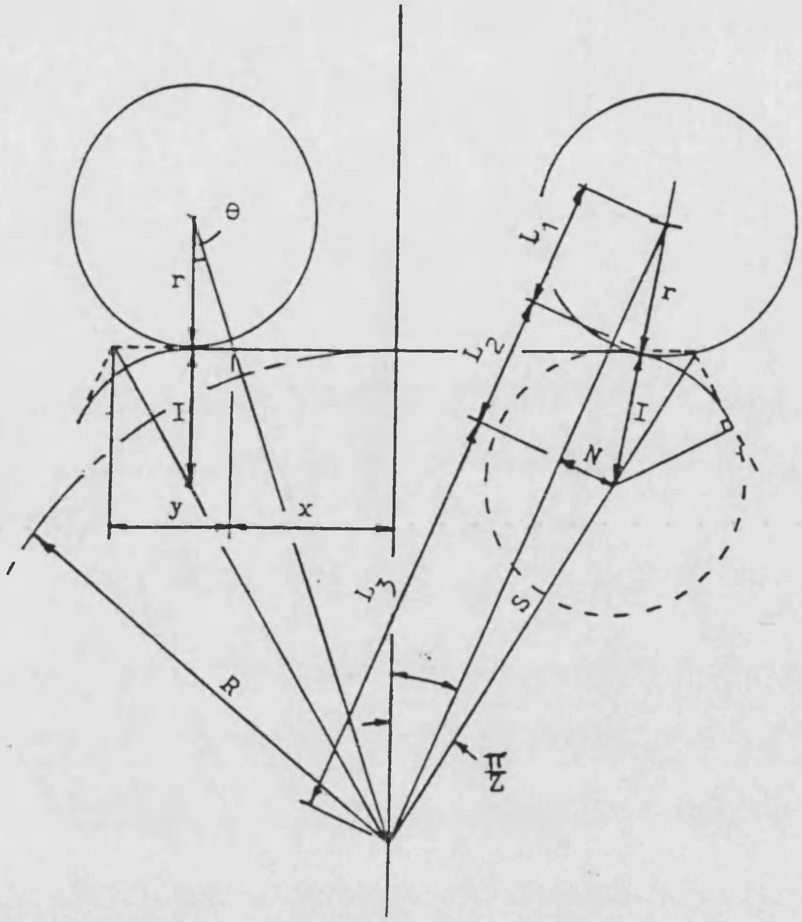
$$\theta = \pm \tan^{-1} \left[\left(\frac{R}{R+r}\right) \tan \left(\frac{\pi}{Z}\right) \right] \qquad (6.2)$$

Equation 6.2 defines the limiting values of θ for which equation 6.1 is valid. Values of θ outside these limiting values and up to $\pm \frac{\pi}{Z}$ are of no interest at this point in the analysis. In a practical application any finite load transmitted between the cam and piston-follower through a sharp edge contact (such as the intersection of adjacent sides) would produce an impracticably high level of contact stress.

Consideration must therefore be given to the situation where straight sides are connected by blend radii to make a smooth transition between the straight portions of the cam profile.

Multi-Straight-Sided Cam with Blend Radii

This arrangement is similar to the previous one and is represented to show the changes in geometry as follows:-



where:-

θ , R and r are as previously defined

I - blend radius between sides of cam.

Now consider the piston-follower to be in contact with some intermediate portion of the blend radius (as shown in the right hand side of the above diagram) then piston displacement is given by:-

$$d = L_1 + L_2 + L_3 - (R+r)$$

as $$S = \frac{R}{\cos\frac{\pi}{Z}} - \frac{I}{\cos\frac{\pi}{Z}} = \sec\left(\frac{\pi}{Z}\right)(R-I)$$

and $$N = S.\sin\left(\frac{\pi}{Z} - \theta\right) = \sec\left(\frac{\pi}{Z}\right)(R-I).\sin\left(\frac{\pi}{Z} - \theta\right)$$

and $\quad L_3 = S.\cos\left(\frac{\pi}{Z} - \theta\right) = \sec\left(\frac{\pi}{Z}\right)(R-I).\cos\left(\frac{\pi}{Z} - \theta\right)$

also $\quad (L_1 + L_2)^2 = (r+I)^2 - N^2$

therefore $L_1 + L_2 = \left[(r+I)^2 - \left(\sec\left(\frac{\pi}{Z}\right)(R-I).\sin\left(\frac{\pi}{Z} - \theta\right)\right)^2\right]^{\frac{1}{2}}$

substituting to find the displacement d gives:-

$$d = \left[(r+I)^2 - \left(\sec\left(\frac{\pi}{Z}\right)(R-I)\sin\left(\frac{\pi}{Z} - \theta\right)\right)\right]^{\frac{1}{2}} + \sec\left(\frac{\pi}{Z}\right)(R-I)\cos\left(\frac{\pi}{Z} - \theta\right) - (R+r) \tag{6.3}$$

Equation 6.3 is only valid for values of θ which produce contact between the piston-follower radius r and the cam blend radius I. The valid limits of θ may be determined as follows:-

consider the point of contact between the piston-follower-radius and the cam blend radius to be at the point of transition from straight cam profile to blend radius, (as shown in the left hand side of the diagram).

Now $\quad \tan\theta = \frac{x}{R}$

but $\quad x = R.\tan\left(\frac{\pi}{Z}\right) - y$

and $\quad y = r.\tan\theta + I.\tan\left(\frac{\pi}{Z}\right)$

therefore $\quad x = R.\tan\left(\frac{\pi}{Z}\right) - I.\tan\left(\frac{\pi}{Z}\right) - r.\tan\theta$

substituting to eliminate x

$$R.\tan\theta = R.\tan\left(\frac{\pi}{Z}\right) - I.\tan\left(\frac{\pi}{Z}\right) - r.\tan\theta$$

$$\theta = \pm \tan^{-1}\left[\left(\frac{R-I}{R+r}\right)\tan\left(\frac{\pi}{Z}\right)\right] \tag{6.4}$$

Equation 6.4 gives the lower limits for which the displacement equation 6.3 becomes valid. The upper limits of validity being $\pm \frac{\pi}{Z}$.

Summary of Displacement Equations for Straight-Sided-Cam with Blend Radii

In the limits of cam angle

$$\theta \leqslant \pm \tan^{-1}\left[\left(\frac{R-I}{R+r}\right) \tan \left(\frac{\pi}{Z}\right)\right]$$

then $d = (R+r)(\sec\theta - 1)$

$$(6.1)$$

for $\pm \tan^{-1}\left[\left(\frac{R-I}{R+r}\right) \tan \left(\frac{\pi}{Z}\right)\right] \leqslant \theta \leqslant \pm \frac{\pi}{Z}$

then

$$d = \left[(r+I)^2 - \left(\sec\left(\frac{\pi}{Z}\right)(R-I)\sin\left(\frac{\pi}{Z} - \theta\right)\right)^2\right]^{\frac{1}{2}} + \sec\left(\frac{\pi}{Z}\right)(R-I)\cos\left(\frac{\pi}{Z} - \theta\right) - (R+r)$$

$$(6.3)$$

where:- Z - number of sides (or pumping elements)

d - piston-follower displacement (from min. position)

R - minimum cam radius

r - piston-follower radius

I - cam blend radius

θ - cam angle

By inspection of equations 6.1 and 6.3 it can be seen from their complexity the reason for selecting the piston-follower and cam blend profile to be of constant curvature. Constant radii obviously makes the analysis simpler and the manufacture of the cam presents no difficulties.

The volume variations produced by the mechanism is the summation of the individual piston-follower displacements multiplied by their area.

$$V = d.A.M \tag{6.5}$$

where V - volume variation

d - piston-follower displacement

A - piston area

M - number of pistons.

6.2.2 - MK I MECHANISM DESIGN CRITERIA

A scheme of the MK I multi-lift cam mechanism installed in a modified Dowty 1P3060 external gear pump is shown in fig. 6.1. A summary of the principal criteria relating to the design of the individual components of the mechanism follows. All identifying part numbers refer to fig. 6.1.

(i) CAM (8005-7)

The cam is attached to the modified pump driveshaft 8005-4 by spigot location to ensure concentricity and squareness of cam faces to the journal diameters. A socket head cap screw provides an attachment clamping force which combined with an adhesive bond (Loctite grade 601) at the location diameter proved completely satisfactory as a means of torque transmission.

The working profile was constrained such that the assembled driveshaft and cam would pass through the journal diameter of the pump bush. The principal dimensions of the cam and piston-followers may be found in A.3.3.

(ii) PISTON-FOLLOWER (8005-10)

The cross-sectional area is of primary importance as this determines the magnitude of the volume variation produced by a given cam profile. Close tolerancing of the outside diameter is important to achieve a spool type fit with the bore in which it operates. The diametral clearance between the piston and bore must be kept small to minimise cross piston leakage and the loss in pump volumetric efficiency. A maximum diametral clearance of 0.025 mm may be maintained by external grinding of the piston and reaming of the bore.

The cylindrical surface which contacts the cam profile must be controlled not only in terms of its radius of curvature, but also in its squareness and symmetry with the external piston diameter. Control of follower contact surface orientation is maintained parallel with the cam face by means of a slot which locates on a garter type spring 8005-11. The peripheral spring also performs a returning function when the mechanism is operating against a mean fluid pressure which is insufficient to overcome acceleration and friction forces.

(iii) PISTON HOUSING (8005-8)

The primary functions of the piston housing are to provide a means of distributing a number of radially oscillating piston-followers around the circumference of the cam and allow the relative radial positions of the piston-followers to be adjusted with respect to some reference point. This adjustment is a means of changing the relative phase of the volume ripple produced by the mechanism and is explained in detail in 6.2.4.

The squareness of the radial bores to the cam surface and therefore to the journal bearings is important. Concentricity of the housing with the driveshaft is not so critical as an inherent feature of the design is the automatic compensation for small driveshaft eccentricities. These movements of the driveshaft are caused by variation in gear tooth loading and pressure distribution, moving the driveshaft journal within the clearances of the bush bore.

The piston-housing, when assembled into a simple bore in the pump body extension 8005-3, also forms an annular shaped cavity through which the volume variations, produced by the individual piston-followers, are conveyed via a short drilled passage to the pump outlet port. This connection port in the body extension makes this arrangement component-handed for either direction of driveshaft rotation. The piston housing is restrained from rotating by axial interference between the pump body extension and the end cover 8005-5.

To enable the modified pump to be quickly converted back to its standard mode of operation, a blank piston housing, not shown was substituted for the piston housing, spring and piston-follower assembly.

(iv) MATERIALS

The material chosen for the cam and piston-followers is case hardened steel (EN36B case hardened 0.75 mm deep 750 to 850 HV) the same as that used for the driveshaft and driven gears. The gear teeth experience high relative surface velocities and contact stresses so it was decided for consistancy and economic reasons to try this combination of surface in the cam/piston environment. As the contact stress reduces with decreasing mean pump pressure it would always be possible to verify the working principle of the device at low mean pressures, should it not be capable of operation at the maximum operating pressure of the pump. Further discussion of the Hertz stress criterion for contact between the cam and piston-followers is given in chapter 8.

6.2.3 - PUMP MODIFICATION DESIGN CRITERIA

The design of a standard Dowty 1P3060 external gear pump required the following modifications to incorporate the multi-lift cam mechanism. Descriptions are again with reference to fig. 6.1.

(i) PUMP BODY (8005-2)

Only two minor modifications were necessary. The first was the addition of two hollow dowel recesses around the body through-bolt holes. The dowels were required to allow precise alignment of the pump body extension (8005-3) with the pump body. The two extra dowel recesses are a standard feature of tandem pump front bodies and hence add a minimal cost penalty. The dowels are not shown in fig. 6.1 but may be seen in the photograph fig. 6.1a, which gives an expanded view of the later MK II design.

The second modification involves removal of body material in the area of the cusp on the high pressure side, also shown in fig. 6.1a. This was done to provide an unrestricted path for flow fluctuations from the mechanism to the meshing gear teeth.

The amount of cusp removal coincides with an extension of the access port to the annular volume in the body extension and extends the full length of the body. This is clearly visible in the photograph fig. 6.1b showing an assembled unit made primarily in clear perspex. The increased cusp removal was thought to have a negligible effect on body strength and modification could be achieved either by machining or pattern and die changes for cast and extruded bodies respectively.

(ii) PUMP BODY EXTENSION (8005-3)

The pump body extension fulfils all the functions of the standard end cover. It provides a contact face, for the body and bush balancing seals, the surface finish, flatness and rigidity of which must be maintained.

The body extension also allows the protrusion of the extended driveshaft through this sealing face and locates the piston housing square and concentric to the cam. Three elastomeric "O" ring seals contain high pressure fluid to the annular volume between the piston housing and the location bore in the pump body extension.

The installation volume available for the addition of the mechanism was intentionally limited by the requirement to minimise the increase in pump overall length. The position of the four pump through bolts proved to be the limiting factor which determined the overall piston housing diameter and the length of the pump body extension. Changing the spacing of the pump through bolts would mean major changes to production tools, fixtures and dies for the mounting flange, body and end cover. This was considered unacceptable.

An alternative to the bolts passing through the pump body extension and therefore limiting the diameter of the piston housing is to increase the body extension so that the bolts may be threaded into the body extension and not have to protrude as far as the piston housing. The required thread depth, tap run out and tapping drill clearance with the high pressure annulus would however increase the length of the body extension and therefore the modified pump by 25 mm over that shown in fig. 6.1. This was also considered unacceptable as the modification, when applied to the smallest pump in the 1P3000 range, would roughly double the body length. It was therefore decided, in this initial study, to develop the mechanism design within the existing bolt pattern.

Manufacture of the pump body extension in this form may easily be achieved by modification of a standard tandem pump centre-section.

(iii) PUMP DRIVESHAFT (8005-4)

Attachment of the MK I cam to the pump driveshaft was described in 6.2.2 (i). The modification consisted simply of facing the journal square to its diameter, to allow a square pick up for the cam and to penetrate the hardened case of the journal. The driveshaft modification was completed by a reamed location diameter, concentric with the journal diameter and a central thread to provide an induced load for cam attachment via a socket head cap screw.

(iv) END COVER (8005-5)

The end cover has the primary function of providing a sealing surface for two face mounted elastomeric "O" ring seals. These prevent high pressure fluid leaking to the low pressure side of the pump or to atmosphere. Face mounting seals are used because of the inherent exclusion of an extrusion gap; a desirable feature in an environment of high mean and fluctuating pressure.

The end cover is made in mild steel of a thickness that is sufficiently stiff to withstand the clamping load imposed by the pump through bolts and the resultant end load produced by fluid pressure on the piston housing without an excessive increase in pump overall length. Fig. 6.2 shows the resultant increase in overall length due to the addition of the mechanism in comparison with a standard unit.

6.2.4 - VOLUME VARIATION PRODUCED BY MK I MECHANISM

The volume variation produced by the MK I multi-lift cam mechanism is determined by substitution of the principle dimensions, shown in A.3.3, in equations 6.1, 6.3 and 6.5 from section 6.2.1. For one complete cycle of piston-follower motion, corresponding to a cam angle interval of $-\frac{\pi}{8} \leq \theta \leq \frac{\pi}{8}$, the peak to peak volume variation produced is 71.13 mm^3. The instantaneous values of volume variation with cam rotational angle are shown in fig. 6.3.

The theoretical volume variation of a Dowty 1P3060 external gear pump, as given by equation 3.2 of section3.3.3 is also shown for comparison. The two curves can be seen to be of similar form but are displaced approximately one quarter of a cycle. The phase relationship between the two curves is a result of the phase references defined in the analysis of the equation representing each curve.

For the pump, zero angle coincides with the mean volume or zero volume variation when the point of mesh of the gears is on the pitch point. For the cam, the angular datum is the point of minimum volume variation.

The volume ripple of the combined pump and mechanism is represented by the summation of the instantaneous amplitudes of each curve at all positions through-out the cycle. Cancellation of pump volume ripple can therefore be achieved by adjustment of the relative phase difference between the individual variations. A phase difference of 11 degrees was found to produce maximum cancellation at pumping frequency. The two curves and their resultant at the condition are shown in fig. 6.4

The resultant volume variation is of considerably reduced amplitude and at fundamental frequency twice that of the original variations. The resultant variation shows a strong component at the second harmonic of pumping frequency because of the obvious absence of the fundamental due to cancellation.

A displacement in mean value of the resultant variation is also apparent. This is due to the volume ripple produced by the mechanism being asymetrical about the zero angle reference of the graph. The lower right hand part of the curve (fig. 6.4) is produced by piston-follower contact with the straight portion of the cam. The upper or left hand side is the result of contact with the curved part of the cam. The displacement characteristics of the straight and curved parts are obviously not the same and the result is effectively a shift in the amplitude axis of the mechanism volume variation. This shift is also seen in the resultant or residual profile.

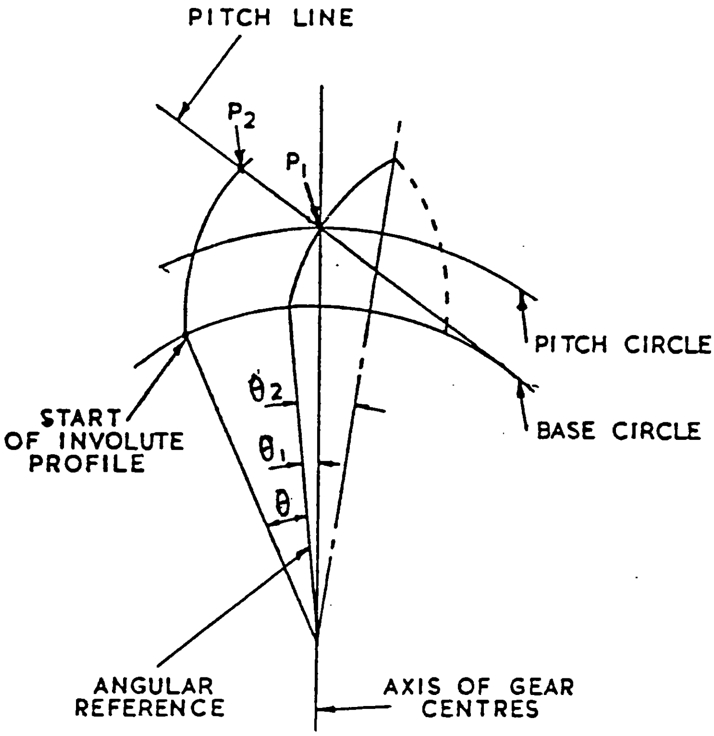
Because of the differences between the pump and mechanism volume ripples at frequencies other than the first harmonic, cancellation or reinforcement will inevitably occur at other harmonics.

The phasing of the volume variations produced by the pump and the mechanism to achieve cancellation of the pump fundamental is described in the following section 6.2.5.

6.2.5 PHASING PUMP AND MECHANISM VOLUME VARIATIONS

The building of a pump and mechanism assembly with fundamental volume ripple component cancellation requires precise orientation of the cam mechanism with respect to the gear teeth. In 6.2.4 it was shown that an 11 degree difference in phase reference between pump and mechanism volume variation is required to effect first harmonic cancellation. To establish this condition for a particular set of hardware it is necessary to examine in detail the angular references assumed in the analysis of the gear and cam mechanisms.

In the derivation of the expression relating the volume variation of external involute spur gearing, equation 3.2, the angular reference, which corresponds to the middle of the tooth cycle and zero amplitude of volume variation, is the start of active involute profile on the base circle. This is shown in the following diagram.

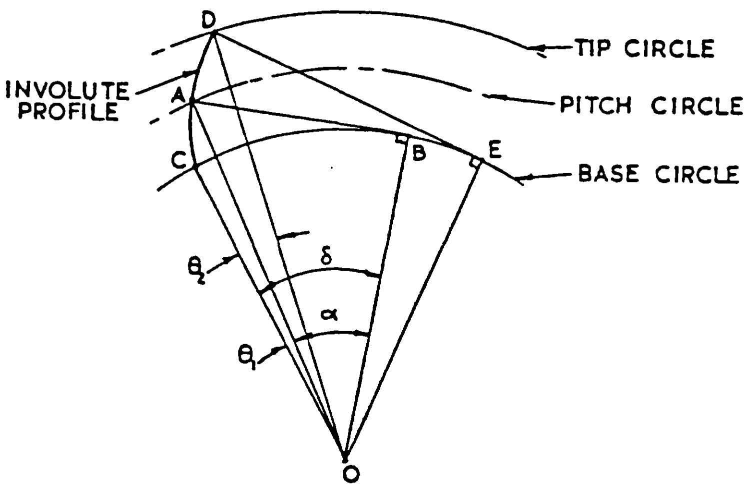


The angular reference is the line drawn from the gear centre to the starting point of involute profile on the base circle, when the two gears are in mesh at the pitch point (P_1).

The angular rotation of the gear (θ) is measured from this line for a mesh point movement along the pitch line from (for example) P_1 to P_2.

To locate this reference line with respect to a dutum which is independent of gear motion, for instance the axis of the gear centres, it is necessary to evaluate the angle θ_1. It is also necessary to locate this reference line with respect to some more easily identifiable point on the gear profile in order to allow accurate orientation of the cam. A convenient point is the tooth centre line and therefore angle θ_2 is also required.

To evaluate θ_1 and θ_2 it is necessary to consider the geometry of the involute gear profile. For the Dowty 1P3000 design the arrangement is as follows:-



In the above diagram let point A be at the pitch point then from Δ ABO

$AB^2 = AO^2 - BO^2$ where AO = Pitch circle radius

BO = Base circle radius.

$$AB = (15.875^2 - 13.747^2)^{\frac{1}{2}}$$

<u>AB = 7.939 mm.</u>

Now a fundamental property of the developed involute is that

$$AB = CB = 7.939 \text{ mm}$$

therefore from sector CBO

$$\angle \delta = \frac{CB}{BO} \cdot \frac{360}{2\pi} = \frac{7.939}{13.747} \cdot \frac{360}{2\pi}$$

$$\therefore \underline{\delta = 33.088^\circ}$$

From Δ ABO

$$\angle \alpha = \tan^{-1} \frac{AB}{BO} = \tan^{-1} \frac{7.939}{13.747}$$

$$\therefore \underline{\alpha = 30.006^\circ}$$

now $\underline{\theta_1 = \delta - \alpha = 3.081^\circ}$

Similarly θ_2 can be evaluated by moving point A to point D.

Now DO = theoretical tooth tip radius = 19.964 mm as before

$$DE = (19.964^2 - 13.747^2)^{\frac{1}{2}}$$

$$\underline{DE = 14.477 \text{ mm}}.$$

then $\angle\alpha$ to the point D i.e. $\angle\alpha D$

$$\alpha D = \tan^{-1} \frac{14.477}{13.747}$$

$$\alpha D = 46.482^\circ$$

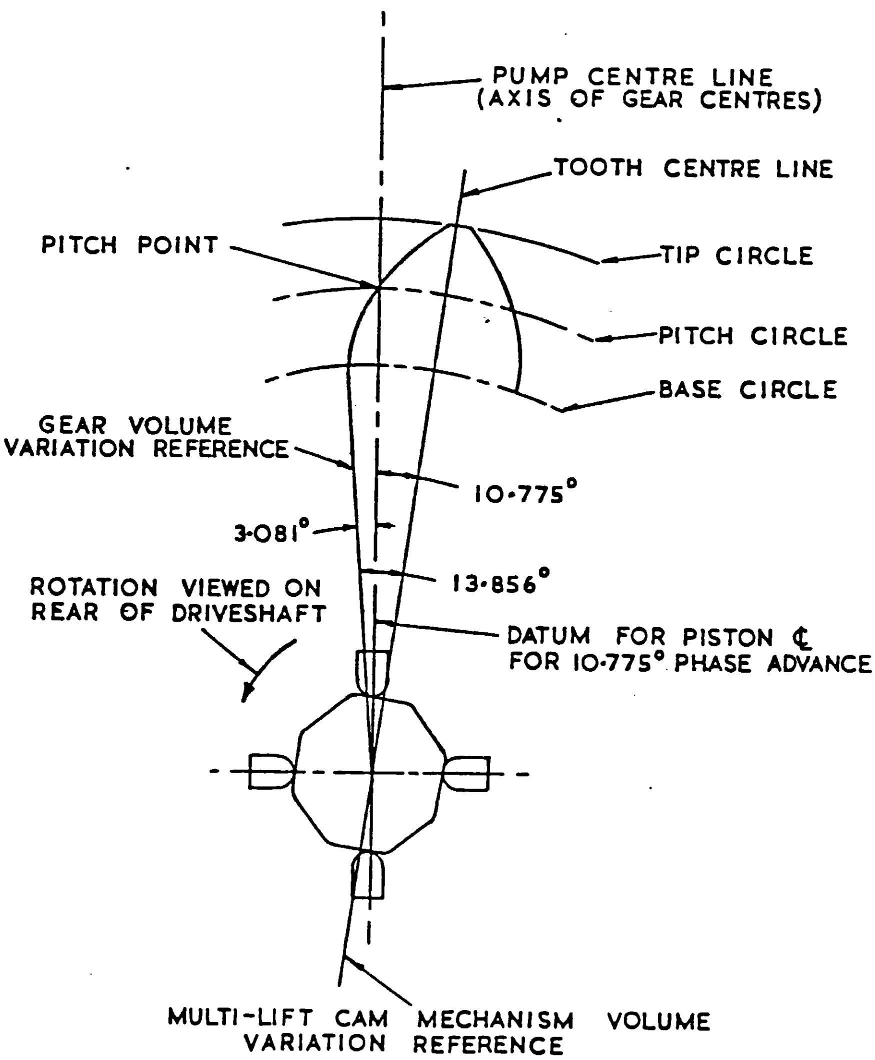
and $\theta_2 = \text{inv}\,\alpha D = \text{inv } 46.482 = 13.856^\circ$

$$\therefore \underline{\theta_2 = 13.856^\circ}$$

If the difference ($\theta_2 - \theta_1$) is considered, then

$$\theta_2 - \theta_1 = 13.856^\circ - 3.081^\circ = 10.775^\circ$$

then it will be seen that this is very close to the estimated 11° phase advance required between the two volume ripples. This means that if the cam zero reference is set coincidental with the centre line of a tooth profile and a piston centre line is set coincident with the line joining the two gear centres, the pump and mechanism are phased for fundamental volume ripple cancellation. This is clearly shown in the following diagram:-



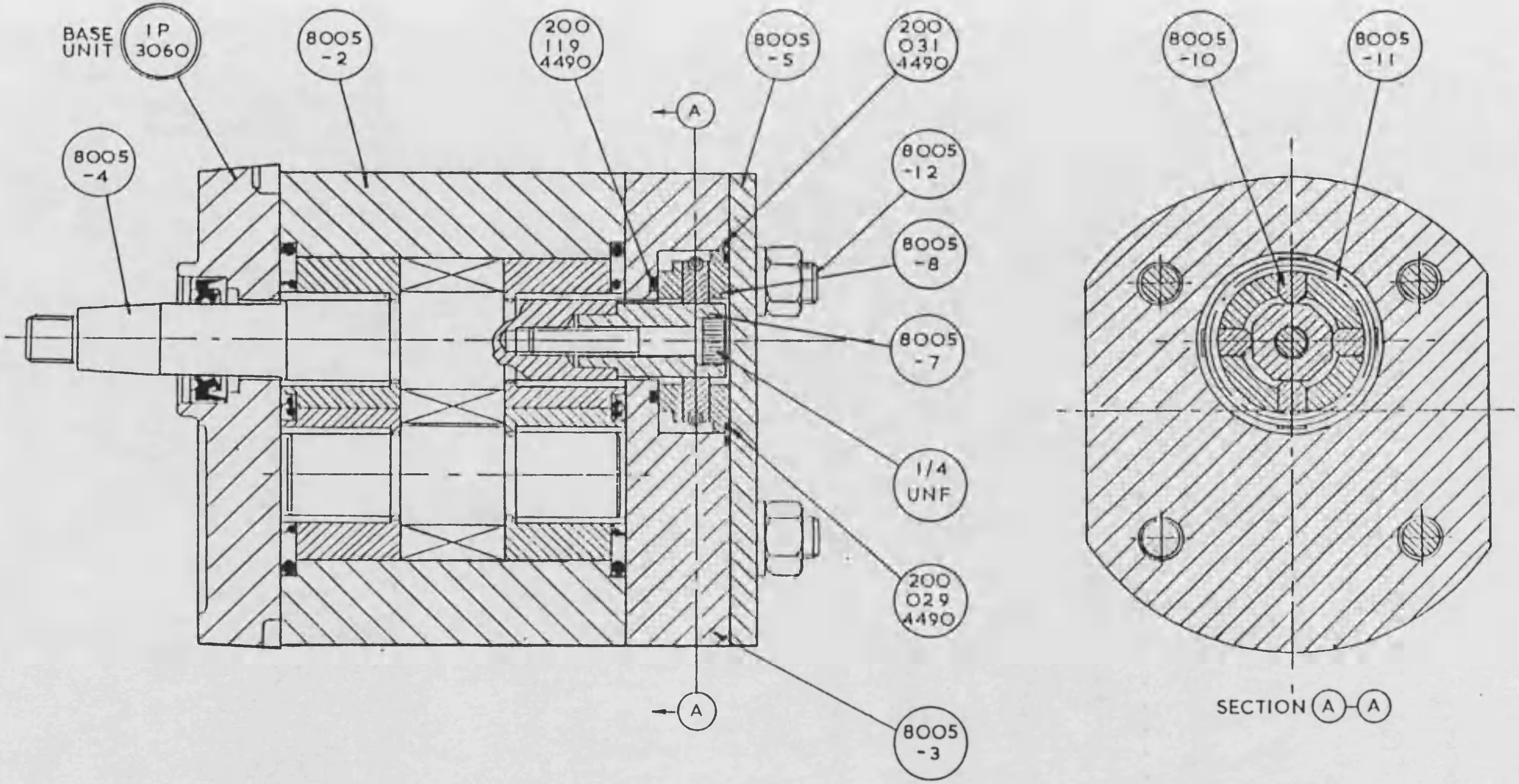


FIG. 6.1 SCHEME OF IP3060 WITH MKI MULTI-LIFT CAM MECHANISM

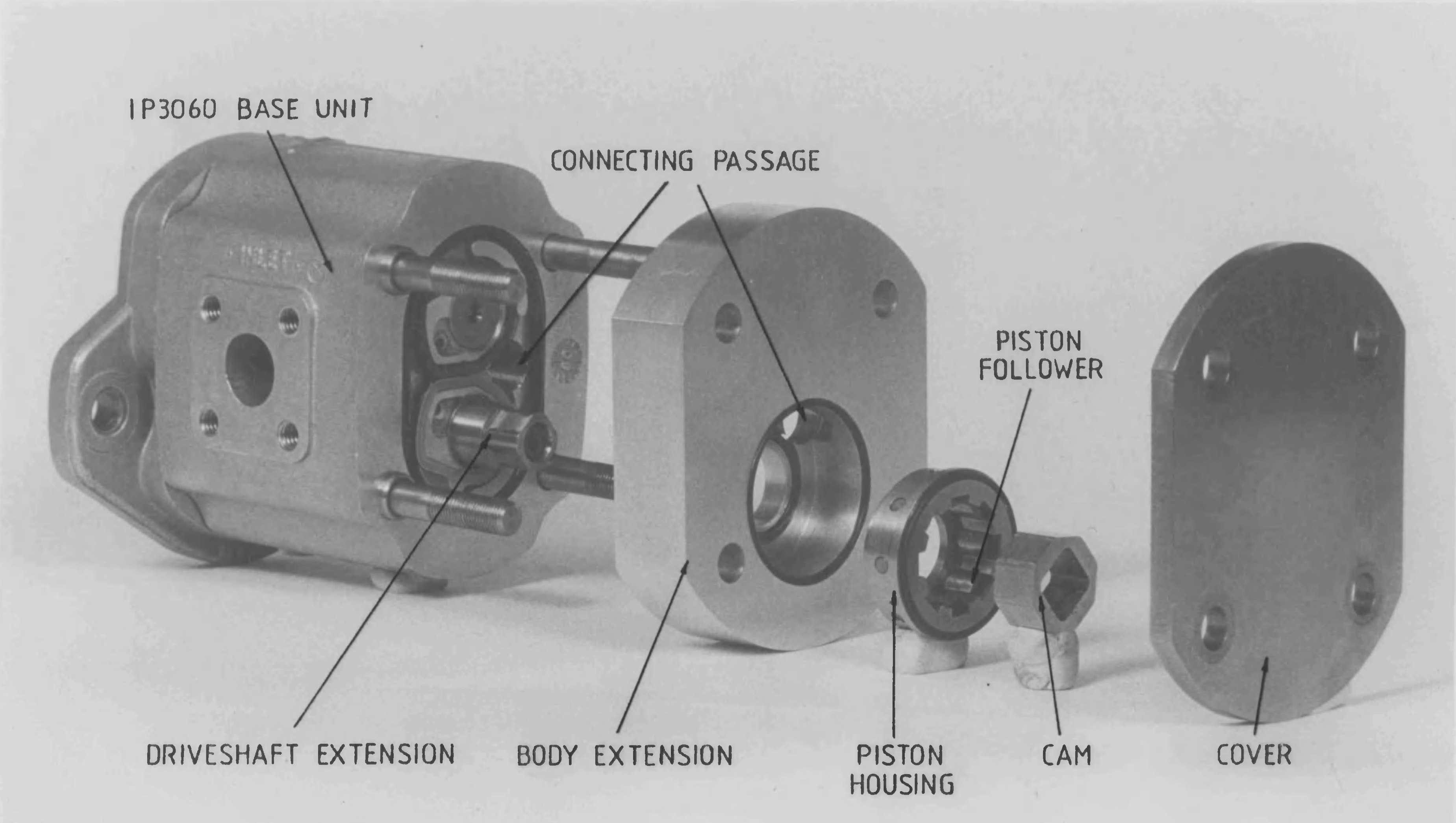


FIG. 6.1a PHOTOGRAPH SHOWING EXPANDED VIEW OF MODIFIED PUMP

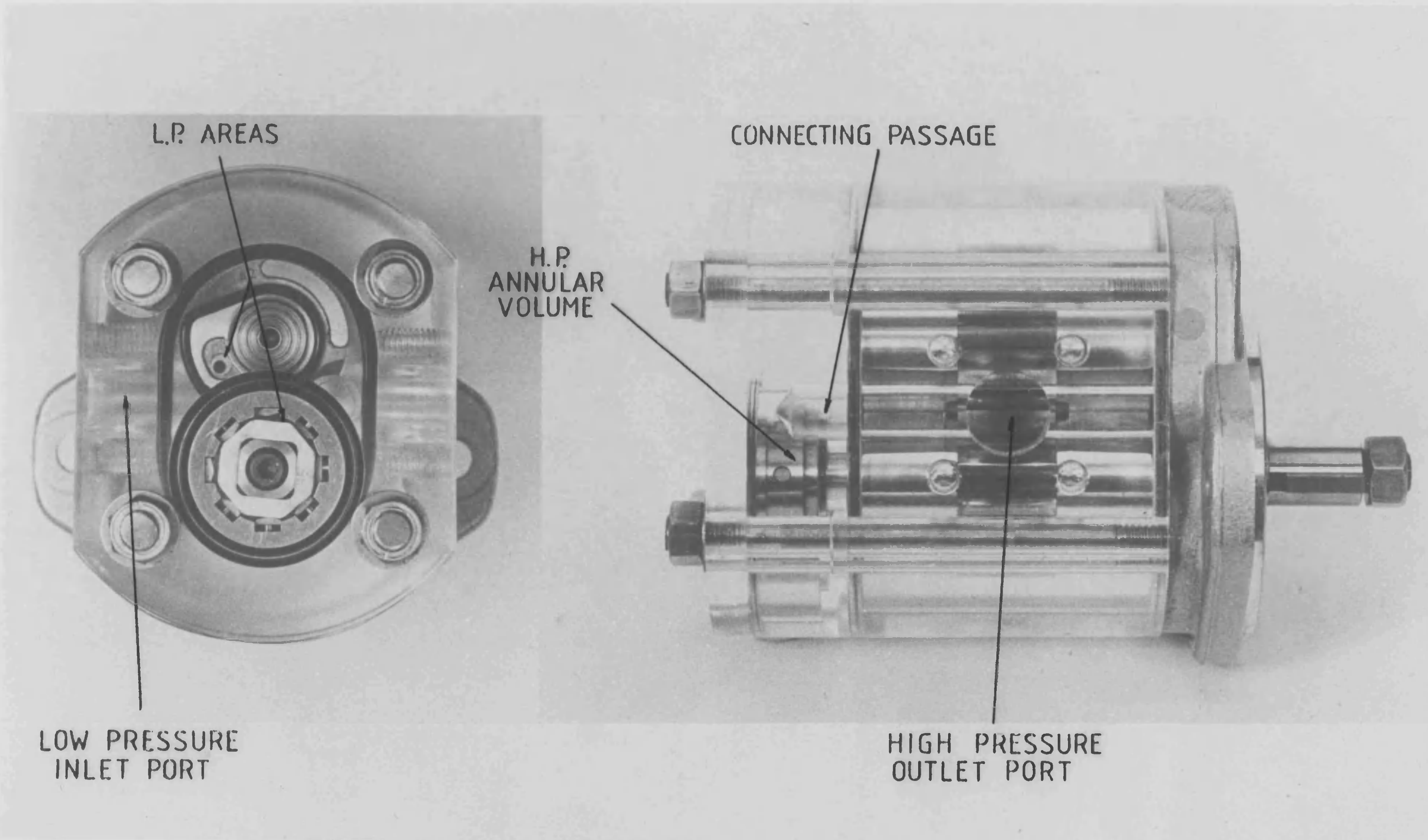
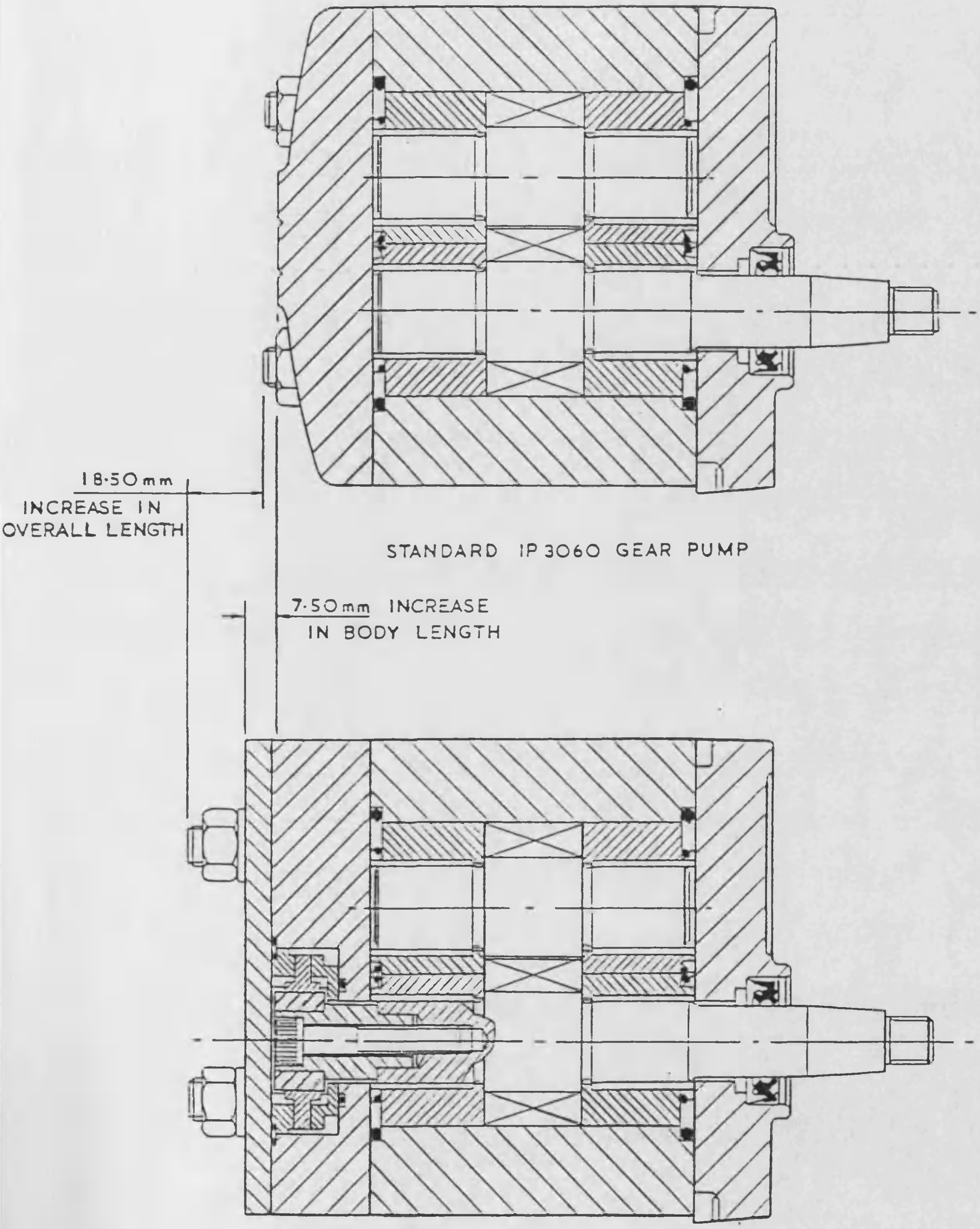


FIG.6.16 PHOTOGRAPH SHOWING VIEWS OF A MODIFIED PUMP MADE IN PERSPEX



IP3060 GEAR PUMP WITH MK III MULTI-LIFT CAM MECHANISM

FIG. 6.2 INSTALLATIONAL LENGTH COMPARISON OF STANDARD AND MODIFIED IP3060 GEAR PUMPS

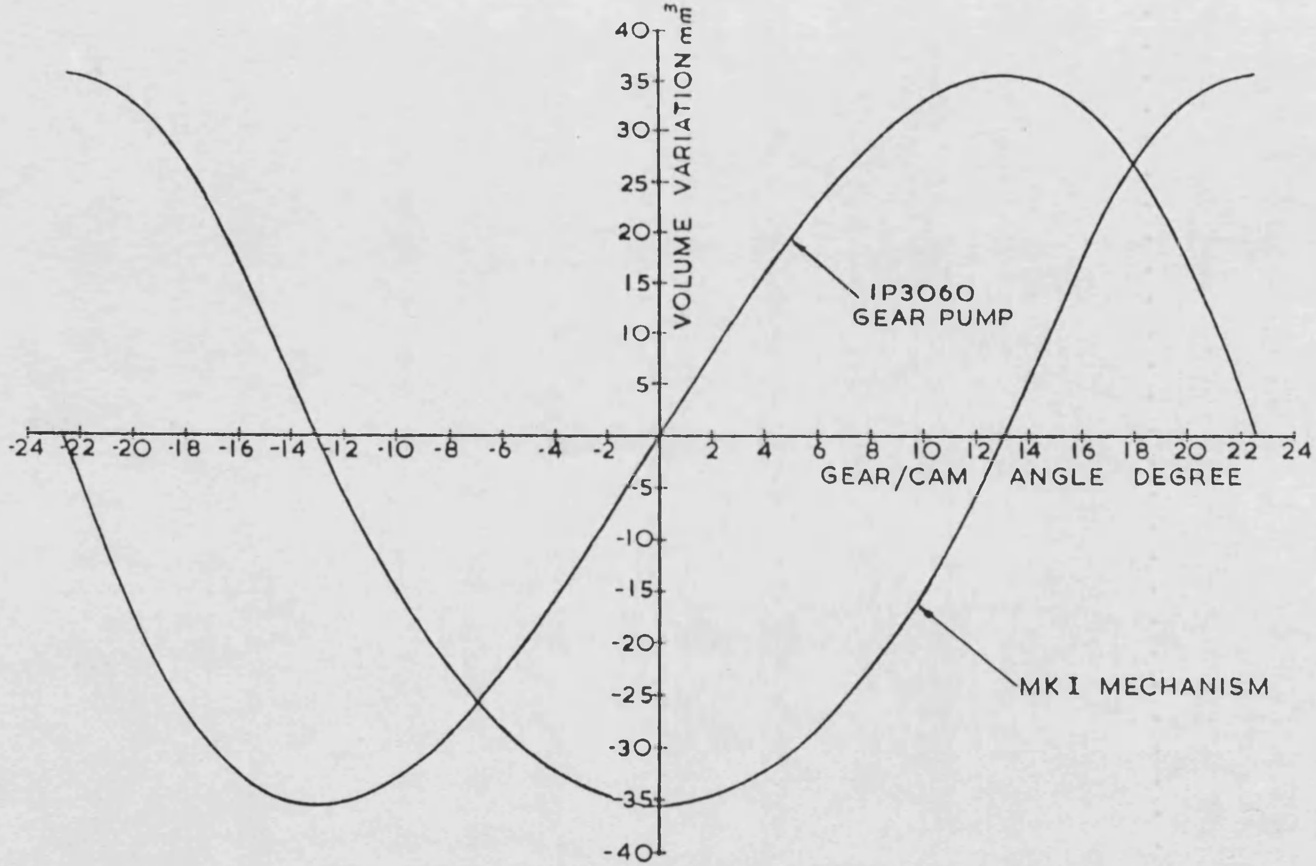


FIG.6.3 COMPARISON OF IP3060 AND MK I MULTI-LIFT MECHANISM THEORETICAL VOLUME VARIATIONS

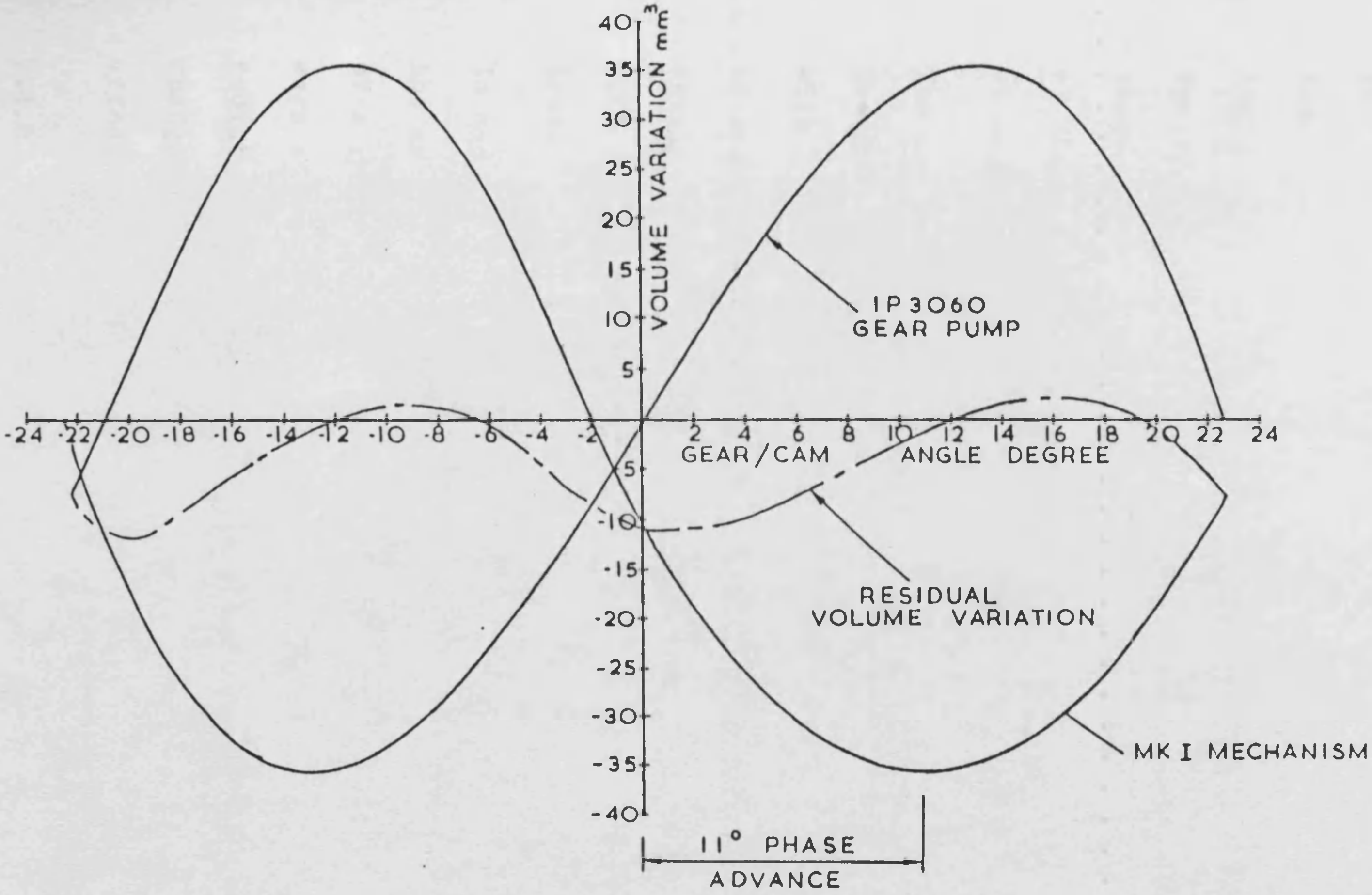


FIG.6.4 INTERFERENCE BETWEEN GEAR PUMP AND MK I MECHANISM THEORETICAL VOLUME VARIATIONS

CHAPTER 7 - TEST RIG AND PRELIMINARY TEST RESULTS

7.1.1 - SCOPE OF EXPERIMENTAL WORK

The experimental work was carried out in order to achieve two basic objectives. The first was to evaluate the flow ripple characteristics of a test pump over its normal range of operating conditions. The evaluated characteristics were then compared with those produced by the theoretical model of the pumping mechanism and used to specify the requirements of a dedicated pump-flow ripple reduction device.

The second objective was to establish the reduction in pump pressure ripple which could be achieved by modifying a pump with a suitable flow-ripple reduction device. The performance of such a device was examined, at different stages of its development, by comparing test results obtained for the pump with and without the device. In this way steady state and dynamic effects could be studied.

In addition to pump flow and pressure ripple, the effect of the mechanism on pump volumetric efficiency and for the case of a motor, torque fluctuation and mechanical efficiency were also investigated. The levels of airborne noise radiated by a hydraulic system, before and after significant changes in fluid borne noise levels in the system were affected, were also compared.

7.1.2 - DESCRIPTION OF TEST RIG

7.1.2 (i) - TEST STAND ASSEMBLY

The test pump was supported by a traditional 'L' shaped mounting bracket which was rigidly attached to the same baseplate as a Crompton-Parkinson 3-phase electric motor with a nominally constant speed of 1500 rev/min. The drive between the test pump and the electric motor was via a U.C.C. drive coupling, consisting of two sintered steel gear hubs engaging in an internally toothed nylon sleeve. This type of coupling was used to isolate the pump driveshaft from sideloads due to errors in driveshaft alignment.

The motor driveshaft incorporated a toothed wheel which was used in conjunction with a proximity detector (magnetic pick-up) to produce a reference signal at pumping element frequency. As this signal may be noisy or of low amplitude, a schmitt trigger network was used to convert the signal into a buffered square wave of the same frequency. This was then used as a frequency and phase reference for harmonic analysis of pump pressure ripple.

A typical set up of the test rig is shown in fig. 7.1. The test stand and reservoir were used in all the tests performed. The only significant changes occur in the pump loading circuit.

7.1.2 (ii) - RESERVOIR

The reservoir was capable of supplying oil at carefully controllable mean pressures between 0.4 and 11 bara. This was achieved by pressurizing the tank with air from a mains air supply. Specified oil temperatures could be maintained automatically to within $\pm$ 2^{o}C by two oil to air heat exchangers. The removal of entrained air was encouraged by a stabilizing tank and particulate contaminate down to 10 μm removed by a full flow return line filter. The oil level was automatically maintained by a make up pump controlled by a float and two level detectors.

The reservoir was in fact specifically designed to facilitate the investigation of pressure fluctuations in pump inlet lines and a full description is given in ref. (17)

7.1.2 (iii) - LOAD CIRCUITS

A number of different loading circuits were used in conjunction with the test pump. Three of these correspond to specialized test methods developed for specific purposes and these are described in 7.1.4.

7.1.2 (iv) - RETURN LINE

The return line to the reservoir consisted of 1.0 inch nominal bore flexible hose. When measurements of mean flow were required a Brodie-Kent positive displacement flow meter was included in this line.

7.1.3 - INSTRUMENTATION

The measurement of pressure fluctuations is of fundamental importance to the study of fluid borne noise in hydraulic systems. The accuracy and reliability of any test method is dependent upon the close control of steady state parameters and the quality of the pressure data acquired.

In general, good repeatability of fluid borne noise measurements is achieved if pump speed, mean flow and mean pressure are maintained within $\pm$ 2%. The mean oil temperature should also not vary by more than $\pm$ 2°C or changes in bulk modulus and hence the speed of sound in the fluid will introduce inaccuracies.

The quality of pressure data is very much dependent upon the use of suitable measuring and analysis equipment. Fig. 7.1a shows a view of the instrumentation used throughout this work.

7.1.3 (i) - DYNAMIC PRESSURE TRANSDUCERS

The measurement of pressure fluctuations may be required over a large range of mean pressures and temperatures, typically up to 400 bar and 120°C, and in environments subjected to mechanical vibration. A transducer is required which is insensitive to these mean variations, yet has good linearity and resolution, low hysteresis, high frequency response and excellent repeatability. These requirements are adequately fulfilled by a piezoelectric pressure transducer. Such a transducer is constructed of small discs of quartz, which has excellent piezoelectric properties, sandwiched between a thin metal diaphram and the transducer body. The quartz discs produce an electrical charge in proportion to the force and hence the pressure applied to the diaphram. The charge produced decays with time so that only pressure variations and not mean pressure levels can be measured. In this application this is an advantage because it means there is no reduction in sensitivity when measuring fluctuations about a high mean pressure, especially if the fluctuating component is small compared to the overall pressure level.

A full description of the piezoelectric pressure tranducers used throughout this experimental work is given in Appendix 3. These particular transducers were very small, with a 6 mm installation thread, which makes precise positioning, flush with the pipe inner wall, possible. Their small size also makes them easily susceptible to damage and they are expensive.

The transducers were used with charge amplifiers so that the transducer calibration is conveniently independent of the length of the connecting cable. However care must be taken to avoid mechanical vibration to the cable as this can affect its capacitance and interfere with the measurements. The output from the amplifier is a voltage which may be analysed in the time or frequency domain.

7.1.3 (ii) - SIGNAL ANALYSIS

Time domain analysis of the pressure signals was done with a two-channel Gould-Advance digital storage oscilloscope. By using the instruments digital storage capability and a plotter interface it was possible to record pressure wave forms. Pressure information in this form is of limited analytical use as most theoretical modelling and evaluation is done in the frequency domain.

Frequency domain analysis of the signals was carried out using continuous and discrete frequency analysers. A spectrum analyser (Hewlett-Packard model 3582 A) was used to display a full amplitude spectrum of one or two signals simultaneously. This instrument also had a storage and plotting capability and was very useful, because of its ability to show, very quickly, amplitude and frequency changes caused by modifications in test conditions or hardware. This was particularly useful in analysing signals which contained harmonic components from two separate sources at different frequencies. This is the case for a constant speed pump driving a motor of variable speed. Signals from a torque transducer were also analysed in this way.

Discrete frequency analysis was done with a Fourier Harmonic Analyser (Solartron FRA 1170) which measured the first ten harmonic components of fundamental pumping frequency in terms of amplitude and phase.

7.1.3 (iii) - DATA ACQUISITION

The amplitude and phase of harmonic components of pressure wave forms tend to fluctuate around a mean value due to slight variations in the pressure wave form. Measurement errors were reduced by averaging five results for each harmonic. For tests which require amplitude and phase information from four separate transducers for up to nine different line lengths, for a range of different mean pressures and speeds, the volume of data to be read is enormous and by hand would be unacceptably time consuming. To speed up the process of data acquisition the harmonic analyser was connected to an on-line computer (PDP-8/E). This permitted fast acquisition, automatic averaging, storage and printing of data.

7.1.4 - SPECIALIZED TEST METHODS

The pump and motor fluid borne noise characteristics, discussed later in this chapter, were evaluated by specialized test methods developed at the University of Bath. The product of pump outlet flow fluctuation and source impedance amplitudes, $|Qs.Zs|$, was determined using the High Impedance pipe method. These parameters were individually evaluated, in terms of amplitude and phase using the Trombone Method, (ref. 27). The Tuned Length Method (ref. 17) was used for the similar evaluation of pump inlet and outlet, and motor inlet parameters. A brief description of these methods will now be given.

7.1.4 (i) - DESCRIPTION OF HIGH IMPEDANCE PIPE TEST

The high impedance pipe test required the measurement, at the pump outlet flange, of the amplitude of the first ten harmonics of pressure fluctuation generated by the pump while connected to a high impedance loading circuit. A circuit diagram of the test rig used is shown in fig. 7.2.

The loading circuit consisted of a pump port adaptor, a straight length of small bore pipe and a load valve adaptor. The basic requirements of the load circuit are shown in fig. 7.3. The two adaptors allow connection of the pipe to the test pump and a restrictor valve to adjust the mean test pressure.

To measure ten harmonics of a particular fundamental frequency required five different pipe assemblies. Four of these were of the same internal diameter and different lengths and a single assembly of larger diameter, and equal in length to the longest of the other four, was used as a test for the correct operation of the method. The diameters and lengths used to test an external gear pump (pump A) at 1500 rev/min. are shown in the following table:-

HARMONIC	LENGTH (m)	INTERNAL dia (mm)
1	1.701	4.0
3	"	AND
5	"	4.5 (CHECK)
7	"	
9	"	
2		
6	0.850	4.0
10		
4	1.275	4.0
8	1.488	4.0

The determination of high impedance pipe assembly dimensions and a full description of the test method may be found in ref. (19).

To avoid difficulties, during cold start-up conditions due to high pressure losses over long, small diameter pipes a special port adaptor was designed. (fig.7.4).

In addition to allowing access for mean and fluctuating pressure measurement it allowed flow to by-pass the high impedance pipe through a needle valve back to tank. This was necessary so that the electric motor could be started off load; without the pump discharging fluid through the restrictive high impedance pipe. This would not have been necessary if the motor had been of the variable speed type. Inclusion of a cartridge relief valve (sterling type A3A125) protected against over pressurizing the test pump.

7.1.4 (ii) - DESCRIPTION OF TROMBONE TEST

The Trombone test utilizes a loading circuit, the effective length of which may be easily changed. This is achieved by a piston-headed section of pipe sliding inside a fixed tube (hence the name) in which are mounted four pressure transducers. A screw down restrictor valve is positioned at the end of the sliding section to allow variation of the mean test pressure. Fig. 7.5 shows a typical test circuit and a layout of the instrumentation used.

After running the pump to stabilise the oil temperature and pressure at the pump inlet a series of measurements of pressure fluctuations were taken for different line lengths. With the sliding piston of the trombone in the fully extended position (longest discharge line condition) the amplitude and phase of the pressure fluctuations at each of the four positions along the discharge line were recorded for ten harmonics of pumping frequency. This was performed at a number of mean pressures up to 200 bar. These measurements were then repeated for a further eight positions of the sliding piston (discharge line lengths).

The data collected from these tests were fed into a digital computer simulation of the system to evaluate the amplitude and phase of Q_s and Z_s of the pump at the pump outlet flange.

7.1.4 (iii) - DESCRIPTION OF TUNED LENGTH TEST

The tuned length test method required pressure fluctuation information from two transducers, positioned close to the source and termination, for eight different line lengths. In this case the various line lengths were achieved by joining together, with union connectors, different pre-cut lengths of steel tubing.

Fig. 7.1 shows a photograph of a test pump connected to a gear motor loaded with a fan. By running the two units at different speeds and collecting pressure data based on their individual fundamental frequencies, each could be used as a source and termination for the other. In this manner the outlet characteristics of the pump and the inlet characteristics of the motor were evaluated.

The tuned length method was also used to evaluate the inlet characteristics of a pump using the reservoir as the termination. This was the purpose for which the method was originally developed. The computer simulation, into which the collected pressure data was fed, was specially designed to detect erroneous data and indicate where additional or replacement data was required for satisfactory results. This was found to be a necessary development in order to model pump suction lines where the levels of pressure fluctuation are often very low and subject to variation because of the presence of air. A full account of the method can be found in ref. (17).

7.2 - DISCUSSION OF EVALUATED SOURCE FLOW CHARACTERISTICS OF AN EXTERNAL GEAR PUMP

In chapter 3 a theoretical model for the output flow fluctuations of an external involute gear pump was developed and subsequent references to flow fluctuation have all implied variations in the pump discharge flow. The gear pump however is a symmetrical unit, in as much as the pumping mechanism geometry causes inlet and outlet flow variations to occur which are ideally of equal amplitude but opposite in sense.

For reasons which will become clear in the following discussion the flow variations in outlet or high pressure fluid flow are of most significance and the greatest effort has been directed toward their reduction.

7.2.1 - PUMP INLET FLOW FLUCTUATION

Presented in fig. 7.6 are the inlet and outlet flow fluctuations for an external gear pump (pump A) running at 1500 rev/min. and various mean pressure levels.

The inlet flow ripples, evaluated by the Tuned Length Method, show a distinct change in form with increasing mean inlet pressure (the outlet pressure was constant at 50 bar and the inlet oil temperature was $50^{\circ}C$).

At 0.0 bar gauge the pump flow ripple appears to be almost non-existent. This of course cannot be the case because the geometry of the gear teeth coming out of mesh must be creating flow variations similar to that shown by the theoretical. What in fact is happening is that the flow ripple, at this suction condition, is being smoothed out or damped by air being released from the fluid. The resulting low level of flow ripple produces a correspondingly low level of pressure ripple to be propagated in the inlet line and the test method evaluates the effective flow ripple at the pump flange.

As the mean inlet pressure is increased the amount of air released from the fluid decreases so that the levels of pressure ripple and evaluated flow ripple also increase. At 1.0 bar the periodic form indicated by the theoretical is just established. At 2.0 bar the wave form is fully developed and shows variations in flow characteristic of the involute gear tooth geometry, but modified by other additional effects. These effects are due to leakage from the higher pressure outlet side of the pump into the inlet volume, particularly the expansion of the high pressure "carry over" volume trapped between meshing teeth.

Tests on a similar pump (pump B) using partially de-aerated oil showed that increased levels of pressure ripple and flow fluctuation, similar to that shown at 2.0 bar in fig. 7.6, could be obtained at pressures down to 1.0 bar. This confirms the dependency of flow and pressure ripple on air release.

Increasing the mean inlet pressure from 2.0 to 4.0 bar produced no further significant increases in pressure ripple and would indicate that the flow ripple is fully developed and independent of air release effects at 2.0 bar.

The significant increase in flow and pressure ripple levels from 0.0 bar to 2.0 bar was also accompanied by a very noticeable increase in the airborne noise radiated from the vicinity of the pump inlet line. Typically a 5 dBA increase was measured 1 m away from and perpendicular to the mid point of the inlet line. This increase, which represents almost double the sound pressure level, was directly attributable to increased fluid borne noise in the pump inlet line brought about by quite a moderate increase in mean inlet pressure. While this shows that an external gear pump under normal suction conditions, (low inlet pressure due to a small head of oil from a reservoir sited above the pump), may have a low flow ripple and hence propagate low levels of fluid borne noise in the inlet line, the opposite may be true if the line is boosted. This is certainly the case for many piston pumps where a boosted suction line is required to avoid cavitation and is also the case, as will be shown later, when the above mentioned gear unit is run as a motor with a high inlet pressure.

7.2.2 - PUMP OUTLET FLOW FLUCTUATION

The outlet flow fluctuations shown in fig. 7.6, again for an external gear pump (pump A) at 1500 rev/min., were evaluated by the Trombone Method. The inlet conditions were maintained at atmospheric pressure and 50°C.

The source flow evaluations shown, for 50, 100, and 200 bar, are very similar to each other in terms of overall form and period. For this reason the evaluations are plotted about mean flows which have been deliberately separated for clarity and are not intended to give an accurate representation of the mean output flow for the range of mean pressures shown. This pump design has a volumetric efficiency which is typically in excess of 96% at 210 bar and the test pump (pump A) actually exhibited 98% at 200 bar.

The generally smooth form of the flow ripples, which represents the summation of the first six harmonic components of pumping frequency, is consistent with what might be expected from an external gear unit. The transfer of fluid from inlet to outlet is smooth and continuous, without the occurence of the sudden back-flows caused by fluid compressibility effects which are characteristic of port plate piston machines. This is due to the comparatively gradual rise in pressure of the transferred fluid by continual leakage, from the higher pressure outlet side of the pump. This leakage occurs through the small clearances between the gear tips and the body and the side faces of the gears and the bushes.

Fig. 7.7, which is reproduced from ref. (13), shows experimental and predicted levels of mean pressure distribution in the body of a ten tooth external involute gear pump with a 55 bar mean pressure drop at 960 rev/min. The non-dimensionalised pressure, which represents the proportion of pressure difference across the unit, shows how mean pressure progressively increases around the gear periphery from inlet to outlet. The flow from the outlet back toward the inlet, although different over each tooth restriction, is in fact a reasonably constant loss which appears as a reduction in the mean output flow rather than a modification to the output flow variation. This is substantiated by the great similarity between the source flow evaluations at 50, 100, and 200 bar (fig. 7.6). The good correlation between these evaluations and the theoretical also indicates that the most significant influence on the outlet flow variation is due to the volume variations inherent in the involute geometry of the meshing gears.

In order to achieve a better understanding of the cause or causes of the flow ripple exhibited by this pump, a more detailed interpretation of the results is necessary. An effective way of doing this is to compare the differences between the theoretical and evaluated flow fluctuations in both the time and frequency domains.

7.2.3 - DIFFERENCES BETWEEN THEORETICAL AND EXPERIMENTAL EVALUATIONS

In fig. 7.8 the evaluated flow and volume variations for pump A are each presented as two separate components. One component is the variation which is theoretically attributable to the involute geometry of the gears and the other is the difference between this theoretical variation and the experimental variation evaluated by the Trombone Method. Separating the experimental wave forms into these two components was achieved by the individual vector subtraction of six theoretical and experimental harmonic components of flow ripple. The individual components of the two wave forms were then reconstructed and plotted as shown. This was done for evaluations at mean pressures of 50, 100, 150, and 200 bar at 50^{o}C.

From both flowand volume plots it is plain that there is some other cause of fluctuation within the test pump, in addition to the predicted effect of gear tooth meshing, which did not change significantly over the pressure range 50 to 200 bar. The volume plot may be interpreted as showing an indication of mean pressure dependency where the peak to peak volume variation increases from 10 to 15 mm^3 at 50 and 200 bar respectively yet there is no measureable difference between the peak to peak variations at 100 and 150 bar.

Looking at comparatively small differences in this way does raise the question of the accuracy of the initial flow ripple evaluation. This is something which will be investigated later but remembering the sophistication and complexity of the evaluation technique there is a remarkable similarity and consistency of results over the mean pressure range represented in the plots of fig. 7.8.

An interesting comparison may be drawn between the resultant peak to peak volume variation of the external gear pump (pump A) after subtraction of the theoretical geometrical fluctuation, and the theoretical volume variation of a typical internal gear pump (pump G) of comparable mean flow. The external gear pump residual volume ripple would still be 2 to 3 times that of the internal gear pump (typically 5 mm^3 peak to peak as shown in fig. 3.3). If it were desired to reduce the peak to peak amplitude of volume variation of an external unit to the level of an internal machine it is apparent that a calculation of the required reduction on the basis of the theoretical model proposed would be insufficiently accurate to achieve the desired result. A method of flow or volume ripple evaluation would in this case be essential.

The outlet flow variation of pump A was also evaluated using the Tuned Length Method. This served the dual purpose of comparing the results of the Tuned Length and Trombone Methods and verification of the outlet characteristics of the pump. Retesting also gave the opportunity to extend the mean pressure range down to 25 bar. The individual amplitudes and phases of the first four harmonics of pumping frequency, evaluated by the two methods, over the mean pressure range 25-200 bar are shown in fig. 7.9. The results of comparable tests show excellent agreement in terms of both the amplitude and phase of the four harmonics shown.

It is also evident that the amplitude and phase of all the harmonics shown are substantially independent of mean pressure.

This is an important revelation as it means that a specific modification to the flow ripple characteristic of the pump will have a similar effect throughout the working pressure range of the unit. The amplitudes of the third and fourth harmonics do show a tendency to increase with increasing mean pressure. However the tendency is small, a variation of approximately 10% about the values for 100 bar mean pressure, and would necessitate a compromise in cancellation by any method generating a fixed amplitude anti-phase component. At least the problem is not compounded by the addition of phase variations. The phase of the third and fourth harmonics is shown to be constant over the pressure range 50-200 bar.

If the evaluated amplitudes and phases are compared with those predicted by the theoretical model marked differences are apparent. The evaluated amplitudes of all the harmonics are consistantly greater than the theoretical for all mean pressures. The results at 25 bar tend to indicate a trend toward the theoretical values at low pressures, also apparent in the phase results, but this cannot be concluded on the basis of a single result.

A typical example of the differences between evaluated and theoretical results for 100 bar mean pressure is shown in table 7.1. These differences may be explained by certain limitations of the theoretical model and dynamic leakage effects.

LIMITATIONS OF THE THEORETICAL MODEL

The analysis of external involute gear geometry, which was presented in 3.3.1 and which has been used as the basis for predicting both flow and volume variations, is a necessarily simplified model of a real pump. This simplification of the analysis causes limitations in the ability to represent accurately a particular manufactured unit and can contribute to the differences in the results which have been shown here.

Inherent in the analysis are the following assumptions:-

i) The gear teeth are perfectly symmetrical, of full unmodified involute form, with a contact ratio of unity and zero backlash.

ii) The driver and driven gears are supported in bearings which maintain meshing contact at the pitch circle radius without variations due to shaft deflections or bearing clearances.

iii) Perfect relief of trapped volumes between meshing teeth.

iv) No internal leakage or compressibility losses.

In practice none of these assumptions are absolutely accurate. Errors in gear flank profile and pitch (the radial tooth spacing) do occur and can in themselves be the cause of severe mechanical vibration and airborne noise as well as producing flow ripple variations from tooth to tooth. The individual tooth profile is not necessarily of true involute form. Indeed as is the case with the test pump (pump A) the involute profile at the tip of the gear tooth flank is deliberately modified to avoid interference with, and undercutting of, the root of the opposing tooth. This is particularly necessary with low numbers of teeth and high pressure angles. Without detailed information of the deviance from a true involute profile it is difficult to assess its effect on flow fluctuation.

It is also impractical to use gears with a contact ratio of unity and no backlash. The condition of unity contact ratio means that there is always single tooth contact between the driver and driven gears. This means that the transfer of the driving load from one tooth to the next is instantaneous as one tooth comes out of mesh and the next comes into mesh. As it is impossible to produce meshing gears with sufficient accuracy to guarantee the exact positioning of the teeth necessary for instantaneous and simultaneous transfer of tooth contact the contact ratio of the gears is increased to ensure a smoother load transfer. A contact ratio of 1·1 means that for 10% of the tooth contact cycle there is an overlap or sharing of contact load bet ween two pairs of teeth. Overlapping the mesh transfer also reduces the chance of port to port leakage through increased tooth flank contact. Because of the necessity for manufacturing tolerances it is usual to produce gears which have clearance between the non-active tooth flanks, that is backlash. The maximum backlash of the gears used in pump A was 0·5 mm measured at the pitch point.

A contact ratio above unity also has the effect of increasing the overall discharge from the unit if suitable relieving grooves are provided so that fluid trapped between the meshing teeth of either gear is expelled into the pump outlet. The trapped volume and the relative meshing points of the gears are shown in fig. 7.10. In position a, the driving tooth is in contact with the driven gear tooth at point A and the following drive tooth is just contacting the next driven tooth at point B. The volume of fluid which is trapped between the two tooth meshes and the gear side face bush contacts is shown by the hatched area.

The trapped volume, which has a maximum value in position a, reduces to a minimum when the backlash is equally displaced about the pitch point and increases to a maximum again in position b. The contact points A and B have now moved to C and D respectively for an angular rotation of the driver gear of 10% of the tooth cycle (4·5 degrees for pump A). Changes in the trapped volume are transferred to the pump outlet port through the area of exposed relief groove (one at each end of the gear) which is shown cross-hatched.

The variation in the trapped volume and hence its effect on pump output volume variation may be determined by analysis of the involute geometry of the gears. The result as shown by Yudin ref. (28) is as follows;-

$$qx_1 = \frac{b}{ro} \cdot to.x_1^2 \qquad (7.1)$$

for x_1 in the range $-\frac{to}{2}(\varepsilon - 1) \leqslant x_1 \leqslant +\frac{to}{2}(\varepsilon - 1)$

where qx_1 - trapped volume variation

b - gear width

to - base pitch

ro - base circle radius

x_1 - distance along line of contact

ε - contact ratio

The substitution of dimensional data for pump A into equation 7.1 indicates that the volume variation follows a parabolic trend as shown in fig. 7.11. The 5·5 mm^3 peak amplitude of this variation coincides with a similar variation at the beginning of the tooth cycle as shown in fig. 7.8. The following negative increase in volume variation which occurs at the end of the trapped volume cycle coincides with a momentary connection of the outlet and inlet ports. This connection which provides the opportunity for dynamic port to port leakage is shown in fig. 7.10b.

The tooth sealing contact length, between points C and D is less than the limits of high and low pressure areas on the bush faces (under lapped) so that the trapped volume is, for a brief period, connected to both the inlet and outlet sides. This explains the mean pressure dependency on volume variation shown in fig. 7.8 as the reverse-flow would depend on the pressure difference across the pump. This may be a source of leakage flow which would vary from one pump build to another. The sealing limits on the bush faces, being subject to positional tolerance variation determine the area and duration of the outlet to inlet port connection. The fact that the high and low pressure limitations are on separate bushes, which may rotate in the body bores, increases the possible underlap variation. A modification to the relief groove or bush design might be necessary to reduce the flow variation caused by this effect over a range of similar pumps.

As the driver tooth contact at point C (fig. 7.10b) ceases and the trapped volume, which is at outlet pressure, is opened fully to the inlet port, there is an expansion of the high pressure volume which is evident as an increase in inlet flow as shown by the evaluated inlet source flow at 2.0 bar in fig. 7.6.

7.3 - THE EFFECTS OF THE MK I MULTI-LIFT CAM MECHANISM ON GEAR PUMP CHARACTERISTICS

The MK I multi-lift cam mechanism, as described in 6.2, was primarily intended to effect a cancellation of the first harmonic of gear pump volume variation. In order to approach 100% reduction at this frequency the design of the mechanism had to take into account the difference between the theoretical and evaluated amplitudes of flow ripple as shown in fig. 7.9. At 50 bar mean pressure, the evaluated amplitude of the first harmonic of flow and volume variation was 10.7% greater than that predicted by the theoretical geometric analysis of the gear teeth. The design of the mechanism was in fact such that it would produce 97% of the evaluated volume ripple, the 3% loss being due to the cam profile being on the bottom limit of the drawing tolerance.

7.3.1 - PUMP OUTLET FLOW AND VOLUME VARIATIONS

The effect of the MK I multi-lift cam mechanism on pump outlet flow and volume variations was determined by testing a modified unit, as shown in fig. 6.1, using the Trombone Method. A comparison of the flow ripple produced by pump A in its standard and modified conditions at 1500 rev/min., 50 bar and 50°C is shown in fig. 7.12. The plots, which represent the reconstruction of the first six harmonics, show a marked reduction in the overall amplitude and almost total removal of the fundamental period as a result of the addition of the mechanism. The peak amplitude of the first harmonic of flow ripple was reduced from $0 \cdot 495 \times 10^{-4}$ m^3/s to $0 \cdot 160 \times 10^{-5}$ m^3/s, a reduction of 96·7%. This corresponds to a reduction in peak volume variation from 39.39 mm^3 to 1.27 mm^3.

The almost total cancellation of the first harmonic is clearly visible in the plot of volume variation shown in fig. 7.13. The volume ripple of the modified pump shows that the second harmonic is now the predominant frequency as was predicted by the residual volume variation shown in fig. 6.4. The direct relationship between the mechanism volume variation and the reduction in evaluated pump volume variation is further confirmed by the fact that the addition of the mechanism caused no change in the evaluated pump impedance. Thus the mechanism must cause a reduction in pressure ripple equal to the reduction in volume ripple.

7.3.2 - PUMP OUTLET PRESSURE RIPPLE

The effect of the MK I mechanism on outlet pump pressure ripple is also shown in fig. 7.13. The plots are the reconstruction of the first ten harmonics of pressure ripple determined by the High Impedance Pipe Method (at 1500 rev/min.

50 bar and 50°C) and therefore represent the product of pump flow fluctuation and source impedance. Again the reduction in the first harmonic is apparent. The RMS values before and after were 4.45 bar and 0.21 bar respectively, a reduction of 95.3%. Comparing this with the 96.7% reduction in flow variation proves conclusively the direct dependence of pressure ripple on flow ripple.

Although the MK I mechanism was primarily intended to investigate the possibilities of reducing the fundamental component of pressure ripple, it also had effects at other frequencies. As the displacement characteristics of the straight-sided cam are not purely sinusoidal changes were effected at higher harmonics as shown in table 7.2. The general effect was to cause a reduction in the amplitude of pressure ripple at each harmonic with the exception of the second and fifth where a slight increase of 4.5% and a condition of no change, respectively, were recorded. An indication of the total effect of the mechanism over the frequency range 200-2000 Hz is given by comparison of the RMS rating of the pump, resulting from the High Impedance Pipe tests, before and after modification. This shows that the pump modified with the MK I mechanism has a rating of fluid borne noise generating potential of 0.76 bar, 83.1% less than the pump in its standard or unmodified condition which had a rating of 4.51 bar. This significant reduction in rating, due mainly to the reduction at the fundamental, highlights the dependency of the rating on the first two harmonics. An additional reduction of 95% at the second harmonic would give a pump rating of 0.3 bar, only 6.8% of the rating of the standard pump.

7.3.3 - PUMP MEAN PRESSURE LIMITATION

The MK I cam design successfully accomplished the objectives set out in 6.2. The near total cancellation of the first harmonic of pressure ripple was achieved and the principle of pressure ripple reduction by source flow modification therefore proven. Testing of the MK I mechanism also revealed two practical limitations of the mechanism design.

The first limitation, as anticipated, was found to be a mean pressure capability significantly less than that of the pump. The mechanism operated successfully, with only a light marking of the contacting surfaces, at 50 bar mean working pressure and 1500 rev/min. However after only a few minutes operation at 100 bar a gradual increase in the amplitude of the first harmonic of pressure ripple was observed. On examination of the mechanism this increase was found to be consistent with a proportional loss in piston-follower lift due to cam wear. Subsequent hardness checks revealed that the cam and piston-followers were below the desired level: 560 Hv instead of 750-850 Hv. A new set of suitably hardened components were run for approximately 10 hours at 50 bar mean pressure and 1500 rev/min., with only slight scuffing of the contacting surfaces and no reduction in cancellation performance.

The second limitation was found to be the garter type spring used to orientate and return the piston-followers. The spring failed, after approximately 7 hours operation, at a point where it contacted one of the piston-followers. The radial extension of the spring was only 0.65 mm and the loading light but the application was severe. Piston-follower extension caused cyclic bending of the spring to occur, at each side of the piston, at a rate of 200 Hz. This corresponds to 5.04×10^6 cycles in 7 hours. As the pump normally has an expected operating life of several hundred hours the low spring life is an unacceptable weakness of the mechanism design.

In order to improve the reliability of the mechanism and to increase its mean pressure capability to that required by the specification certain design changes were made and incorporated in the MK II design, which was still aimed at the removal of the first harmonic of pressure ripple.

HARMONIC	DIFFERENCE FROM THEORETICAL AMP. % OF THEORETICAL	DIFFERENCE FROM THEORETICAL PHASE DEGREE
1	+12	0
2	+23	-17
3	+60	-37
4	+103	+65
5	+101	-80
6	+138	-117

RESULTS FOR PUMP A - 100 BAR MEAN PRESSURE 50°C AND 1500 REV/MIN.

TABLE. 7.1 DIFFERENCES IN THEORETICAL AND EVALUATED HARMONICS OF GEAR PUMP FLOW RIPPLE

HARMONIC OF 200 Hz	STANDARD PUMP $Q_S.Z_S$ BAR RMS	STANDARD PUMP + MKI MECH. $Q_S.Z_S$ BAR RMS	DIFFERENCE %
1	4·45	0·21	−95·3
2	0·67	0·70	+4·5
3	0·25	0·17	−32·0
4	0·11	0·10	−9·0
5	0·076	0·076	0
6	0·038	0·020	−47·4
7	0·023	0·018	−21·7
8	0·018	0·009	−50·0
9	0·045	0·028	−37·7
10	0·032	0·011	−65·6
R.M.S RATING	4·51	0·76	−83·1

HIGH IMPEDANCE PIPE TEST RESULTS AT 50 BAR AND 50°C

TABLE. 7.2 EFFECT OF MKI MECHANISM ON TEN HARMONICS OF GEAR PUMP PRESSURE RIPPLE

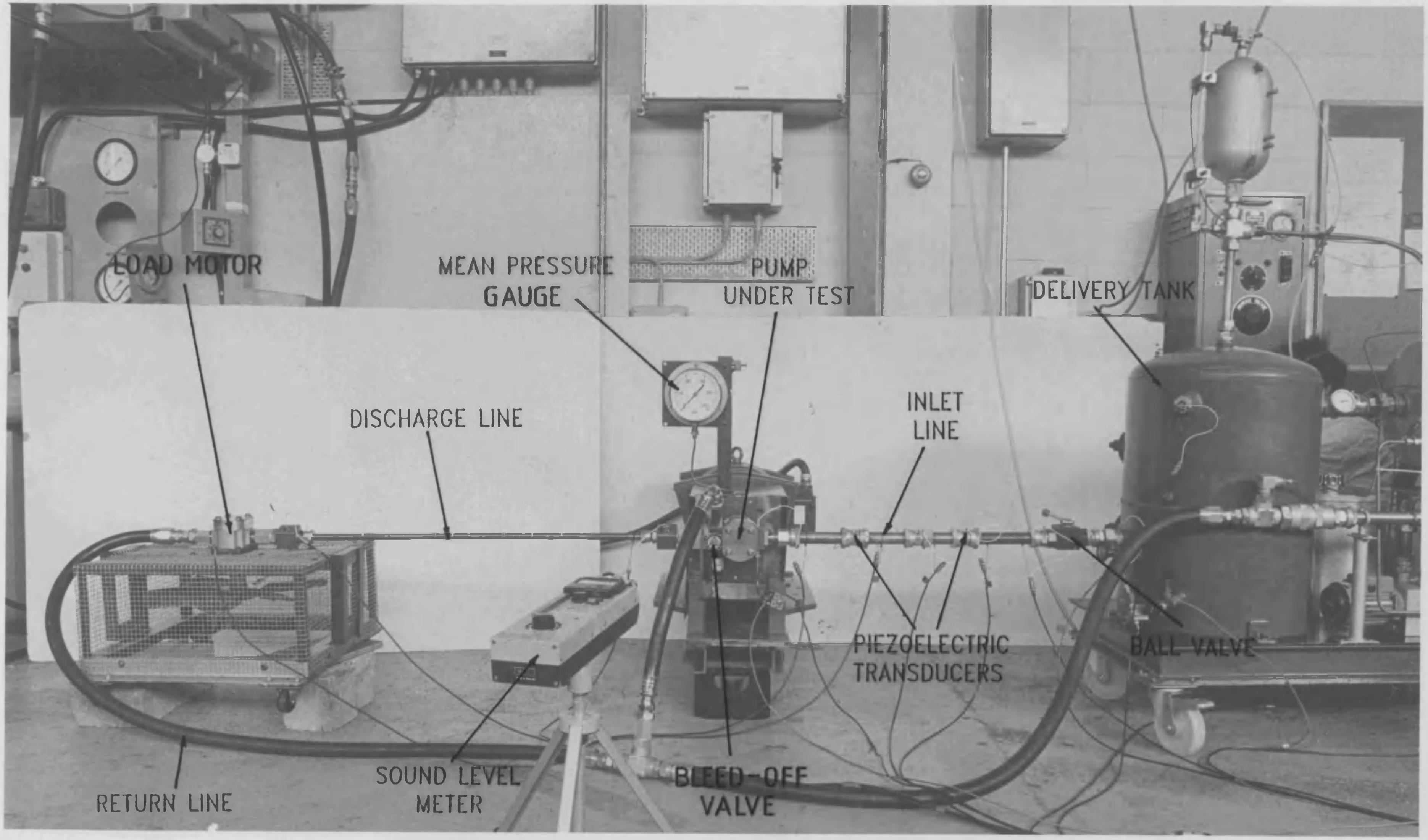


FIG.7.1 PHOTOGRAPH SHOWING TYPICAL ARRANGEMENT OF TEST RIG

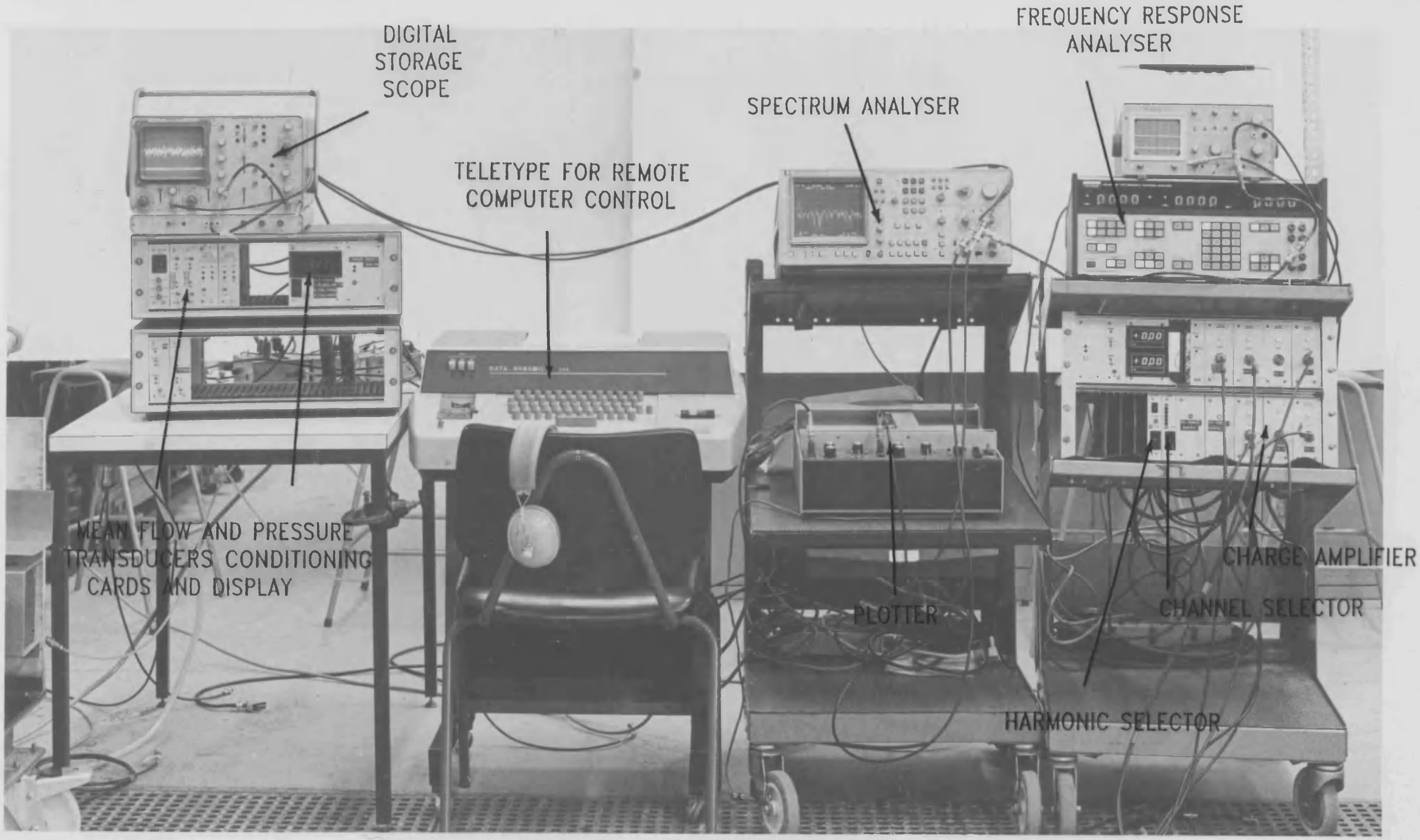


FIG.7.1a PHOTOGRAPH SHOWING TYPICAL ARRANGEMENT OF INSTRUMENTATION USED

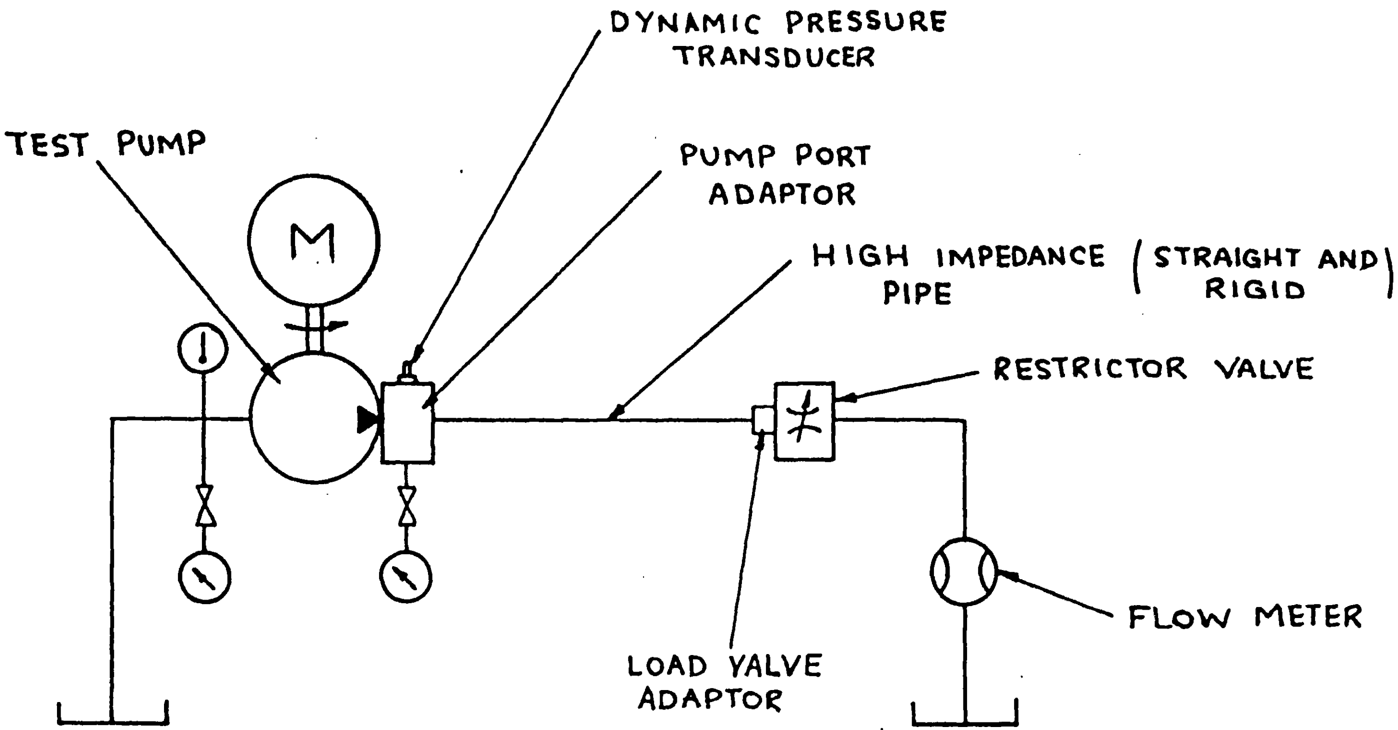


FIG. 7.2 CIRCUIT DIAGRAM FOR HIGH IMPEDANCE PIPE TEST RIG

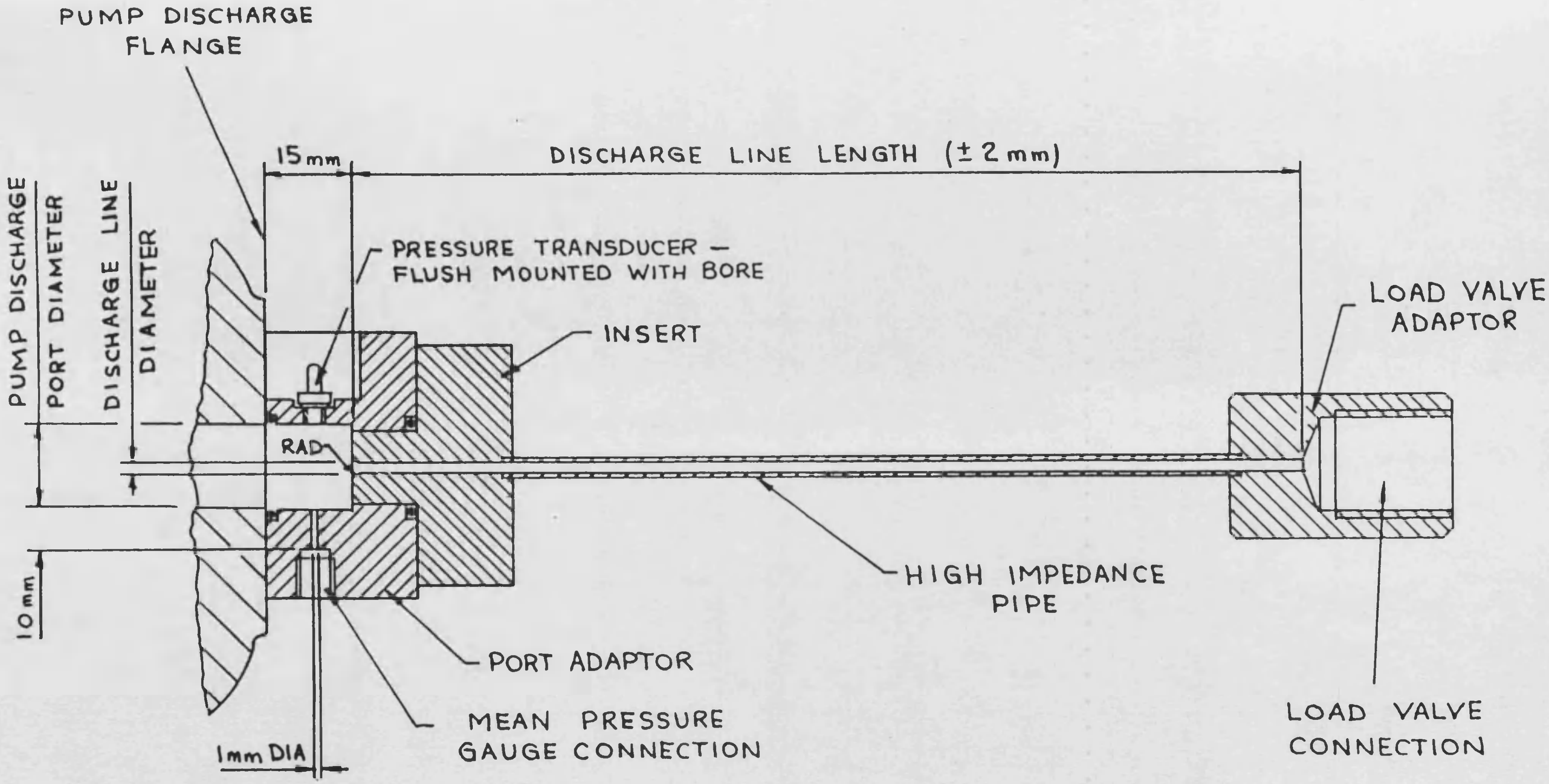


FIG. 7.3 PUMP PORT ADAPTOR AND HIGH IMPEDANCE PIPE ASSEMBLY

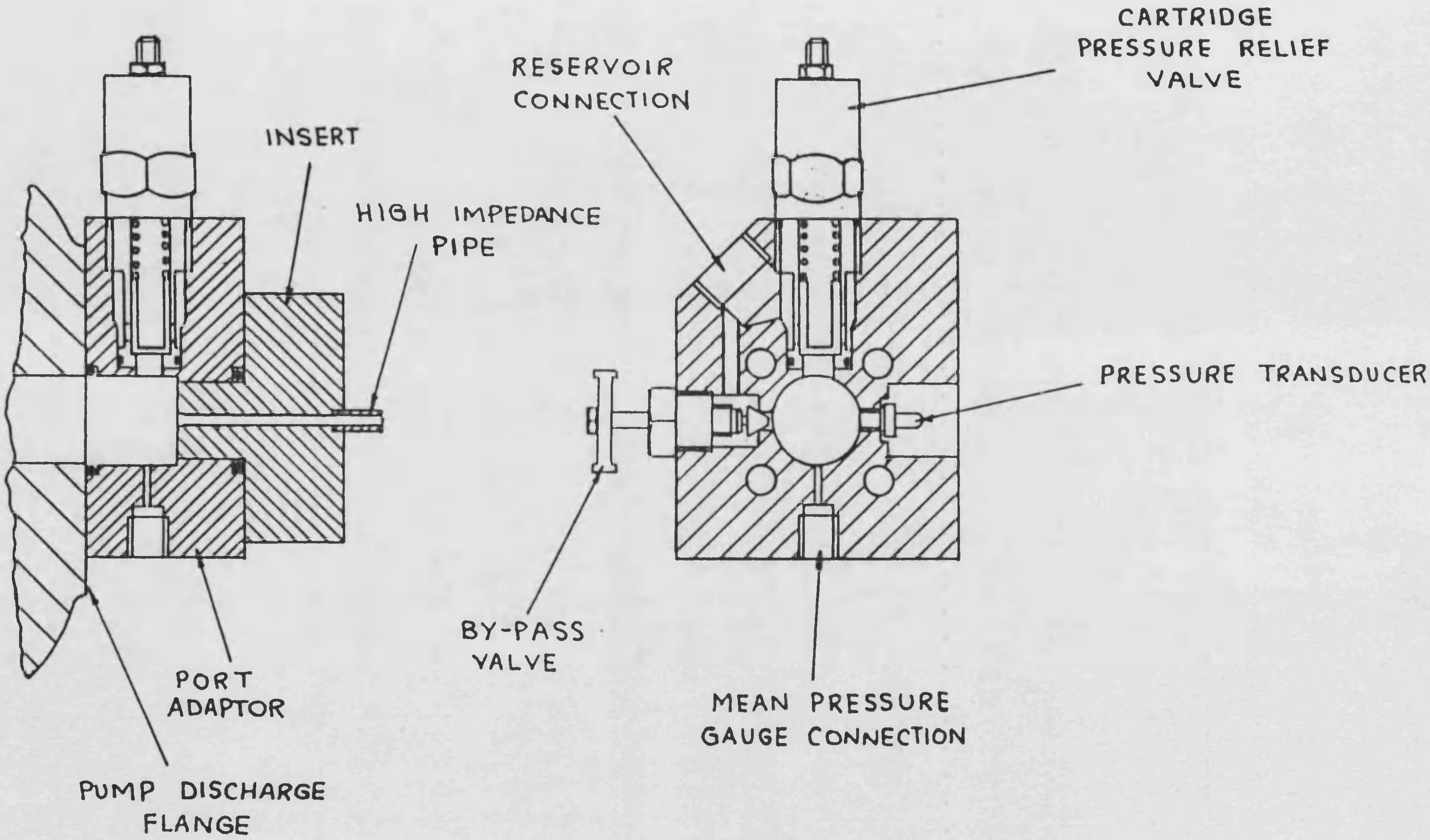


FIG. 7.4 PORT ADAPTOR INCORPORATING SAFETY FEATURES

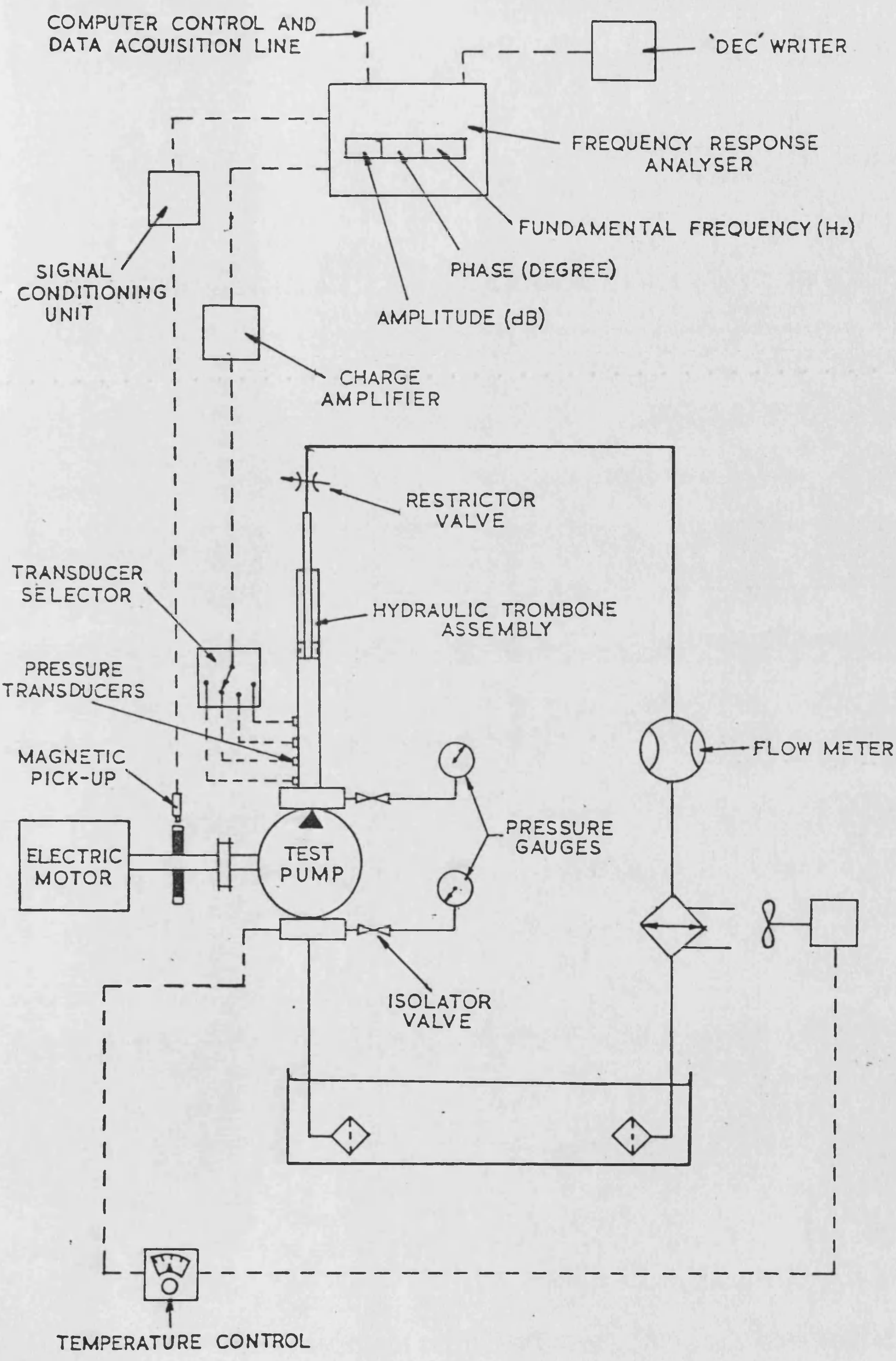


FIG. 7.5 CIRCUIT. DIAGRAM FOR TROMBONE TEST RIG

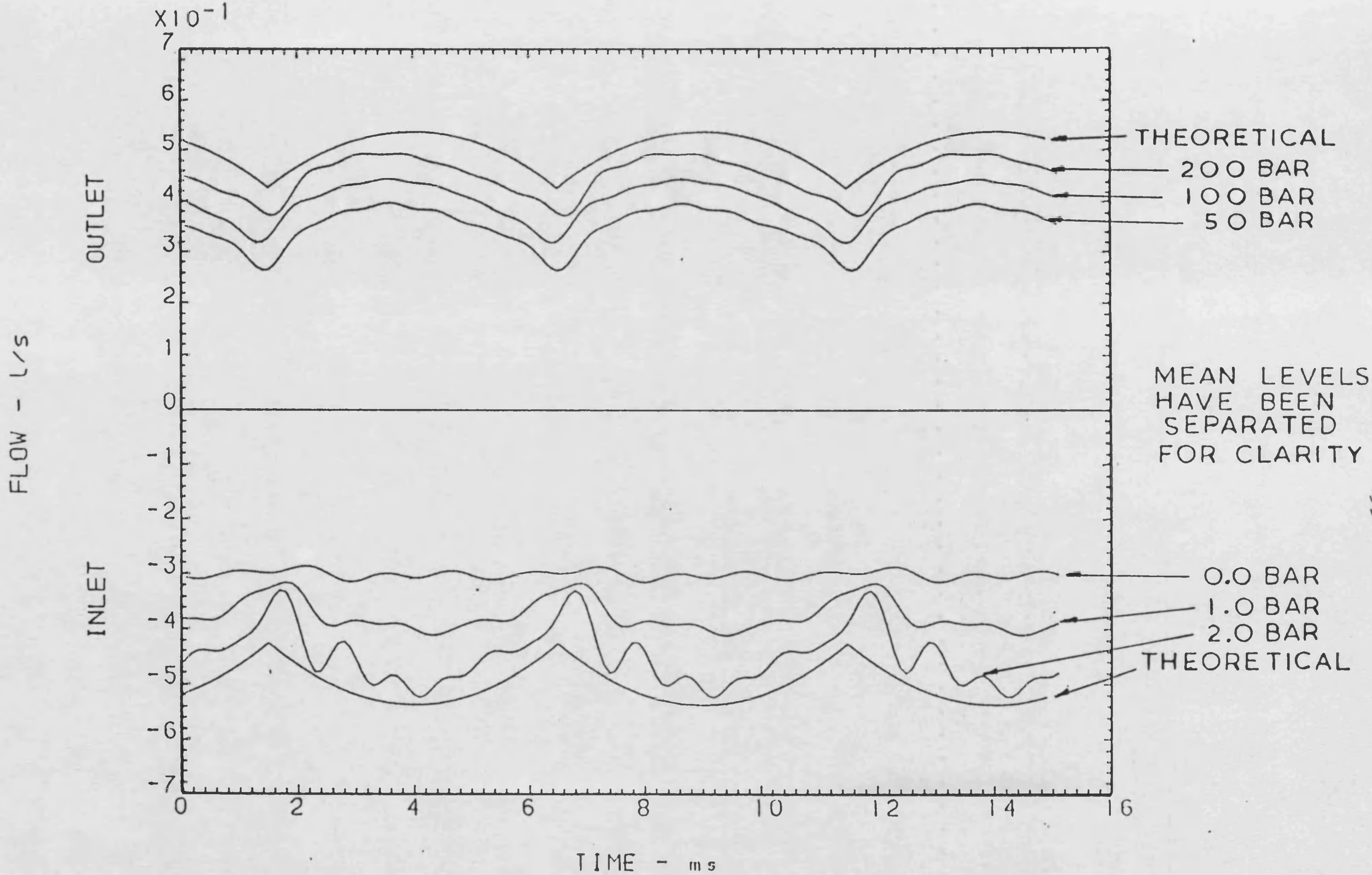


FIG. 7.6 INLET AND OUTLET SOURCE FLOW FLUCTUATIONS FOR EXTERNAL GEAR PUMP A

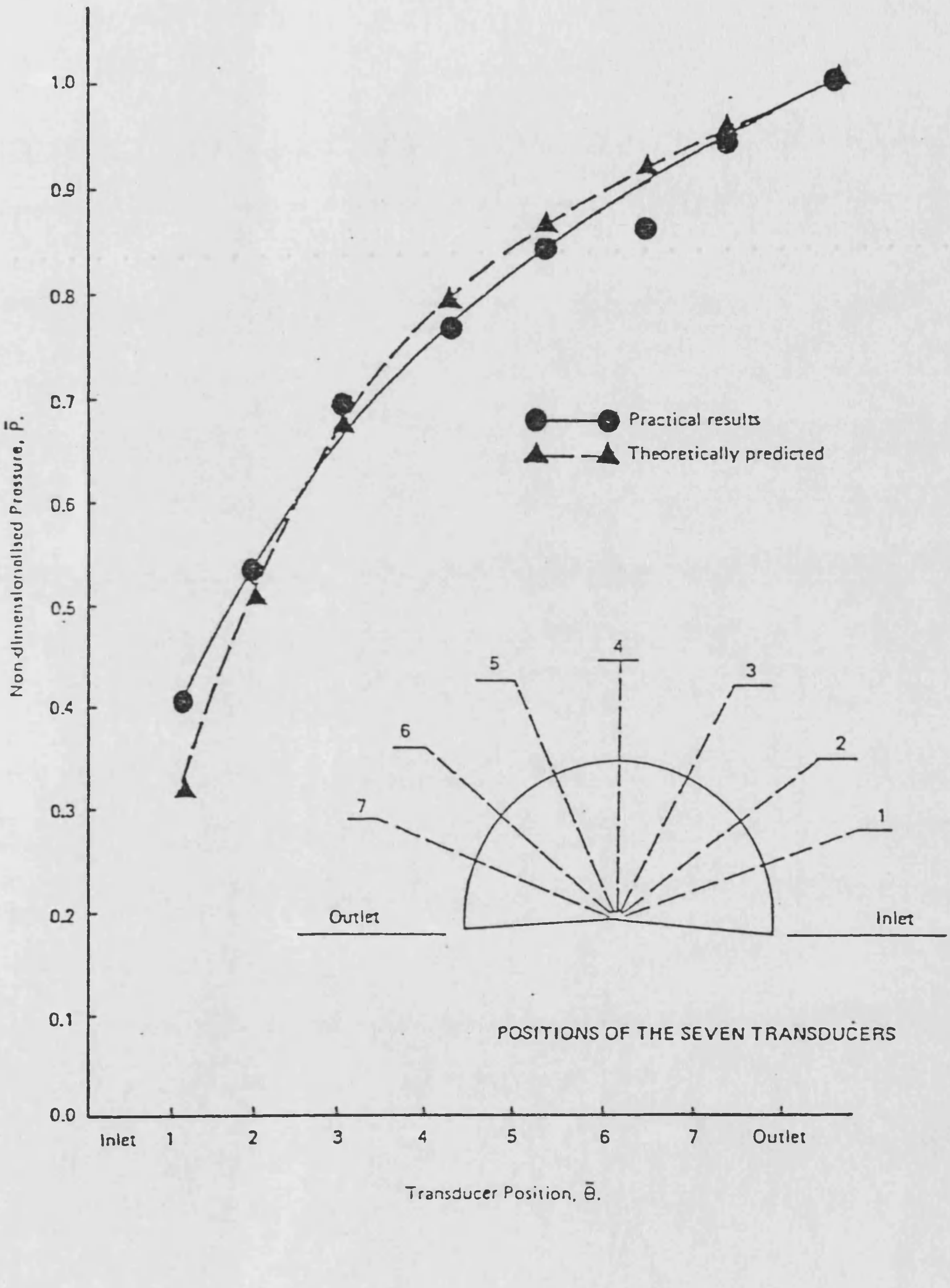


FIG.7.7 PRACTICAL AND THEORETICAL PRESSURE DISTRIBUTIONS IN AN EXTERNAL GEAR PUMP

REPRODUCED FROM REF. (13)

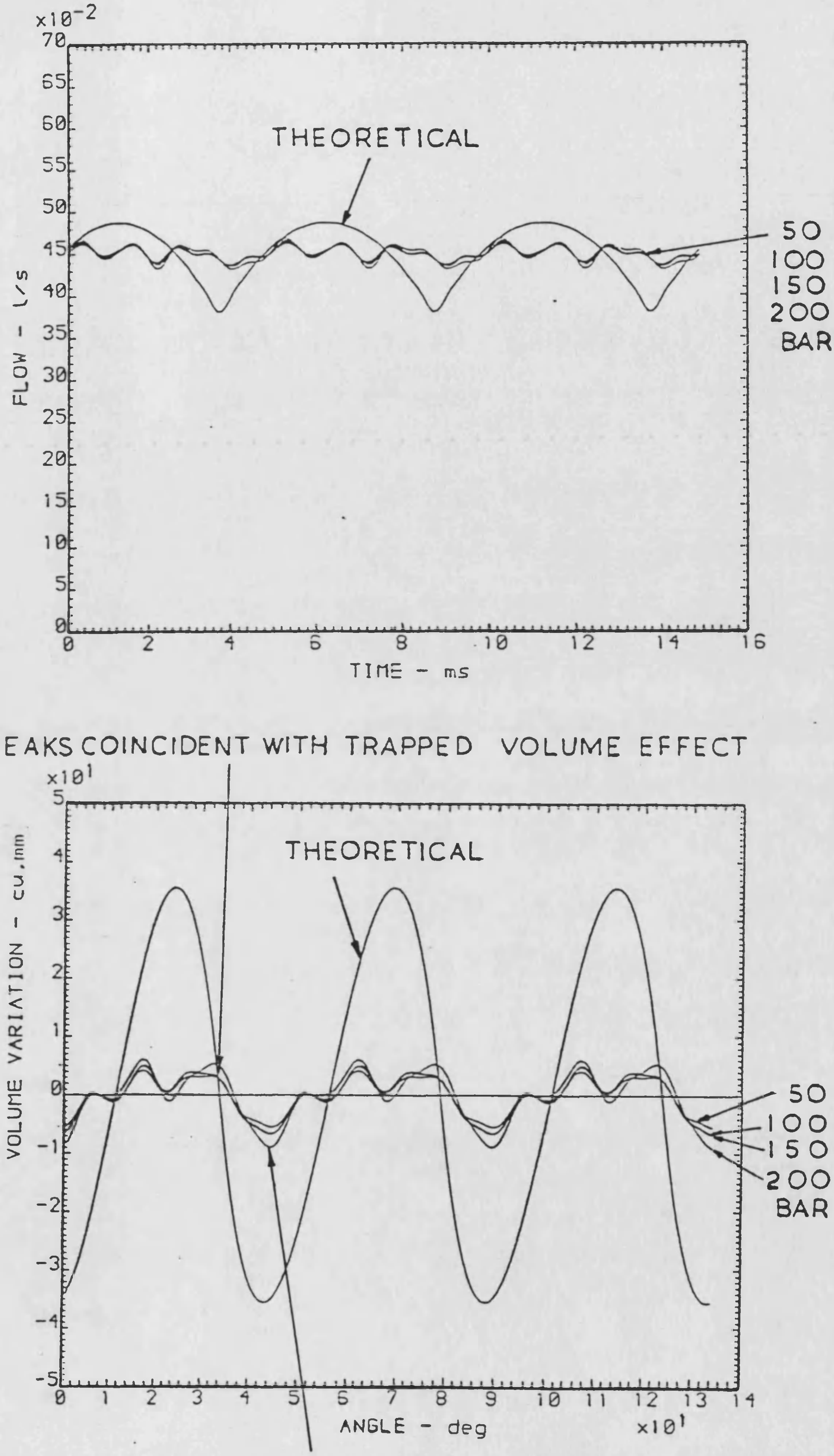


FIG. 7.8 DIFFERENCES BETWEEN THEORETICAL AND ACTUAL FLOW AND VOLUME VARIATIONS (PUMP A)

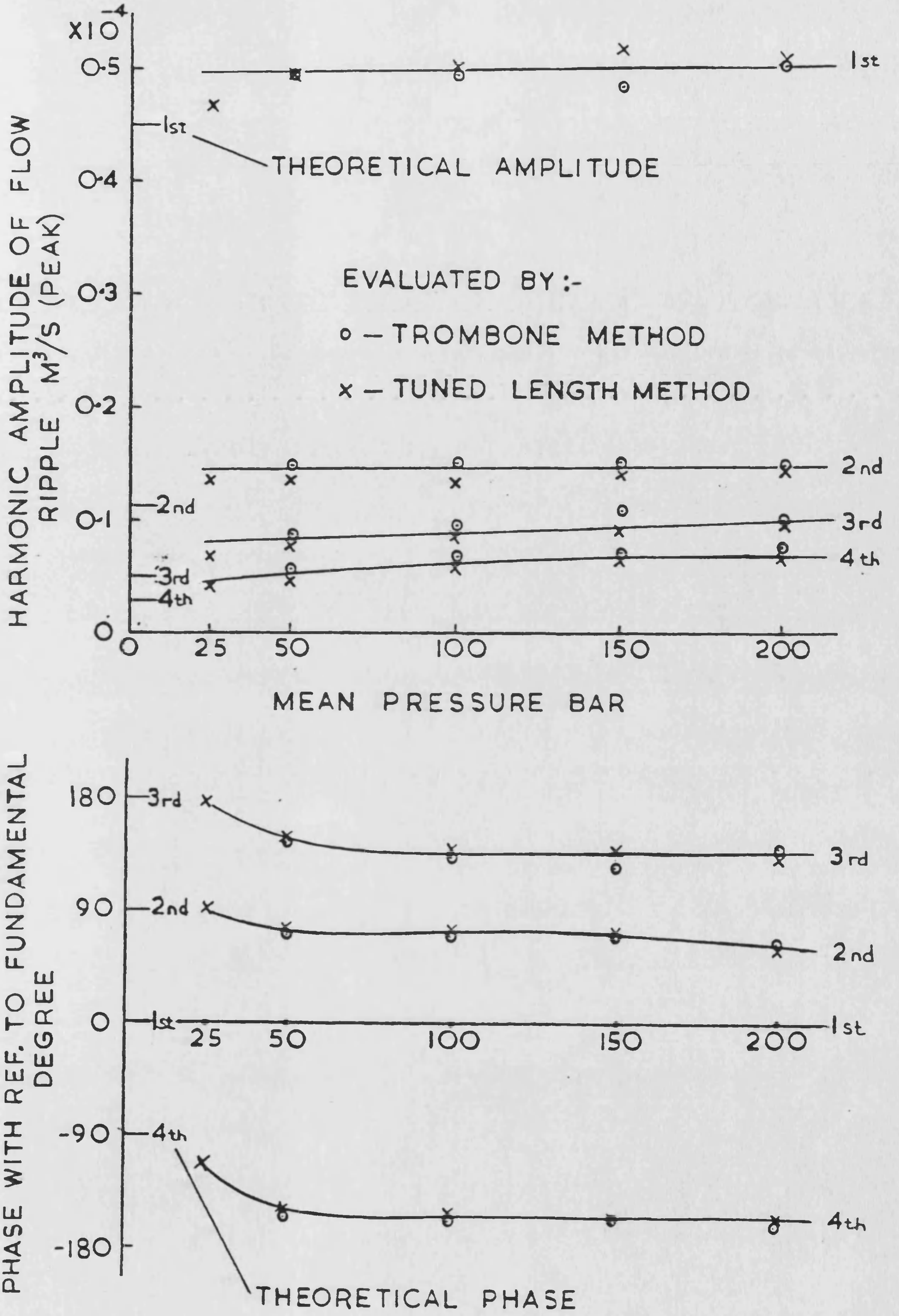
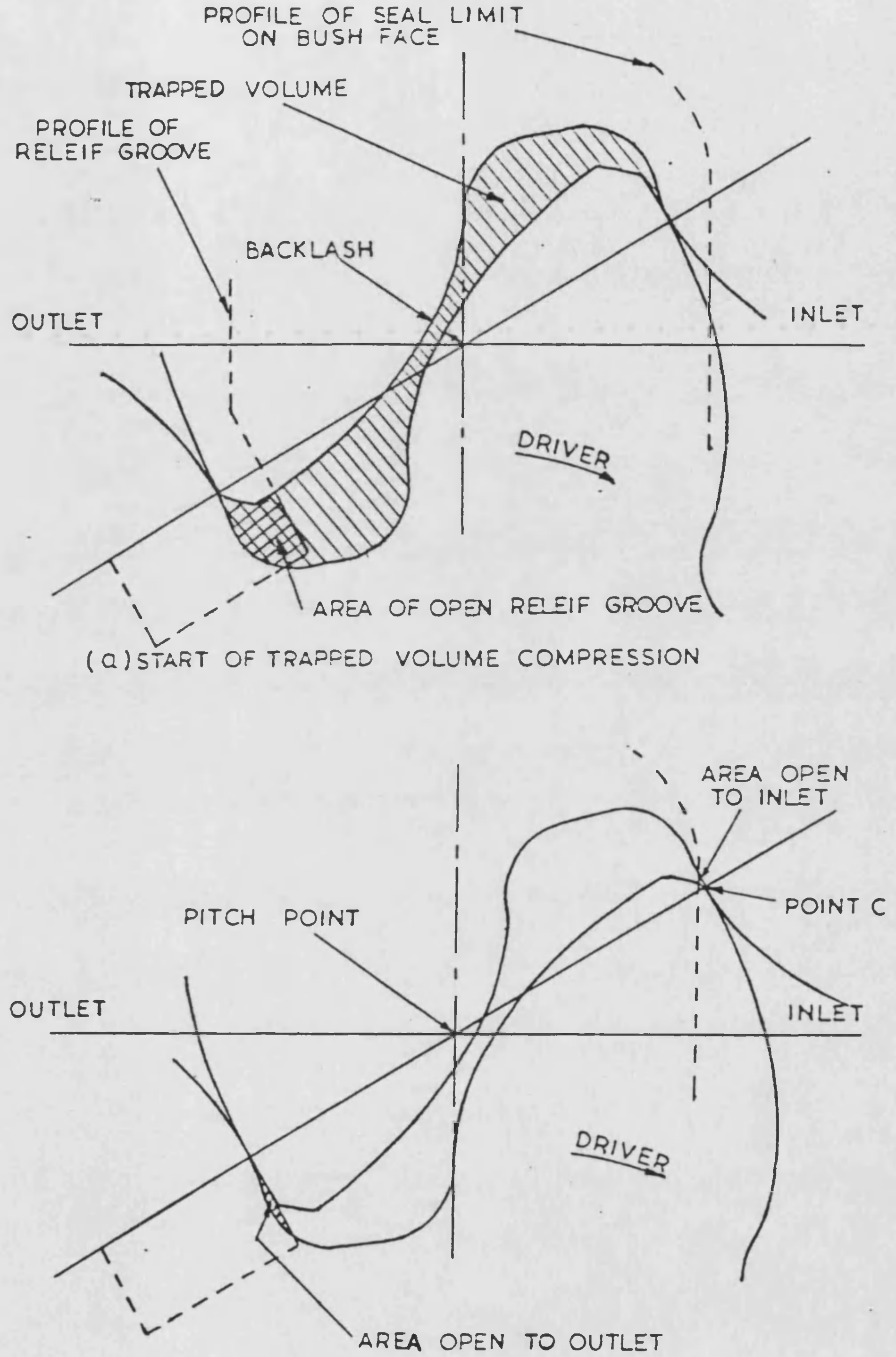


FIG. 7.9 VARIATION IN FOUR HARMONIC COMPONENTS OF PUMP A FLOW RIPPLE WITH MEAN PRESSURE



(b) FINISH OF TRAPPED VOLUME EXPANSION

FIG. 7.10 EFFECT OF CONTACT RATIO ON TRAPPED VOLUME

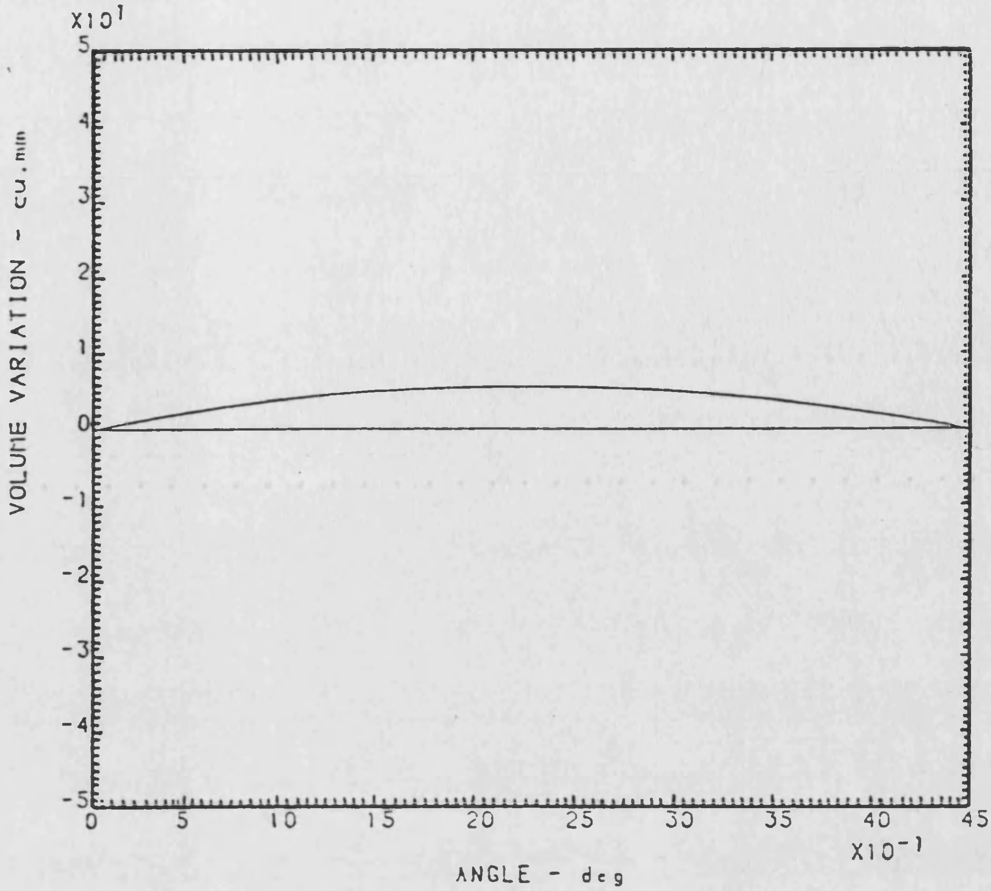


FIG. 7.11 VOLUME VARIATION DUE TO EFFECT OF CONTACT RATIO (PUMP A)

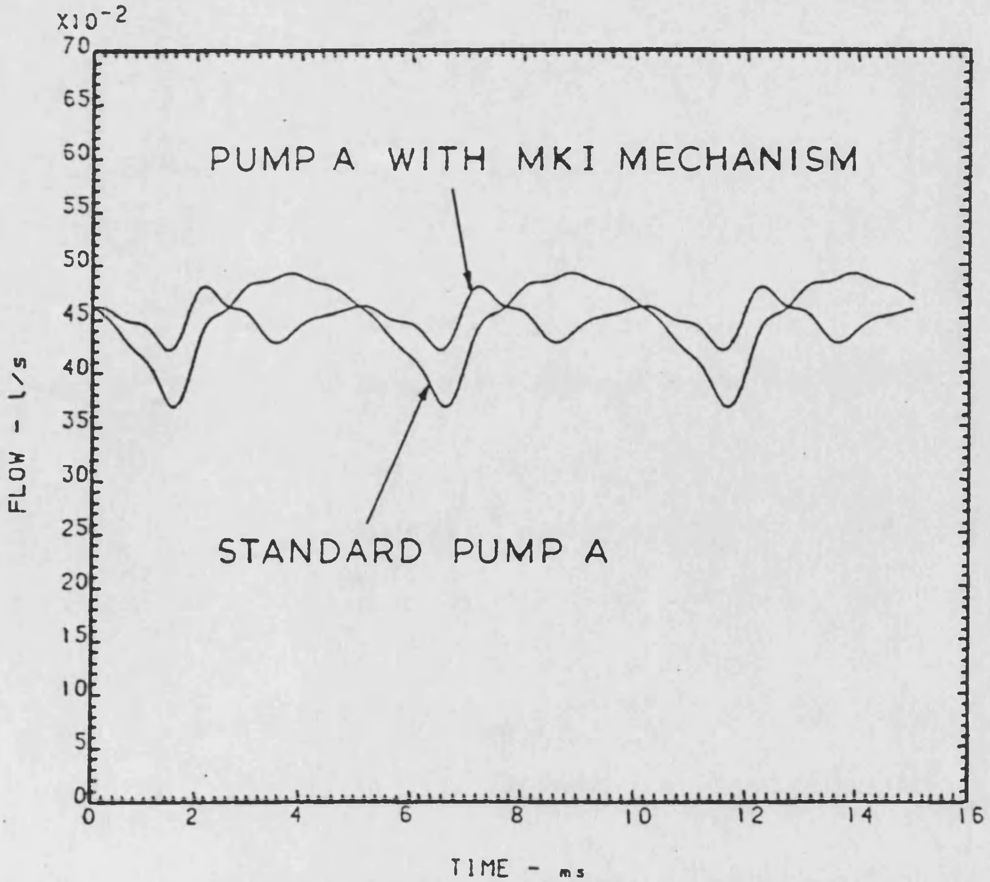
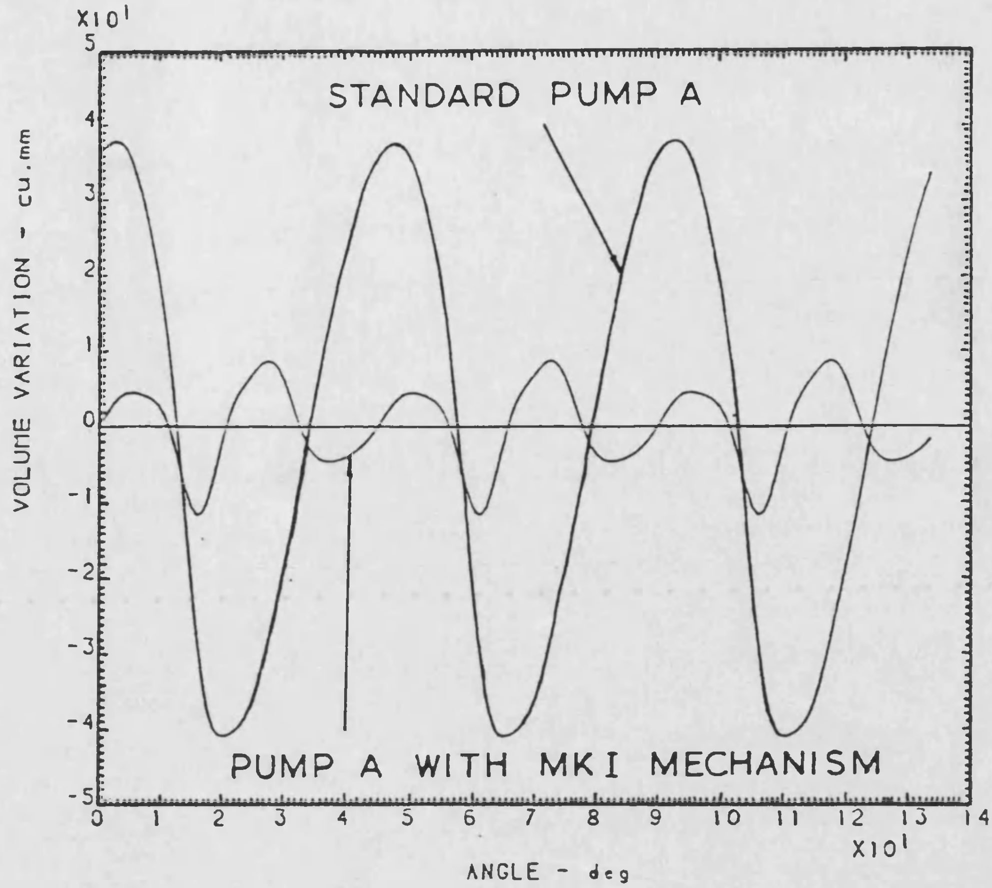


FIG. 7.12 GEAR PUMP FLOW RIPPLE BEFORE AND AFTER MODIFICATION BY MKI CAM MECHANISM



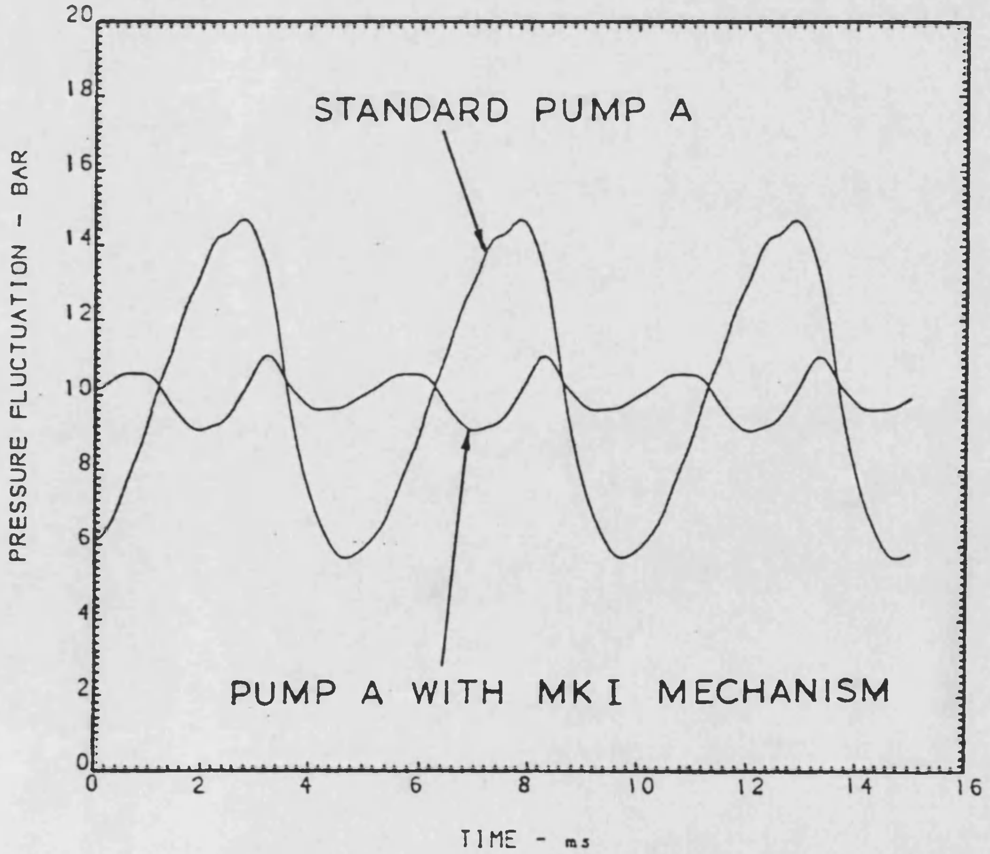


FIG. 7.13 GEAR PUMP A VOLUME AND PRESSURE VARIATIONS BEFORE AND AFTER MODIFICATION WITH MKI CAM MECHANISM

CHAPTER 8 - IMPROVEMENTS IN MECHANISM DESIGN AND EXPERIMENTAL WORK

8.1 - MK II MULTI-LIFT CAM MECHANISM DESIGN

The successful operation of the MK I design at a mean working pressure of 50 bar was used as a datum for piston-follower to cam contact stress in the design of the MK II mechanism. In order to evaluate the contact stress at this condition the Hertz stress between the cam and piston follower was adopted as the design criterion.

8.1.1 - PISTON-FOLLOWER TO CAM HERTZ STRESS CRITERION

The Hertz stress between the piston-follower and cam may be calculated using the following equation, from ref. (37):-

$$\sigma = \left[\frac{F.\sec\,(fi)\left(\frac{1}{Cc} + \frac{1}{r}\right)}{\pi.c1\left(\frac{1 - U_c^{\,2}}{E_c} + \frac{1 - U_f^{\,2}}{E_f}\right)} \right]^{\frac{1}{2}} \qquad (8.1)$$

The contact or Hertz stress derived from equation 8.1 is not a valid criterion of strength because it assumes static conditions and perfect alignment. Relative rolling or sliding induces significantly greater stresses which are aggravated by misalignments caused by manufacturing or operational errors and elastic deflections under varying loads. However a comparison of experimental results with values calculated from equation 8.1 for similar working conditions makes it possible to use the Hertz stress as an index of loading severity for design purposes.

A full design and development analysis of the mechanism would require such a loading index to include a relative sliding or scuffing parameter based on cam peripheral velocity. However as most of the test work was to be carried out at a fixed speed of 1500 rev/min., the Hertz stress index or criterion was considered a good initial parameter for comparison.

The variation in the maximum value of Hertz stress, corresponding to piston-follower contact with the cam blend radius, with mean working pressure for the MK I design is shown in fig. 8.1. The 100 bar condition where failure of the cam and piston contacting surfaces was evident corresponds to a peak Hertz stress of 16.5×10^3 bar. The 50 bar working pressure condition, which was considered successful, corresponds to a Hertz stress of 11.5×10^3 bar. The dotted line at 14×10^3 bar represents the Hertz stress between the meshing gear teeth of the pump at 210 bar working pressure. This would suggest that the cam mechanism and the gear teeth could have similar limiting Hertz stress values, which is not unexpected as they are all made of the same material, even though the mechanism contact is predominantly sliding rather than rolling.

For the MK II cam design it was decided to adopt a peak Hertz stress criterion of 10^4 bar. This would leave a margin of reserve over the MK I working condition, to allow for the inevitable increase in surface sliding velocity incurred by the necessary cam modifications. Hertz levels of the order of 10^4 bar are also not uncommon for the cams and followers which operate the valves in automotive internal combustion engines, ref. (38). The rotational speeds of engine cam shafts are also typically up to 3000 rev/min. which corresponds to the required maximum pump drive shaft speed.

8.1.2 - MK II MECHANISM DESIGN CRITERIA

A scheme of the MK II multi-lift cam mechanism installed in a modified Dowty 1P3060 external gear pump is shown in fig. 8.2. A summary of the changes in principal criteria relating to the design of the individual components of the mechanism follows. All identifying part numbers refer to this figure.

(i) - CAM (8007-4)

The MK II cam differs from the MK I design in that it has a larger profile for increased lift and is not a bonded fixture to the pump driveshaft. The cam is driven and located through a square spigot on a driveshaft extension (8007-5) which is attached to the pump driveshaft in the same manner as the MK I cam. A square drive was chosen to permit good angular location at the cam while maintaining ease of phase setting and assembly. Separating the cam from the pump driveshaft also facilitated easy removal and exchange of the entire mechanism. Axial location of the cam is achieved with a shoulder produced by the square on the driveshaft extension and the end cover (8005-5). This means there is a possibility of rubbing contact between the cam and end cover, but as there is no axial load on the cam, this was not found to be a problem.

The profile of the cam is again of the straight-sided form but the dimensional parameters are modified to achieve the 10^4 bar Hertz stress criterion, see fig. 8.1. The principal dimensions of the mechanism are shown in A.3.3.

(ii) - PISTON-FOLLOWER (8007-2)

The MK II piston-follower design reduces the contact force, and therefore the Hertz stress, because of the following changes. Firstly the displacement area of the piston-follower is reduced by 55%. This is achieved by doubling the number of active piston-followers from 4 to 8 and increasing the maximum lift of the cam by 13.6%. The contact load is further reduced by omission of the garter type piston-follower return spring. Return of the piston-follower by fluid pressure alone was subsequently shown to be acceptable by analysis of the acceleration forces, ref. (A.1.2), acting on the piston-follower. The piston-follower and cam remain in contact with a minimum working pressure of the order of 1 bar. This was considered acceptable and proved convenient as the additional space created by the spring removal was used to accommodate the larger cam.

The second important change in piston-follower design was the increase in contact length with the cam surface. The contact length of the MK I pistons was the same as the displacement diameter. The MK II piston-follower has a contact length independent of displacement diameter. This produces a head on the piston-follower which orientates the cylindrical contact surface with respect to the cam, a function previously provided by the slot and return spring.

The piston-follower contact radius is no longer full, but forms a blend radius between two flat faces each inclined at 75° to the centre line of the displacement diameter. This has two beneficial effects. Firstly it provides two flat surfaces, for orientating the piston-follower with the piston-housing, while maintaining small piston mass. Secondly it is likely to improve the ability of the piston-follower to create an oil film between the contacting surfaces because of the wedge effect that it creates.

(iii) - PISTON-HOUSING (8007-3)

The MK II piston-housing is very similar to the MK I design. The essential differences are that the number of piston-follower bores has increased from 4 to 8. The working position of the pistons is moved radially outward to accommodate a larger cam, but the installation of the MK I and MK II housings in the pump body extension is unchanged. The circumferential groove positioning the garter spring is replaced by individual slots to orientate the piston-followers. Clearance between the piston-follower and slot is kept small to ensure accurate orientation. This requires accurate positional tolerancing of the slots relative to the piston-housing bores which is difficult to achieve without appropriate jigs and fixtures.

8.1.3 - PUMP MODIFICATION DESIGN CRITERIA

The modifications to the design of the standard pump to incorporate the MK II mechanism are identical to those required by the MK I (as described in 6.2.3). The only minor difference is in the driveshaft and gear sub-assembly. Instead of the driveshaft and gear being directly attached to the cam, the driveshaft is attached to a driveshaft extension (8007-5 see fig. 8.2) which supports and drives the cam through a locating square.

8.1.4 - VOLUME VARIATION PRODUCED BY MK II MECHANISM

The principal dimensional parameters of the MK II design, which is intended primarily for cancellation of the first harmonic of flow ripple, are shown in A.3.3. Substituting these values in equations 6.1, 6.3 and 6.5 for one cycle of piston-follower motion results in a peak to peak volume variation of 84.00 mm^3. The instantaneous volume variation with cam angle is shown in fig. 8.3 and is of similar form to that produced by the MK I design. The increase in peak to peak variation, from 71.13 to 84.00 mm^3, is required to account for the differences between the theoretical and experimental pump volume variations which were described earlier in 7.2.3.

8.2 - MK III MULTI-LIFT CAM MECHANISM DESIGN

The mechanism designs MK I and MK II were both successful in achieving their objectives: the MK I in establishing a method of pump pressure ripple reduction, the MK II by creating a practical means of producing at least the first harmonic component of the total source flow spectra necessary to produce a pressure ripple free pump. The problem then was to improve the cam design to effect the desired degree of cancellation at frequencies in addition to the fundamental. This required a complete change in the approach to cam design used for the MK I and MK II versions.

The displacement characteristics of the straight-sided cam were very useful in allowing the approximate matching of the fundamental peak to peak volume variations produced by the pump and mechanism, but a fixed format of straight sides and blend radii does not allow the addition of specific displacement variations with particular phases for successive frequencies. It was therefore necessary to achieve a direct method for defining the profile of the cam from the source flow of the pump.

Because of the differences between the theoretical source flow variations derived from the pump model and the actual variations measured using the "Trombone" technique it is convenient to define the flow fluctuation in a general form, as a frequency spectrum. It is then possible to consider the summation of a number of significant harmonics to be representative of the pump flow characteristic. Henderson ref. (4) suggests that an external gear pump may be represented by the first two harmonics, but as the third harmonic is of the same order of magnitude as the fundamental component of an internal gear pump it was decided the MK III cam should attempt the cancellation of the first four harmonics of the 1P3060 gear pump. This was thought to be a good test of the cam profile designing technique and would provide useful hardware with which to investigate the effect of pump pressure ripple reduction on pump dependent airborne noise.

8.2.1 - MULTI-HARMONIC CAM DESIGN ANALYSIS

The purpose of the following analysis is to show how the summation of a number of spectral components of pump flow ripple may be converted, by logical progressive steps, into the definition of a cam profile. This cam, when used in conjunction with cylindrical contacting radial piston-followers, will then produce a flow ripple of opposite form to that of the pump.

Consider the flow ripple of a positive displacement pump to be represented by a Fourier series of the following form:-

$$Qs = \sum_{i=1}^{i=K} a(i).\sin(iZ\theta + P(i)) \qquad (8.2)$$

where a - amplitude of harmonic component of pumping frequency

P - phase of harmonic component

Z - number of pumping elements

θ - angle of shaft rotation at pumping frequency

i - counter

K - number of significant harmonics.

The flow ripple which the mechanism must produce is of opposite amplitude and identical phase. It therefore follows that the mechanism flow ripple is given by:-

$$Qm = -Qs = \sum_{i=1}^{i=K} -a(i).\sin(iZ\theta + P(i)) \qquad (8.3)$$

The volume variation produced by the mechanism Vm can now be evaluated by integrating the expression for mechanism flow ripple w.r.t time:-

$$Vm = \int Qm.dt$$

$$Vm = \int \left(\sum_{i=1}^{i=K} -a\,(i).\sin\,(iZ\theta + P\,(i)) \right) dt \qquad (8.4)$$

as $wt = Z\theta$

$$\frac{t}{\theta} = \frac{Z}{w}$$

$$\therefore \frac{dt}{d\theta} = \frac{Z}{w}$$

substituting for dt in equation 8.4

$$Vm = \int \left(\sum_{i=1}^{i=K} -a\,(i).\sin\,(iZ\theta + P\,(i))\,\frac{Z}{w} \right) d\theta \qquad (8.5)$$

$$Vm = \sum_{i=1}^{i=K} \frac{a\,(i)}{iw}.\cos\,(iZ\theta + P\,(i)) \qquad (8.6)$$

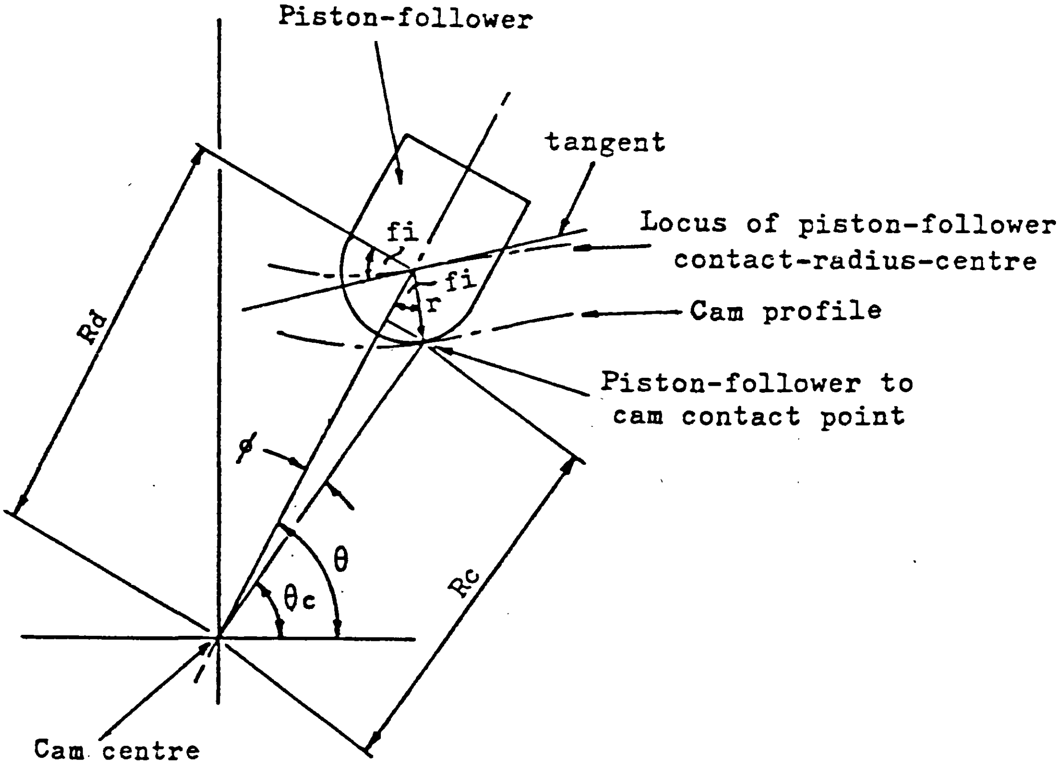
Equation 8.6 shows how the pump cancelling volume variation, to be produced by the cam mechanism, changes with cam rotational angle. In order to define the cam profile it is now necessary to remove, from equation 8.6, the effects of the piston-followers. This may be done in two steps; firstly to determine the displacement characteristics of a single piston-follower and secondly to determine the necessary cam profile for a piston-follower of a particular contact geometry, the simple radius type.

The piston-follower displacement characteristics may be determined by dividing equation 8.6 by the total displacement area of all active pistons. If m is the number of active pistons, and Ap the individual piston area, then the displacement characteristic of the piston-followers is given by:-

$$d = \sum_{i=1}^{i=K} \frac{a\,(i).\cos}{i.w.m.Ap}\,(iZ\theta + P\,(i)) \tag{8.7}$$

Equation 8.7 now describes the displacement of the centre of curvature of the piston-follower contact radius. Account must now be taken of the effect on piston-follower displacement of the movement of the cam to piston-follower contact point along the contact radius.

This is shown in the following figure:-



where θ – cam rotational angle turned through

ϕ - angular deviation in cam angle to point of contact

fi - pressure angle

r - piston-follower radius

Rc - cam profile radius

Rd - radius to contact radius centre

The position of the point of contact on the piston-follower radius determines the pressure angle (fi), which may be evaluated from the slope of the tangent to the piston-follower contact-radius-centre displacement profile. The displacement profile at cam angle θ is defined by Rd which superimposes the piston-follower displacement characteristic determined by equation 8.7 on some specified mean cam radius, i.e.

$$Rd = d + C + r \tag{8.8}$$

where C - mean cam radius

r - piston-follower radius

d - piston displacement

The variation in radius, from the cam centre to the centre of piston-follower radius, with cam angle is thus given by equation 8.8. To define the actual cam profile now only requires a slight geometrical adjustment to account for contact point movement and produce the relationship between Rc and θ (the cam profile radius with cam angle).

The angle ϕ, between the centre of piston-follower radius and the point of contact is given by:-

$$\phi = \tan^{-1}\left(\frac{r.\sin(fi)}{Rd - r.\cos(fi)}\right) \tag{8.9}$$

and the radius Rc, to the point of contact is given by:-

$$Rc = \frac{Rd - r.\cos(fi)}{\cos(\phi)} \tag{8.10}$$

Hence for a particular value of cam rotational angle θ, the corresponding angle to the point of contact on the cam profile, θc, may be determined from:-

$$\theta c = \theta \pm \phi \tag{8.11}$$

The sign of ϕ is determined by the sign of the pressure angle, fi, indicating on which side of the piston-follower centre line contact is taking place.

The cam profile radius corresponding to a cam angle θc may be determined by substitution of equations 8.7 and 8.8 in equation 8.10 as follows:-

$$Rc = \sec(\phi) \left(\sum_{i=1}^{i=K} a(i).\cos(iZ\theta + P(i)) + C + r(1-\cos(fi)) \right) \quad (8.12)$$

Equation 8.12 evaluates the cam profile radius Rc, corresponding to values of cam rotational angle θc for instantaneous incremental changes in cam rotational angle from zero to $\frac{2\pi}{Z}$ (one tooth cycle at pumping frequency).

Thus it is possible to define a cam profile which in conjunction with specified radial piston-followers, will produce a flow ripple the exact opposite of that evaluated for a pump. This method of defining a cam profile is a significant advancement over that used for the MK I and MK II designs. However, profile definition is only a partial solution to the cam design problem. To establish the likely mechanical performance of a cam and follower a number of other parameters require evaluation. Some of the parameters considered essential to the design for the MK III, 4-harmonic, cam are as follows:-

i) cam profile definition

ii) cam profile curvature

iii) cam pressure angle

iv) piston acceleration

v) cam to piston-follower Hertz stress

The evaluation of these parameters for a number of increments in cam angle is most conveniently achieved with the use of a computer. A digital computer program, CAM, was written to perform the computations necessary to design and evaluate a multi-harmonic cam mechanism. The program is documented and listed in Appendix A.1.2.

8.2.2 - MK III MECHANISM DESIGN CRITERIA

The MK III mechanism design is identical to MK II, as shown in fig. 8.2, with the exception of the cam. The MK III cam is interchangeable with the MK II, the only difference being the external profile. Details of the MK III design are as follows:-

(i) - CAM (8008-11)

The cam profile was designed such that it would, in conjunction with MK II piston-followers and piston housing, produce a flow ripple which would cancel the first four harmonics of pump flow ripple. The amplitude and phase of the first four harmonic components of flow and the corresponding volume ripples are shown in table 8.1. These were determined using the Trombone Method (described in 7.1.4 (ii)) at a mean pressure of 50 bar and a shaft rotational speed of 1500 rev/min.

The cam profile produced by the computer program, CAM, is shown in fig. 8.4 (c) which also shows the profiles of the MK I and MK II cams.

Fig. 8.5 (a) shows the variation in Hertz stress over one complete revolution of the MK III cam at a mean working pressure of 210 bar. It can be seen that the maximum Hertz stress is approximately 1.65×10^4 bar. This is significantly higher than the previously proposed limit of 10^4 bar. The high Hertz stress condition is a result of producing a four harmonic cam within the physical restraints of the MK II mechanism design. It was decided that a reduction in the maximum working pressure capability of the MK III cam was acceptable as an alternative to producing a completely new set of hardware to test the design feasibility of multi-harmonic cams. The Hertz stress variation of the MK II cam at a working pressure of 80 bar is shown in 8.5 (b). The maximum Hertz stress at this working pressure is 1.02×10^4 bar.

The piston-follower (8007-2) and piston housing (8007-3) used in the MK III mechanism design are identical to those used in the MK II and are described in 8.1.2 (ii) and 8.1.2 (iii) respectively.

8.2.3 - PUMP MODIFICATION DESIGN CRITERIA

The modifications to the design of the standard pump to incorporate the MK III mechanism are identical to those required by the MK II. Details are given in 8.1.3.

8.2.4 - VOLUME VARIATION PRODUCED BY MK III MECHANISM

The principal dimensional parameters of the MK III design, for four harmonic cancellation, are shown in A.3.3. The zero to peak amplitudes of the four harmonics of volume ripple which dictate the cam profile are shown in table 8.1 and the MK III cam profile produced is shown in fig. 8.4 (c). In fig. 8.6 the summation of the four harmonic components of volume ripple, which the mechanism is designed to produce, is compared with the summation of the first six harmonics of pump volume ripple evaluated experimentally. Adding the two profiles together produces a very small net volume ripple. This residual ripple is the combined effects of the fifth and sixth harmonics of pump volume ripple which are approximately 1% of the amplitude of the fundamental. Incorporation of harmonics higher than the sixth into the representation of pump volume ripple is unlikely to alter significantly the residual ripple. The amplitude of the theoretical components of volume ripple for an external gear pump are inversely proportional to the cube of the harmonic number. Thus the amplitude of the 10th is 0.1% of the fundamental.

8.3 - THE EFFECTS OF THE MK II AND MK III MULTI-LIFT CAM MECHANISMS APPLIED TO PUMPS AND MOTORS

8.3.1 - OUTLET AND INLET PUMP PRESSURE RIPPLE

The effect of the various multi-lift cam mechanisms on gear pump pressure ripple was examined by the measurement of the inlet and outlet pressure fluctuations produced in a simple system before and after modification by the cam mechanism. The system consisted of the pressurised reservoir connected to the pump inlet by a steel pipe 1.1 m long and 25 mm internal diameter. The pump outlet line consisted of a steel pipe

2.0 m long of 15 mm internal diameter terminated by a screw type restrictor valve. Pressure fluctuations were measured at the inlet and outlet flanges of the pump.

The effect of the MK II mechanism on the outlet pressure ripple produced by pump B (a similar unit to pump A) at 1500 rev/min. 50 bar mean pressure and 50°C is shown in fig. 8.7. Comparison of the frequency spectrum before and after modification shows the almost total cancellation of the first harmonic of pressure ripple at 200 Hz. The actual reduction was from 0.95 to 0.05 bar rms, 94.4% of the original pressure fluctuation. This compares very well with the 95.3% reduction effected by the MK I design on pump A.

The MK II mechanism was also found to effect similar reductions at other mean pressures throughout the range 50 to 200 bar. This is to be expected as the outlet flow and hence the outlet volume variation of this type of pump is virtually independent of mean pressure as shown by fig. 7.9. The evaluated inlet flow variation as shown in fig. 7.6 does however show a marked dependency on mean pressure level and this is also apparent in the inlet pressure ripple comparison shown in fig. 8.8.

At 0 bar g inlet pressure the mechanism appears to have negligible effect on inlet pump pressure ripple. At first this was thought to be due to the lack of a clear connecting passage between the low pressure side of the piston-followers and the inlet side of the pump, but this was found to be incorrect. The occurance of air release from the fluid, which caused low levels of flow and pressure variation in the standard pump also smoothed out the volume changes caused by the mechanism.

Increasing the mean pressure level to 2.0 bar shows the ceasation of the damping effect of air release and the re-established levels of flow and pressure fluctuation. The modification of the pump design to accept the mechanism was biased to give a clearer connecting passageway to the outlet side of the pump than to the inlet. The inlet connection was via a 0.5 mm annular clearance around the driveshaft extension and a 3 mm diameter lubrication hole through the length of the bushes; the outlet connection was direct via a 15 mm diameter hole. However the reduction in the amplitude of the first harmonic of pressure ripple at 2.0 bar of 93.4% (0.67 to 0.04 bar rms) was similar to that effected on the outlet. This ability of the mechanism to effect the inlet and outlet pressure ripples simultaneously is obviously a distinct advantage especially in the case of a gear pump with a pressurised inlet or gear motor running with moderate back pressure.

Examination of the effects of the MK III mechanism on the outlet pressure ripple of pump A was conducted in a similar manner to that described above. The cam profile of the MK III mechanism was designed by the computer program CAM described in A.1.2, to cancel the first four evaluated harmonics of flow ripple produced by pump A at 1500 rev/min., a mean pressure of 50 bar and a fluid temperature of 50°C. A comparison of the outlet pressure ripples with and without the MK III mechanism for pump A, at the same running conditions, is shown in fig. 8.9. The loading system consisted of 2.1 metres of 19 mm internal diameter high pressure flexible hose terminated with a screw type restrictor valve and pressure fluctuations were measured at the pump outlet flange.

Comparison of the two time domain pressure traces reveals a significant reduction in peak to peak amplitude and an increase in high frequency content of the modified pump result. The frequency spectrum shows the expected reductions in the components at 200, 400, 600 and 800 Hz, the first four harmonics of pumping frequency.

To show more clearly the effect of the mechanism the amplitudes of the first five harmonics before and after modification are presented in the following table:-

Harmonic of 200 Hz	Pump A	Pump A with MK III mechanism	Change %
	Amp. bar rms.		
1	0.580	0.008	-98.6
2	0.140	0.035	-75.0
3	0.126	0.020	-84.1
4	0.078	0.007	-91.0
5	0.045	0.053	+17.7

The high levels of pressure ripple reduction achieved by the MK III mechanism, at all the frequencies where a modification was expected shows quite clearly the high degree of accuracy with which the components of flow fluctuation were evaluated and demonstrates the ability of the mechanism to produce compensating components.

The evaluation of source flow by the Trombone Method has been shown by Wing ref. (27) to be reasonably insensitive to random errors in the amplitude and phase of the input pressure data. The evaluation of source impedance is shown to be more sensitive to such errors and both parameters are directly dependent upon accurate representation of the properties of the working fluid, in particular the effective bulk modulus.

The 17.7% increase in pressure ripple of the fifth harmonic does show the sensitivity of the mechanism to small dimensional inaccuracies in the cam profile. The peak to peak displacement of the MK III cam at the first harmonic is 0.704 mm and at the fourth is only 0.020 mm. This demands a high degree of accuracy of the profile production method. This is possible, as the above results show, but may be expensive on a large scale production basis. The cam lift for a particular harmonic may be increased, there by reducing its sensitivity to dimensional inaccuracies, by reducing the effective area of the piston-followers. Thus two or more separate profiles each combining one or more harmonic displacements may be used with suitably selected piston-followers to achieve the effect that the MK III design produces with a single profile. This may not be practical as a means of producing a low ripple production pump design, but may be very useful in enabling the mechanism to be used for the evaluation of pump flow ripple. This will be discussed in chapter 9 where the multi-lift cam mechanism is proposed as a measuring device.

8.3.2 - PRESSURE RIPPLES IN A SIMPLE PUMP-MOTOR SYSTEM

The results presented up to now have shown how a reduction in the flow ripple of a pump causes a proportional reduction in pump pressure ripple. Using a simple pump and motor system it can be shown that the pressure ripple produced by a motor may also be reduced in a similar manner.

The system, shown by the photograph, fig. 7.1, consisted of an external gear pump B of 19.21 cm^3/rev. displacement connected via steel pipe of 15 mm internal diameter to an external gear motor E, of 9.47 cm^3/rev. displacement. The motor E was in fact a pump of similar design to pump B but with approximately half the gear width. To provide a load for the motor a fan, similar to the type used for cooling truck diesel engines, was attached directly to the motor driveshaft. A magnetic pick-up was used in conjunction with a suitable number of slots on the attachment flange of the fan to determine the rotational speed of the motor. The facility to bleed off flow at the pump outlet, with a restrictor valve incorporated in the pump outlet port adaptor (shown in fig. 7.4), allowed the speed of the motor to be steplessly variable up to 2800 rev/min.

By collecting pressure data from two transducers, one near the pump and the other near the motor, and varying the length of line between the pump and motor, the inlet source flow characteristics of the motor were evaluated using the Tuned Length Method. Fig. 8.10 shows the comparison between theoretical and evaluated source flows for motor E at a fundamental tooth frequency of 325 Hz (2437 rev/min), a mean pressure of 45 bar and a fluid temperature of 50°C. The evaluated ripple correlates very well in terms of peak to peak amplitude, period and general form, with that predicted by the theoretical model. The two ripples have been superimposed upon slightly different mean flows for clarity.

The amplitude of the first harmonic of flow ripple at 325 Hz was 0.336×10^{-4} m^3/s. This is equivalent to a peak volume variation of 16.45 mm^3 which is 5.3% less than the 17.38 mm^3 predicted by theory. This result for a motor is in contrast to the results obtained for a pump which consistently showed a fundamental component approximately 12% greater than the theoretically predicted. It is possible that in the case of a motor, where the leakage over the gear tips is in the same direction as the tooth motion, a lower variation in flow may result but it was thought unlikely that this could be lower than the theoretical. However modifying the motor with a multi-lift cam mechanism, similar to the MK II design previously described but with only four pistons of 3.63 mm dia, which produced a peak volume variation of 15.36 mm^3 reduced the motor pressure ripple by 94.4%. Assuming that at this condition the motor and mechanism volume variations were exactly out of phase, this implies a motor ripple of only 16.22 mm^3, and that the difference of the original evaluation from the theoretical may not have been due to experimental error.

An example of the combined pressure ripple generating effect of the pump and motor source flows is shown in fig. 8.11. This shows the amplitude of the pressure ripple in the line joining pump and motor in both time and frequency domains.

The fundamental frequency of the pump is 200 Hz (1500 rev/min. shaft speed) and that of the motor is 285 Hz (2140 rev/min.). The mean pressure is 35 bar.

In fig. 8.11 (a), with both pump and motor in a standard or unmodified condition, it is apparent that the majority of the pressure ripple is due to excitation of the largest components of flow ripple at the pump and motor fundamental frequencies. The 15 bar peak to peak amplitude of the combined pressure ripples is almost a half of the 35 bar mean pressure level.

Fig. 8.11 (b) and (c) show the effect of applying MK II cam mechanisms to the pump and motor respectively. In each case the fundamental frequency component of the modified unit is reduced by approximately 95% leaving the fundamental of the unmodified unit clearly dominant. Comparing fig. 8.11 (a) and 8.11 (d) shows the significant reduction achieved by modifying both units simultaneously.

8.3.3 - PRESSURE AND TORQUE FLUCTUATIONS IN A VARIABLE SPEED PUMP-MOTOR-PUMP SYSTEM

As the input and output torque of a pump and motor respectively are a function of the displacement of the particular unit, reducing the variation in instantaneous displacement should also effect a reduction in torque fluctuation.

To investigate the effect of the multi-lift cam mechanism on torque variation a test rig was constructed which allowed measurement of the torque transmitted between a pump and motor. A circuit diagram of the arrangement is shown in fig. 8.12. The motor and load pump were rigidly connected to a torque transducer and the whole assembly mounted in a test stand. The test stand was normally used for the calibration of torque transducers and therefore maintained accurate alignment of the driveshafts. This is important when the coupling has to be rigid in order to make dynamic measurements.

The electric motor and drive pump used were those described in 7.1.2 and the pump outlet port adaptor had the pressure relief and flow bleed-off facilities shown in fig. 7.4. Because the motor used was in fact a pump design, run as a motor, it would not start by the gradual application of inlet pressure. This was due to the unit having plain lead-bronze journal bearings with a relatively high friction coefficient which reduces starting torque efficiency. Motor units are normally built with low friction P.T.F.E coated lead-bronze bearings which greatly improve starting ability. However, it was possible to start the motor by applying a rapid increase in supply pressure. This was achieved by closing a ball valve, close to the motor inlet, increasing the pressure in the drive pump delivery line to approximately 50 bar by using the bleed-off restrictor valve and then quickly opening the ball valve. Once started the motor could be run at speeds between 375 and 2850 rev/min. by varying the amount of flow bled from the drive pump.

The torque transducer was a Vibro-meter type TG-2/B with a nominal capacity of 19 Nm. Using a maximum motor inlet pressure of 75 bar produced a mean torque approximately half the nominal and allowed a reasonable margin to accommodate dynamic torque fluctuations. This inlet pressure also permitted the MK III mechanism to be used within its Hertz stress design limit. The torque transducer, which works on an inductive principle, outputs a signal via a slipringless rotary transmission system using an 8 KHz carrier frequency. This system makes the transducer suitable for measuring high frequency torque variations without the signal noise associated with sliprings. The measurement of variations in torque at first harmonic of gear tooth frequency at maximum motor speed (380 Hz) was well within the capability of the transducer.

The mean torque was displayed in analogue form by a torque meter (Vibro-meter type B-CT-1/A). This meter also had an output which could be connected to a continuous spectrum analyser. This enabled mean and fluctuating torque measurements to be made and a plotting facility coupled to the analyser allowed permanent recording.

Fig. 8.13 (a) shows the variation in the torque transmitted between the load pump C and motor E in their unmodified conditions. The steady state running conditions were as follows:-

fundamental motor frequency	380 Hz
pressure differential across motor	70 bar
pressure differential across load pump	30 bar
measured mean torque	9.45 Nm
mean fluid temperature	50°C

The frequency plot shows a very strong torque ripple component of the load pump and motor fundamental frequency of 380 Hz. Both units are of similar design (see A.3.1) each having eight teeth. Theamplitude of this fundamental component is the resultant effect of the interaction of the individual motor and load pump torque ripple components at this frequency and as the phase of the gear teeth of one unit relative to the other was not known the measured component can only be viewed as the combined effect of the two units. Components with smaller amplitudes are also evident at 332, 428, 760 and 855 Hz. These correspond to the seventh, ninth, sixteenth and eighteenth harmonics of the motor shaft frequency of 47.5 Hz. The sixteenth harmonic of shaft frequency is also the second harmonic of tooth frequency. The components at 200 and 400 Hz are the first and second harmonics of drive pump fundamental frequency, showing that pressure fluctuations from the drive pump cause torsional variations in the output torque of the motor. This effect would also occur in the case ofa linear actuator and emphasizes the need to reduce pump pressure ripple if precise control of a torsional or linear load is required.

The time domain plot shows the modulating effect often called "beating" produced by two components at similar but not identical frequencies. Here the effect is caused by the 380 Hz fundamental component of the motor and the 400 Hz component of the drive pump.

The effect of modifying the instantaneous displacement of the motor with a MK III mechanism is shown in fig. 8.13 (b). Monitoring the pressure ripple in the motor inlet line showed that the mechanism reduced the fundamental component of motor pressure ripple by 86%. The fundamental component of the motor-pump torque ripple was reduced by 79% from 1.68 to 0.36 Nm rms. If it is assumed that there is some loss in torque application due to piston-follower to cam contact then the modification in displacement variation effected by the mechanism appears to have an equal effect on torque and pressure ripples, and the fundamental component of torque ripple is mainly attributable to the motor. To verify this a MK II mechanism was applied to the load pump C. This was seen to produce a 95% reduction in outlet pressure ripple at the fundamental of 380 Hz, but only reduced the torque ripple by a further 0.12 Nm rms, equivalent to a total reduction of 85% of the original. As the remaining torque and pressure ripples, after the motor modification, are of similar proportions the mechanism must be having a torque ripple reduction effect proportional to its reduction in flow ripple.

Fig. 8.13 (c) shows the effect of MK III mechanisms in both the drive pump B and motor E. The torque ripple components at 200 and 400 Hz due to drive pump pressure ripple have been effectively cancelled and the time trace no longer exhibits the modulation seen in (a).

To reduce the torque ripple of a pump or motor the piston-followers have to extract torque from the driveshaft during periods when the instantaneous torque is greater than the mean torque and provide a torque input to the shaft when the instantaneous value is less than the mean. The input and output of torque to and from the driveshaft is achieved through contact between the piston-followers and the cam. Any frictional losses within the cam mechanism, due to motion of the piston-followers relative to the cam or piston-housing will appear as an increase in pump input torque or decrease in motor output torque.

The mean input torque required to drive gear pump D, at a mean pressure of 30 bar, with and without a MK II mechanism is shown in fig. 8.14 for a range of pump speeds. The effect of viscous friction within the pump is apparent by the gradual increase in mean torque with increase in rotational speed. Adding the mechanism, which in this case had four piston-followers, had the effect of increasing the mean torque by an amount which was reasonably constant over the range of speeds tested, approximately equivalent to 2.5% of the mean. Drive system limitations prevented the investigation of mechanism torque losses over the full pressure range of the pump, but doubling the mean pressure to 60 bar produced a 3% increase in the mean torque. Obviously future mechanism development must include testing at higher pressures to assess typical mechanical efficiencies of modified pumps. As mechanism losses are proportional to the number of piston-followers, for a given piston diameter, varying the number of piston-followers with the displacement of the pump is a means of reducing the variation in mechanical efficiency of a range of pump sizes.

The Effect of Rotational Speed on Mechanism Pressure Ripple Reduction

The fixed volume variations produced by the mechanism at the fundamental tooth frequency create flow variations which vary in amplitude dependent upon the rotational speed of the cam. As the cam is attached directly to the pump driveshaft the flow ripples of the pump and mechanism maintain constant synchronization. Hence the reduction effect of the mechanism is constant if the pump volume variation is independent of rotational speed.

Fig. 8.15 shows the effect of MK II mechanisms on the fundamental components of pressure ripple produced by two pumps C and D over a range of fundamental frequencies. The mechanism applied to pump D was deliberately selected to produce a reduction which would leave a larger residual component to ease identification and measurement. The tests show that the mechanism had a constant effect, an 82% reduction, over the whole range of speeds at a constant mean pressure of 30 bar. This shows that the fundamental component of pump volume variation is virtually independent of rotational speed. Other harmonics up to the fourth also exhibited similar characteristics. This combined with the fact that the flow ripple and hence volume ripple of this type of pump does not vary significantly with mean pressure (fig. 7.9) makes the external gear pump ideally suited for modification with a simple constant displacement mechanism.

The results for pump C in fig. 8.15 show the modification in fundamental pressure ripple effected over a range of speeds and pressures. The outlet flow was discharged through a fixed restriction so that the mean pressure varied with the pump speed up to a maximum of 55 bar at 300 Hz. Here again the reduction is reasonably constant over the range of speeds and pressures with reductions in fluctuating pressure levels varying by only 4% from 95 to 99%.

8.3.4 - VOLUMETRIC EFFICIENCY LOSS DUE TO MK II MECHANISM

The additional pump internal leakage caused by modifying the pump with a typical multi-lift cam mechanism was measured for a number of mean pressures up to 200 bar for a fluid temperature of 50°C.

The output flow from a gear pump B, of 19.21 cm^3/rev. displacement, was measured using a positive displacement flow meter (Brodie-Kent model 31 X) at a nominal speed of 1500 rev/min. A restrictor valve in the outlet line was used to vary the mean outlet pressure. The pump was initially tested with a blank in place of the piston-follower housing so that retesting with the mechanism in place allowed the leakage past the piston-followers to be determined in isolation. The difference between the flow rate with the blank and mechanism at each pressure was expressed as a proportion of the theoretical pump flow and is shown in fig. 8.16. The loss in pump volumetric efficiency would appear to be linear with increasing mean pressure with a maximum loss of 3.3% at 200 bar. The linear relationship implies laminar flow in the small clearances between the piston-followers and the housing.

The estimated leakage past 8 pistons, assuming a maximum diametral clearance of 0.023 mm was 0.0062 l/s at 200 bar mean pressure and 50°C. This was calculated using Reynolds' equation for laminar flow through small annular clearances with a factor of 2.5 included to account for axial eccentricity of the piston, ref. (39). The measured leakage for the same conditions was 0.013 L/s indicating that a factor of 5 is more appropriate to the operating situation of the pistons. Because of the action of the cam the eccentricity of the piston is likely to oscillate within the piston clearance in the bore making the assumption of axial eccentricity invalid.

There is no reason why a reduction in volumetric efficiency of the order of 3% cannot be achieved over a range of pump displacements. As volume ripple is proportional to mean displacement, for a gear pump, smaller displacement units can have proportionally fewer piston-followers and hence proportionally less leakage.

8.4 - THE EFFECT OF PRESSURE RIPPLE REDUCTION ON AIRBORNE NOISE

The airborne noise radiated by hydraulic systems is a direct result of the excitation of component exterior surfaces in contact with the air. These surface vibrations are caused by the transmission of load variations or structure borne noise produced directly by mechanical oscillation or indirectly via pressure fluctuations in the working fluid. To achieve low levels of airborne noise it is therefore necessary to reduce both the direct and indirect sources of structural vibration.

The effect of significantly reducing the fluid borne noise of a pump, on pump and system radiated airborne noise was investigated in a non-anechoic environment.

8.4.1 - PUMP RADIATED AIRBORNE NOISE

The level of airborne noise produced by a gear pump B was measured, before and after modification with a MK III multi-lift cam mechanism, using a sound level meter (Bruel and Kjaer type 2204). The pump installation was similar to that shown in fig. 7.1, the pump being rigidly mounted and connected by rigid steel pipes on both inlet and outlet. The 2.1 metre length of outlet line was terminated with a restrictor valve to enable control of the pump mean pressure level. The electric motor was covered, with 50 mm thick rubber foam supported on 3 mm thick hardboard to form a semi-circular enclosure, as shown in fig. 8.17, to reduce the influence of electric motor noise on the measurement of airborne noise radiated by the load system. The load system will be described in 8.4.2.

Airborne noise measurements were made at a distance of 0.5 metre directly behind the pump for operating conditions of 75 bar mean pressure, 1500 rev/min. and 50°C. Although the mechanism effected significant reductions in the amplitude of the first four harmonics of pump outlet pressure ripple, similar to those shown in fig. 8.9, there was no change in the overall airborne noise level. This was not completely unexpected as the measurements included airborne noise generated by the electric motor, the test stand and the pipework. What it does show is that the airborne noise radiated from the pump is caused predominantly by direct mechanical vibration rather than pump pressure ripple.

This was verified by positioning the microphone of the sound level meter close to the rear face of the pump, approximately 30 mm away, so that effects from sources other than the pump would be minimised. Observing the output from the sound level meter on a continuous spectrum analyser showed that although the mechanism had a reduction effect at the first and second harmonics it also caused an increase in the high frequency content of the signal.

The application of an accelerometer to the centre of the rear face of the pump also showed an increase in the velocity of vibration of the cover due to the mechanism. This increase in structural vibration of the pump occurred at harmonics of pumping frequency over the range 2 to 4 KHz and proved to be dependent upon the method of orientating the piston-followers to the cam. A significant reduction in the structural vibration attributable to the mechanism was achieved by aligning the fast moving piston-followers with a moving rather than stationary reference. This was achieved by transferring piston-follower alignment from the radial slots in the piston housing to a duplicate set of piston-followers positioned in close axial proximity to the first. In this manner each row was made to align the other, with each piston-follower making only a single contact with the piston housing through the piston diameter.

To evaluate fully the effects of pressure ripple reduction on pump radiated airborne noise it would be necessary to test the unit in isolation in an anechoic chamber. Only then could the additional structure borne effects of the mechanism on the airborne noise of the modified unit be determined. The indications are that considerable reductions in gear pump pressure ripple may be achieved by a mechanism of the type described without significantly increasing the airborne noise radiated from the pump casing.

8.4.2 - SYSTEM RADIATED AIRBORNE NOISE

In hydraulic systems where the pump casing is a comparatively small part of the total surface area which may be excited by fluid borne noise or where the pump is situated some distance away from the system, airborne noise from the system can be of greater importance than that from the pump.

Fig. 8.17 shows a hydraulic system which was used to investigate the effect of pump pressure ripple on system radiated airborne noise. The system was originally designed as a portable teaching rig intended to demonstrate certain characteristics of a pressure relief valve. To facilitate this the relief valve was rigidly attached to a vertical steel panel mounted on an all metal trolley of angle iron construction. The 1.7 mm thick steel panel was reinforced at the top and bottom by 15 mm square steel bars along its length. The relief valve was rigidly mounted on a "L" shaped bracket which was attached via bolts through the panel to a steel bar 8 mm thick, 80 mm wide welded to the steel bars at the top and bottom of the panel.

Four differential pressure transducers were rigidly bolted to the panel, but none of the 3 mm internal diameter tungum tube interconnecting the transducers passed through or contacted the panel. The electrical power supplies and signal conditioning boards for the pressure transducers and the flow meter were housed together with inlet and outlet mean pressure gauges (snubbed Bourden type) in an eye level steel canopy.

The canopy, made of sheet steel similar to the back panel, and only open at the bottom was supported on an angle iron frame extending down to the top of the trolley.

The outlet flow from the test pump was delivered to the relief valve via a 2.1 metre length of 15 mm internal diameter steel tubing and a further 1.8 metres of nominally 12.5 mm bore flexible hose. The return line from the flow meter to the reservoir was also via flexible hose in order to reduce the transmission of structural vibration from the pump and test stand to the load system. Thus the airborne noise produced by the panel system would be predominantly due to excitation of the structure by the pressure fluctuations in the fluid.

The effect, of pressure ripple, on the airborne noise from the system was measured with a sound level meter positioned 1 metre in front of and at the same height as the relief valve as shown in fig. 8.17. The pump delivered 0.45 l/s at 1500 rev/min. at 50°C and the test valve was set to relieve at a mean pressure of 50 bar.

The unweighted overall noise level with the pump in its standard condition was 91 dB. Modifying the pump with a MK III mechanism reduced this to 81 dB. Fig. 8.18 shows the output from the sound level meter, for these two conditions, as frequency spectra and also presents the corresponding pressure ripple frequency spectra measured at the pump outlet flange.

The strong correlation between pump pressure ripple and system airborne noise is clearly evident. Pressure fluctuations at the first three harmonics of the 200 Hz pump fundamental produced structural excitation of the hydraulic system and corresponding pressure fluctuations in the air. Removing the source of excitation by reducing the pressure ripple produced by the pump caused a proportional reduction in the airborne noise radiated by the system at the frequencies affected.

HARMONIC OF 200 Hz	FLOW RIPPLE Q_s		VOLUME RIPPLE V_s	
	AMPLITUDE $\times 10^{-3}$ M^3/S	PHASE DEGREE	AMPLITUDE mm^3	PHASE DEGREE
1	·0495	173	40·00	83
2	·0144	57	5·82	-123
3	·00805	-60	2·17	30
4	·00562	175	1·13	175

RESULTS AT 50 BAR AND 1500 REV/MIN.

TABLE. 8.1 FLOW AND VOLUME RIPPLE HARMONIC COMPONENTS FOR IP3060 GEAR PUMP A

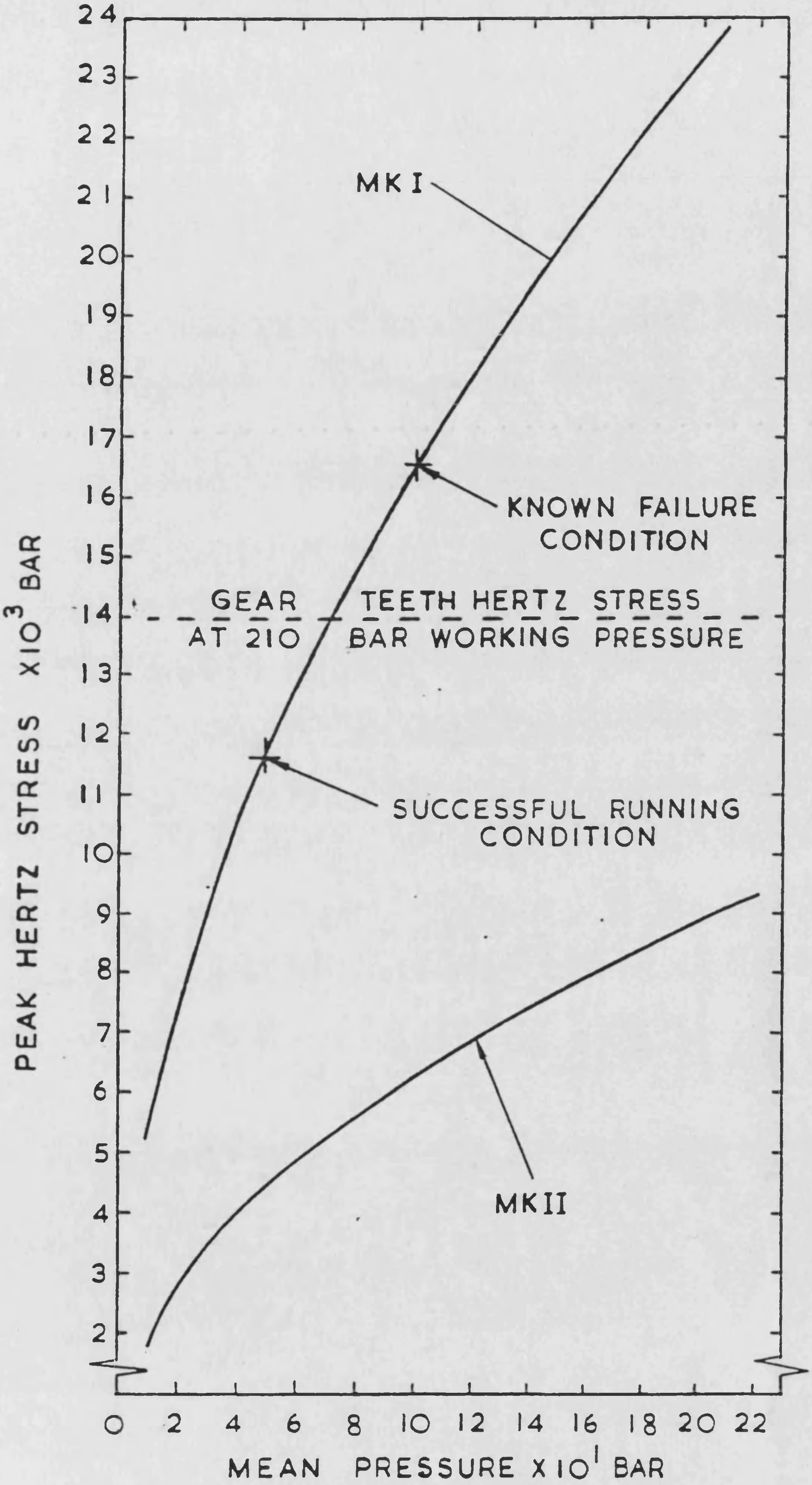


FIG. 8.1 VARIATION IN PEAK HERTZ STRESS WITH MEAN PRESSURE FOR MKI AND MKII MULTI-LIFT CAM MECHANISMS

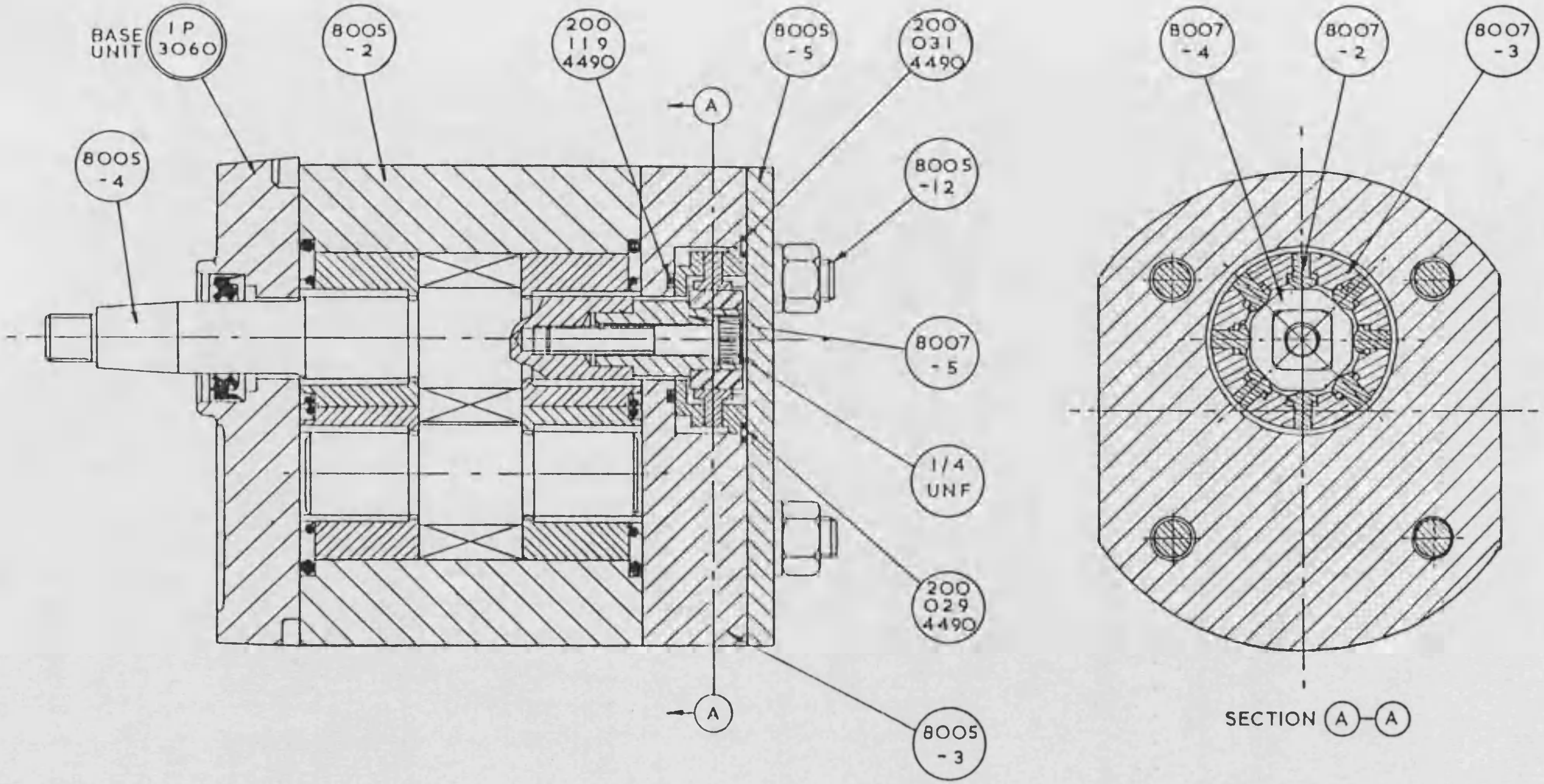


FIG. 8.2 SCHEME OF IP3060 WITH MKII MULTI-LIFT CAM MECHANISM

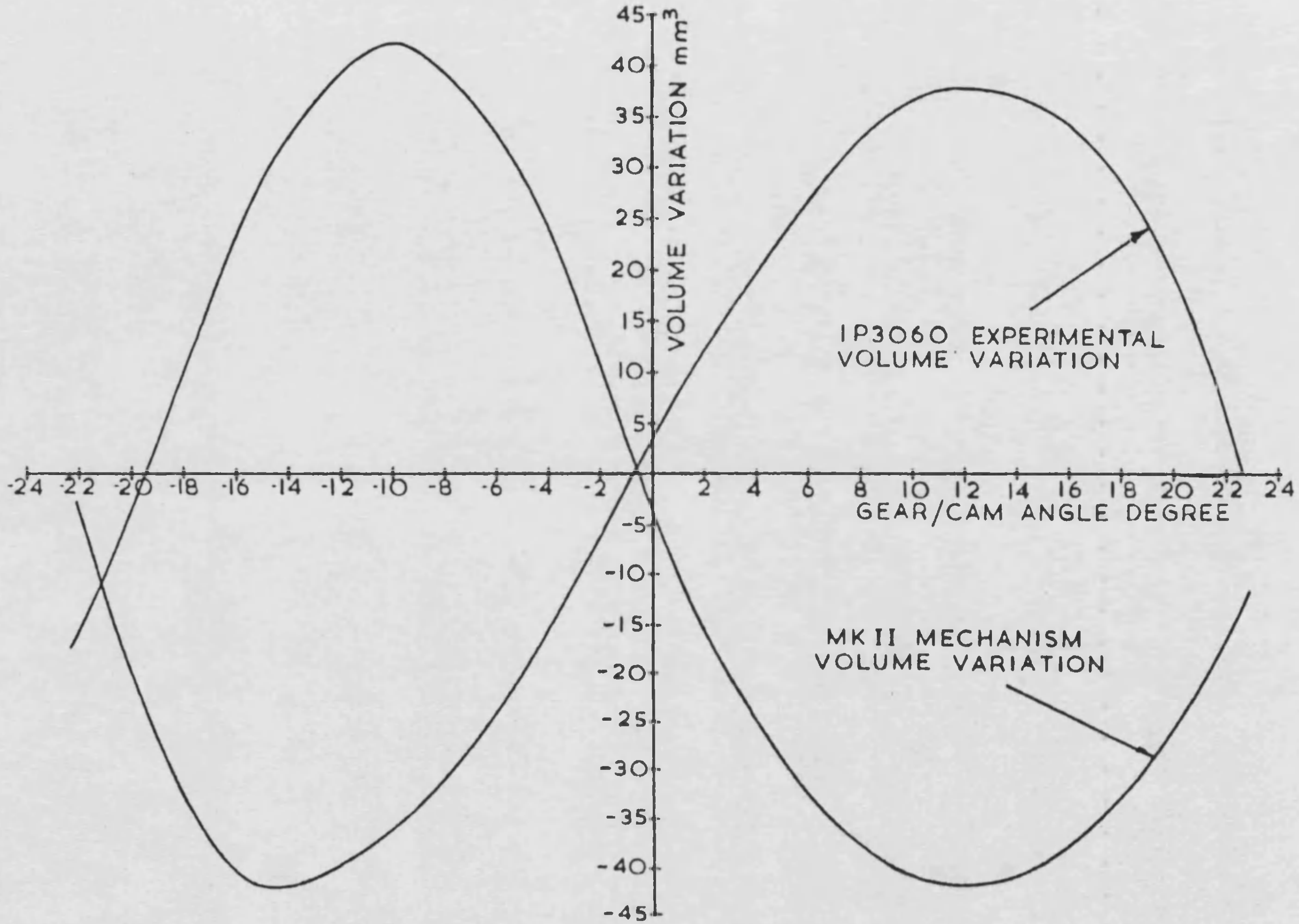
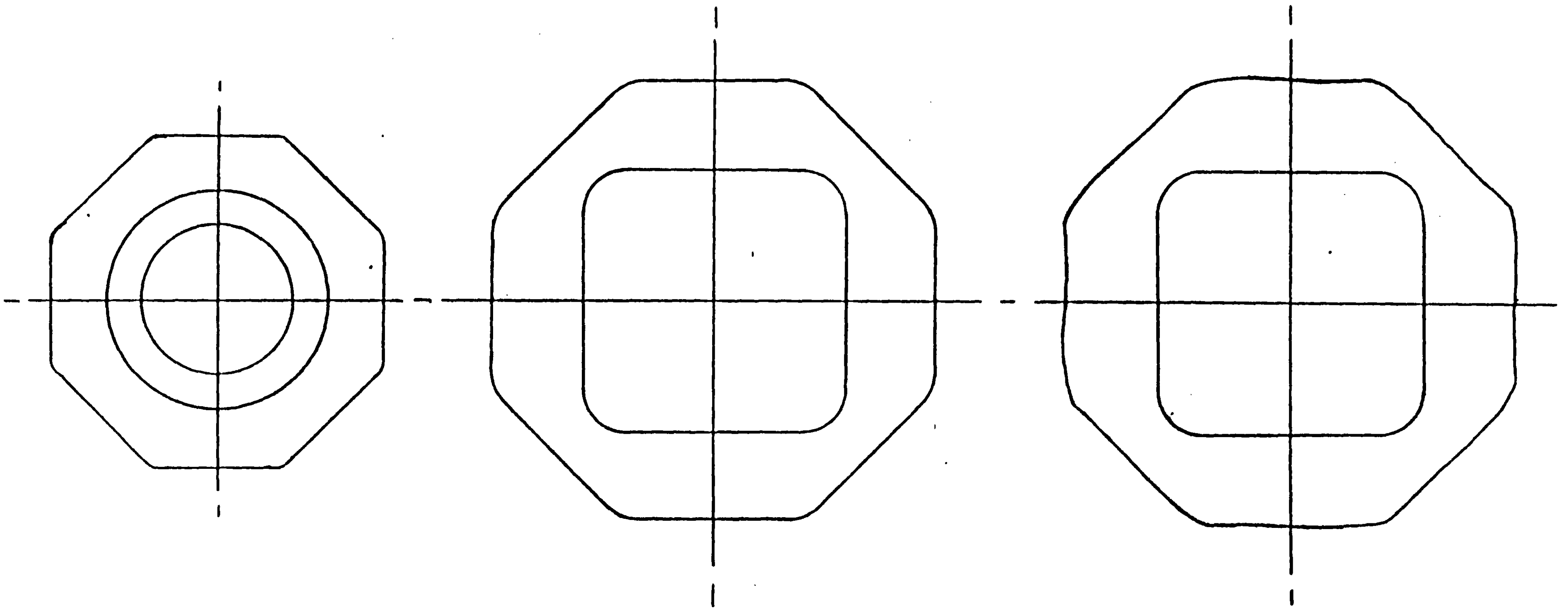
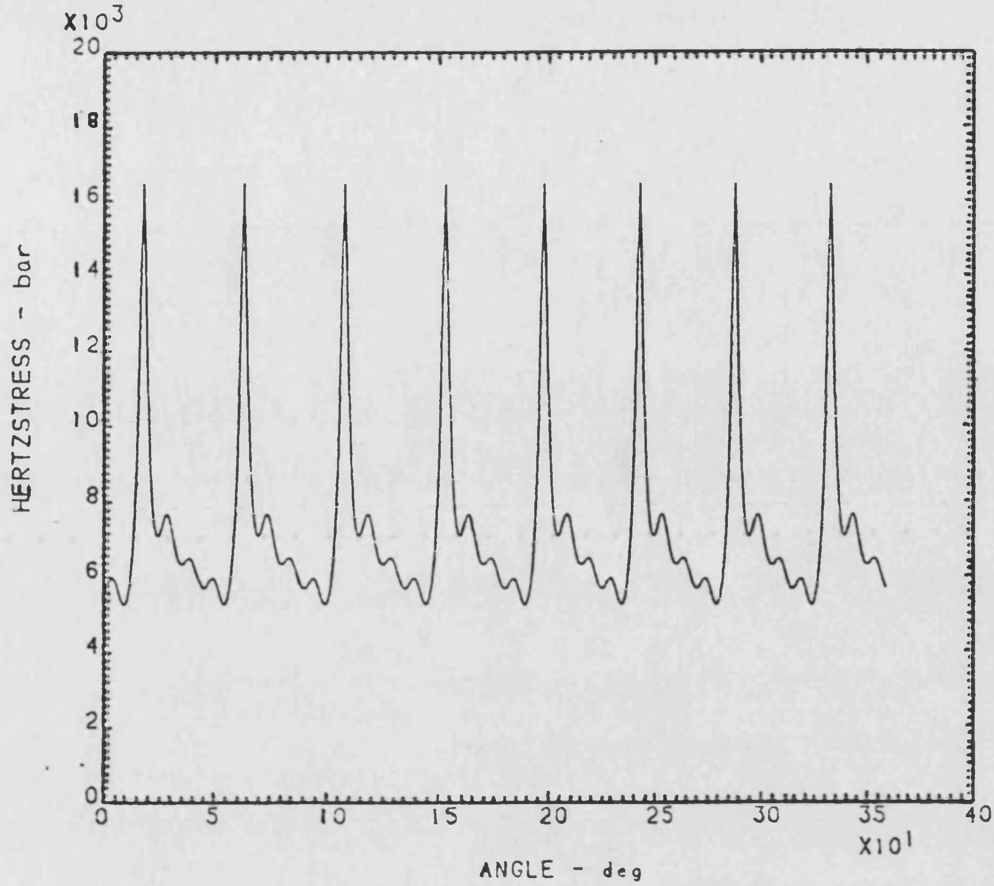


FIG. 8.3 COMPARISON OF IP3060 AND MKII MULTI-LIFT CAM MECHANISM VOLUME VARIATIONS

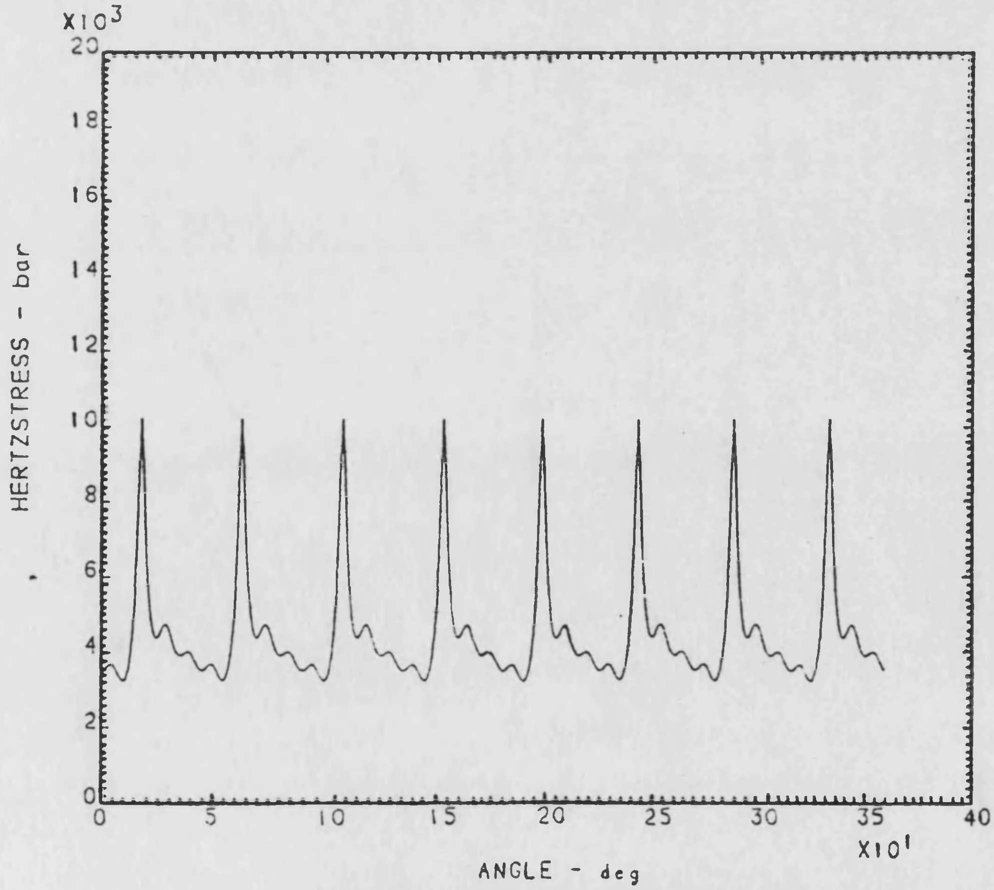


(a)MK I (b)MK II (c)MK III

FIG. 8.4 MK I, MK II AND MK III CAM PROFILES



(a) 210 BAR WORKING PRESSURE



(b) 80 BAR WORKING PRESSURE

FIG. 8.5 MKIII CAM HERTZ STRESS CHARACTERISTICS

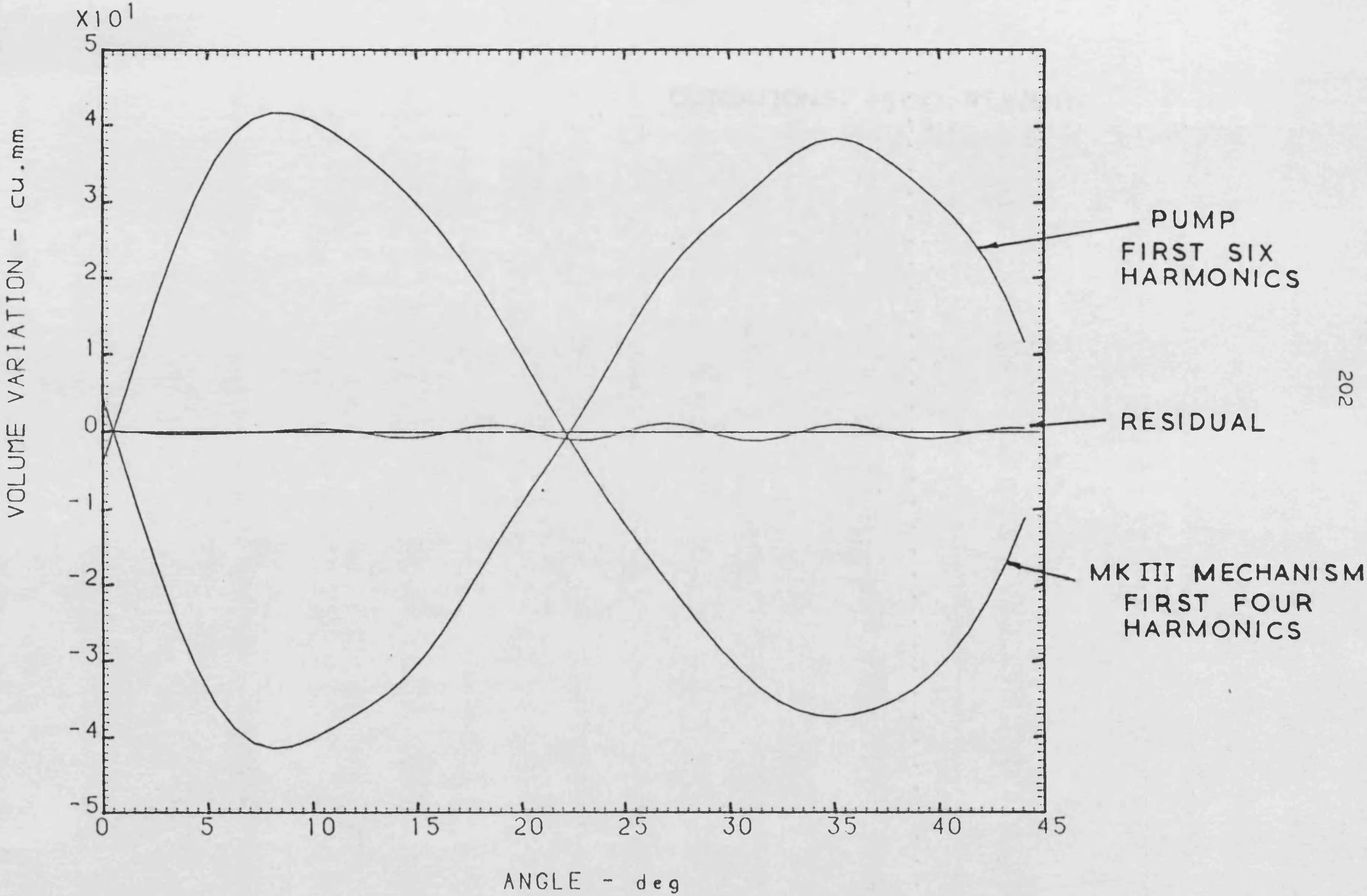


FIG. 8.6 COMPARISON OF IP3060 AND MK III MECHANISM VOLUME VARIATIONS

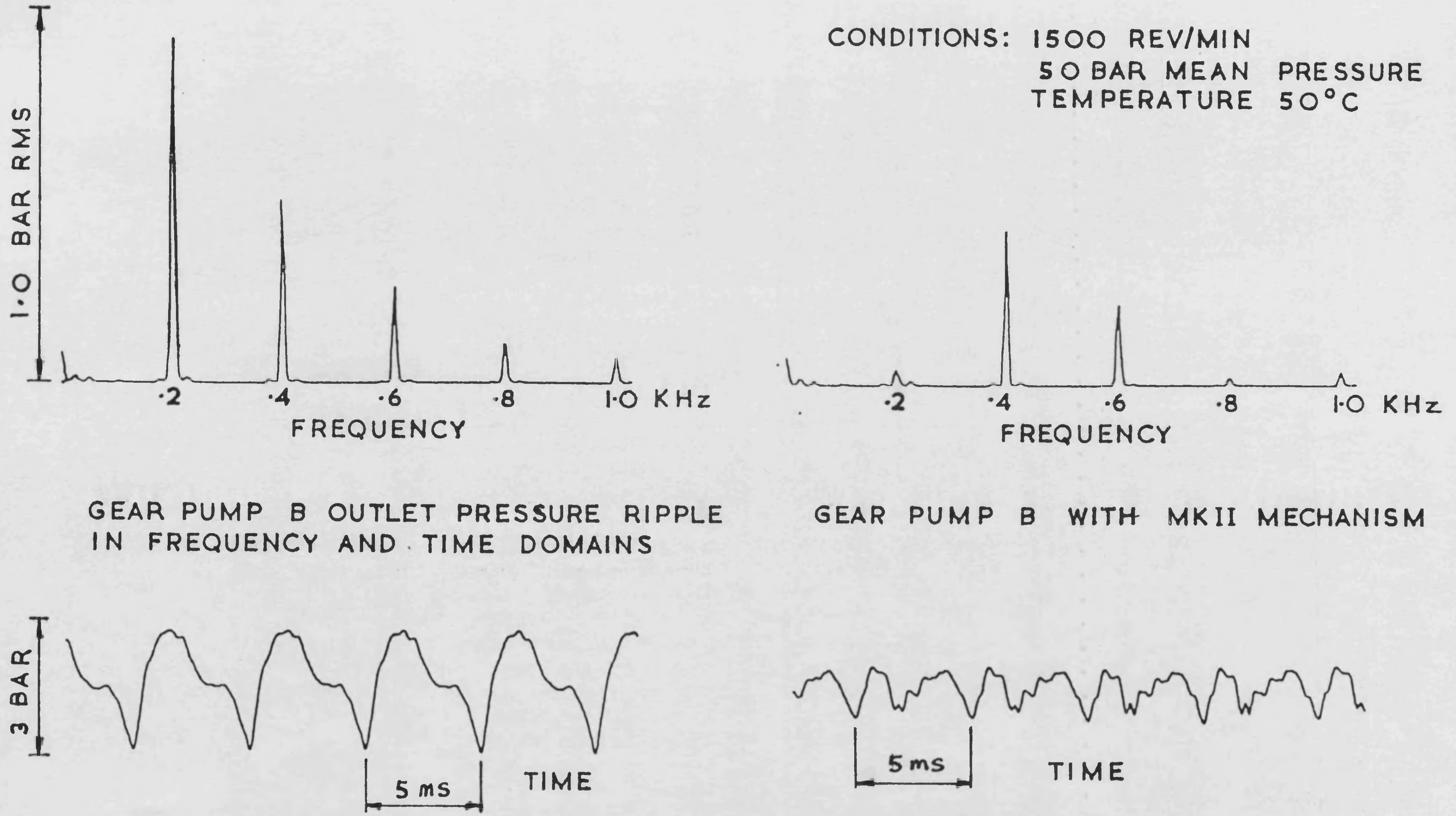


FIG. 8.7 GEAR PUMP B OUTLET PRESSURE RIPPLE BEFORE AND AFTER MODIFICATION WITH MK II MECHANISM

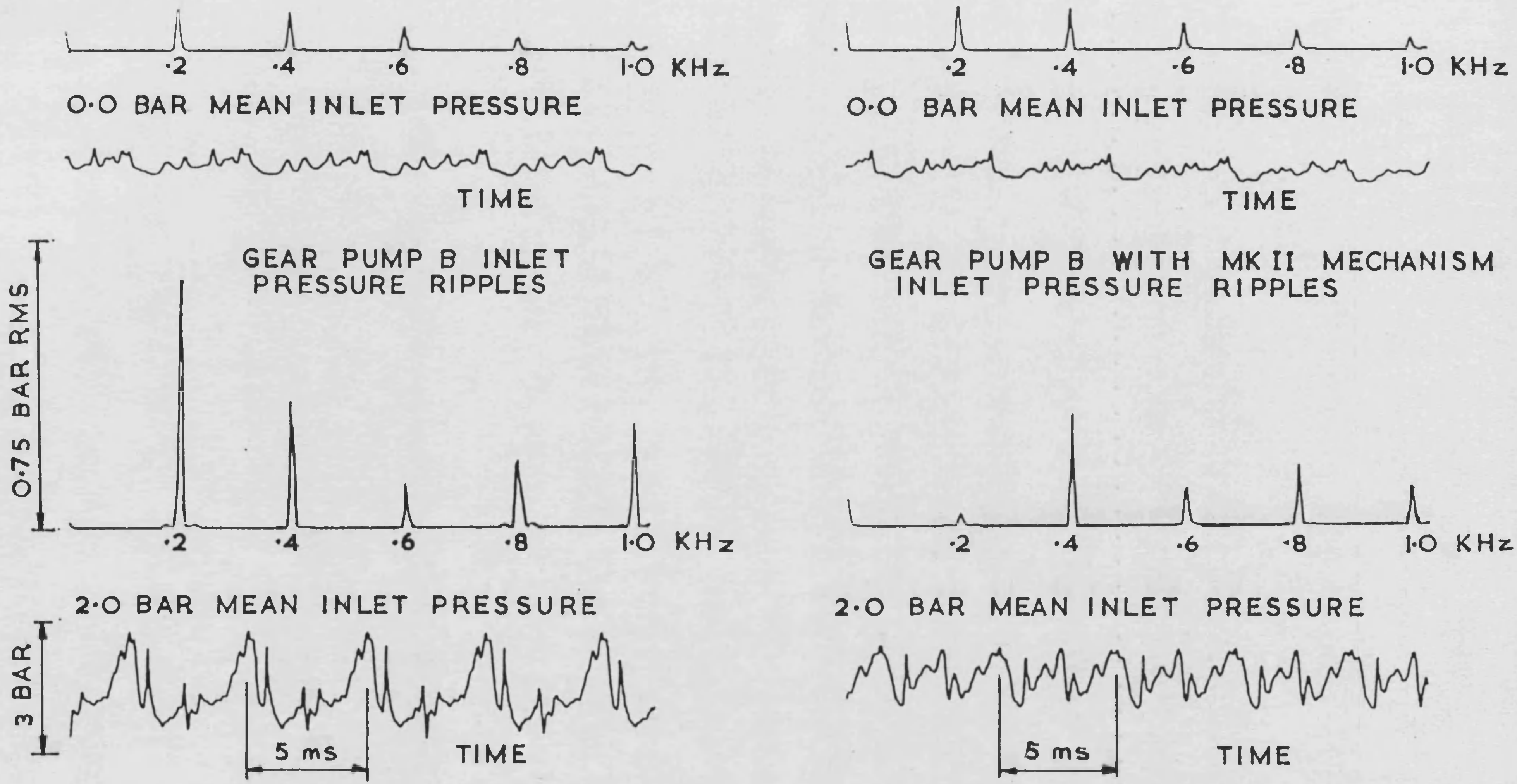


FIG. 8.8 GEAR PUMP B INLET PRESSURE RIPPLE BEFORE AND AFTER MODIFICATION WITH MK II MECHANISM

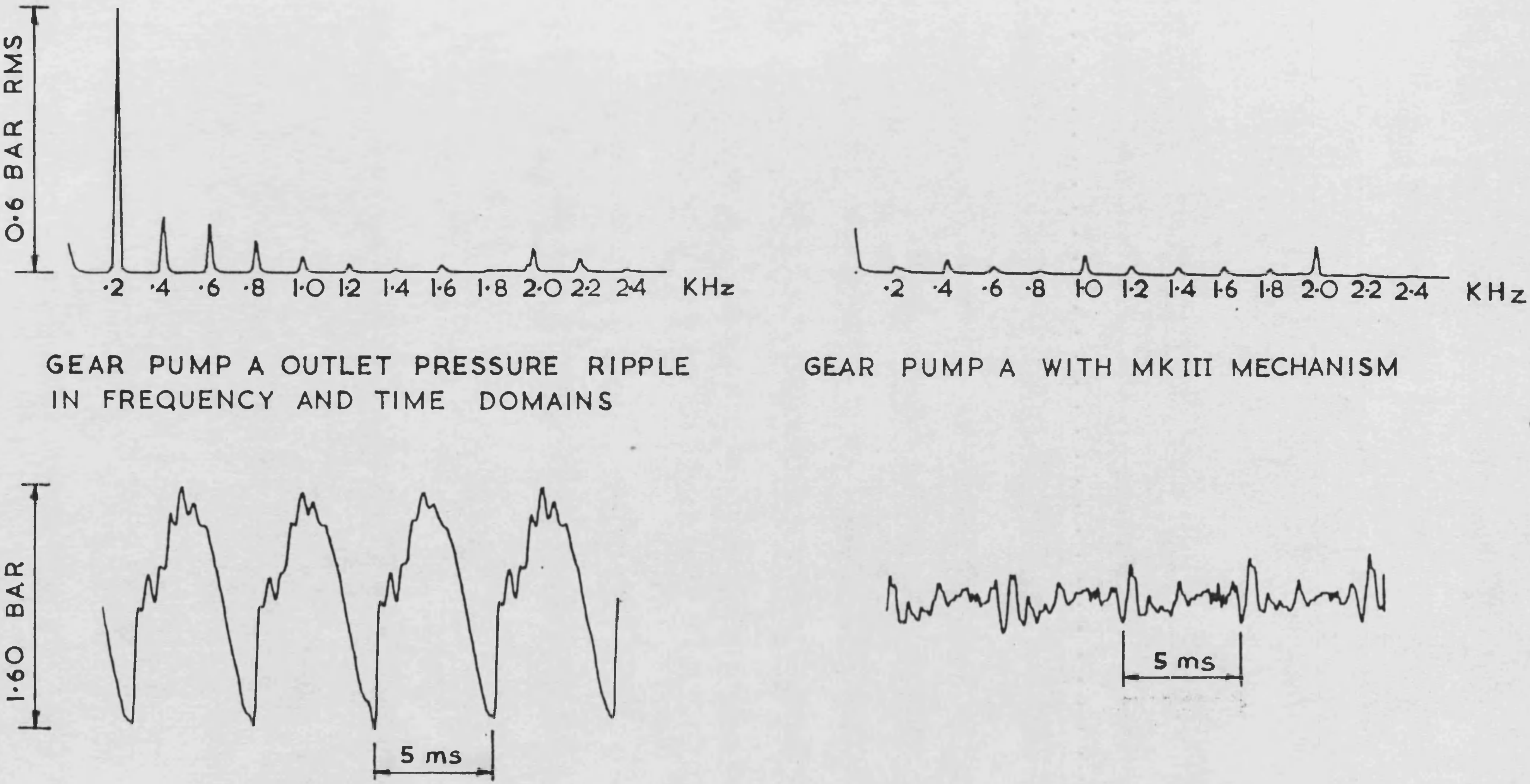
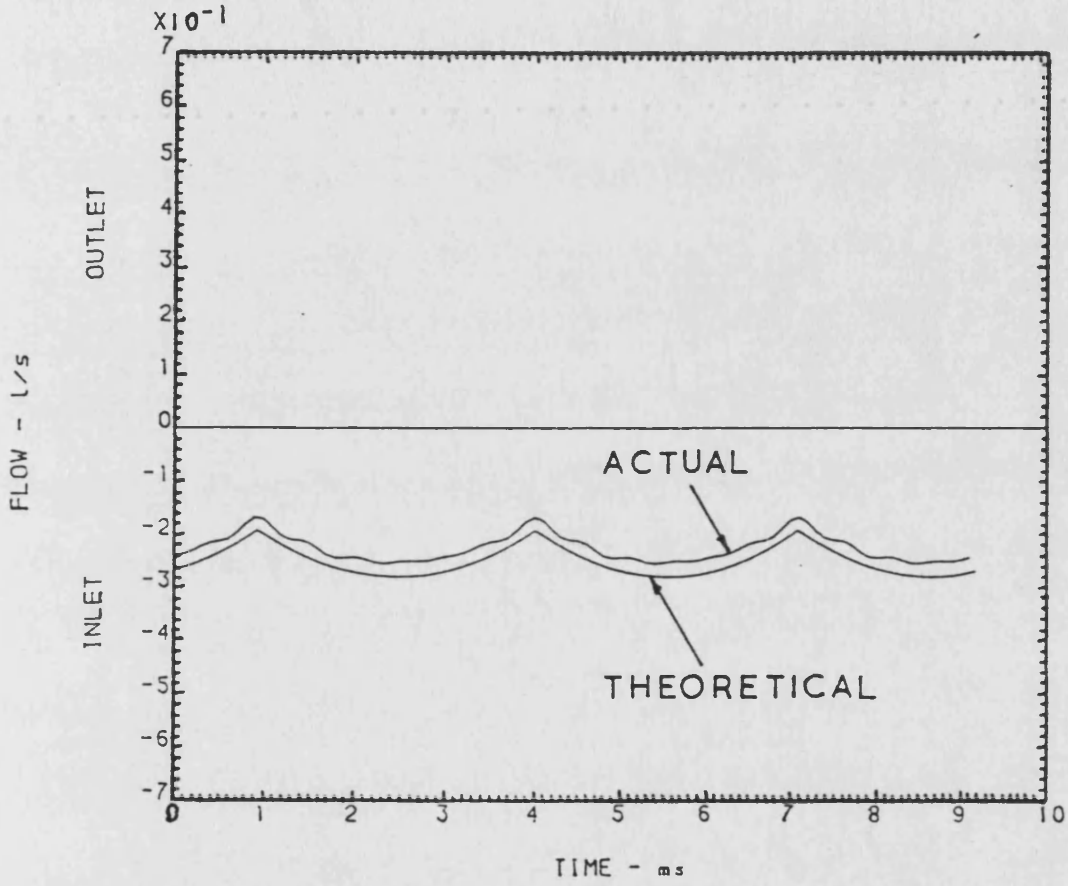


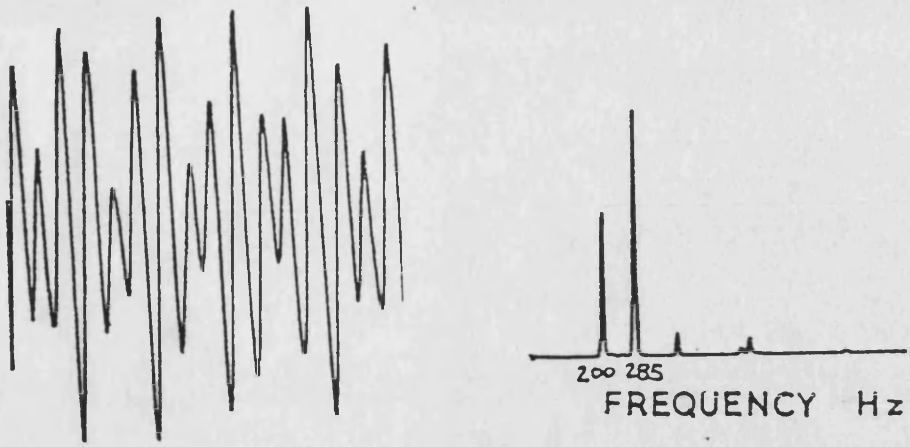
FIG. 8.9 GEAR PUMP A OUTLET PRESSURE RIPPLE BEFORE AND AFTER MODIFICATION WITH MK III MECHANISM



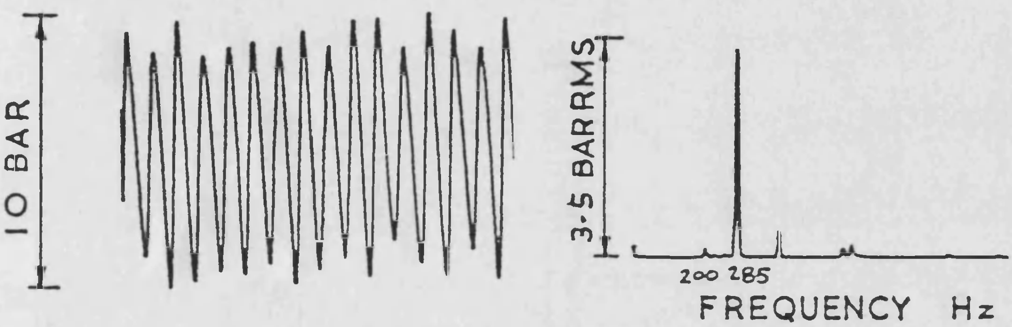
SPEED - 2437 REV/MIN

MEAN PRESSURE - 45 BAR

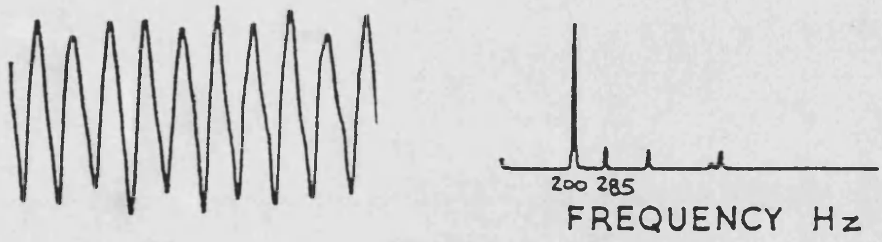
FIG. 8.10 INLET FLOW VARIATION OF EXTERNAL GEAR MOTOR E



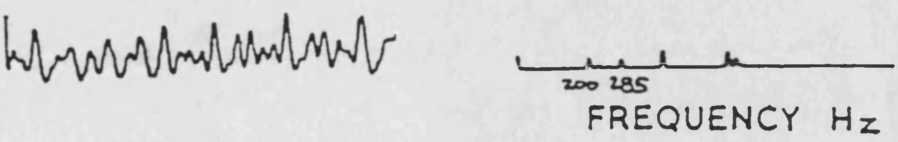
(a) STANDARD PUMP B AND MOTOR E



(b) STANDARD MOTOR E AND PUMP B WITH MKII MECHANISM



(c) STANDARD PUMP B AND MOTOR E WITH MKII MECHANISM



(d) PUMP B AND MOTOR E WITH MKII MECHANISMS

FIG. 8.11 TYPICAL PRESSURE RIPPLES IN A PUMP-MOTOR SYSTEM BEFORE AND AFTER SOURCE FLOW MODIFICATION

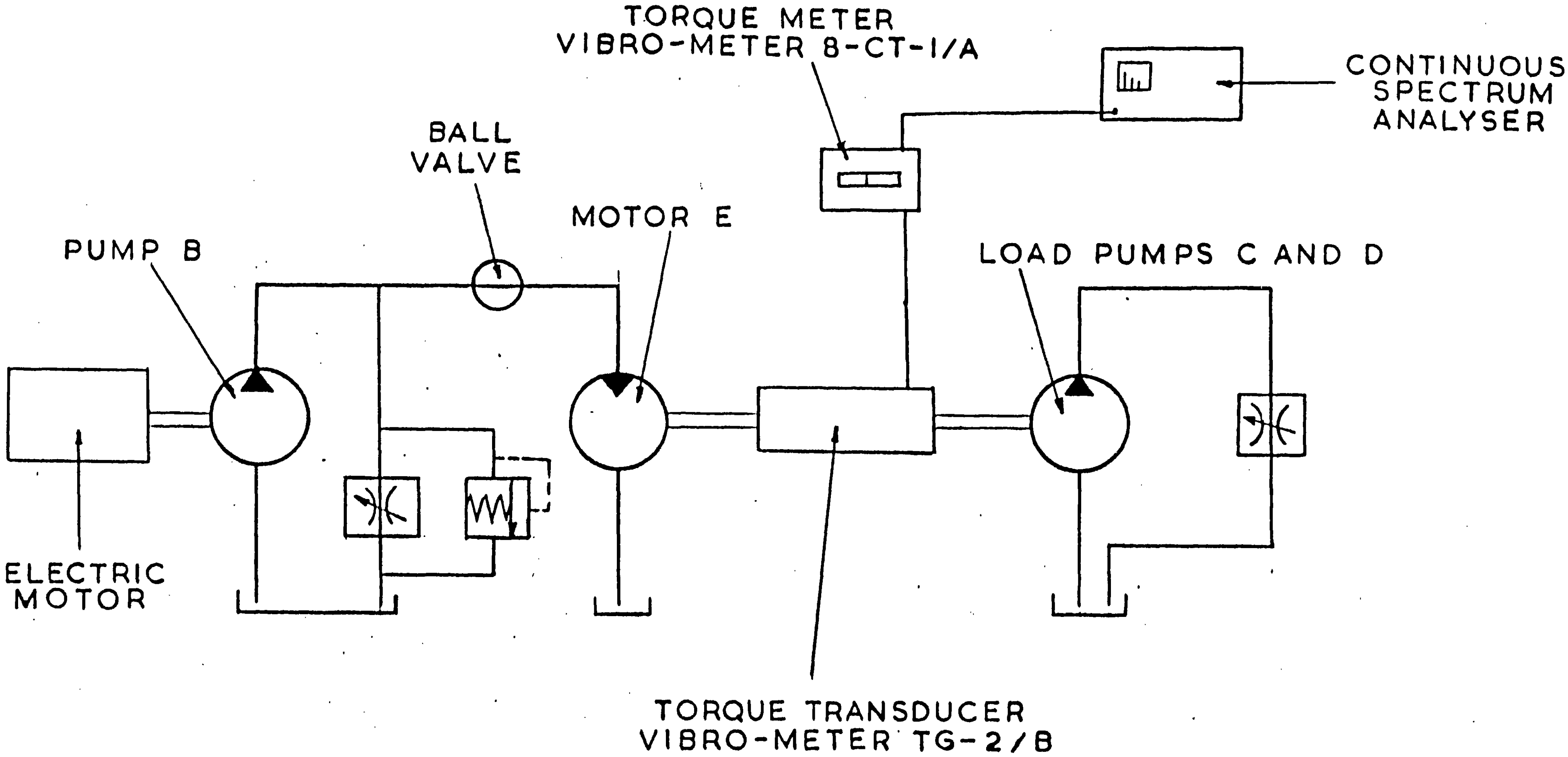
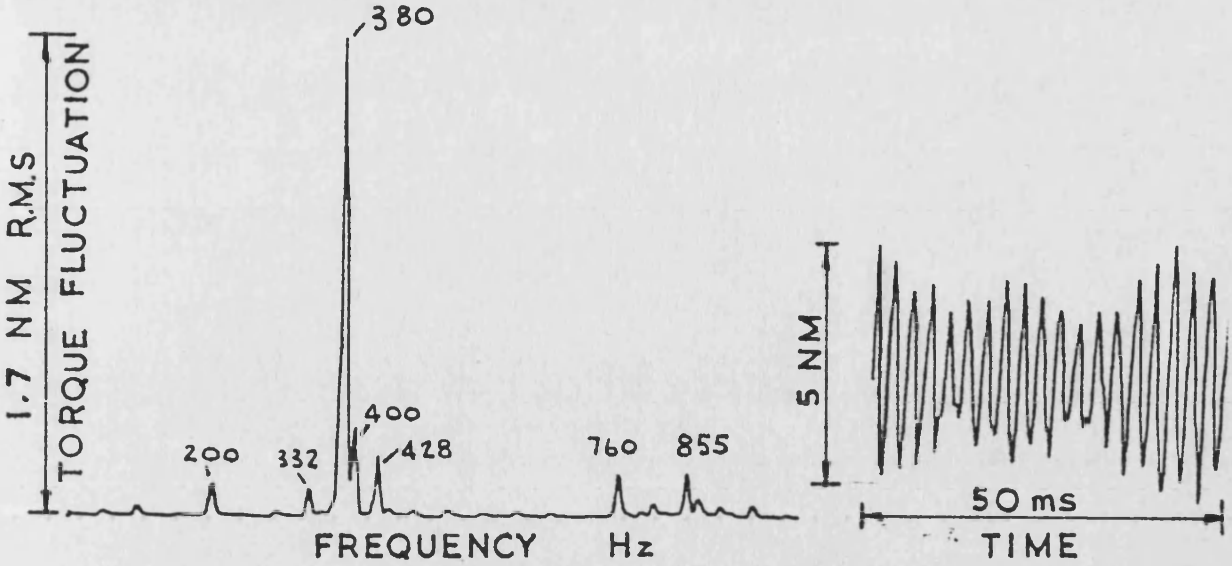
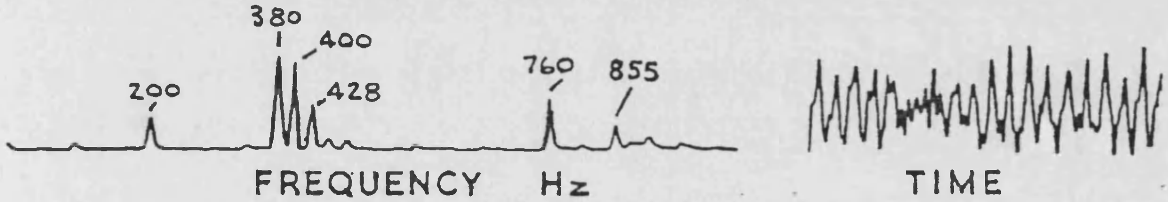


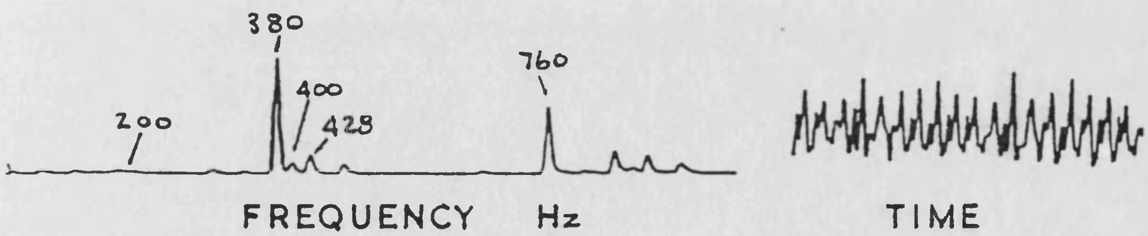
FIG. 8.12 CIRCUIT DIAGRAM OF VARIABLE SPEED RIG USED FOR TORQUE RIPPLE MEASUREMENTS



(a) STANDARD DRIVE PUMP B AND MOTOR E



(b) STANDARD DRIVE PUMP B AND MOTOR E WITH MKIII MECHANISM



(C) DRIVE PUMP B AND MOTOR E WITH MKIII MECHANISMS

FIG. 8.13 TORQUE FLUCTUATIONS BETWEEN A PUMP AND MOTOR

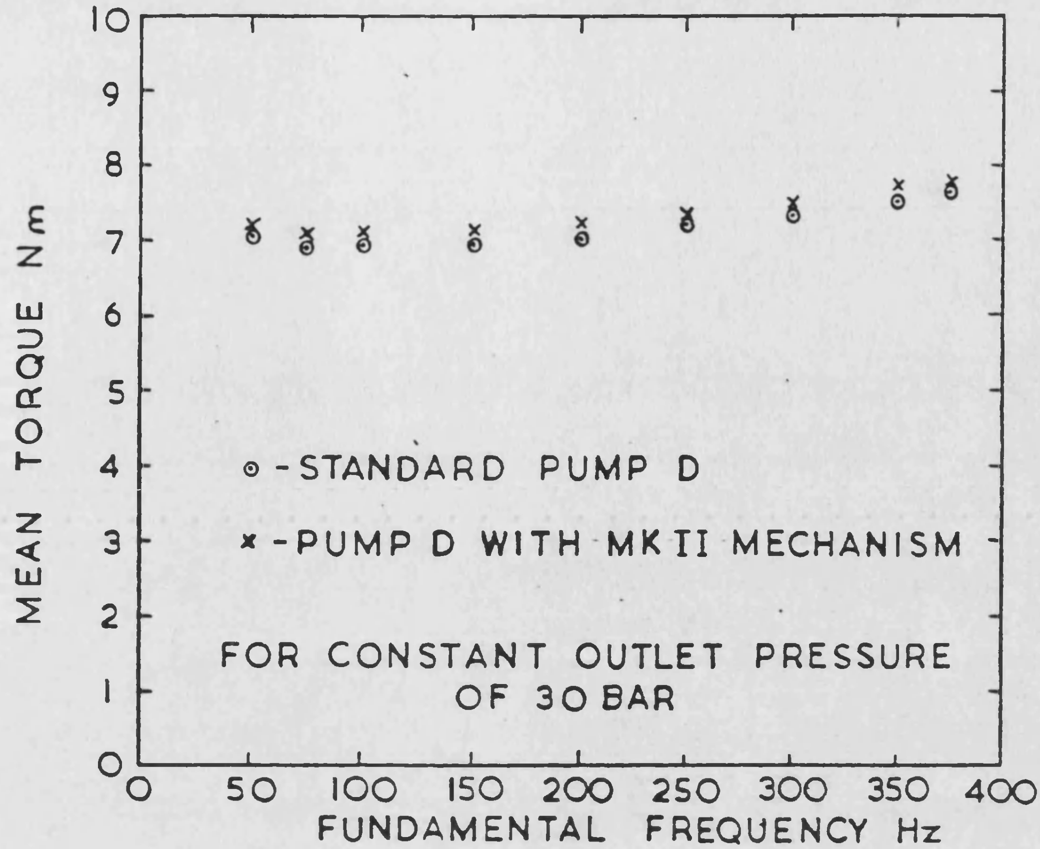


FIG. 8.14 EFFECT OF MECHANISM ON MEAN TORQUE OVER PUMP SPEED RANGE

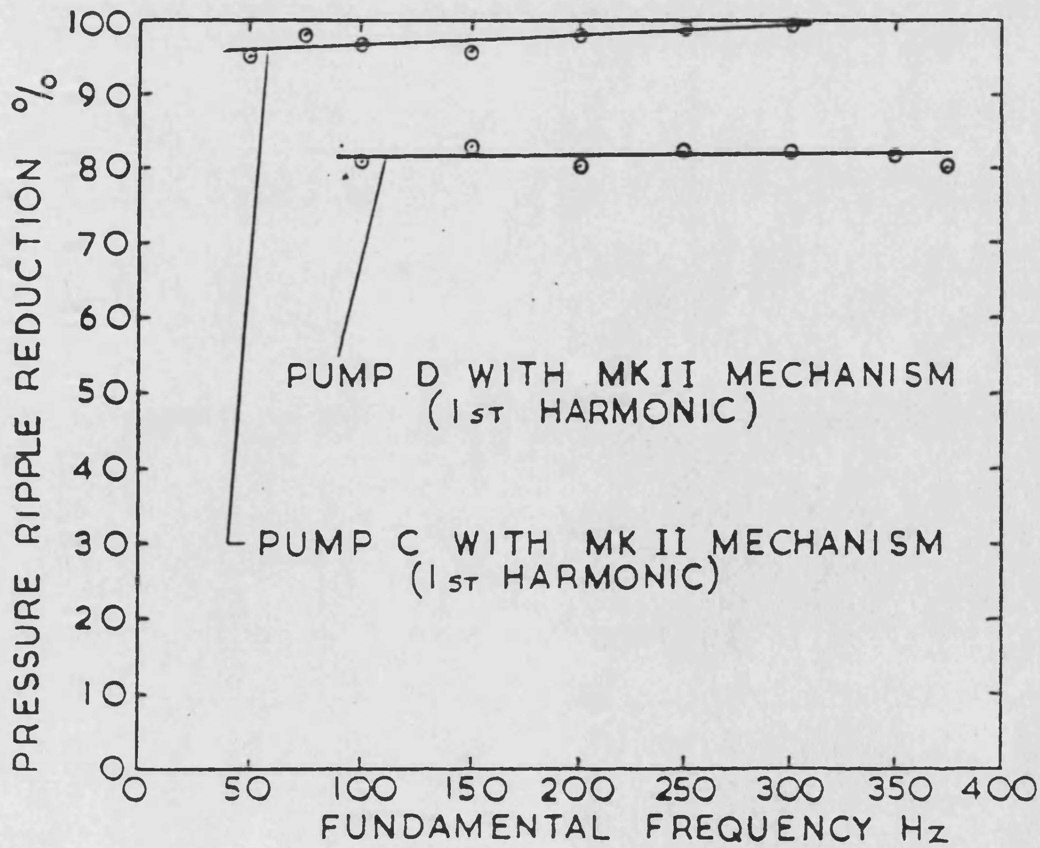


FIG. 8.15 EFFECT OF MECHANISM ON PRESSURE RIPPLE OVER PUMP SPEED RANGE

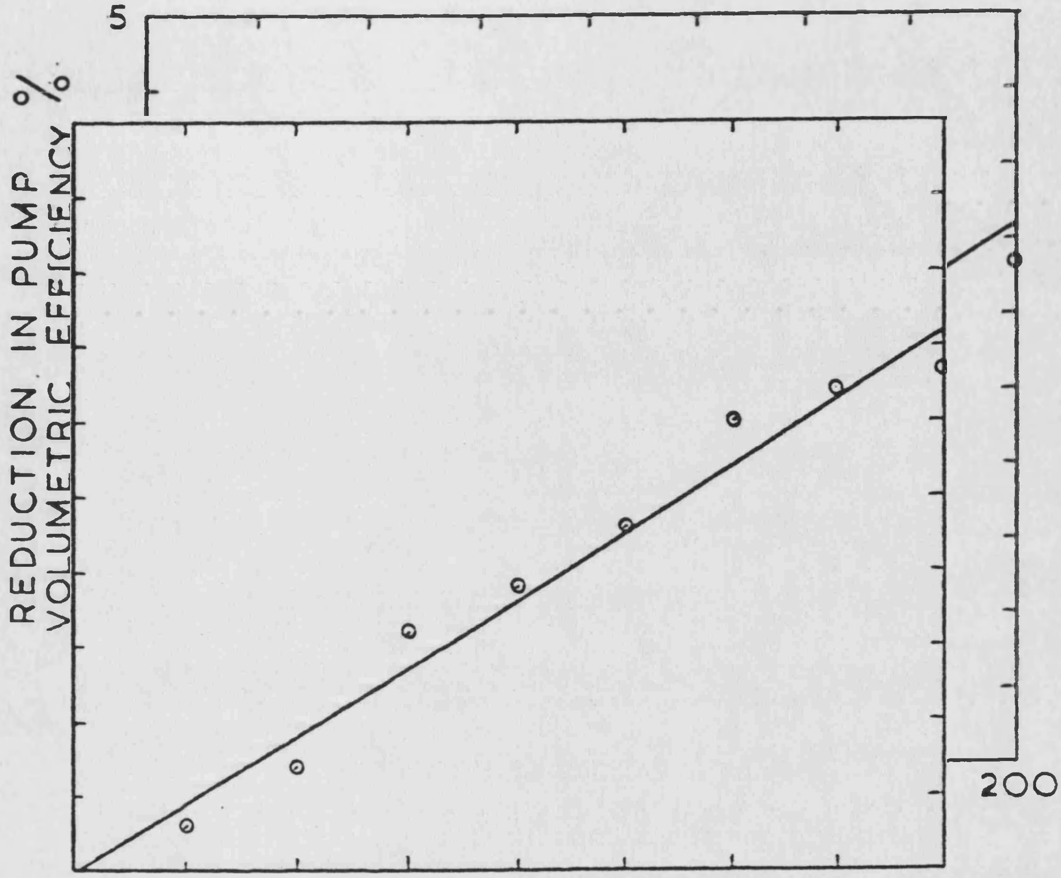


FIG. 8.16 EFFECT OF MKII MECHANISM ON PUMP B VOLUMETRIC EFFICIENCY

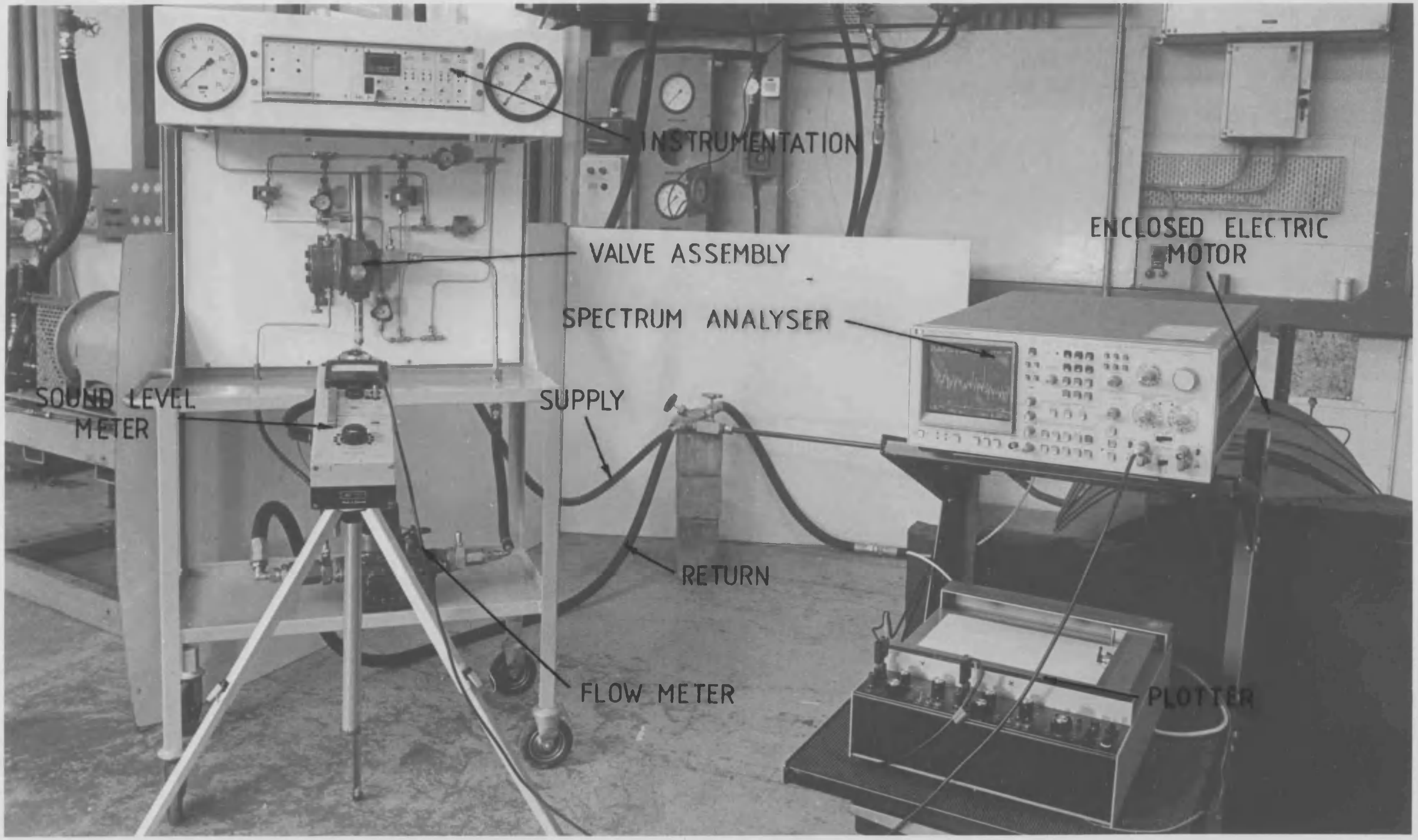
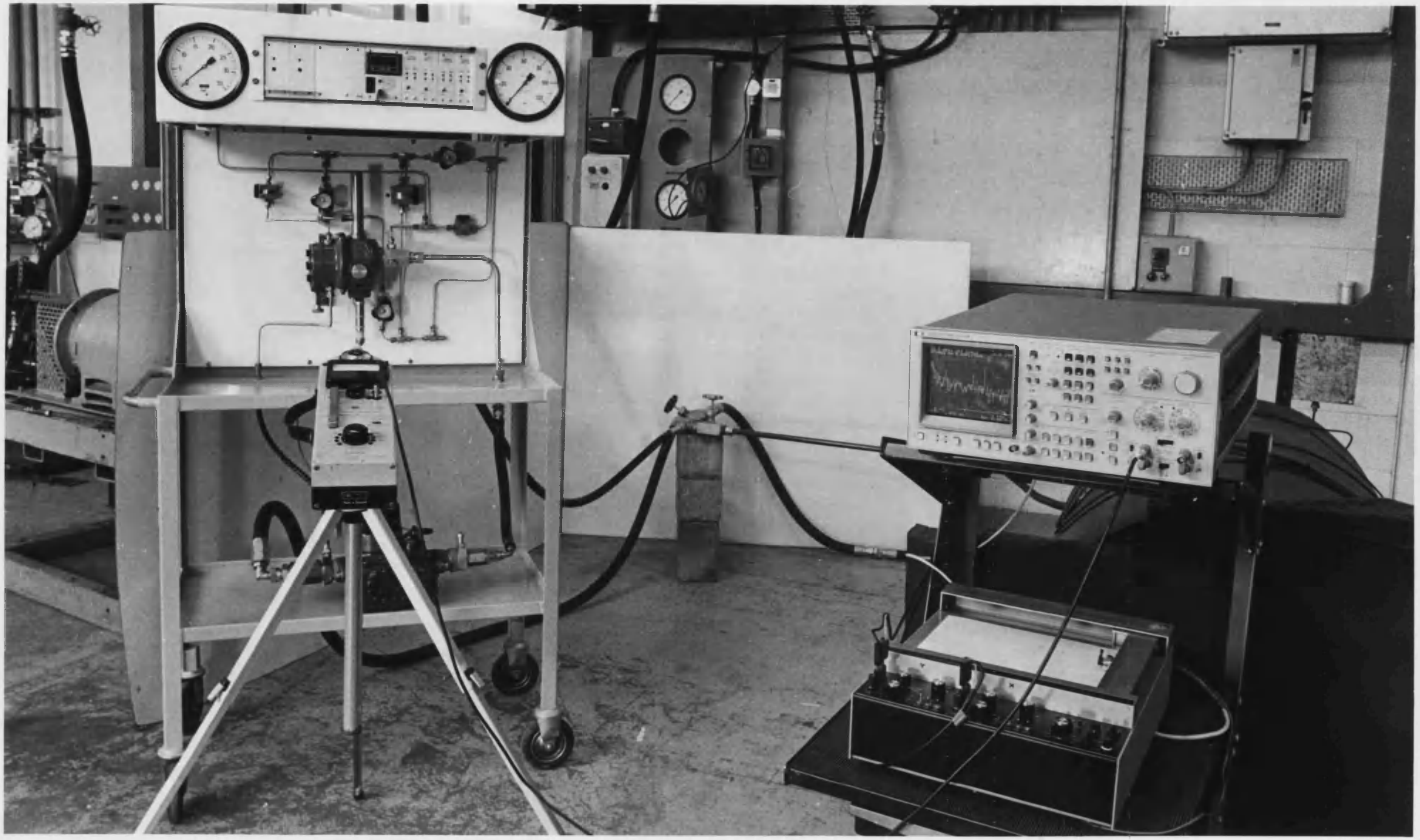
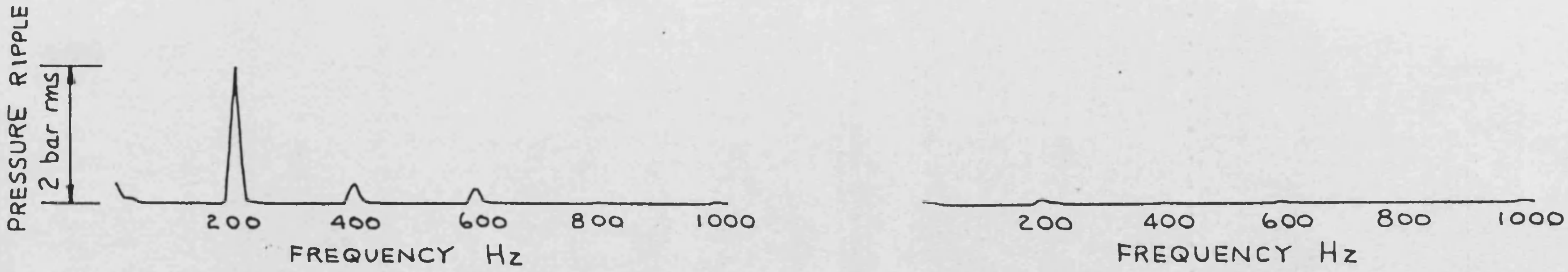
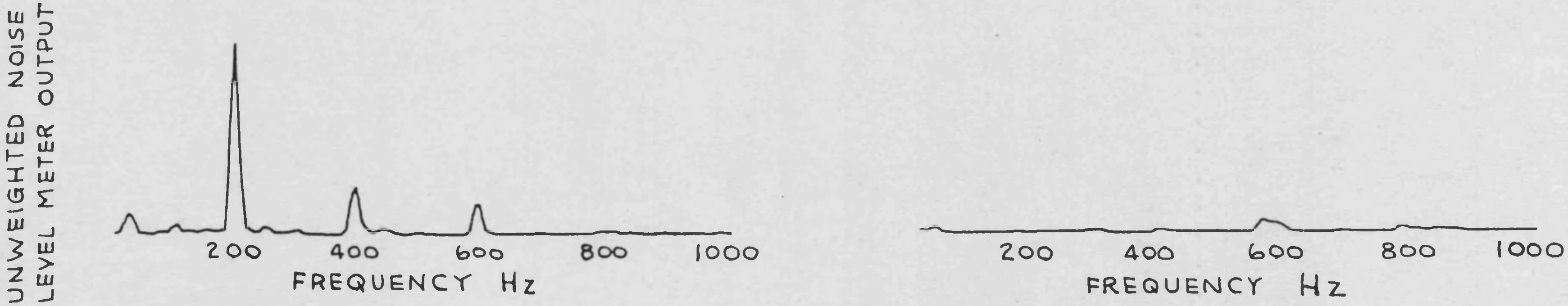


FIG. 8.17 PHOTOGRAPH SHOWING HYDRAULIC SYSTEM USED FOR AIRBORNE NOISE MEASUREMENTS





PUMP PRESSURE RIPPLE BEFORE AND AFTER MODIFICATION WITH MK III MECHANISM



SYSTEM AIRBORNE NOISE BEFORE AND AFTER MODIFICATION WITH MK III MECHANISM

FIG. 8.18 PUMP PRESSURE RIPPLE AND SYSTEM AIRBORNE NOISE COMPARISON

CHAPTER 9 - A NOVEL METHOD OF DETERMINING PUMP PRESSURE RIPPLE GENERATING POTENTIAL

The pressure ripple generating potential of a pump is not easily determined because neither the pump flow fluctuation or pump impedance can be measured directly. However, it is the opinion of the author that an active equivalence method of assessing pump flow ripple, by observing the neutralization of pressure ripples, is a significant advancement in evaluating pump fluidborne noise characteristics.

An important advantage of such a method is the inherent independence of any evaluation of either fluid or system properties. In particular the calculation of the speed of sound in the fluid for various combinations of mean pressure and temperature is not required. Before describing in detail how the ability to produce flow ripples may be used as a means of measurement, it is interesting to review existing flow ripple evaluation methods.

9.1 - EVALUATION OF PUMP FLOW FLUCTUATIONS

One method of measuring flow fluctuation is the hot film anemometer as developed by Tilley ref. (40). This method utilizes the measurement of the convective heat transfer from a heated sensor, which is exposed to the fluid flow, to determine velocity changes. Under established laminar flow conditions variations in flow can be measured at frequencies up to 700 Hz. However it is not suitable for measuring the flow fluctuations inside the outlet chamber of a pump where flow conditions are almost certainly turbulent.

The difficulty in direct measurement of flow fluctuation has been overcome by the development of test methods which allow the flow ripple to be calculated from the measured pressure fluctuation produced by a pump in a load system where the impedance characteristics are known.

Szerlag, ref. (41) and Unruh ref. (42) have both suggested that a simple load system connected to the pump may be used to determine the flow fluctuation. The load system consisted of a short line, in which standing wave effects may be kept negligibly small, and a restrictor valve of known impedance characteristic. Standing wave effects can usually be neglected if the line length is less than 1/20th of the wave length of the highest frequency considered. It must also be assumed that the pump impedance exhibits the same characteristics as a lumped volume equal to the internal volume of the pump. If these conditions exist the pump flow ripple ($|Qs|$) can be determined from the measured pressure fluctuation at the pump outlet flange as follows:-

$$|Qs| = \frac{|Po|}{Z}$$

where $|Qs|$ - amplitude of source flow

$|Po|$ - amplitude of pressure fluctuation at pump flange

Z - calculated effective impedance of pump, line and restrictor valve.

The amplitudes of the source flow evaluated by this method are obviously only as accurate as the prediction of the combined or effective impedance of the system (Z). It has been shown by a more sophisticated measurement and analytical technique ref. (27) that pump impedance cannot be represented by a lumped or distributed parameter model without incurring errors of up to 100%. This casts serious doubt upon the accuracy of $|Qs|$ determined by this simple method.

In contrast to the exclusion of standing wave effects, some methods rely entirely on the fact that the distribution of pressure fluctuation along a pipe line may be accurately predicted by the wave equation (e.q. 2.1). Theoretically, pressure measurements at two positions on a pipe line of a particular length are sufficient to allow evaluation of the termination impedance (Z_T) of a simple system. Such a system may consist of a pump, a straight section of pipe and a restrictor valve termination. A further measurement at a different position for an extended pipe length provides enough information for the evaluation of the pump source flow (Qs) and source impedance (Zs) for all harmonics for which pressure measurements are taken. It was found however, ref. (12), that Z_T is particularly sensitive to experimental scatter in the pressure information making it necessary to test a large number of pipe lengths in order to obtain mean values of Qs, Zs and Z_T .

The method was found to be very time consuming and required the use of high speed data aquisition by computer to make it practical. Freitas, ref. (17) has shown that by careful choice of pipe lengths the accuracy of the Extending Pipe Length Method can be greatly improved and that fewer pipe lengths may be used. The Tuned Length Method, devised by Freitas, is a development of the Extending Pipe Length Method and uses two pressure transducers, one at the source and the other at the termination. This method enables Qs, Zs and Z_T to be evaluated at ten harmonics of pumping frequency with only eight pipe lengths.

The evaluation of pump parameters for a range of mean pressures and speeds still requires vast amounts of pressure data to be collected and processed. This can only be done practicably in a reasonable time by computer. This is also true of the Trombone Method developed by Wing ref. (27). The Trombone Method, so called because of a sliding pipe section to permit ease of pipe length variation, utilizes pressure data from four transducers at nine different line lengths to evaluate ten harmonics of Qs and Zs. The pressure standing wave, in terms of amplitude and phase, defined by the four transducers is matched, by a least squares error fitting method, to the theoretical standing wave given by equation 2.1. The estimated amplitudes and phases of Qs, Zs and Z_T are varied until unique values of each are achieved which give the best fit of the standing waves for all the measurements taken.

This is a very powerful technique and may be used to evaluate the characteristics of passive components such as valves as well as pumps and motors.

In conclusion, it is apparent that, at present, there is no suitable method of directly measuring pump flow fluctuation. Methods using indirect measurement or evaluation tend to be simple and inaccurate or complex and highly dependent upon computer facilities for data acquisition and manipulation. A method of evaluating pump source flow will now be described which is simple, potentially very accurate and does not require computer facilities.

9.2 - THE MULTI-LIFT CAM MECHANISM AS A MEASURING DEVICE

The ability of the multi-lift cam mechanism to produce accurately controllable variations in fluid flow, which may be used to cancel pressure variations produced by a pump, inherently provides a means of measuring the source flow characteristics of a pump.

If a known volume variation can be applied at or very close to the point of production of the source flow variation within a pump and with such relative phase to cause neutralization, then the pump volume variation is also a known quantity.

This simple deduction provides the basis of an active method for the experimental evaluation of pump volume variation. Pump flow variation may then be obtained by simple differentiation. To enable the MK II or III cam mechanisms to be used as a practicable measurement device, two additional facilities must be provided. Firstly the amplitude of the volume variation produced must be easily variable and secondly, measurement of the phase of the volume variation produced with respect to a convenient datum must be possible. Assuming these two facilities can be incorporated into a practical design, see discussion in 9.3, the actual test method for determining the volume variations of a pump would be as follows:-

a) Run the pump at the desired operating conditions (speed, mean pressure, temperature and fluid type) for which volume ripple evaluation is required. Monitor the harmonic amplitudes of the pressure fluctuations produced when the pump discharges into a suitable loading circuit. High levels of pressure ripple would improve the accuracy of measurement and these could easily be produced by designing a loading circuit with a high entry impedance at the frequencies under consideration. These conditions are exactly those required by the High Impedance Pipe Test and a load circuit as described in ref. (19) would also be ideal for this test method.

b) Introduce into the pump outlet port a volume variation of known amplitude and frequency. For an external gear pump a good starting value of amplitude would be that given by the theoretical model as shown by equation 3.2. This could be produced by a suitably designed multi-lift cam mechanism with a single harmonic cam of known lift and a number of active piston-followers of known area.

c) Vary the phase of the additional volume variation until the amplitude of the pressure fluctuation, at the selected frequency, is a maximum. At this condition the volume variation produced by the mechanism is in-phase with that of the pump and the two ripples are additive. The phase of the pump volume ripple at this frequency is now determined.

d) Vary the phase of the additional volume variation until the amplitude of the pressure fluctuation at the selected frequency is now a minimum. At this condition the volume variation produced by the mechanism is in anti-phase with that of the pump and the two ripples are cancelling each other.

e) Now vary the amplitude of the volume variation produced by the mechanism until the amplitude of the pressure fluctuation, at the selected frequency, is zero. The two volume ripples are now of equal amplitude and opposite phase. Hence the pump volume ripple is determined in amplitude and phase.

f) The above may be repeated for the second and subsequent harmonics of pumping frequency to determine the pump characteristic.

9.3 - MECHANISM ADAPTION FOR AN EXTERNAL GEAR PUMP

The design scheme, fig. 9.1, shows how the basic cancellation mechanism may be adapted to provide a measuring capability. As a measuring device the mechanism requires the additional facility to vary, easily, both the amplitude and phase of the volume variation produced so that cancellation conditions may be achieved with an unknown pump volume variation. The means of providing these parameter variations are best explained with reference to fig. 9.1.

The amplitude of the volume variation is dependent upon the following three variables:-

i) the lift of the cam

ii) the area of the piston-followers

iii) the number of active piston-followers.

i) The amplitude spectrum of a particular cam are fixed and to alter the cam lift requires the substitution of a different cam. This is done simply by sliding the cam off its square location after removing the plug.

ii) The area of the piston-followers would be fixed for a given piston-follower and piston carrier assembly. Different assemblies would have different piston areas and it is possible to have an assembly in which all the piston areas are not equal. Due consideration must be given to the resultant side load on the pump driveshaft if unequal area diagonally opposed piston-followers are used.

iii) The number of active piston-followers (those in contact with the cam) may be varied by the use of a simple piston retraction device. Undoing the locking collar permits the internal thread of the sliding piston retractor to engage with the threaded end of the piston-follower. The subsequent tightening of the locking collar allows the piston-follower to be held clear of the cam surface rendering it inactive.

The number of combinations of these three parameters allows a considerable range of amplitudes of volume ripple to be produced. The magnitude of the step change between each value of amplitude produced is dependent upon the number of combinations of components available (different area piston and piston housings and different lift cams) for each frequency of interest.

Table 9.1 shows how eight piston-followers of equal area and two cams of different lift can be used to approximate the fundamental volume variations of eight pumps in the Dowty 1P3000 range of external gear pumps.

Ideally, to cover a wide range of pump sizes and match volume ripples exactly, in order to produce total pressure ripple cancellation, the mechanism would need to be infinitely variable. This of course is not physically possible and, because of the direct relationship between volume and pressure ripples, is not essential to accurate evaluation. Partial pressure ripple cancellation by a known neutralizing volume variation may be used to determine the required volume variation by direct proportions. For instance suppose a 25 mm^3 volume variation produced a 50% reduction in the pressure ripple produced by a pump, then the volume variation of the pump is 50 mm^3. Care must be taken however to ensure that the volume variation introduced by the mechanism is less than the pump volume variation to be determined or the residual pressure ripple will be due to the mechanism and not the pump, thus making any evaluation void. This situation is easily detected by observance of a phase reversal in the residual pressure ripple. Alternatively two partially neutralizing volume ripples may be used, the second larger in amplitude than the first. For a valid evaluation these must produce proportional reductions in the residual pressure ripple.

The mechanism does however permit the phase of the volume ripple to be varied steplessly. This is achieved by rotation of the piston carrier assembly with respect to a selected reference point. A suitable reference is described in 6.2.5. A method (not shown in fig. 9.1) allowing precise control of piston carrier angular position is essential and fundamental to the accuracy of results. The principle of angular measurement depicted in fig. 9.1 by a graduated angular scale is intended only as an illustration of what is required. An idea of the accuracy of measurement required may be obtained by consideration of the effect that a one degree change in piston carrier position has on the phase of volume variations produced at harmonics of fundamental frequency.

Table 9.2 shows the increasing sensitivity of volume ripple phase with increasing frequency. This is due to the fact that in an eight piston unit 45 degrees of cam angle represent 360 degrees or 1 cycle of volume variation at the first harmonic, hence the 8:1 relationship. This increases linearly with integer increases in frequency. Hence to evaluate the fourth harmonic to within 5 degrees the method of angular position and measurement must have a resolution not less than 18.75 minutes of arc. It is not intended to give details of a means of achieving this here but a possible solution may be found by application of the principles used in a precision engineering dividing head or by using an optical shaft encoder.

Although this volume ripple producing mechanism is shown as an integral part of a modified gear pump, because of its logical adaption from the simple cancellation device, there is no reason why it could not be designed as a free standing unit. Such a unit, with suitable cam shaft support bearings, could be used with a twin head driveshaft arrangement to test pumps without the need for any special adaption or modification.

9.4 - VOLUME VARIATION AS A MEASURE OF PUMP PRESSURE RIPPLE GENERATING POTENTIAL

To accurately represent the fluidborne noise generating potential of a pump, an independent evaluation of the two pump related parameters flow fluctuation (Qs) and impedance (Zs) is required. However, the inability to measure either of these parameters directly has led to the proposition that a measure of generating potential may be determined by the direct measurement of the pressure ripple produced at the pump. A measurement of this kind is a valid representation of pump generating potential only if system parameters can be excluded or made negligibly small.

This situation is achieved with the High Impedance Pipe test ref. (19) where, with the use of a specially designed loading circuit, the pressure ripple measured at the pump flange is the product Qs.Zs. If the values of $|Qs.Zs|$ are measured for all harmonics, at specified conditions of speed, mean pressure, temperature and oil type, then the pump may be characterized, at these conditions, by the R.M.S. value of all harmonics as follows:-

$$\text{R.M.S. pressure rating} = \sqrt{\sum_{i=1}^{i=n} \left| Qs.Zs \right|_i^2} \quad \text{where}$$

i = harmonic number.

Although this method can be used to determine accurately a pump rating on the basis of a Qs and Zs product, it is questionable whether such a rating is accurately representative of the likely pressure ripple generating potential of a wide range of pump types operating in real systems. The basis for this doubt is the fact that the amplitude of the pressure ripple generated in a simple system, as shown by eq.2.1, is not directly and exclusively proportional to $|Qs.Zs|$. The appearance of Zs in the denominator and again implicity in ρs effects the sensitivity of pressure ripple to changes in Zs. A detailed investigation by Freitas ref. (17) concluded that Zs is most significant in determining the fluidborne noise generating potential of a pump if it has a value $|Zs| < |Zo|$. For pumps of compact design and hence low internal volume this condition is most likely to occur at higher frequencies where both gear and piston pumps exhibit relatively low values of $|Zs|$.

The $|Qs.Zs|$ rating is however useful in comparing pumps of similar source impedance (or internal volume) with different source flow characteristics. Under these conditions a meaningful comparison of the pressure ripple potential of internal and external gear pumps could be achieved. Variations in the rating could also be used to determine the effectiveness of port plate timing or precompression groove modifications for a particular development pump as changes in flow ripple would be seen as proportional changes in pressure rating.

The pump parameter which is the most appropriate for the basis of a pump pressure ripple rating is Qs. This is the view held by many researchers in the field including Willikins ref. (43) and Unruh ref. (42) who both consider it to be the singularly most fundamental pump parameter influencing pump fluidborne noise characteristics by reason of the direct relationship between pressure and flow ripple, as shown in chapter 2, it follows that doubling $|Qs|$ in a given system will double the amplitude of pressure ripple. It is this simple fact which makes Qs a suitable basis for the comparison of the fluidborne noise generating potential of different pumps.

To determine the Qs characteristics of a pump for 10 harmonics of pumping frequency over a wide range of speeds and mean pressures, for a particular fluid of a given temperature, would require a considerable amount of data. For a variable swash piston pump, where Qs varies with swash angle, this adds even further complexity.

A considerable reduction in the amount of data required may be achieved by using pump volume variation as a basis for comparison. As shown in 3.3.3 Vs may be obtained by the summation of the instantaneous values of Qs over a given time period. This allows the fundamental changes in pump output to be interpreted independently of the speed of shaft rotation. For pumps, with output volume variations which are substantially constant irrespective of shaft rotational speed, this means that the output variation of each harmonic may be represented by a single number instead of a number for each particular speed of interest. The following example will clarify this point:-

If the amplitude of the first harmonic of Qs evaluated for a gear pump with 8 teeth at 1500 rev/min. at a given pressure is 0.05 l/S, at 300 rev/min. and 3000 rev/min. it would be 0.01 l/S and 0.1 l/S respectively. Obviously the value of Qs at any other speed would be in a similar proportion. However the value of Qs at the fundamental frequency at the initial speed is equivalent to a volume variation of 39.78 mm^3 and is constant for all speeds of rotation.

If Qs is substantially constant with mean pressure as well as speed of rotation it means that the fluidborne noise generating characteristics of a pump over its normal range of operating conditions may be represented by a single value of volume variation amplitude at each harmonic.

As an example, the characteristic amplitudes of volume variation of the external gear pump A are as follows:-

Harmonic	Peak Amplitude mm^3
1	40.00 ± tol
2	5.82 "
3	2.16 "
4	1.13 "
5	0.73 "
6	0.44 "

Given the number of pumping elements the above information predicts the volume variation, for any shaft rotational speed, at the first six harmonics of pumping frequency. When connected to a system with any effective entry impedance, the pump will produce pressure fluctuations which are directly proportional in amplitude to these volume variations. A pump with the same number of pumping elements as pump A but with a first harmonic component of 80 mm^3 would therefore produce twice the pressure ripple of pump A at the same frequency. Alternatively if a pump had a first harmonic of 20 mm^3 but twice the number of pumping elements of pump A, then for the same rotational speed it would produce no pressure ripple at the first harmonic of pump A but 3.4 times its value at the second. This method of presentation is clearly useful in comparing the fluidborne noise generating potential of similar pumps.

The six amplitudes of volume variation shown, although typical for pump A, would benefit by an individual tolerance showing the maximum deviation from the stated value over stated speed and pressure ranges for a number of different examples of the same unit.

To represent pumps which have volume ripple characteristics which vary with both speed and mean pressure requires significantly more data. Piston type pumps for instance have this characteristic and often show strong volume variations over ten or more harmonics of pumping freqnency. The volume variation data for this type of pump would require an extensive amount of testing and results would need to be presented in tabular or graphical form.

There is no doubt however, that pump volume variation data would be very useful to the system designer in providing a means of comparing the likely pressure ripple generating potential of different pumps.

PUMP SIZE CODE	THEORETICAL 1st HARMONIC VOLUME VARIATION mm^3	0·6 mm CAM LIFT 15 mm^2 PISTON AREA		1·0 mm CAM LIFT 15 mm^2 PISTON AREA	
		VOLUME VARIATION mm^3	Nº OF ACTIVE PISTONS	VOLUME VARIATION mm^3	Nº OF ACTIVE PISTONS
IP3020	26·30	27	3	15	1
IP3028	35·47	36	4	30	2
IP3036	44·65	45	5	45	3
IP3044	53·82	54	6	-	-
IP3052	62·38	63	7	60	4
IP3060	71·56	72	8	75	5
IP3072	85·58	-	-	90	6
IP3090	105·22	-	-	105	7

TABLE. 9.1 VOLUME VARIATIONS PRODUCED BY VARIOUS CONFIGURATIONS OF CAM MECHANISM GEOMETRY

HARMONIC OF FUNDAMENTAL FREQUENCY	PHASE CHANGE PER DEGREE OF PISTON CARRIER ROTATION DEGREE
1	8
2	16
3	24
4	32
5	40
6	48
7	56
8	64
9	72
10	80

TABLE. 9.2 VOLUME VARIATION PHASE SENSITIVITY TO CHANGE IN PISTON CARRIER ANGULAR POSITION

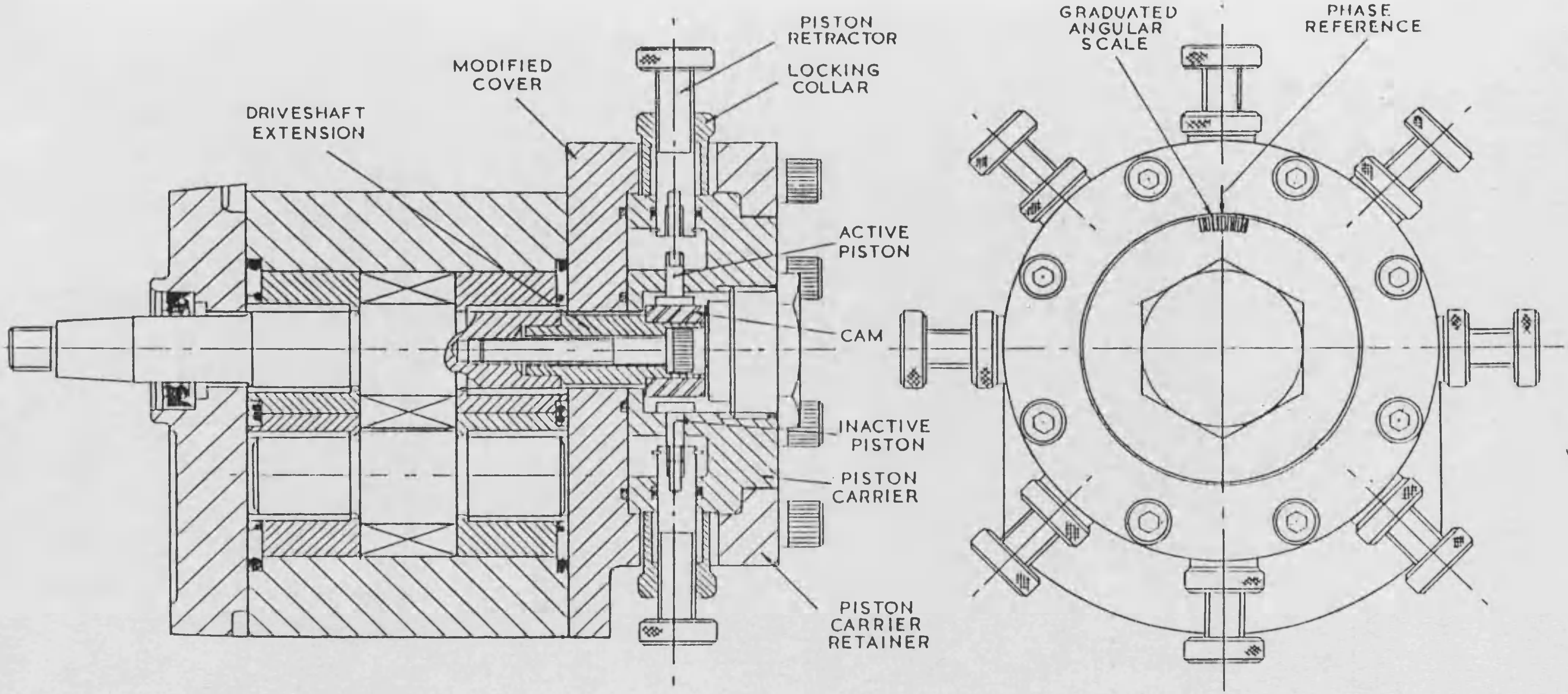


FIG. 9.1 GEAR PUMP VOLUME RIPPLE MEASURING DEVICE

CHAPTER 10 - CONCLUSIONS

10.1 - THE EXTERNAL GEAR PUMP

The variation in the instantaneous flow produced by external gear pumps and motors may be predicted with reasonable accuracy by a simple model of involute gear geometry. This has been established with the aid of two digital computer modelling techniques which were used to evaluate the source flow fluctuations of several gear units from the experimental pressure data collected under specialized test conditions.

Gear pump inlet flow and pressure ripples have been found to vary considerably with mean inlet pressure. Under atmospheric pressure inlet conditions the levels of the pressure standing waves in the inlet line were considerably less than what might be expected from the tooth geometry. By progressively increasing the mean inlet pressure it was found that pressure and flow ripple levels also increased, until at 2 bar they were fully developed and similar to that normally found in the outlet line. This apparent reduction in pump flow fluctuation is caused by air being released from the fluid at low pressures. The air release occurs inside the pump inlet port and has a damping effect on the normal flow fluctuation produced by the pump.

From this it is apparent that boosting the pump inlet or operating with a pressurized reservoir will increase the level of fluidborne noise in the inlet line making it an airborne noise source of equal potential to that of the outlet. This is of course exactly the case of a motor operating with moderate return line pressure and indicates the need of a method to reduce both inlet and outlet pressure fluctuations.

The outlet flow ripple of the gear pumps tested did however show a marked insensitivity to mean pressure level, making the level of pressure ripple developed in the outlet line dependent upon load impedance rather than operating pressure level. This is because the gear pump transfers fluid from inlet port to outlet port without incurring the pressure dependent variations in mean flow, associated withport plate timing, which are so characteristic of most piston pumps and motors. This, together with well designed pressure trapping relief grooves and effective minimization of internal leakage (volumetric efficiency can be typically 96% to 98% at 210 bar), makes the output flow variation primarily dependent upon the characteristics of the involute gear geometry.

Pump flow fluctuations, measured over a range of operating pressures up to 200 bar, showed great similarity with those predicted by involute gear geometry. However differences in harmonic amplitudes of measured and predicted values have been found which tend to increase with frequency. Measured amplitudes are consistantly greater than those predicted by amounts which vary from 10% at the first harmonic of pumping frequency, to 100% at the fourth. This obviously limits the usefulness of the geometric model as a basis for design. It would not be sufficiently accurate, for instance, to allow the design of a ripple compensation device that would reduce the peak to peak flow fluctuation of an external gear pump to less than that of a comparable internal gear unit which is approximately 3% of the mean flow.

In order to account for the deviations, of particular pump designs, from the model it is therefore necessary to employ a flow ripple measurement technique.

10.2 - THE METHOD OF PRESSURE RIPPLE REDUCTION

A method of reducing pressure ripple in hydraulic systems has been established which has the potential to virtually eliminate the pressure fluctuations produced by a known source. The method utilizes the fact that the pressure fluctuations produced by a particular source are directly proportional to the fundamental variations in flow which are characteristic of the source. The application at the source of equal and opposite flow fluctuations from a secondary source produces a net reduction in flow and hence pressure fluctuation from the two sources combined.

Implementing such a method requires the solution of two problems; the design of a suitable secondary flow ripple generating device and the evaluation of the flow ripple characteristics of the source.

A flow ripple generating device has been designed and developed for use in conjuction with external gear pumps and motors. The device consists primarily of a number of radial piston-followers controlled by a multi-lift cam mounted directly on the pump driveshaft. A computer program has been developed which converts harmonics of pump flow ripple at any fundamental frequency to equivalent volume variations with cam rotational angle, and for a given piston-follower area the cam profile is then defined. The number of harmonics which can practicably be incorporated on one cam depends on the relative amplitudes and phases of the harmonic piston-follower displacements, which cause limitations due to manufacturing tolerances and contact stress respectively.

A cam incorporating the first four harmonics of pump volume ripple has been manufactured and in operation significantly reduced the pressure ripple at each harmonic, the best performance being a reduction of 98.5% at the first. The predominance of the lower harmonic amplitudes of gear pump volume variation means that a typical pump pressure ripple rating as determined by the High Impedance Pipe method, may be reduced from 4.5 to 0.1 bar rms by cancelling just the first four harmonics.

Mechanisms designed to modify the first harmonic of pump volume variation have achieved pressure ripple reductions of the order of 96% over the full working range of pump speeds up to 3000 rev/min. and at pressures up to 200 bar. Because the piston-followers access both inlet and outlet ports, volume and pressure ripple reductions are effected in the inlet and outlet lines simultaneously. The direct action of the mechanism through the drive shaft also reduces pump input torque fluctuations and motor output torque fluctuations.

The difficulties of evaluating pump flow ripple characteristics have been overcome by the use of digital computer modelling techniques which are sophisticated, time consuming and dependent upon the accurate representation of the test system and the properties of the working fluid. It is feasible, however, to measure the most significant harmonics of gear pump flow ripple using a suitably adapted secondary flow ripple generator, by the observance of pressure ripple cancellation for a known secondary input. This active method of evaluation has the advantage of reducing test time and is independent of system and fluid properties.

The airborne noise measurements carried out indicate that the level of airborne noise radiated from the gear pump casing is more dependent upon direct mechanical vibration than excitation due to pressure fluctuations. However, the airborne noise radiated by a load system, isolated from pump structural vibration, can be dominated by structural excitation caused by pump pressure ripple. In this situation modifying the pump flow ripple has been found to reduce system airborne noise levels by 10 dB.

The penalties of modifying a pump or motor with a multi-lift cam mechanism are comparitively modest. Reductions in volumetric and mechanical efficiencies of up to 3% are to be expected and installationally the modification could add 6% to the length of the average pump body. It is unlikely that the additional cost, over a standard unit, will be less than 25% unless the mechanisms are produced in numbers in excess of an estimated 10,000 per annum.

The commercial viability of this method of controlled pressure ripple reduction will depend greatly on the overall level of performance and life expectancy future development can achieve and the demand in the hydraulics industry for pumps with low pressure ripple characteristics.

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A P P E N D I C E S

A.1.1 - COMPUTER PROGRAM DOCUMENTATION FOR EXGEAR PLOT.

Library classification

BENG (F) EXGEAR PLOT. 246.01

TITLE - Theoretical and Experimental External Involute Gear Pump Source Flow and Volume Variation Plotting Program.

FORTRAN IV Honeywell Multics 19.1.82

AUTHOR - B. Lipscombe

Hardware required - Plotter

Purpose

This program produces experimental source flow and volume plots from supplied harmonic data and theoretical external gear pump source flow and volume fluctuations from simple mathematical models. These may be viewed in isolation or superimposed on the same axes.

Associated subroutines

Gino plotting subroutines (Gino-f mk 2.6)

INPUT INFORMATION (via terminal)

amp	Harmonic amplitude of source flow	l/s
ffreq	Fundamental pumping frequency	Hz
nc	Desired number of cycles to be plotted	-
nh	Number of harmonics to be included	-
phase	Harmonic phase of source flow	deg.
qm	Mean flow	l/s
psp	Pump speed	rev/min.
b	Gear width	m
pcd	Pitch circle diameter of gears	m
z	Number of teeth on gears	-
beta	Pressure angle of gears	deg.
no	Number of 1/50ths of cycle offset	-
ians	Denotes form of output selected	-
inumber	Number of executions of selected output	-

OUTPUT INFORMATION (via terminal)

r	Experimental amplitude of source flow	l/s
qs	Theoretical amplitude of source flow	l/s
s	Experimental amplitude of volume fluctuation	mm^3
vs	Theoretical amplitude of volume fluctuation	mm^3
t	Time	ms
tangle	Shaft rotational angle	deg.
n	Number of points in plotting array	-

INSOURCE VARIABLES

i.m.n.l	Counters	-
j	Number of time increments	-
pi	Pi - 3.14159	-
tmpry	Temporary variable	-
w	Angular frequency	rad/s
xmax	High bound of time axis	s
ymax	High bound of flow axis	l/s
amax	High bound of angle axis	deg.
am	Module of gears	m
theta	Gear rotational angle	rad.

LIMITATIONS AND ACCURACY OF PROGRAM

Single precision arithmetic is used throughout. The number of experimental harmonics which can be included is limited to ten by the size arrays "amp" and "phase".

INBUILT ERROR MESSAGES

Those standard to gino plotting subroutines.

PROGRAM ACTION AND ALGORITHM

The program first sets up the desired output and presentation from the following selections by setting "ians" equal to 0 1 2 3 4 5 or 6.

0 for single theoretical Qs - Vs plots

1 for single experimental Qs - Vs plots

2 for single theoretical and experimental Qs - Vs plots superimposed

3 for multiple experimental Qs plots on same axes

4 for multiple experimental Vs plots on same axes

5 for multiple theoretical Qs plots on same axes

6 for multiple theoretical Vs plots on same axes.

The number of executions of a particular selection is set by the value of "inumber" as input via the terminal.

Input information is then required, which is dependent upon whether the selected output is theoretical or experimental. A typical example of input data, for each, is shown in table A1.1 for a selected output "ians" = 2. The corresponding graphical output is shown in fig. A.1.1.

Evaluated experimental flow fluctuations are reconstructed by the summation of a number of harmonics of pumping frequency according to the following equation:-

$$Qs = \sum_{i=1}^{i=nh} amp(i) \cdot \sin(iwt + phase(i)).$$

where i denotes harmonic of pumping frequency

w denotes gear rotational frequency

t denotes time.

Experimental volume variations are determined by the integral of Qs w.r.t time:-

$$Vs = \sum_{i=1}^{i=nh} -\frac{amp}{iw}(i) \cdot \cos(iwt + phase(i))$$

Theoretical flow and volume variations are determined by the following equations derived from an analysis of involute gear geometry shown in chapter 3.

$$Qs = \frac{wbm^2 \cos^2 B}{12} (\pi^2 - 3 z^2 \theta^2)$$

for $-\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$

and $$Vs = \frac{b\, m^2 \cos^2 B}{12} (\pi^2 \theta - z^2 \theta^3)$$

for $-\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$

where w - rotational frequency

b - gear width

m - gear module

B - pressure angle of gears

Z - number of teeth

θ - angle of gear rotation.

```
c       PROGRAM NAME - EXGEARPLOT
c
c
c       TITLE - Theoretical & Experimental External Involute Gear
c               Pump Source Flow & Volume Variation Plotting Program
c
c       Fortran IV     Honeywell Multics      19.1.82
c
c       AUTHOR - B.Lipscombe
c
c       Hardware Required - Plotter
c
c       No Normal Error Messages
c
c       Purpose - This program produces experimental source flow
c                 and volume plots from supplied harmonic data
c                 and theoretical external gear pump source flow
c                 and volume fluctuations from simple mathematical
c                 models.These may be viewed in isolation or
c                 superimposed on the same axes.
c
c       ASSOCIATED SUBROUTINES
c        Gino plotting routines
c
c       INPUT VARIABLES (Via terminal)
c           amp       Harmonic amplitude of source flow         l/s
c           ffreq     Fundamental pumping frequency             hz
c           nc        Desired number of cycles to be plotted    -
c           nh        Number of harmonics to be included        -
c           phase     Harmonic phase of source flow             deg
c           qm        Mean flow                                 l/s
c           psp       Pump speed                                rev/min
c           b         Gear width                                m
c           pcd       Pitch circle diameter of gears            m
c           z         Number of teeth on gears                  -
c           beta      Pressure angle of gears                   deg
c           no        Number of 1/50ths of cycle offset         -
c
c       OUTPUT VARIABLES (via plotter)
c           n         Number of points in plotting array        -
c           r         Experimental amplitude of source flow     l/s
c           qs        Theoretical amplitude of source flow      l/s
c           s         Experimental volume fluctuation           cu.mm
c           vs        Theoretical volume fluctuation            cu.mm
c           t         Time                                      ms
c           tangle    Shaft rotational angle                    deg
c
c       INSOURCE VARIABLES
c           i,m,n,l   Counters                                  -
c           j         Number of time increments                 -
c           pi        Pi - 3.14159                              -
c           tmpry     Temporary variable                        -
c           w         Angular frequency                         rad/s
c           xmax      High bound of time axis                   s
c           ymax      High bound of flow axis                   l/s
c           amax      High bound of angle axis                  deg
c           am        Module of gears                           m
c           theta     Gear rotational angle                     rad
c
c       **************************************************************
```

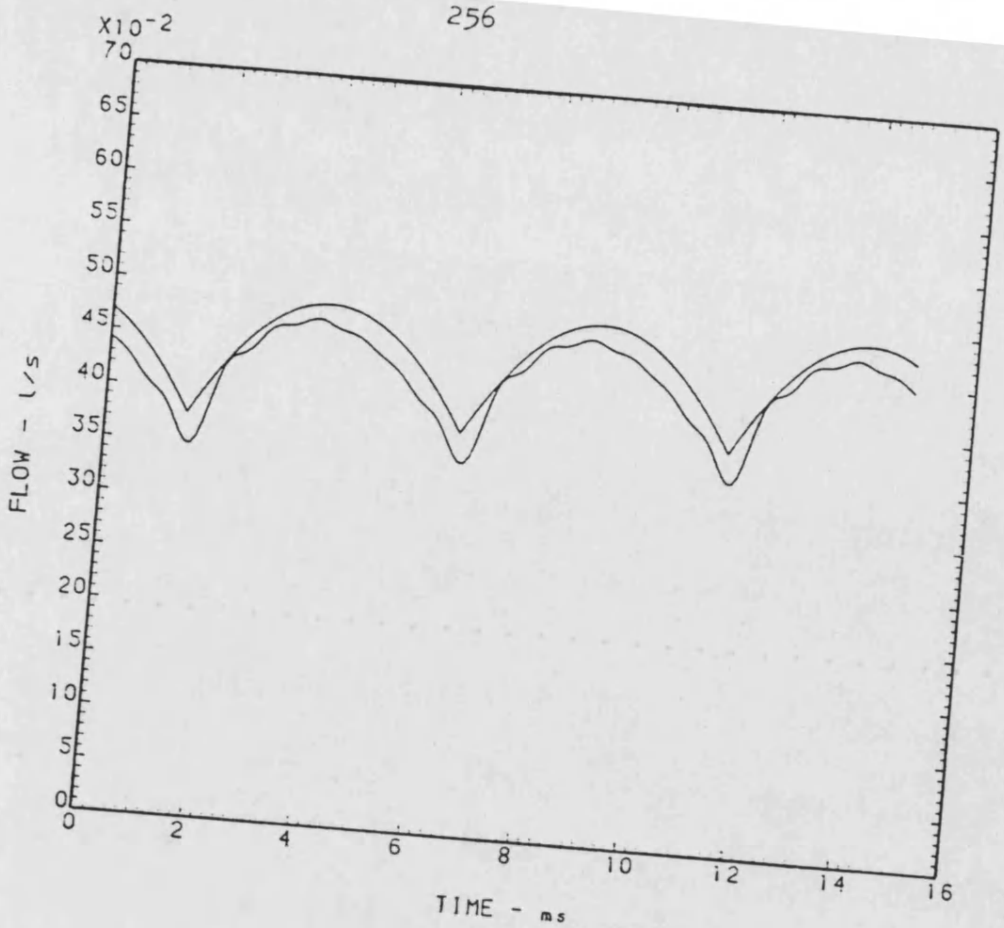
```
c
c
          external chahol(descriptors)
          dimension amp(10),phase(10),r(500),t(500),tt(500),s(500)
          dimension qs(250),vs(250),tangle(500),ttangle(500)
          call vterm
          call units(6.4)
          pi=3.14159
          write(6,601)
 601      format(1h ,'Type 0 for single theoretical Qs-Vs plots'/
&         1h ,'1 for single experimental Qs-Vs plots'/
&         1h ,'2 for single theoretical and experimental Qs-Vs
&         plots superimposed'/
&         1h ,'3 for multiple experimental Qs plots on same axes'/
&         1h ,'4 for multiple experimental Vs plots on same axes'/
&         1h ,'5 for multiple theoretical Qs plots on same axes'/
&         1h ,'6 for multiple theoretical Vs plots on same axes'/)
          read(5,501)ians
 501      format(v)
c
 223      write(6,614)
 614      format(1h ,'How many executions of the selected program
&         do you want ?')
          read(5,501)inumber
          do 401 l=1,inumber
c    If only theoretical plots requied skip experimental
c         section and go to statement 700
          if(ians.eq.0.or.ians.eq.5.or.ians.eq.6)goto 700
c
c   SUMMATION OF EXPERIMENTAL DATA
c     Enter data
          write(6,602)
 602      format(1h ,'Enter:- Fundamental frequency (hz)'
&         /1h ,8x,'Number of teeth'
&         /1h ,8x,'Mean flow (l/s)'/1h ,8x,'Number of harmonics'
&         /1h ,8x,'Number of cycles to be plotted'/1h ,8x,
&         'high bound flow axis(l/s)')
          read(5,501)ffreq,z,qm,nh,nc,ymax
          write(6,603)
 603      format(1h ,'Enter:- Amplitude (l/s) & Phase (deg)'/1h ,
&         8x,'of each harmonic in turn')
          read(5,501)(amp(i),phase(i),i=1,nh)
c     Set step length
          j=10000*nc/ffreq
c     Convert phase from degrees to radians
          do 100 i=1,nh
          phase(i)=phase(i)*pi/180.
 100      continue
c     Start time step loop
          do 200 n=1,j
          r(n)=0.0
          s(n)=0.0
          t(n)=(n-1)/10.
          tangle(n)=t(n)*360.e-3*ffreq/z
c     Start loop to sum components of each frequency
          do 300 i=1,nh
c     Convert frequency to radians/second
          w=2.*pi*ffreq*i
          tmpry=w*t(n)/1000+phase(i)
          r(n)=r(n)+amp(i)*sin(tmpry)
```

```
      s(n)=s(n)-1.e+6*amp(i)*cos(tmpry)/w
c     write(56,12)r(n)
c     write(57,12)s(n)
 300  continue
c     Superimpose fluctuations on mean flow
      r(n)=qm+r(n)
      m=n
12    format(v)
 200  continue
c     Set max and min scale values
      xmax1=j/10.
c     If only experimental plots required skip theoretical
c     section and goto statement 333
      if(ians.eq.1.or.ians.eq.3.or.ians.eq.4)goto 333
c
c     THEORETICAL GEAR PUMP SOURCE FLOW MODEL
c     Input data
 700  continue
      if(ians.eq.0.or.ians.eq.1.or.ians.eq.2)number=1
      write(6,605)
 605  format(1h ,'Pump speed (rev/min)'/1h ,'Gear width
&     (m)'/1h ,'Pitch circle dia (m)'/1h ,'No. of teeth'
&     /1h ,'Press angle (deg)'/1h ,'Mean flow (l/sec)'
&     /1h ,'High bound for flow axis'/1h ,'No. of cycles'
&     /1h ,'Offset (50ths of cycle, must be => 1)')
      read(5,501)psp,b,pcd,z,beta,qm,ymax,nc,no
c     Calculate gear module
      am=pcd/z
c     Calculate angular velocity of gears
      w=psp*2.*pi/60.
c     Calculate time period of pumping frequency
      tp=60000/(psp*z)
c     Start plotting loop
      do 500 j=no,(nc*50+no)
c     Divide each cycle into 50 steps
      tt(j-no+1)=(j-no)*tp/50.
      ttangle(j-no+1)=((j-no)*tp/50)*psp*6.e-3
c     Calculate theta, must reset at start of each time period
      k=j/50
      theta=-pi/z+(j-50*k)*pi/(z*25.)
c     Calculate Qs and add to mean flow
      qs(j-no+1)=qm+1000*w*b*am*am*(cos(beta/57.3))**2.*
&     (pi*pi-3.*z*z*theta*theta)/12.
      vs(j-no+1)=1.e+9*b*am*am*(cos(beta/57.3))**2.*
&     (pi*pi*theta-z*z*theta*theta*theta)/12
c     write(56,12)qs(j-no+1)/1000
c     write(57,12)vs(j-no+1)
 500  continue
c     Plot results
      xmax=tp*nc
 333  continue
      if(ians.eq.1.or.ians.eq.2.or.ians.eq.3.or.ians.eq.4)
&     xmax=xmax1
      amax=xmax*360.e-3*ffreq/z
      if(ians.eq.0.or.ians.eq.2.or.ians.eq.5.or.ians.eq.6)
&     amax=xmax*psp*6.e-3
      if(ians.eq.4.or.ians.eq.6)goto 447
c     If not the first plot skip setting up axes
      if(l.gt.1)goto 555
c     set up the graphics output for pump flow ripple
```

```
          call axipos(0,-50.,-50.,120.,1)
          call axipos(0,-50.,-50.,100.,2)
          call axisca(1,10,0.,xmax,1)
          call axisca(1,10,0.,ymax,2)
          call chahar(1,1)
          call movto2(-60.,-11.)
          call chahol("FLOW - l/s*.")
          call chahar(1,0)
          call movto2(1.,-60.)
          call chahol("TIME - ms*.")
          call grid(-2,1,1)
 555      continue
c     If only theoretical plots required skip experimental
c     plot statement
          if(ians.eq.0.or.ians.eq.5)goto 444
          call grapol(t,r,m)
c     If only experimental plots required skip theoretical
c     plot statement
          if(ians.eq.1)goto 447
          if(ians.eq.3)goto 401
 444      continue
          call grapol(tt,qs,nc*50)
 447      continue
          if(ians.eq.3.or.ians.eq.5)goto 401
c     If not the first plot skip setting up axes
          if(l.gt.1)goto 446
c     Set up the graphics output for pump volume ripple
          call piccle
          call axipos(0,-50.,-50.,120.,1)
          call axipos(1,-50.,-50.,100.,2)
          call axisca(1,10,0.,amax,1)
          call axisca(1,10,-50.,50.,2)
          call chahar(1,1)
          call movto2(-60.,-11.)
          call chahol("VOLUME VARIATION - cu.mm*.")
          call chahar(1,0)
          call movto2(1.,-60.)
          call chahol("ANGLE - deg*.")
          call grid(-2,1,1)
 446      continue
c     If only theoretical plots required skip experimental
c     plot statement
          if(ians.eq.0.or.ians.eq.6)goto 445
          call grapol(tangle,s,m)
c     If only experimental plots required skip theoretical
c     plot statement
          if(ians.eq.1.or.ians.eq.4)goto 401
 445      continue
          call grapol(ttangle,vs,nc*50)
 401      continue
          call piccle
          call devend
          stop
```

```
(input)   EXGEARPLOT
Gino-f Mk 2.6
    Type  0 for single theoretical Qs-Vs plots
          1 for single experimental Qs-Vs plots
          2 for single theoretical and experimental
                    Qs-Vs plots superimposed
          3 for multiple experimental Qs plots on same axes
          4 for multiple experimental Vs plots on same axes
          5 for multiple theoretical Qs plots on same axes
          6 for multiple theoretical Vs plots on same axes

(input)   2

How many executions of selected program do you want ?

(input)   1

Enter:- Fundamental frequency (hz)       (input)   200
        Number of teeth                            8
        Mean flow (l/s)                            .43
        Number of harmonics                        6
        Number of cycles to be plotted             3
        High bound flow axis(l/s)                  .7

Enter:- Amplitude (l/s) & Phase (deg)
of each harmonic in turn

(input)   .0495     173
          .0144     57
          .00805    -60
          .00562    175
          .00453    58
          .00339    -84

Enter:- Pump speed (rev/min)             (input)   1500
        Gear width (m)                             .02423
        Pitch circle dia (m)                       .03175
        No. of teeth                               8
        Press ang. (deg)                           30
        Mean flow (l/s)                            .45
        High bound for flow axis                   .7
        No. of cycles                              3
        Offset (50ths of cycle, must be => 1)      35
```

TABLE A1.1 Example of screen input for program EXGEARPLOT



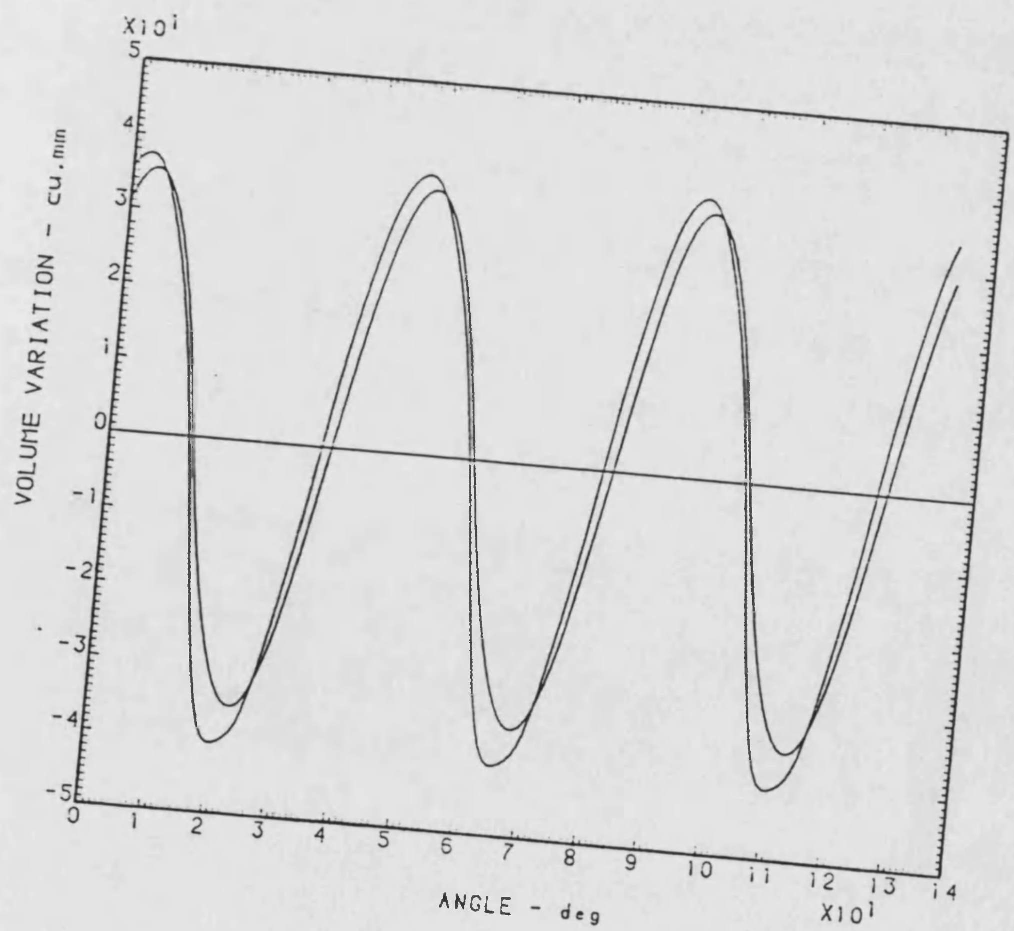


FIG. A1.1 EXAMPLE OF GRAPHICAL OUTPUT FROM PROGRAM EXGEARPLOT

A.1.2 - COMPUTER PROGRAM DOCUMENTATION FOR CAM

Library classification

BENG (F) CAM. 247.01

TITLE - Cam Profile Design Program For Flow Ripple Generating Mechanism.

FORTRAN IV Honeywell Multics 20.7.81

AUTHOR - B. Lipscombe

Hardware required - Plotter

Purpose

This program designs the profile of a cam which, in conjunction with a number of radial piston-followers, will cancel pump flow fluctuations supplied as harmonic data. The flow and volume fluctuations effected by the mechanism and the cam to piston-follower contact stress are also presented graphically.

Associated subroutines

Gino plotting routines (gino-f mk 2.6).

Subroutines "conversion", "piston" and "fit" are included in the main program.

Input information (via file 16)

j	Number of gear teeth	-
ffreg	Fundamental pumping frequency	Hz
n	Number of harmonics to be cancelled	-
m	Number of piston-followers	-
d	Piston-follower diameter	mm
r	Piston-follower radius	mm
cl	Piston-follower contact length	mm
pmass	Piston-follower mass	gram
c	Mean cam radius	mm
Qm	Mean pump flow	l/s
Ec	Young's modulus of cam material	N/m^2
Ef	Young's modulus of piston-follower material	N/m^2
Uc	Poisson's ratio of cam material	-
Uf	Poisson's ratio of piston-follower material	-
working pressure	Mean pump outlet working pressure	bar
a	Harmonic amplitude of source flow	m^3/s
p	Harmonic phase of source flow	deg.

This input information is applicable to a particular mechanism design format which is explained in the program action. For application to pumps other than gear units, replace number of gear teeth by number of pumping elements.

A typical example of input data is shown in table A1.2. This example corresponds to the design of a cam profile which cancels the first three harmonics of pump flow ripple.

The output from the program is in three forms. A screen message indicates the violation of one or more preset design criteria. This occurs if the cam pressure angle is greater than 15^{o}, the Hertz stress is greater than 10^{4} bar or the cam radius is greater than 12.6 mm. A typical example is shown in table A1.3. This is particularly useful as it enables a program run to be terminated, the input file to be modified and the program re-run until the cam is acceptable.

The normal graphics output, which may be viewed on a suitable graphics terminal or by use of a plotter, consists of the definition of the cam radius, the flow and volume variations effected and the cam to piston-follower Hertz stress for one complete revolution of the cam. The generated cam profile may be viewed by invoking the subroutine "piston". This routine was originally used to check the directly calculated profile during the development of the program. Examples of typical graphical output are shown in fig. A1.2.

The program also outputs, via file 07, the values of a number of variables at discrete intervals of cam rotational angle. This output was used for evaluation of the cam characteristics and the subsequent preparation of detailed manufacturing drawings.

<u>Information output</u> (via terminal-screen)

When a design criterion is exceeded.

theta	Angle to centre of piston radius	deg.
fi	Cam pressure angle	deg.

curvc	Radius of cam curvature	mm
R	Cam radius	mm
Hertz-stress	Cam to piston-follower contact stress	bar

Information output (via plotter or screen)

xxa	x ordinate of piston-cam contact point	mm
yya	y ordinate of piston-cam contact point	mm
Qs	Pump flow ripple	l/s
Vs	Pump volume variation	mm^3
thetad	Cam angle	deg.
Hertz-stress	Cam to piston-follower contact stress	bar
number	Number of points on cam profile	-

Information output (via file 07)

theta	Angle to centre of piston radius	deg.
fi	Cam pressure angle	deg.
curv	Radius of curvature of piston displacement profile	mm
curvc	Radius of cam curvature	mm
aminpress	Minimum pressure to ensure piston-cam contact	bar
force	Net radial force on piston	N
Hertz-stress	Cam to piston-follower contact stress	bar
Qs	Pump flow ripple	l/s
Vs	Pump volume variation	mm^3
angle	Uniformly incremented cam angle	deg.
value	Cam radius corresponding to angle	mm
theta 1	Non-uniformity incremented cam angle	deg.
R	Cam radius corresponding to theta 1	mm

<u>In source variables</u> (not shown in input or output)

D	Piston-radius-centre displacement profile	m
D1	1st derivitive of displacement profile	
D2	2nd derivitive of displacement profile	
V	Piston-follower radial velocity	m/s
ACC	Piston-follower radial acceleration	m/s^2
freq	Fundamental pumping frequency	rad/s
area	Piston-follower displacement area	m^2
i	Counter	

<u>Limitations and Accuracy of Program</u>

The program translates a given pump flow variation into the profile of a cam which will impart equivalent displacement variations to a number of radially disposed, radiused-piston-followers. To ensure the piston-followers can reproduce the required volume variations the minimum concave radius of cam curvature must be greater than the radius of curvature of the piston-followers.

The accuracy of profile definition or profile resolution can be varied by changing the number of points at which evaluation is performed. Generally evaluations at degree intervals ("number" = 360) have been used but half degree intervals ("number" = 720) were found necessary to provide the specific information required for numerically controlled grinding of a four harmonic profile.

<u>Storage</u> - 2 records.

Program Action and Algorithm

The program stores the values of seventeen input variables which are read from file 16. A typical example of file 16 is shown in table A.1.2. The amplitude and phase of harmonic components of pump flow ripple are stored in two arrays which can hold information for a maximum of ten harmonics. The phase of the selected harmonics is converted from degrees to radians by do loop 20.

The necessary parameters are then evaluated at intervals of cam rotational angle. The number of points of evaluation in a complete cycle (360 degrees) of cam rotation is determined by the variable "number" and is executed by do loop 30. The size of array (k) for each parameter is sufficient to allow evaluations at intervals of 0.25 degree (equivalent to "number" = 1440).

Evaluated parameters:-

(i) The angle to the centre of the piston-follower radius theta(k) = (2.pi.(k-1)/number, in radians is used for in source calculations
thetad(k) = theta(k).180/pi, in degrees is used as a convenient base for graphics output.

(ii) The pump flow ripple. The instantaneous value of pump flow ripple at increments of cam (and pump drive-shaft) rotational angle is given by:-
Qs(k) = Qs(k) + a(i).sin(j.i.theta(k) + P(i)).
Initially Qs(k) is set to zero. The variable Qs(k) is then the summation of a selected number of

harmonics, n, of individual amplitude a(i) and phase P(i). The summation of the selected number of harmonics of flow ripple and other dependent parameters is done by do loop 40.

(iii) The pump volume ripple. The instantaneous value of pump volume ripple is given by the $\int Qs\ dt$

$$Vs(k) = Vs(k) - \frac{a(i)}{freq.i}.cos(j.i.theta(k) + P(i))$$

(iv) The mechanism volume ripple. The volume variation to be produced by the mechanism, which will cancel the volume variation produced by the pump, is identical in amplitude and phase to that of the pump but opposite in sense. The relationship between mechanism displacement volume and cam angle is given by:-

$$D(k) = D(k) + \frac{a(i)}{freq.i}.cos(j.i.theta(k) + P(i))$$

To enable the subsequent evaluation of cam profile curvature and pressure angle the first and second derivitives of displacement colume w.r.t. theta are required. Therefore:-

$$D1(k) = D1(k) - \frac{a(i)}{freq}.j.sin(j.i.theta(k) + P(i))$$

and $$D2(k) = D2(k) - \frac{a(i).i}{freq}.j^2.cos(j.i.theta(k) + P(i))$$

(v) The piston-follower radial velocity. The instantaneous radial velocity of an individual piston-follower is determined by dividing the total instantaneous flow variation by the total area of all the active piston-followers:-

$$V(k) = V(k) - \frac{a(i).\sin(i.j.theta(k) + P(i))}{m.area}$$

(vi) The piston-follower radial acceleration. The radial component of piston-follower acceleration is determined using, from ref. (44):-

$$ar = \frac{d^2r}{dt^2} - r\left[\frac{d\theta}{dt}\right]^2$$

where the piston follows the curve $r = f(\theta)$.

After making the appropriate substitutions

$$ACC(k) = ACC(k) + \frac{a(i)}{m.area}.\cos(j.i.theta(k) + P(i).j.freq - D(k) . \left[\frac{freq}{j}\right]^2$$

(vii) The minimum working pressure to ensure continuous contact between the piston-follower and the cam is assumed to be totally dependent upon piston-follower acceleration. Frictional effects are not taken into account.

$$aminpress(k) = \frac{ACC(k).pmass}{area}$$

The sign of aminpress is determined by the sign of ACC. Only +ve values are significant where a minimum pressure-force is required to provide a piston acceleration sufficient to maintain piston contact with the cam.

(viii) The resultant force on the piston-follower is assumed to be dependent upon the effect of the mean working pressure and the force required to accelerate the piston-follower mass.

Force(k) = working pressure.area + ACC(k).pmass

The calculation of parameters which require the direct summation of flow ripple harmonic components is complete and do loop 40 ends. The calculation of subsequent parameters is carried out at increments of cam angle by the continuation of do loop 30.

(ix) The addition of pump flow ripple to pump mean flow. This produces the instantaneous pump output flow as shown in the graphics output.

Qs(k) = Qs(k) + Qm

(x) The piston-radius-centre displacement profile. The displacement of the centre of a piston-follower contact radius is determined by dividing the instantaneous displacement volume by the combined area of all the active pistons. The displacement of the piston-follower is then added to the selected mean cam radius and the piston-follower contact radius to produce the piston-radius-centre displacement profile. A typical example is shown in fig. A.1.3

$$D(k) = \frac{D(k)}{area.m} + c+r$$

The first and second derivitives of piston-radius-centre displacement profile are

$$D1(k) = \frac{D1(k)}{area.m} \quad \text{and} \quad D2(k) = \frac{D2(k)}{area.m} \quad \text{respectively.}$$

(xi) The slope of the tangent to the displacement profile. Fig. A.1.3 shows the piston-follower and cam contact geometry. As the piston-follower radius is normal to both the cam profile and the piston-radius-centre displacement profile, tangents to the piston-radius at the contact point and the centre of piston radius are also tangents to both profiles. Hence the cam pressure angle $\phi = \phi_1 - \frac{\pi}{2}$

From ref. (44) the angle ϕ_1 between the radius vector and the tangent at a point on a curve of the general form $r = f(\theta)$ is $\phi_1 = \text{atan}\left[\frac{r}{\frac{dr}{d\theta}}\right]$. From this it follows that the cam pressure angle

$$\text{fi} = -\ \text{atan}\left[\frac{\text{D1(k)}}{\text{D(k)}}\right]$$

where D1(k) is the instantaneous value of the first derivitive of D(k)w.r.t theta(k).

(xii) The angle between the piston-centre radius and the point of contact. This is held as a temporary variable, theta1(k) which subsequently becomes the angle to the point of contact. By the geometry shown in Fig.A.1.3

$$\text{theta1(k)} = \text{atan}\left[\frac{\text{r.sin(fi)}}{\text{D(k)-r.cos(fi)}}\right]$$

The sign of temporary variable theta1(k) is +ve or -ve depending upon which side of the piston centre line contact with the cam is made.

(xiii) The radius to the point of contact. By simple geometry $\text{R(k)} = \frac{\text{D(k)-r.cos(fi)}}{\text{cos(theta1(k))}}$

(xiv) The angle to the point of contact. The angle theta(k) to the centre of the piston-follower radius, is now modified to theta1(k), such that the cam profile is now defined by theta1(k) and R(k).

theta1(k) = theta(k)-theta1(k) temporary.

(xv) The radius of curvature on the piston-radius-centre displacement profile. From ref. (44) the curvature at a point (r, θ) of the curve $r = f(\theta)$ is

$$K = \frac{r^2 + 2r'^2 - rr''}{(r^2 + r'^2)^{3/2}}$$

where r' and r" are respectively the first and second deritives of r w.r.t θ . Hence the radius of curvature at the centre of piston radius is given by:-

$$\text{curv(k)} = \frac{(\text{D(k)}^2 + \text{D1(k)}^2)^{3/2}}{\text{D(k)}^2 + 2.\text{D1(k)}^2 - \text{D(k).D2(k)}}$$

The corresponding radius of curvature at the point of contact with the cam profile is given by

curvc(k) = curv(k) - r

(xvi) The Hertz stress criterion for cam to piston-follower contact. The magnitude of the contact stress between the cam and piston-follower was calculated for incremented values of cam angle using the following equation from ref. (37)

$$\text{Hertzstress} = \left[\frac{\text{Force(k).sec(fi)}\left(\dfrac{1}{\text{curvc(k)}} + \dfrac{1}{\text{r}}\right)}{\text{pi.cl}\left(\dfrac{1 - \text{Uc}^2}{\text{Ec}} + \dfrac{1 - \text{Uf}^2}{\text{Ef}}\right)}\right]^{1/2}$$

All the variables are as previously defined.

```
c      PROGRAM NAME - CAM
c
c
c      TITLE - Cam Profile Design Program For Flow
c              Ripple Generating Mechanism
c
c      Fortran IV    Honeywell Multics    20.7.81
c
c      AUTHOR - B.Lipscombe
c
c      Hardware Required - Plotter
c
c      No Normal Error Messages
c
c      Purpose:-
c
c       This program designs the profile of a cam which, in
c       conjunction with a number of radial piston-followers,
c       will cancel pump flow fluctuations supplied as harmonic
c       data.The flow and volume fluctuations affected by the
c       mechanism and the cam to piston-follower contact stress
c       are also presented graphically.
c
c     ASSOCIATED SUBROUTINES
c
c     Gino plotting routines plus CONVERSION,
c          PISTON and FIT included in main program
c
c     INPUT VARIABLES (via file16)
c
c          j          Number of gear teeth                      -
c          ffreq      Fundamental pumping frequency             Hz
c          n          Number of harmonics to be cancelled       -
c          m          Number of piston-followers                -
c          d          Piston-follower diameter                  mm
c          r          Piston-follower radius                    mm
c          cl         Piston-follower contact length            mm
c          pmass      Piston-follower mass                      gram
c          c          Mean cam radius                           mm
c          Qm         Mean pump flow                            l/s
c          Ec         Young's mod. cam matl.                    N/m**2
c          Ef         Young's mod. follower matl.               N/m**2
c          Uc         Poisson's ratio cam matl.                 -
c          Uf         Poisson's ratio follower matl.            -
c   workingpressure   Mean pump outlet workingpressure          bar
c          a          Harmonic amplitude of source flow         cu.m/s
c          p          Harmonic phase of source flow             deg.
c
c      OUTPUT VARIABLES (via plotter)
c
c          xxa        x ordinate of piston-cam contact point    mm
c          yya        y ordinate of piston-cam contact point    mm
c          Qs         Pump flow ripple                          l/s
c          Vs         Pump volume variation                     cu.mm
c          thetad     Cam angle                                 deg.
c       Hertzstress   Cam to piston-follower contact stress     bar
c          number     Number of points on cam profile           -
c
c      OUTPUT VARIABLES (via screen)
c          When a design criterion is exceeded
```

```
c
c          theta      Angle to centre of piston radius         deg.
c          fi         Cam pressure angle                       deg.
c          curvc      Radius of cam curvature                  mm
c          R          Cam radius                               mm
c      Hertzstress    Cam to piston-follower contact stress    bar
c
c      OUTPUT VARIABLES (via file07)
c
c          theta      Angle to centre of piston radius         deg.
c          fi         Cam pressure angle                       deg.
c          curv       Rad.of curvature of piston disp.profile  mm
c          curvc      Radius of cam curvature                  mm
c        aminpress    Min.press. to ensure piston-cam contact  bar
c          force      Net radial force on piston               N
c       Hertzstress   Cam to piston-follower contact stress    bar
c          Qs         Pump flow ripple                         l/s
c          Vs         Pump volume variation                    cu.mm
c          angle      Uniformly incremented cam angle          deg.
c          value      Cam radius corresponding to angle        mm
c          theta1     Non-uniformly incremented cam angle      deg.
c          R          Cam radius corresponding to theta1       mm
c
c          ******************************************************

           dimension Qs(1500),Vs(1500),thetad(1500)
           dimension Hertzstress(1500),Force(1500),curvc(1500)
           dimension V(1500),ACC(1500),aminpress(1500)
           dimension xx(1500),yy(1500),theta(1500),a(10),p(10)
           dimension xxa(1500),yya(1500),R(1500),theta1(1500)
           dimension D(1500),D1(1500),D2(1500),curv(1500)
           dimension xxxa(1500),yyya(1500),angle(1500),value(1500)
           external conversion(descriptors),fit(descriptors)
           external piston(descriptors),chahol(descriptors)
           call vterm
           call units(6.4)
c    Enter data
           write(6,11)
11         format(1h ,"Enter:- Number of gear teeth",/1h ,8x,
&          "Fundamental pumping frequency (Hz)",/1h ,8x,
&          "Number of harmonics to be cancelled",/1h ,8x,
&          "Number of pistons",/1h ,8x,"piston diameter (mm)",
&          /1h ,8x,"piston radius (mm)",/1h ,8x,"piston contact
&          length (mm)",/1h ,8x,"piston mass (gram)",/1h ,
&          8x,"mean cam radius(mm)",/1h ,8x,"mean pump flow(l/s)",
&          /1h ,8x,"Young's mod. cam matl.(N/m**2)",/1h ,8x,
&          Young's mod. follower matl.(N/m**2)",/1h ,8x,
&          "Poisson's ratio cam matl.",/1h ,8x,"Poisson's
&          ratio follower matl.",/1h ,8x,"working pressure(bar)")
           read(16,10)j,ffreq,n,m,d,r,cl,pmass,c,Qm,Ec,Ef,Uc,Uf,
&          workingpressure
           write(6,12)
12         format(1h ,"Enter amplitude (m**3/s) and (degree)
&          of harmonics")
           read(16,10)(a(i),p(i),i=1,n)
c    output to screen values which exceed design criterion
           write(6,22)
22         format(1h ,/////1h ,10x,"A DESIGN CRITERION HAS BEEN
&          EXCEEDED",//1h ,6x,"theta",6x,"fi",6x,"curvc",5x,"R",
&          4x,"Hertzstress")

           write(7,18)
18         format(1h ,2x,"theta",8x,"fi",8x,"curv",6x,"curvc",
&          4x,"aminpress",3x,"force",4x,"Hertzstress",6x,"Qs",
&          8x,"Vs")
```

```
      pi=3.141592
      c=c/1000.
      r=r/1000.
      d=d/1000.
      pmass=pmass/1000
      cl=cl/1000.
      Qm=Qm/1000
c   convert Hz to rad/s
      freq=ffreq*2*pi
c   calculate piston area
      area=(pi*d**2)/4.
c   designate number of points on cam profile
      number=360
c   convert phase from degree to radian for all harmonics
      do 20 i=1,n
      p(i)=p(i)*pi/180.
20    continue
c   start loop for incrementing cam angle
      do 30 k=1,number+1
      theta(k)=(2*pi)*(k-1)/number
      thetad(k)=theta(k)*180/pi
c   set initial values to zero
      Qs(k)=0.
      Vs(k)=0.
      D(k)=0.
      D1(k)=0.
      D2(k)=0.
      V(k)=0.
      ACC(k)=0.
c   start loop for summing all components of flow ripple
      do 40 i=1,n
c   calculate pump flow ripple
      Qs(k)=Qs(k)+a(i)*sin(j*i*theta(k)+p(i))
c   calculate pump volume ripple
      Vs(k)=Vs(k)-(a(i)/(freq*i))*cos(j*i*theta(k)+p(i))*1.E+9
c   calculate displacement volume profile
      D(k)=D(k)+(a(i)/(freq*i))*cos(j*i*theta(k)+p(i))
c   calculate velocity profile
      D1(k)=D1(k)-a(i)*j/freq*sin(j*theta(k)*i+p(i))
c   calculate acceleration profile
      D2(k)=D2(k)-a(i)*i*j**2/freq*cos(j*i*theta(k)+p(i))
c   calculate piston radial velocity
      V(k)=V(k)-a(i)*sin(i*j*tneta(k)+p(i))/(m*area)
c   calculate piston radial acceleration
      ACC(k)=ACC(k)+a(i)/(area*m)*cos(j*i*theta(k)+p(i))
&     *i*freq-D(k)*((freq/j)**2)
c   calculate minimum working pressure for contiuous
c     cam-piston contact
      aminpress(k)=ACC(k)*pmass/(area*100000)
c   calculate resultant force on piston
      Force(k)=workingpressure*1.E+05*area+ACC(k)*pmass
40    continue
c   add pump flow ripple to mean flow
      Qs(k)=(Qs(k)+Qm)*1000
c   calculate piston-radius-centre displacement profile
      D(k)=D(k)/(area*m)+c+r
c   calculate 1st derivative of displacement profile
      D1(k)=D1(k)/(area*m)
c   calculate 2nd derivative of displacement profile
      D2(k)=D2(k)/(area*m)
```

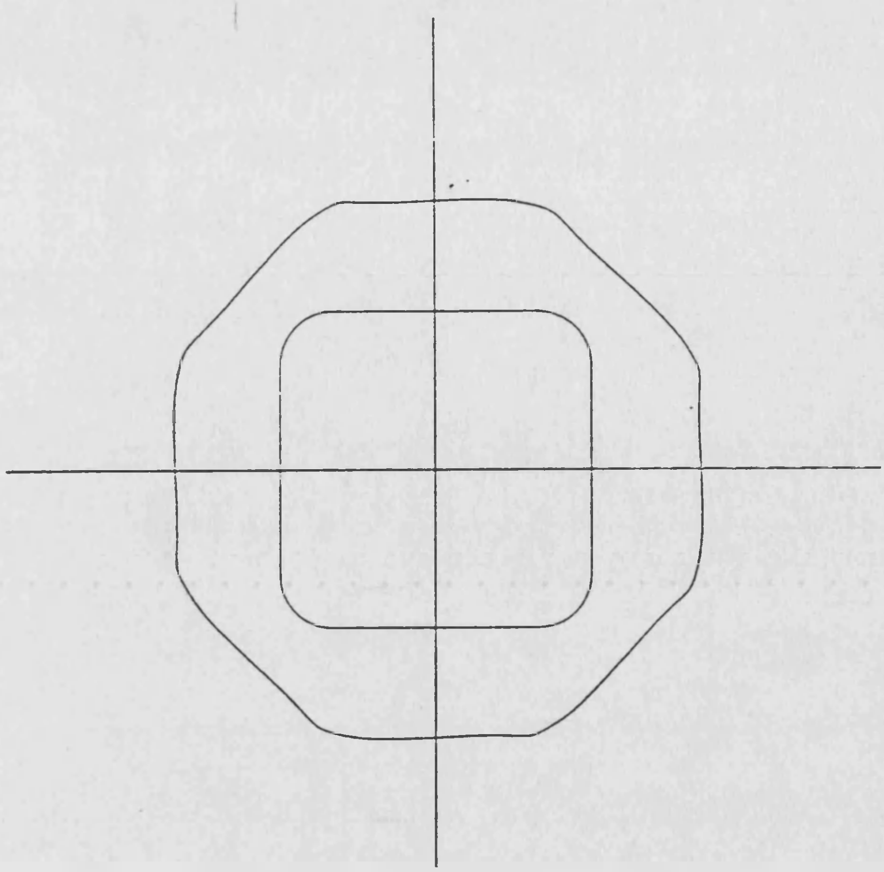
```
c   calculate slope of tangent to displacement profile
          fi=-atan(D1(k)/D(k))
c   calculate angle between piston-radius-centre and point
c         of contact
          theta1(k)=atan(r*sin(fi)/(D(k)-r*cos(fi)))
c   calculate radius to point of contact
          R(k)=(D(k)-r*cos(fi))/cos(theta1(k))
c   calculate angle to point of contact
          theta1(k)=theta(k)-theta1(k)
c   calculate radius of curvature at point theta on piston
c         displacement profile
          curv(k)=sqrt((D(k)**2+D1(k)**2)**3)/
c                    (D(k)**2+2*D1(k)**2-D(k)*D2(k))
c   calculate radius of curvature of cam profile
          curvc(k)=curv(k)-r
c   calculate Hertz stress criterion for cam-follower contact
          Hertzstress(k)=sqrt(abs((Force(k)*1/cos(fi)*(1/curvc(k)
&         +1r))/(pi*cl*((1-Uc**2)/Ec+(1-Uf**2)/Ef))))/100000
c    if design criterion exceeded output values to screen
          if(abs(fi).gt.pi/12..or.Hertzstress(k).gt.10000..
&         or.R(k).gt.0.0126)write(6,13)theta(k)*180./pi,
&         fi*180./pi,curvc(k)*1000.,R(k)*1000.,Hertzstress(k)
13        format(1h ,4x,f7.2,2x,f7.2,2x,f7.2,2x,f7.3,2x,f9.2)
          write(7,17)theta(k)*180./pi,fi*180./pi,curv(k)*1000,
&         curvc(k)*1000
&         aminpress(k),Force(k),Hertzstress(k),Qs(k),Vs(k)
17        format(1h ,6(f7.2,4x),f9.2,4x,f7.4,4x,f7.2)
30        continue
c   convert x-y to polar co-ordinates
          call conversion(xx,yy,D,theta,number+1)
          call conversion(xxa,yya,R,theta1,number+1)
          call conversion(xxxa,yyya,value,angle,number)
c   interpolate between theta1 values to allow equal
c         division of angle
          call fit(angle,value,theta1,R,number+1,j)
c   output numerical data
          write(7,21)
21        format(1h ,//////////,5x,"angle",6x,"radius(mm)",
&         4x,"radius(in)",6x,"theta1",8x,"R(mm)")
          write(7,19)(angle(i)*180./pi,value(i)*1000.,value(i)
&         *1000/25.4,theta(i)*180./pi,R(i)*1000.,i=1,number+1)
19        format(1h ,f9.3,5x,f9.3,5x,f9.5,5x,f9.3,5x,f9.3)
c   setup graphics output
c   cam graphics output
c   plotting scale factors(sf=120 for normal,40 for actual)
          sf=40.
          a1=0.+sf*0.00464/0.04
          a2=0.-sf*0.00464/0.04
          a3=0.+sf*0.007/0.04
          a4=0.-sf*0.007/0.04
          call axipos(0,0.,0.,sf,1)
          call axipos(0,0.,0.,sf,2)
          call axisca(1,10,-0.02,0.02,1)
          call axisca(1,10,-0.02,0.02,2)
          call axidra(0,0,1)
          call axidra(0,0,2)
          call gramov(0.007,-0.00464)
          call gralin(0.007,0.00464)
          call arcto2(a1,a1,a1,a3,1)
          call gralin(-0.00464,0.007)
```

```
        call arcto2(a2,a1,a4,a1,1)
        call gralin(-0.007,-0.00464)
        call arcto2(a2,a2,a2,a4,1)
        call gralin(0.00464,-0.007)
        call arcto2(a1,a2,a3,a2,1)
c   generate cam profile (for visual check)
c       call broken(0)
c       call grapol(xx,yy,number+1)
c       do 50 k=1,number+1
c       call piston(xx(k),yy(k),r)
c       50 continue
c   plot cam profile
        call broken(0)
        call grapol(xxa,yya,number+1)
        call grapol(xxxa,yyya,number)
c   pump flow ripple graphics output
        call piccle
        call axipos(0,-50.,-50.,120.,1)
        call axipos(0,-50.,-50.,100.,2)
        call axisca(1,10,0.,360.,1)
        call axisca(1,10,0.,1.0,2)
        call chahar(1,1)
        call movto2(-60.,-11.)
        call chahol("FLOW - l/s*.")
        call chahar(1,0)
        call movto2(1.,-60.)
        call chahol("ANGLE - deg*.")
        call grid(-2,1,1)
        call grapol(thetad,Qs,360)
c   pump volume ripple graphics output
        call piccle
        call axipos(0,-50.,-50.,120.,1)
        call axipos(1,-50.,-50.,100.,2)
        call axisca(1,10,0.,360.,1)
        call axisca(1,10,-100.,100.,2)
        call chahar(1,1)
        call movto2(-60.,-25.)
        call chahol("VOLUME VARIATION - cu mm*.")
        call chahar(1,0)
        call movto2(1.,-60.)
        call chahol("ANGLE - deg*.")
        call grid(-2,1,1)
        call grapol(thetad,Vs,360)
c   cam-follower Hertzstress graphics output
        call piccle
        call axipos(0,-50.,-50.,120.,1)
        call axipos(0,-50.,-50.,100.,2)
        call axisca(1,10,0.,360.,1)
        call axisca(1,10,0.,20000.,2)
        call chahar(1,1)
        call movto2(-60.,-11.)
        call chahol("HERTZSTRESS - bar*.")
        call chahar(1,0)
        call movto2(1.,-60.)
        call chahol("ANGLE - deg*.")
        call grid(-2,1,1)
        call grapol(thetad,Hertzstress,360)
        call devend
10      format(v)
        stop
        end
```

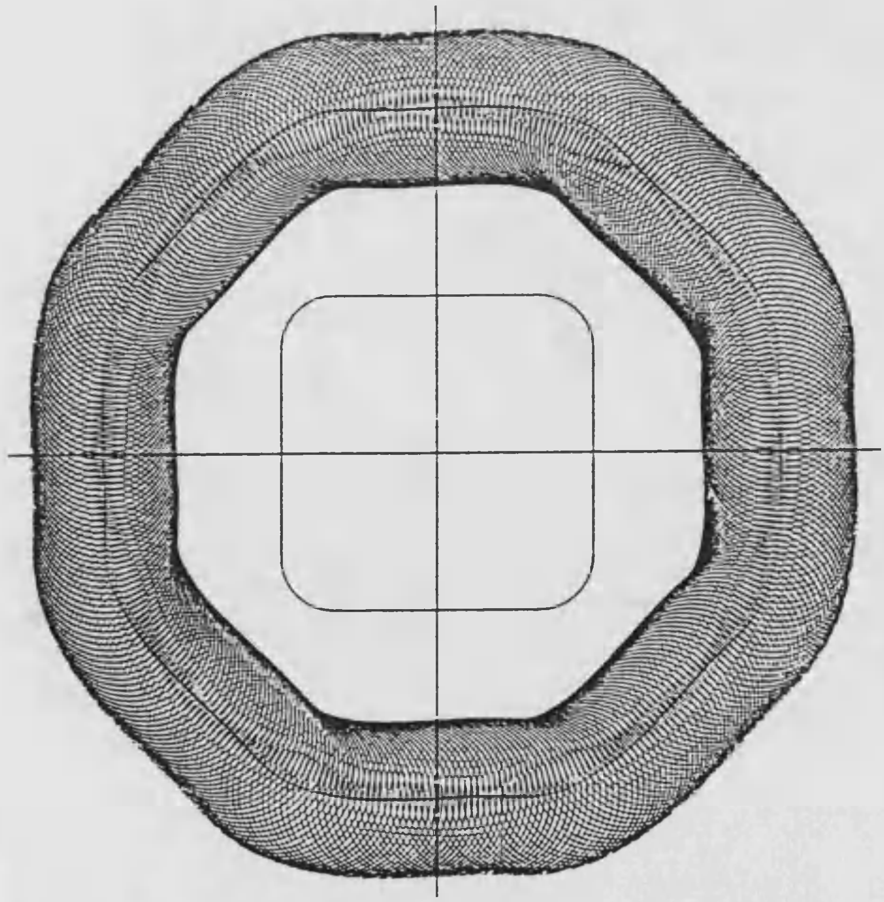
```
      subroutine conversion(xx,yy,rr,tt,n)
      dimension xx(1500),yy(1500),rr(1500),tt(1500)
      do 10 i=1,n
      yy(i)=rr(i)*sin(tt(i))
      xx(i)=rr(i)*cos(tt(i))
10    continue
      return
      end
```

```
      subroutine piston(x1,y1,r)
      dimension x(100),y(100)
      nn=40
      pi=3.141592
      do 10 i=1,nn
      x(i)=x1+r*cos(2*pi*i/nn)
      y(i)=y1+r*sin(2*pi*i/nn)
10    continue
      x(nn+1)=x(1)
      y(nn+1)=y(1)
      call grapol(x,y,nn+1)
      return
      end
```

```
      subroutine fit(xx,yy,tt,zz,number,j)
      dimension xx(1500),yy(1500),tt(1500),zz(1500)
      pi=3.141592
      ainterval=360./(number-1)
      acycle=(number-1)/j
      do 10 i=1,acycle
      xx(i)=(360./j+(i-1)*ainterval)*pi/180.
      do 20 k=1,number
      if(xx(i).gt.tt(k)) go to 20
      yy(i)=zz(k-1)+(zz(k)-zz(k-1))/(tt(k)-tt(k-1))*
&     (xx(i)-tt(k-1))
      go to 10
20    continue
10    continue
      do 40 kk=1,j
      do 30 i=1,acycle
      nn=i+acycle*(kk-1)
      yy(nn)=yy(i)
      xx(nn)=((nn-1)*ainterval)*pi/180.
30    continue
40    continue
      return
      end
```



Direct definition of cam profile



Generated definition of cam profile

FIG. A1.2 EXAMPLE OF GRAPHICAL OUTPUT FROM PROGRAM CAM

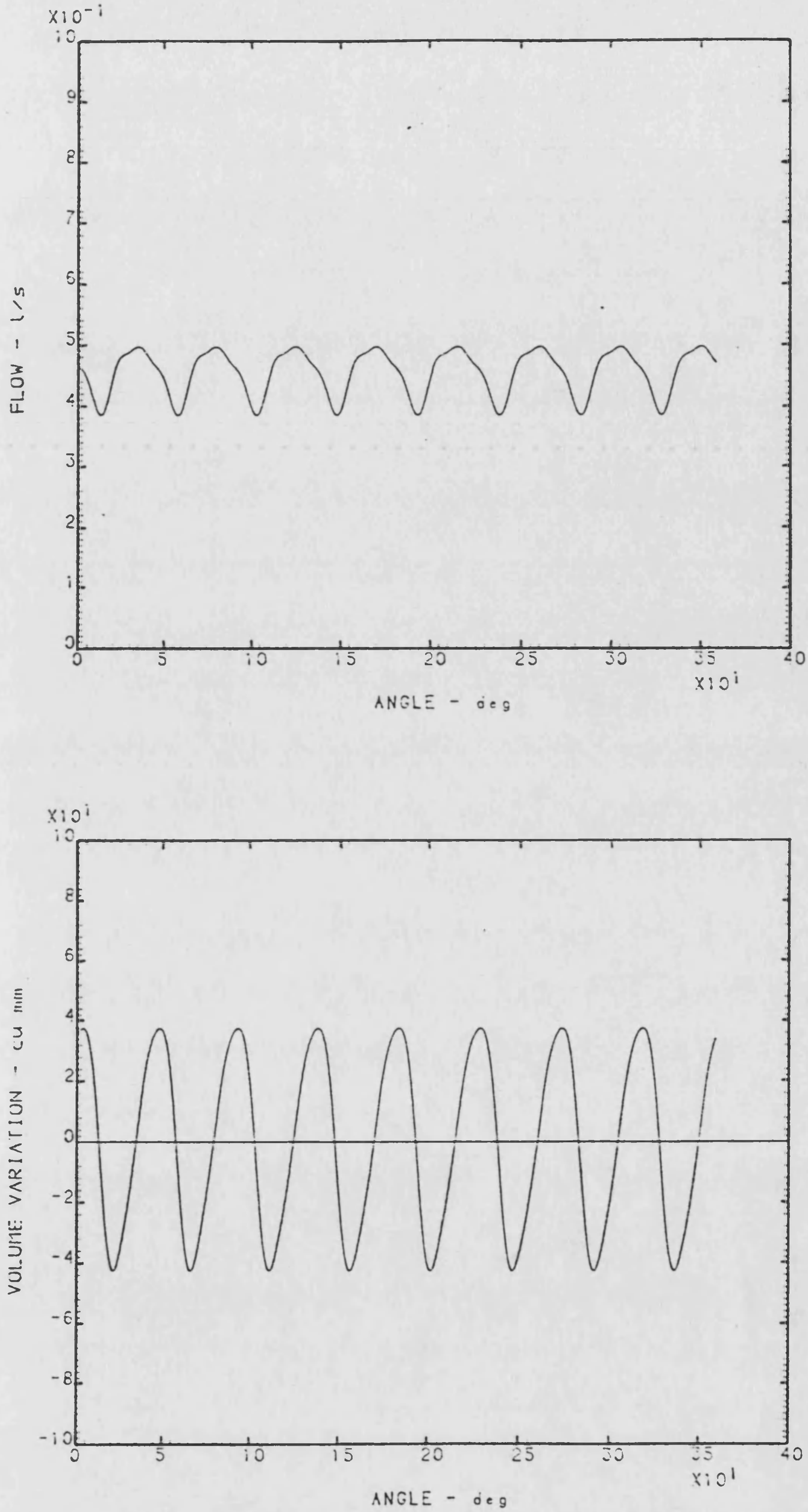


FIG. A1.2 CONTINUED

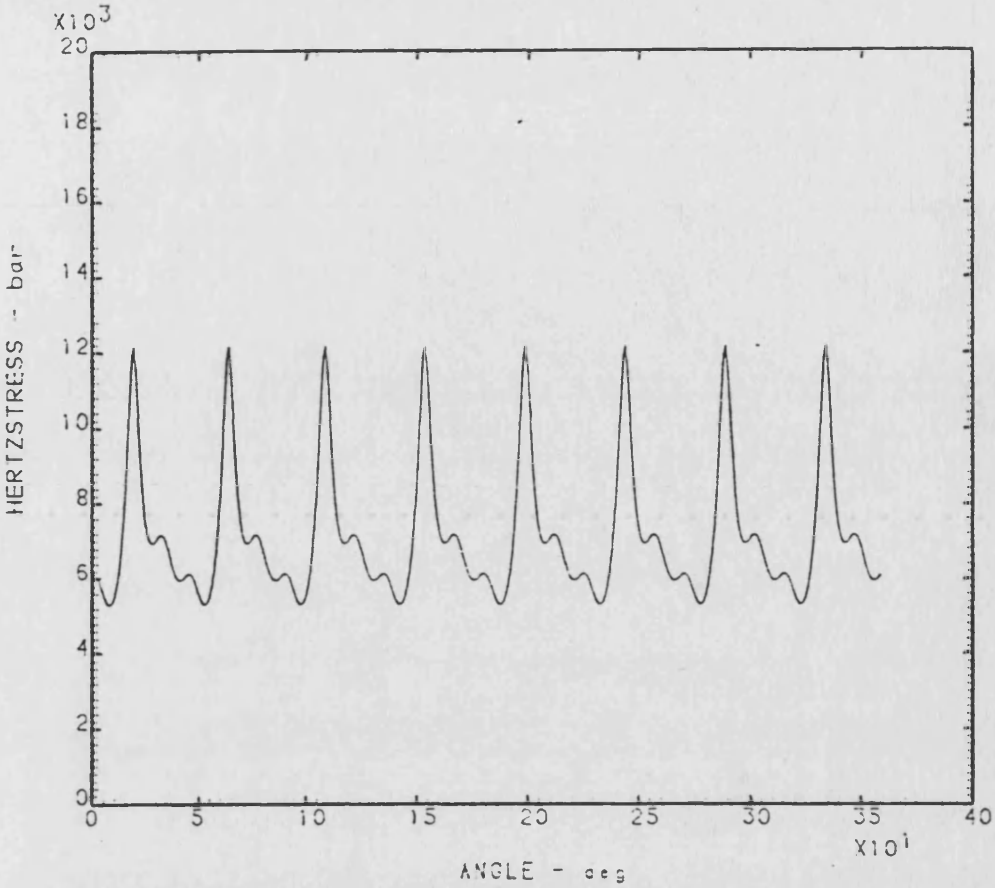


FIG. A1.2 CONTINUED

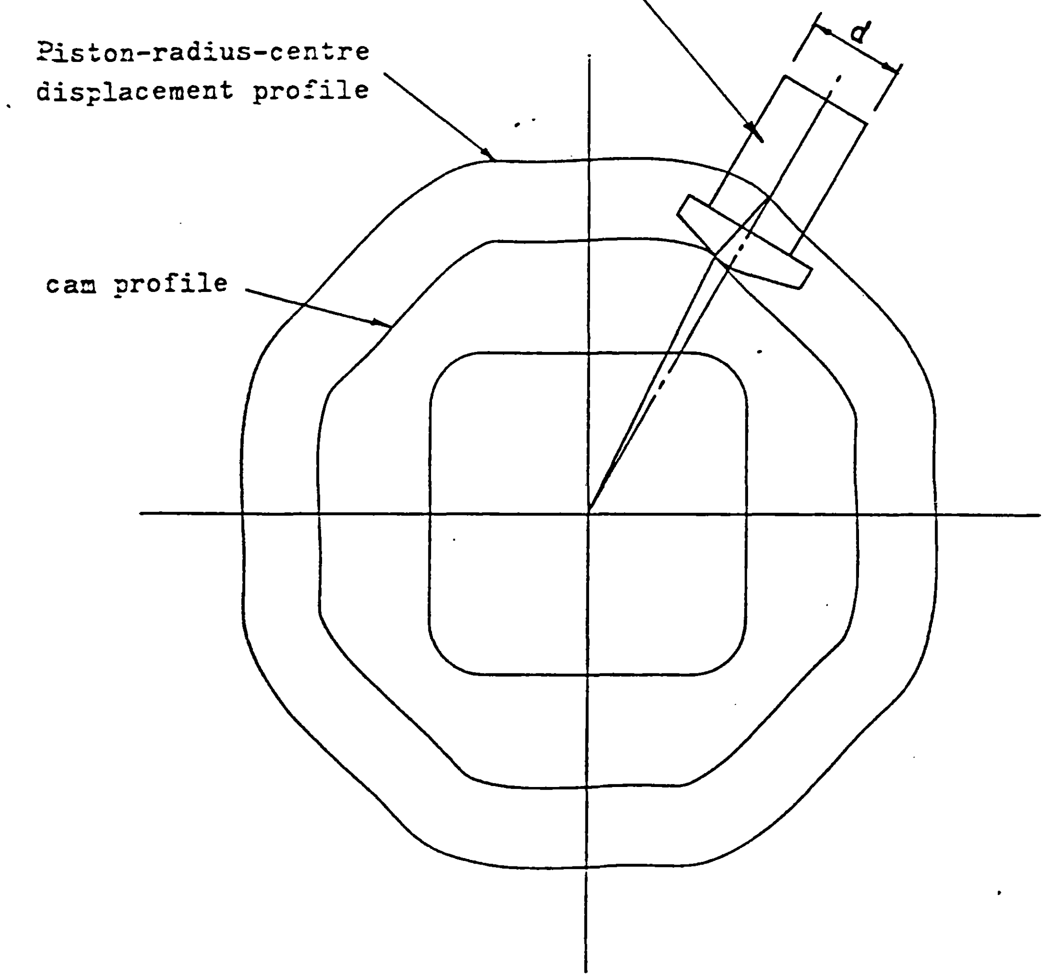
(Terms in brackets would not normally be seen)

(Number of gear teeth)	8
(Fundamental pumping frequency)	197
(Number of harmonics to be cancelled)	3
(Number of piston-followers)	8
(Piston-follower diameter)	4.242
(Piston-follower radius)	3.5
(Piston-follower contact length)	8.89
(Piston-follower mass)	1.3905
(Mean cam radius)	12.2
(Mean pump flow)	.455
(Young's mod. cam matl.)	2.08e+11
(Young's mod. follower matl.)	2.08e+11
(Poisson's ratio cam matl.)	.29
(Poisson's ratio follower matl.)	.29
(Mean pump outlet workingpressure)	210

(Pump source flow characteristics)

(Harmonic)	(Amplitude)	(Phase)
(1)	.495e-4	173
(2)	.144e-4	57
(3)	.805e-5	-60
(4)	.562e-5	175
(5)	.453e-5	58
(6)	.339e-5	-84

TABLE A1.2 EXAMPLE OF INPUT (FILE 16) FOR PROGRAM CAM



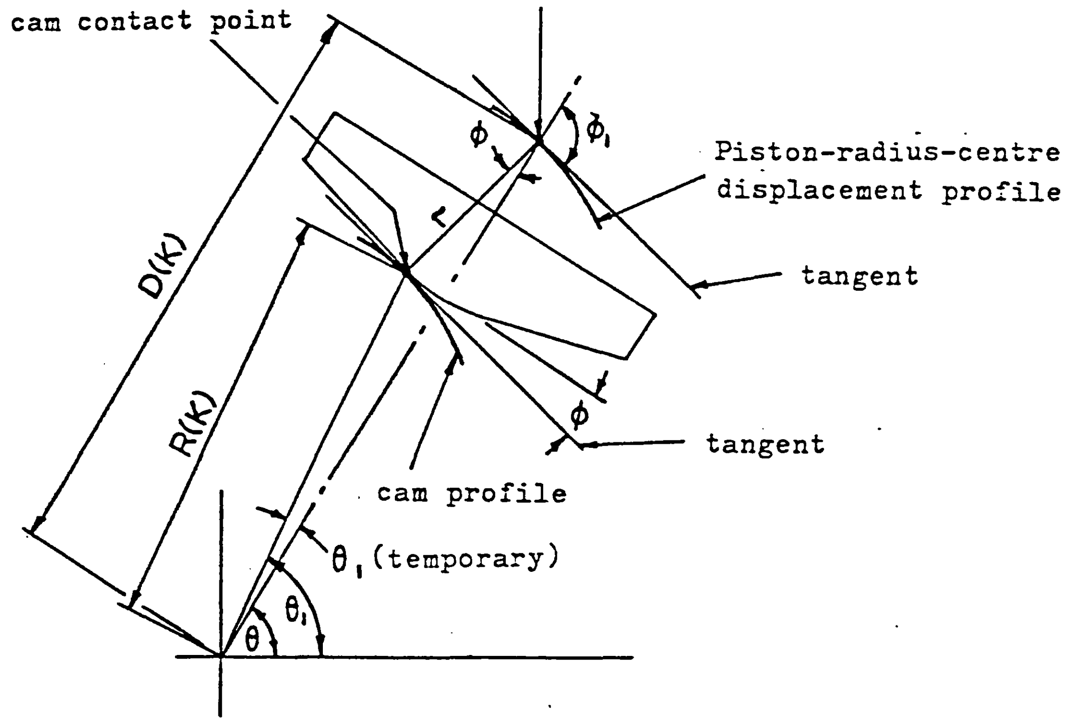


FIG. A1.3 DIAGRAMS SHOWING CAM AND PISTON-FOLLOWER GEOMETRY

```
Gino-f Mk 2.6
Enter:- Number of gear teeth
        Fundamental pumping frequency (Hz)
        Number of harmonics to be cancelled
        Number of pistons
        Piston diameter (mm)
        Piston radius (mm)
        Piston contact length (mm)
        Piston mass (gram)
        Mean cam radius (mm)
        Mean pump flow (l/s)
        Young's mod. cam matl. (N/m**2)
        Young's mod. follower matl. (N/m**2)
        Poisson's ratio cam matl.
        Poisson's ratio follower matl.
        Working pressure (bar)
Enter:- amplitude (m**3/s) and phase (degree) of harmonics

        (input from file16)
```

A DESIGN CRITERION HAS BEEN EXCEEDED

theta	fi	curvc	R	Hertzstress
17.00	-9.65	1.51	12.549	10797.03
18.00	-7.28	1.15	12.563	11891.69
19.00	-4.76	1.08	12.572	12156.21
20.00	-2.30	1.26	12.578	11441.92
21.00	-0.05	1.70	12.580	10280.36
62.00	-9.65	1.51	12.549	10797.02
63.00	-7.28	1.15	12.563	11891.68
64.00	-4.76	1.08	12.572	12156.21
65.00	-2.30	1.26	12.578	11441.93
66.00	-0.05	1.70	12.580	10280.37

TABLE A1.3 EXAMPLE OF SCREEN OUTPUT FROM PROGRAM CAM

Example of output (file07) from program CAM

theta	fi	curv	curvc	aminpress	force	Hertzstress	Qs	Vs
0.00	2.37	41.89	38.39	-0.23	296.47	6132.71	0.4661	35.44
1.00	1.74	42.61	39.11	-0.23	296.46	6126.78	0.4632	36.53
2.00	1.06	58.29	54.79	-0.27	296.41	6052.94	0.4600	37.28
3.00	0.24	212.02	208.52	-0.34	296.32	5915.98	0.4561	37.63
4.00	-0.82	-79.02	-82.52	-0.43	296.18	5740.13	0.4512	37.49
5.00	-2.16	-30.86	-34.36	-0.54	296.02	5559.35	0.4448	36.71
6.00	-3.81	-19.64	-23.14	-0.65	295.87	5406.94	0.4371	35.14
7.00	-5.71	-15.58	-19.08	-0.73	295.75	5309.75	0.4280	32.62
8.00	-7.75	-14.57	-18.07	-0.77	295.70	5287.00	0.4182	29.03
9.00	-9.79	-16.12	-19.62	-0.75	295.73	5351.77	0.4083	24.31
10.00	-11.64	-22.68	-26.18	-0.66	259.85	5513.51	0.3990	18.51

angle	radius(mm)	radius(in)	thetal	R(mm)
0.000	11.884	0.46786	-0.697	11.890
1.000	11.876	0.46754	0.487	11.879
2.000	11.870	0.46733	1.687	11.871
3.000	11.867	0.46722	2.930	11.867
4.000	11.869	0.46727	4.242	11.869
5.000	11.874	0.46749	5.638	11.879
6.000	11.884	0.46786	7.120	11.899
7.000	11.898	0.46841	8.672	11.934
8.000	11.919	0.46925	10.258	11.985
9.000	11.944	0.47025	11.831	12.051
10.000	11.976	0.47151	13.339	12.129

TABLE A1.4 EXAMPLE OF OUTPUT (FILE 07) FROM PROGRAM CAM

A.1.3 - COMPUTER PROGRAM DOCUMENTATION FOR fft.FORTRAN

Library classification

BENG (F) fft. 248.01

TITLE - Fast Fourier Transform Evaluation Program

NEW FORTRAN 9 Honeywell Multics 20.3.82

AUTHOR - B. Lipscombe

Hardware required - None

Purpose

This program calculates the discrete Fourier Transform of a sequence of n real data values.

Associated subroutines

NAG routine c06eaf is basis of program.

Input information (via file 56)

n	Number of data points	integer
x	Amplitude data in sequence	real

Output information (via screen)

jm1	Number of harmonic	-
A	Harmonic amplitude	as input
P	Harmonic phase	deg.

Limitations and Accuracy of Program

The c06eaf routine evaluates the same number of harmonic components as the number of real data points supplied; this program outputs only the first ten. Accuracy is dependent upon the number of data points given.

Inbuilt error messages

IFAIL 1 - at least one of the prime factors of n is greater than 19

IFAIL 2 - n has more than 20 prime factors.

IFAIL 3 - n is less than or equal to 1

Program Action and Algorithm

Given a sequence of n real data values xj, j=0, 1...n-1, this program calculates their discrete Fourier Transform defined by

$$\hat{z}_k = \frac{1}{\sqrt{n}} \sum_{j=0}^{n-1} xj \cdot \exp\left[\frac{-i\ 2\ \pi\ jk}{n}\right]$$

$k = 0,1,\ldots n-1$

where $\frac{1}{\sqrt{n}}$ is a scale factor in this definition.

The complex double sided spectrum is converted to a single sided spectrum and the first ten harmonic components are defined in terms of harmonic peak amplitude and phase. A listing of the program follows and an example of the input and output is shown in table A.1.5

```
c     PROGRAM NAME - fft.fortran
c
c
c     TITLE - Fast Fourier Transform Evaluation Program
c
c     New Fortran 9    Honeywell Multics    20.3.82
c
c     AUTHOR - B.Lipscombe
c
c     Hardware Required - none
c
c     Normal Error Messages:
c
c     IFAIL 1 - at least one primefactor of n>19
c     IFAIL 2 - n has more than 20 primefactors
c     IFAIL 3 - n < or = 1
c
c     Purpose:-
c
c          This program calculates the discrete Fourier
c          Transform of a sequence of n real data values.
c
c     ASSOCIATED SUBROUTINES
c
c     NAG routine c06eaf is basis of program
c
c     INPUT VARIABLES(via file56)
c
c          n          Number of data points          integer
c          x          Amplitude data in sequence     real
c
c     OUTPUT VARIABLES(via screen)
c
c          jm1        Number of harmonic             -
c          A          Harmonic amplitude             as input
c          P          Harmonic phase                 deg.
c
c    ************************************************
           double precision x(500)
           dimension a(500),b(500),A(500),P(500)
c     Input data
           write(6,11)
11         format(1h ,"Enter number and amplitude of
&          ordinates in sequence")
           read(56,12)n,(x(j),j=1,n)
           ifail=0
           call c06eaf(x,n,ifail)
           a(1)=x(1)
           b(1)=0.0
           n2=(n+1)/2
           do 20 j=2,n2
           nj=n-j+2
           a(j)=x(j)
           a(nj)=x(j)
           b(j)=x(nj)
           b(nj)=-x(nj)
20         continue
           if(mod(n,2).ne.0)goto 30
           a(n2+1)=x(n2+1)
           b(n2+1)=0.0
```

```
        write(6,14)
14      format(1h ,2x,"HARMONIC",9x,"AMP",8x,"PHASE(deg)")
c    Convert double sided complex spectrum to single sided
c    spectrum and compensate for inherent routine scale factor
30      do 40 j=1,n
        a(j)=a(j)*2/sqrt(n)
        b(j)=b(j)*2/sqrt(n)
        jm1=j-1
c    Convert real and imaginary parts to polar form
        A(j)=sqrt((a(j))**2+(b(j))**2)
        if(abs(b(j)).lt.1e-10)go to 155
        P(j)=(atan2(b(j),a(j))*180/3.14159)+90
        go to 158
155     P(j)=0.
158     if(jm1.eq.1)const=P(2)
        P(j)=P(j)-const*jm1
        if(jm1.gt.10)stop
530     if(abs(P(j)).lt.180.)go to 510
        if(P(j).lt.0.)go to 520
        if(P(j).gt.0.)P(j)=P(j)-360
        go to 530
520     P(j)=P(j)+360
        go to 530
c    Output harmonic amplitude and phase
510     write(6,13)jm1,A(j),P(j)
13      format(1h ,4x,i2,4x,f15.9,4x,f7.2)
40      continue
12      format(v)
        stop
        end
```

INPUT

```
50
  -.65264088E-04
  -.56921173E-04
  -.48933274E-04
  -.41300393E-04
  -.34022530E-04
  -.27099685E-04
  -.20531857E-04
  -.14319047E-04
  -.84612545E-05
  -.29584809E-05
   .21892762E-05
   .69820150E-05
   .11419737E-04
   .15502440E-04
   .19230126E-04
   .22602794E-04
   .25620445E-04
   .28283078E-04
   .30590693E-04
   .32543290E-04
   .34140870E-04
   .35383432E-04
   .36270976E-04
   .36803503E-04
   .36981012E-04
   .36803503E-04
   .36270976E-04
   .35383432E-04
   .34140870E-04
   .32543290E-04
   .30590693E-04
   .28283078E-04
   .25620445E-04
   .22602794E-04
   .19230126E-04
   .15502440E-04
   .11419737E-04
   .69820156E-05
   .21892762E-05
  -.29584809E-05
  -.84612532E-05
  -.14319046E-04
  -.20531856E-04
  -.27099684E-04
  -.34022530E-04
  -.41300393E-04
  -.48933274E-04
  -.56921173E-04
  -.65264088E-04
  -.73962023E-04
```

OUTPUT

HARMONIC	AMP	PHASE(deg)
0	0.000000059	0.00
1	0.000045023	0.00
2	0.000011300	90.00
3	0.000005056	-180.00
4	0.000002870	-90.00
5	0.000001859	0.00
6	0.000001310	90.00
7	0.000000979	-180.00
8	0.000000765	-90.00
9	0.000000618	0.00
10	0.000000514	90.00

TABLE A1.5 EXAMPLE OF INPUT AND OUTPUT FROM PROGRAM fft.FORTRAN

A.2 - PATENT SPECIFICATION OF FLOW RIPPLE REDUCTION DEVICE.

ABSTRACT OF THE INVENTION

"PUMPS AND MOTORS"

(with ref. to figures A2.1 and A2.2)

A pump or motor 11 comprising a casing 12, housing at least one rotary member 15 and having an inlet port and an outlet port, cam means 34 rotatable by said member, and a fluid displacement device driven by the cam means and including at least one reciprocal element 36. This element, upon rotation of said member through each revolution, so repeatedly projects into chamber 41 which is in communication with one of said ports as to effect a series of momentary decreases in the effective volume of that chamber and the element is also so repeatedly retracted from that chamber as to effect a series of momentary increases in the effective volume of the chamber. Thus momentary flow of fluid is caused repeatedly to occur out from, and back into, the chamber thereby substantially reducing pulsation of flow through the pump or motor, and also attendant noise, during operation.

"PUMPS AND MOTORS"

This invention relates to pumps and motors.

Hitherto in certain forms of pumps and motors, particularly those of gear type, it has been found that during operation thereof pulsation, or "ripple", in fluid flow has occurred. In gear pumps or motors for example the extent of such pulsation is dependent upon the shapes of the teeth of the gears, the number of teeth and the positions of the contact points on the flanks of the intermeshing teeth. Here ideally there should be no variation in flow with gear rotation at any designed operating speed, but in practice the actual operating characteristics produced have, graphically, for each gear revolution included a series of peaks and a series of troughs with such consequent flow pulsation that the pumps and motors have proved to be unacceptable for certain applications due to the noise level reached during operation.

In an endeavour to reduce noise, for example in gear pumps and motors, it has already been arranged for the gears thereof to have relatively large numbers of teeth. In order to reduce noise further gear pumps and motors of dual type have been produced in which one pair of gears have their teeth meshing one half tooth pitch out-of-phase with respect to a second pair of gears arranged in parallel with the first pair, and smoother operating characteristics have been achieved.

However the results have still left much to be desired from the noise standpoint.

The invention as claimed is intended to provide a remedy. It solves the problem of how to design a pump or motor in which means are provided for at least substantially reducing the said peaks and for at least substantially removing the said troughs in the operating characteristics of the pump or motor thereby substantially to reduce pulsation of flow and attendant noise during operation.

According to the invention a pump or motor comprises a casing, which houses at least one rotary member and which has an inlet port and an outlet port, cam means connected to be rotated by said member and a fluid displacement device which is driven by said cam means when rotating and which includes at least one reciprocable element which, upon rotation of said rotary member through each revolution, so repeatedly projects into a chamber in said casing which is in communication with one of said ports as to effect a series of momentary decreases in the effective volume of that chamber and which is so repeatedly retracted from that chamber as to effect a series of momentary increases in the effective volume of said chamber thereby to cause momentary flow of fluid repeatedly to occur out from, and back into, said chamber.

Each momentary flow of fluid out from said chamber, which is towards at least one of said ports, has the effect of at least substantially removing a trough otherwise in the operating characteristics of the pump or motor and each momentary flow of fluid back into said chamber has the effect of at least substantially reducing a peak otherwise in said characteristics. As a result pulsation of flow through the pump or motor is reduced.

The pump or motor may be of gear type in which case at least two of said rotary members may be provided which are in the form of intermeshing gears. Here said cam means preferably has the same number of lobes as there are teeth on each gear and thus each projection of said element into said chamber coincides with a respective trough in said characteristics and each retraction of said element from said chamber coincides with a respective peak in said characteristics.

The advantages offered by the invention are mainly that as a result of the reduced pulsation of flow through the pump or motor, the pump or motor operates at a much lower noise level and, since in consequence wear of the components thereof is reduced, it has a longer life.

One way of carrying out the invention is described in detail below with reference to the accompanying drawings which illustrate only one specific embodiment, in which:-

Figure A.2.1 is a cross-sectional elevation of a gear pump in accordance with the invention, and,

Figure A.2.2 is a cross-section taken along the line II-II in Figure A.2.1

The gear pump 11 shown includes a main casing 12 having overlapping bores 13, 14 which in conventional manner house a pair of rotary members in the form of inter-meshing gears 15, 16, the shafts 17, 18, 19, 20 of which are supported for rotation in bushes 21, 22, 23, 24 respectively of D-shaped cross-section. The flats of the bushes interengage as shown at 25 and 26 and each bush is provided with a liner 27 of suitable longwearing material.

The shaft 17, which forms the driving shaft of the pump, extends to the exterior of the pump through an end cover member 28 suitably secured to the left-hand face in Figure 2.1 of the main casing 12.

A subsidiary casing 29 having a cavity 30 of circular cross-section formed therein is secured to the right-hand face in Figure 2.1 of the main casing 12, and a cover plate 31 is secured to the right-hand face of the subsidiary casing.

In this embodiment the casing 12, cover member 28, casing 29 and cover plate 31 are held in unit assembly by a plurality of bolts, as at 32, which pass through them all, and by nuts as at 33, applied to the bolts.

Cam means in the form of a member 34 of octagonal cross-section at its right-hand-end portion in Figure 2.1 is secured in coaxial manner to the shaft 18, so as to be rotatable therewith, by a bolt 35 suitably provided with locking means (not shown). The member 34 projects into the cavity 30 and its octagonal end portion is engaged by four reciprocable elements 36 which are radially slidable in respective bores 37 provided in an annular insert 38 suitably non-rotatably located in the cavity 30 coaxially with respect to the shaft 18 and member 34. the elements 36 and the insert 38 form a fluid displacement device.

The bores 37 and thus the elements 36 are equispaced circumferentially of the insert 38. The end portions of the elements 36 which engage the octagonal cam formation of the member 34 are of semi-circular cross-section as shown in Figure2.2 and the outer end portion of each of these elements has a groove 39 also of semi-circular cross-section. An endless spring ring 40 surrounds the insert 38 and seats in the grooves 39 thus to bias the elements 36 into engagement with the cam formation of the member 34.

The annular portion 41 of the cavity 30 surrounding the insert 38 forms a chamber which is in communication by way of a passage (not shown) in the pump casing structure with the outlet or delivery port (also not shown) of the pump.

The annular clearance 42 between the member 34 and the insert 38 is open to a low pressure region of the pump which in turn is in communication with the inlet port (also not shown) of the pump.

Pressure-balancing means of known form are embodied in the pump and are generally indicated at 43 and 44 in Figure2.1.

In this embodiment the gears each have eight teeth.

During operation of the gear pump 11 the shaft 17 is driven by suitable means (not shown), for example an electric motor, and liquid is drawn in through the inlet port and is elevated in pressure by the action of the rotating intermeshing gears 15, 16. Liquid under high pressure is discharged from the delivery port of the pump to a point of usage, and during operation the pressure-balancing means 43 and 44 hold the bushes 21, 22, 23, 24 in adequate sealing contact with the faces of the gears, the bushes being thereby so hydraulically balanced that undue wear thereof is avoided.

During pumping operation, since the member 34 is being driven from the shaft 18 of gear 15 the four elements 36 are reciprocating in their bores 37. Since the cam formation of the member 34 has eight lobes, each of the elements is reciprocated in its bore eight times for each revolution of the shaft 18 and gear 15. Thus the elements all move outwardly simultaneously against the spring ring 40 thereby projecting into the annular portion or chamber 41 of the cavity 30 and then move back into their bores to the positions shown in the drawings in which their radially-outer end faces are flush with the exterior surface of the insert 38. In simultaneously moving in the radially-outward direction from a mean stroke position these elements together effect such momentary decrease in effective volume of the annular chamber as to cause a momentary flow of fluid from the chamber into the delivery port of the pump to occur thereby increasing overall flow. Such momentary flow in this direction occurs eight times per revolution of the gears.

Further, the elements 36, in moving also, and simultaneously, in the radially-inward direction from the mean stroke position to the position in which their radially-outer end faces are each flush with the exterior surface of the insert 38, together effect such momentary increase in effective volume of the annular chamber 41 that a momentary flow of fluid from the delivery port of the pump into the chamber occurs thereby reducing overall flow. Such momentary flow in this direction also occurs eight times per revolution of the gears.

Considering conventional gear pump operation, that is operation of a gear pump without a cam member 34 and without elements 36, the delivery flow characteristics are such, evident when graphically plotted against gear rotation, that a pulsating or "ripple" effect is produced due to the occurrence of a series of flow peaks and a series of flow troughs in the resultant performance curve. These peaks and troughs result from the successive meshing and unmeshing of the co-operating teeth of the gears, and with eight teeth on each gear a series of eight peaks and a series of eight troughs are graphically evident for each revolution of the gears.

It has been proven in practice that the pulsating or "ripple" effect produced by such peaks and troughs causes the pump to be noisy in operation.

However, as explained above, the cam member 34 and elements 36 included in the construction of the present invention provide means for momentarily increasing the flow of liquid passing into the delivery port of the pump and for momentarily decreasing the flow of liquid passing into the delivery port of the pump, in this embodiment each eight times during one revolution of the gears. The cam means is so arranged that such momentary increases are caused to occur at the eight positions in the one revolution cycle at which the said troughs would otherwise occur, and such momentary decreases are caused to occur at the eight positions in the one revolution cycle at which the said peaks would otherwise occur. Since these momentary increases and decreases in flow, within the pump, into the delivery port have the effect of avoiding, to a substantial extent at least, the setting up of such peaks and troughs, the pulsating or "ripple" effect in the pump delivery flow is substantially reduced and a significant reduction in operational noise level of the pump is obtained.

During operation of the pump the elements 36 when reciprocating also simultaneously project radially-inwardly into, and move radially-ourtwardly from, the annular clearance 42 which is in communication with the low pressure port of the pump. The flow characteristics, which to a certain extent reflect those on the delivery side of the pump, are therefore modified in a manner likewise to that on the high pressure side and this contributes to the reduction in operational noise level of the pump.

Although in the embodiment above-described with reference to the drawings the four said elements 36 are provided in association with the cam member 34, in other embodiments any other suitable number of elements may instead be provided. Further, the invention is not limited to the gears each having eight teeth as in other embodiments they are provided with any other suitable number of teeth provided the cam member has a similar number of lobes whereby the said momentary flows of fluid into, and from, the delivery port respectively coincide with the said troughs and peaks during pump operation.

The invention is not limited to the said elements being radially-disposed, as in other embodiments they may be axially disposed and in this case the cam means be suitably designed to co-operate therewith.

Again, although in the embodiment above-described with reference to the drawings the invention is applied to a gear pump, in alternative embodiments the invention may with advantage be applied to gear motors.

Finally, although in the embodiment above-described with reference to the drawings the invention is applied to a pump of gear type, in alternative embodiments the invention may with advantage be applied to pumps, and also motors, of other type, for example of axial-piston type, where flow pulsation problems with attendant noise in operation otherwise occur.

CLAIMS

1) A pump or motor comprising a casing, which houses at least one rotary member and which has an inlet port and an outlet port, cam means connected to be rotated by said member and a fluid displacement device which is driven by said cam means when rotating and which includes at least one reciprocable element which, upon rotation of said rotary member through each revolution, so repeatedly projects into a chamber in said casing which is in communication with one of said ports as to effect a series of momentary decreases in the effective volume of that chamber and which is so repeatedly retracted from that chamber as to effect a series of momentary increases in the effective volume of said chamber thereby to cause momentary flow of fluid repeatedly to occur out from, and back into, said chamber.

2) A pump or motor as claimed in claim 1 and of gear type, wherein at least two of said rotary members are provided and are in the form of intermeshing gears.

3) A pump or motor as claimed in claim 2, wherein said cam means has the same number of lobes as there are teeth on each gear.

4) A pump or motor as claimed in any one of the preceding claims, wherein said chamber is formed by the annular outer portion of a cavity of circular cross-section formed in the casing.

5) A pump or motor as claimed in claim 4, wherein an annular insert, surrounded by said chamber, is non-rotatably located in said cavity and said cam means is disposed in the central opening of said insert.

6) A pump or motor as claimed in claim 5, wherein the or each said reciprocable element is radially slidable in a respective bore or the like formed in said insert and the inner end portion of the or each said element is directly engaged by said cam means.

7) A pump or motor as claimed in claim 6, wherein said inner end portion of the or each said element is of semi-circular cross-section.

8) A pump or motor as claimed in either claim 6 or claim 7, wherein the outer end portion of the or each said element is provided with a groove or semi-circular cross-section.

9) A pump or motor as claimed in claim 8, wherein a spring ring surrounds said insert and seats in said groove or grooves of said semi-circular cross-section.

10) A pump or motor as claimed in any one of the preceding claims, wherein said chamber is in communication with that one of said inlet port and said outlet port which is at high pressure.

11) A pump or motor as claimed in any one of claims 5 to 10, wherein a generally annular clearance is formed between said cam means and said insert, said clearance being in communication with that one of said inlet port and said outlet port which is at low pressure.

12) A pump or motor substantially as herein before described with reference to the accompanying drawings.

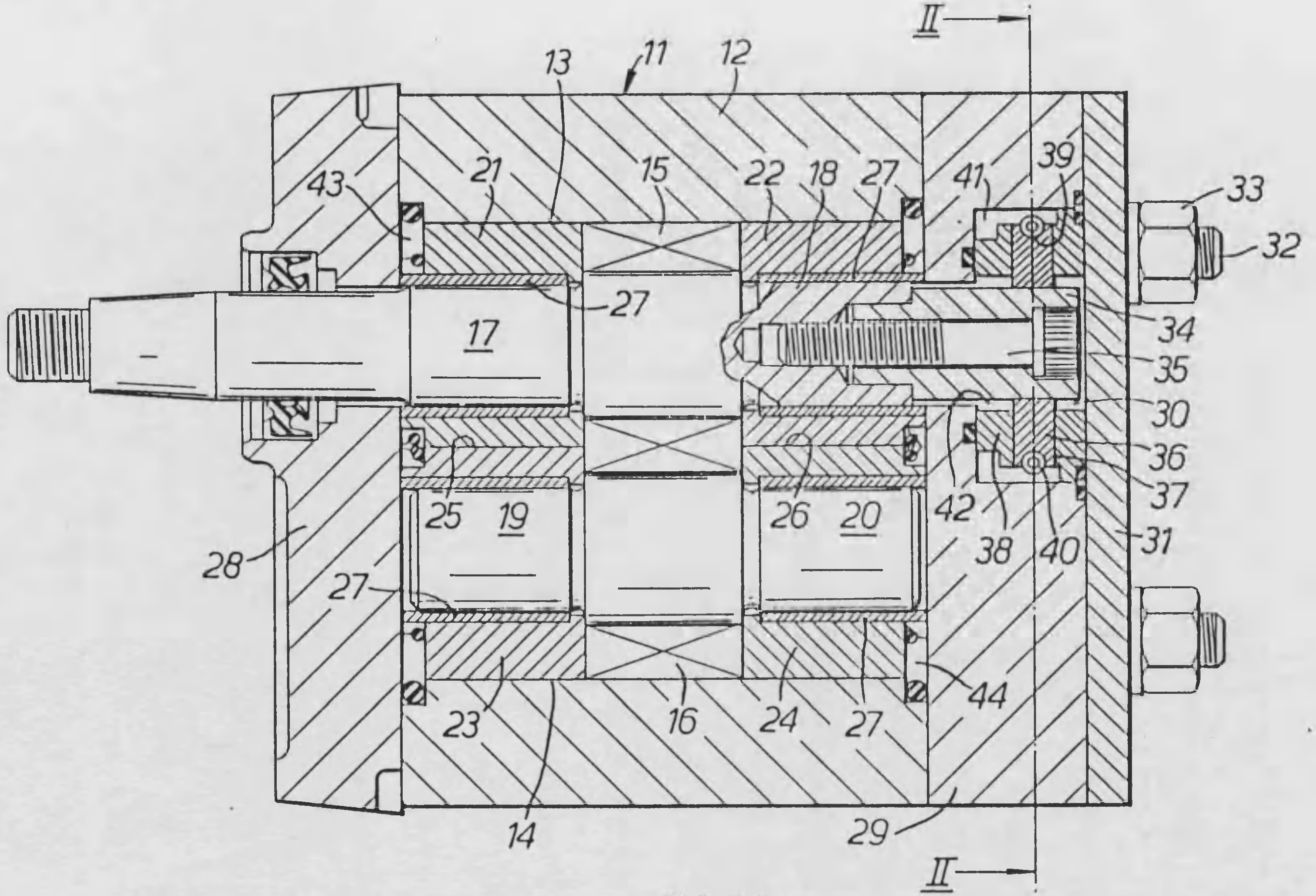

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FIG. A 2.1

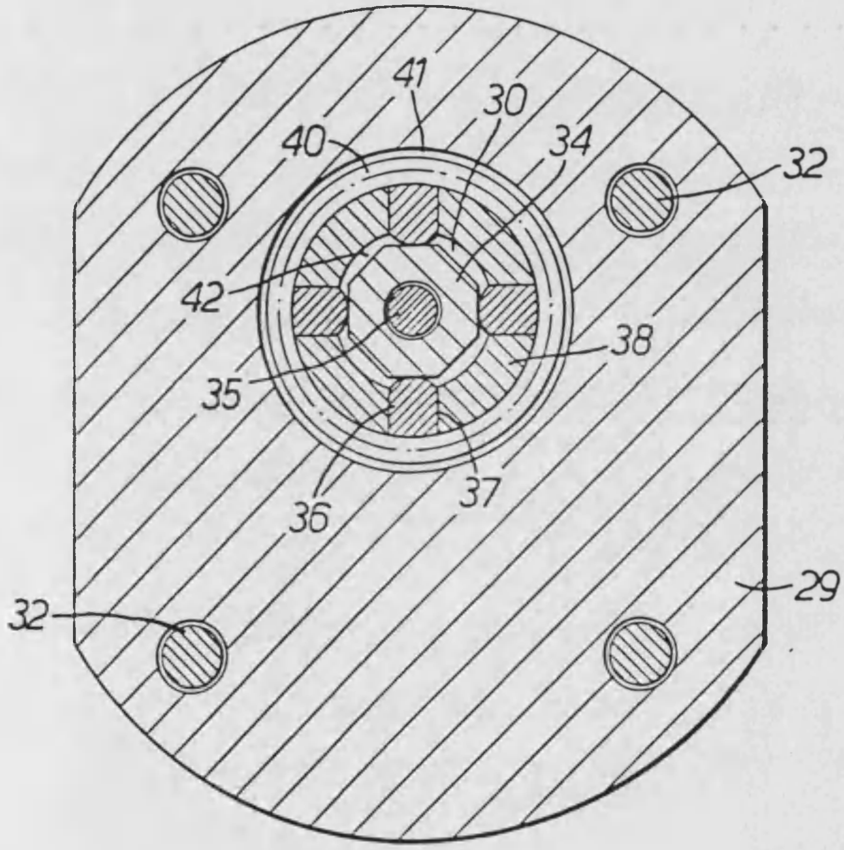


FIG. A 2.2

A.3 - DESCRIPTION OF EXPERIMENTAL HARDWARE

A.3.1 - Hydraulic Gear Units

Pumps A, B and C.

These units were Dowty external gear pumps modified, as described in chapter 6, to allow operation with or without a flow ripple modifying device. These three examples of the same pump type had the following standard characteristics:-

type	1P3060
theoretical displacement	19.21 cm^3/rev
maximum working pressure	207 bar
maximum rotational speed	3500 rev/min.
number of involute gear teeth	8
gear pressure angle	30 degree
gear width	0.02423 m.
Pitch circle diameter	0.03175 m.

Pump D and motor E

These two units were also modified Dowty pumps. The motor E was, in fact, a pump design run as a motor. Unfortunately, due to the bush balancing, the unit would not start without a sudden increase in input flow. Once started however it operated satisfactorily. The basic characteristics are as follows:-

type	1P3028
theoretical displacement	9.47 cm^3/rev
maximum working pressure	207 bar

maximum rotational speed	3500 rev/min.
number of involute gear teeth	8
gear pressure angle	30 degree
gear width	0.01188 m.
Pitch circle diameter	0.03175 m.

Details of the installation and construction of the standard 1P3000 range of pumps are shown in figs. A.3.1 and A.3.2 respectively.

Pump F

This external gear pump was manufactured by Hamworthy and had the following characteristics:-

type	PA2113
theoretical displacement	57 cm^3/rev
maximum working pressure	172 bar
maximum working speed	2500 rev/min.
number of involute gear teeth	10
gear pressure angle	30 degree
gear width	0.03302 m.
Pitch circle diameter	0.05148 m.

Pump G

This internal involute gear pump, manufactured by Voith (under licence from Eckerle), was not tested but was dismantled and inspected. The data obtained was then used to predict flow and volume variations using a model of internal involute gear geometry. These variations were compared with an external gear unit of similar displacement.

The principal characteristics of the unit are as follows:-

type	1PH 3-16-101
theoretical displacement	15.8 cm^3/rev
maximum working pressure (continuous)	250 bar
maximum working speed	3000 rev/min.
number of involute gear teeth on pinion	13
number of teeth on wheel	20
gear pressure angle	30 degree
gear width	0.031 m.
Pitch circle diameter	0.0356 m.

A.3.2 - Instrumentation

Dynamic Pressure Transducer.

The transducers used for the measurement of pressure fluctuations were of the piezoelectric type manufactured by Vibrometer Ltd. and had the following characteristics:-

type	6QP500
maximum working pressure	470 bar
maximum working temperature	240 C
mass	negligible
linearity	$\ll \pm 1\%$
natural frequency	67 KHz
damping	0.3
insulation resistance	2 kΩ

These transducers were small with a 6 x 0.5 mm installation thread and a sensitivity of approximately 7 pcoulomb/bar.

Digital spectrum analyser.

The spectrum analyser used was a two channel Hewlett Packard instrument capable of performing a Fast Fourier Transform on two signals simultaneously. The amplitude and phase specta of the analysed signal could be displayed over a selected frequency range, stored, and then plotted using an x-y plotter which was interfaced with the instrument. The spectrum analyser had the following characteristics:-

type	3582A
frequency range	0.02 Hz to 25 KHz
choice of amplitude display	10 dB scale 2 dB scale linear scale
amplitude display	256 points (one channel)
amplitude range	+30 to -50dB (rel. to 1V) 30V rms to 3 mV

During experimental work the spectrum analyser was used to examine the pumping frequency components of signals from the following tranducers:- pressure, torque, accelerometer and microphone. The ability to examine the frequency content of two different parameters, such as fluidborne and airborne noise, simultaneously was particularly useful.

Storage oscilloscope.

This instrument was a gould digital storage oscilloscope with the following specifications:-

type	OS4000
time base range	1×10^{-6} sec to 20 sec/cm
input range sensitivity	5mV/cm to 20V/cm
maximum input voltage	400V DC or peak AC

screen resolution:

single trace	100 points/cm
double trace	50 points/cm
digital storage size	1024x8 bits
plotter output (OS4001)	100mV/cm of screen

This instrument was also interfaced to the x-y plotter and was used to record time domain pressure and torque signals.

Frequency Response Analyser (F.R.A.)

The F.R.A. used, a solartron 1170, was used primarily to perform Fourier analysis on the signals from pressure transducers. The basic instrument had been modified so that analysis could be carried out up to the tenth harmonic of an input reference signal. (This reference signal was a square wave produced by conditioning the output of a magnetic pulse generator attached to the pump drive shaft.) Up to six separate channels could be selected manually or remotely via a digital interface (solartron 1181) driven by a DEC PDP8/E computer. Computer control of the F.R.A. enabled fast, automatic data acquisition.

The output from the F.R.A., which is a complex number, could be displayed in any of the following forms:-

(i) Cartesian coordinates (a, b)

(ii) Polar coordinates (A, θ)

(iii) log polar (db, θ)

The logarithmic scale (dB) is defined relative to a 10μV reference voltage as follows:-

$$amp(dB) = 20 \log_{10} (amp(volts) \times 10^5)$$

The phase result displayed corresponds to the phase lag of the evaluated harmonic component relative to the square wave reference signal.

Torque Transducer.

The torque transducer used was an inductive device manufactured by Vibro-Meter SA. Connection to the rotating elements was via rotary transformers (instead of slip rings) and the transducer was capable of measuring both static and dynamic torques. The device had the following characteristics:-

type	TG-2/BP
rated torque	20 Nm
maximum speed	22000 rev/min
moment of inertia	0.0004 Kgm^2
elasticity	$5x10^3$ Nm/rad
sensitivity (of rated torque)	3-6 mV/V
linearity (of rated torque)	0.25-0.5%
supply (via Vibro-Meter torque meter type 8-CT-1/A)	10V/8KHz
temperature range	-10^oC to $+70^oC$

Sound Level Meter.

The sound level meter used was a Brüel and Kjaer instrument type 2204 having a ½" condenser microphone. A number of accessories were also used:-

octave filter set 1613

sound level calibrator 4230

integrator ZR0020 giving displacement, velocity and acceleration

Accelerometer (piezoelectric compression type)

A.3.3 - MULTI-LIFT CAM MECHANISMS - PRINCIPAL DIMENSIONS

(i) MK I mechanism

minimum cam radius	8.75 mm
cam blend radius	0.825 mm
piston-follower diameter	5.89 mm
piston-follower radius	3.50 mm
piston-follower contact length	5.89 mm
number of piston-followers	4
number of cam sides	8

(ii) MKII mechanism

minimum cam radius	11.54 mm
cam blend radius	2.54 mm
piston-follower diameter	4.24 mm
piston-follower radius	3.50 mm
piston-follower contact length	8.89 mm
number of piston-followers	8
number of cam sides	8

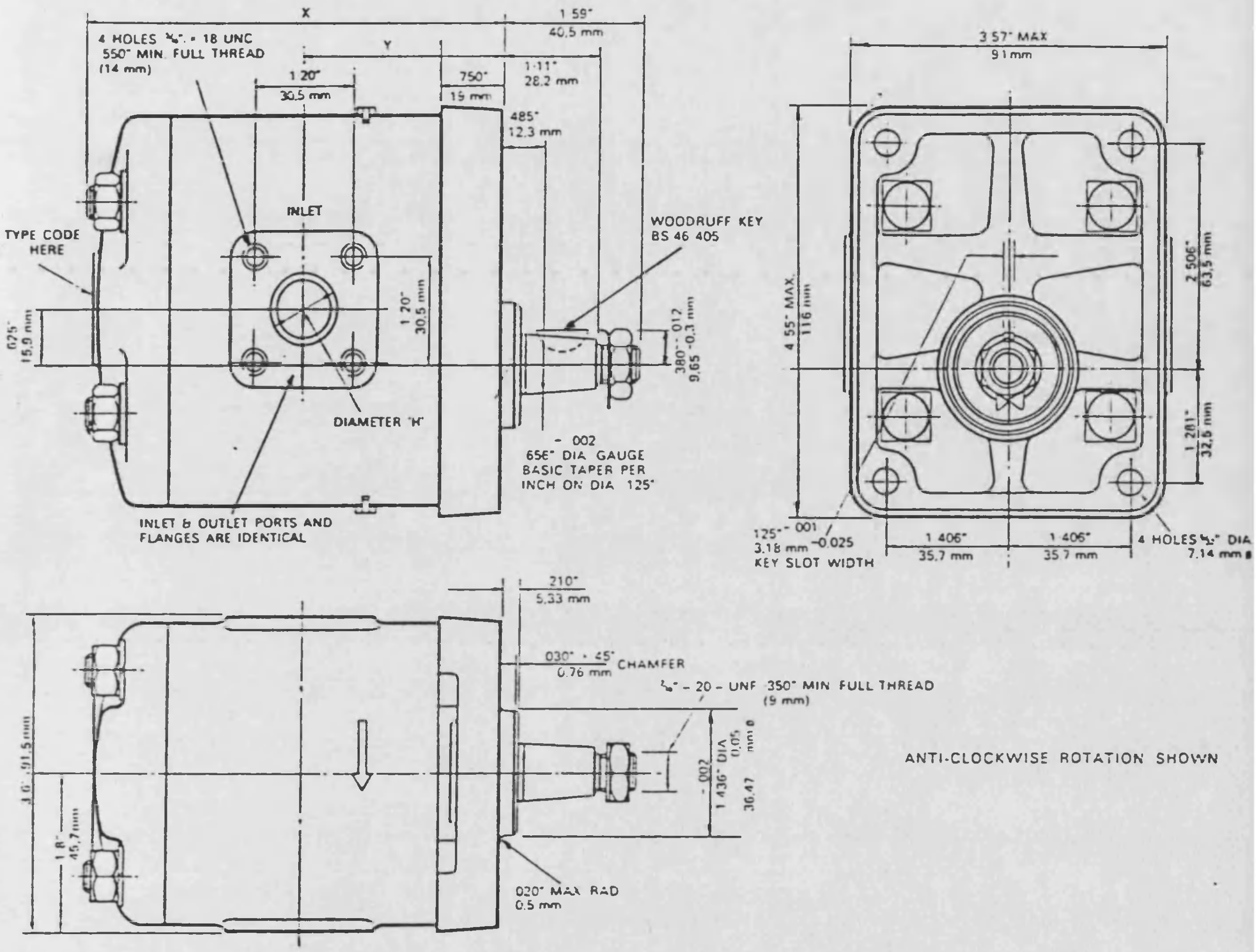
(iii) MK III mechanism

mean cam radius	12.2 mm
piston-follower diameter	4.24 mm
piston-follower radius	3.50 mm
piston-follower contact length	8.89 mm
number of piston-followers	8

DOWTY

GROUP
1P

P3000 SERIES GEAR PUMPS



NOTE: The variable dimensions X, Y and H are tabulated below. All other dimensions are common to group 1 pumps. Design improvements may lead to dimensional changes and these figures should be confirmed before use.

Pump Type	Nominal Flow at 1500 rev/min		Maximum Pressure		Max Speed at Max Pressure rev/min	Min Speed at Max Continuous Pressure rev/min	Dimension X -0.050 in (1,27 mm)		Dimension Y -0.030 in (0,76 mm)		Dimension H +0.020 in (0,5 mm)	
	Imp gal/min	Litres min	lbf/in²	kgf/cm²			in	mm	in	mm	in	mm
IP 3020	2.0	9,1	3000	210	3500	500	3.58	91,0	1.69	43,0	0.68	17,3
IP 3028	2.8	12,7	3000	210	3500	500	3.71	94,3	1.75	44,5	0.68	17,3
IP 3036	3.6	16,4	3000	210	3500	600	3.83	97,3	1.81	46,0	0.68	17,3
IP 3044	4.4	20,0	3000	210	3500	500	4.54	115,3	2.16	55,0	0.80	20,3
IP 3052	5.2	23,6	3000	210	3500	600	4.66	118,4	2.23	56,6	0.80	20,3
IP 3060	6.0	27,3	3000	210	3500	650	4.78	121,4	2.29	58,2	0.80	20,3
IP 3072	7.2	32,7	2525	177	3000	650	4.96	126,0	2.38	60,5	0.80	20,3

Pressures quoted are relief valve maximum by-pass pressures obtained using SAE 20 oil at 50°C.

Average weight, without connectors, 6 lb (2,7 kg)

FIG. A3.1 MANUFACTURERS INSTALLATION AND DISPLACEMENT DATA FOR IP3000 RANGE

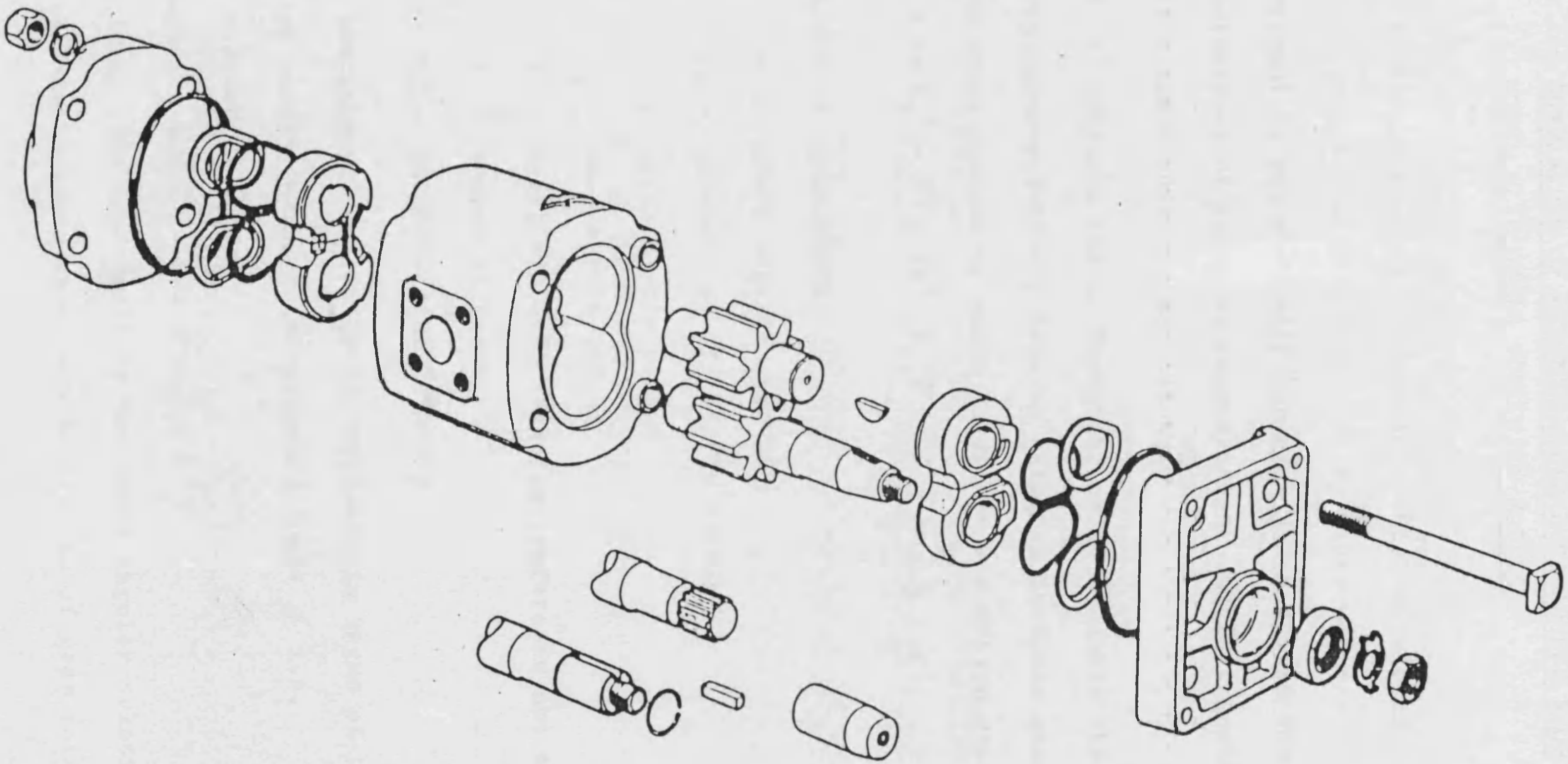


FIG. A3.2 P. 3000 GENERAL CONSTRUCTION

A.4 - COMPARISON OF EXTERNAL GEAR PUMP SOURCE FLOW MODEL WITH OTHER PUBLISHED MODELS

The external gear pump source flow model developed in 3.3.1 agrees entirely with models used by other workers. The agreement is not obviously apparent and requires the substitution of inherent properties of involute geometry to verify their equality as follows:-

(i) T. Ichikawa and K. Yamaguchi ref.(30) quote the instantaneous delivery from an external involute gear pump with equal numbers of teeth on driver and driven gears as:-

$$Qi = bw(R_T^2 - R^2 - Rg^2 \quad \theta^2) \quad \text{for} \ -\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$$

where b = gear width

w = shaft angular velocity

R_T = addendum or tip circle radius

R = pitch circle radius

R_g = base circle radius

θ = angle of tooth rotation (reference not stated)

Z = number of teeth

Qi = instantaneous delivery

If the base circle radius is expressed in terms of the pitch circle radius R and pressure angle ϕ i.e.

$Rg = R\cos\phi$

then $Qi = bw(R_T^2 - R - R^2\cos^2\phi \ \theta^2)$

dividing this expression by the shaft angular velocity w gives the instantaneous delivery/radian of gear rotation which is identical to that shown in 3.3.1.

(ii) E.M. Yudin ref. (28) gives the instantaneous output as

$$Qi = bw(R_T^2 - R^2 - x^2)$$

where x = distance of mesh point along pitch line from pitch point

and all other parameters are as previously defined substitution for the distance of the point of mesh along its pitch line in terms of pitch circle radius, pressure angle and rotational angle from the start of active profile on the base circle i.e.

$x = R.\cos\phi.\theta$ gives

$$Qi = bw(R_T^2 - R^2 - R^2\cos^2\phi.\theta^2) \text{ as before.}$$

(iii) K.T. Duke and P. Dransfield ref. (29) use the following equation to represent the fluctuating component of instantaneous output flow

$$Qs = \frac{wbm^2\cos^2\phi}{12}\left[\pi^2 - 3Z^2\theta^2\right] \quad \text{for } -\frac{\pi}{Z} \leqslant \theta \leqslant \frac{\pi}{Z}$$

where m = module of gears = $\frac{2R}{Z}$

ϕ = pressure angle

other parameters as previously defined.

This is shown in section 3.3.1 to be identical to the expression for Qs derived from Qi developed therein and is therefore synonymous with the models of refs. (30 and 28) also.