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ACCUMULATORS
AND
ITS APPLICATIONS

TRAN THIEU HANH

A TECHNICAL REPORT
in
The Department
of
Mechanical Engineering

Presented in Partial Fulfillment of the Requirements
for the Degree of Master of Engineering at
Concordia University
Montreal, Quebec, Canada

September, 1977

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ABSTRACT

ACCUMULATORS AND ITS APPLICATIONS

Tran Thieu Hanh

Theoretical and practical aspects of hydraulic accumulators have been developed through this technical report. Selected materials and dimensional data from various handbooks and papers have been gathered to give the identity and value of accumulators.

Hydraulic accumulators are devices to store fluid under pressure. This pressure is exerted by either a dead weight a compressed spring or compressed gas. We therefore consider weight-loaded accumulators, spring-loaded accumulators gas-loaded non separated accumulators and gas-loaded separated accumulators.

An experimental and theoretical analysis of the effect of frequency on the polytropic exponent in a gas charged piston type accumulator are presented. Relying on Daniels' theories concerning the theoretical value for the polytropic exponent as a function of the signal frequency, an initial series of tests on an unmodified piston type accumulator is made, and show good agreement between them. Also, an attempt is made to obtain isothermal conditions within the accumulator through the introduction of copper mesh in the cylinder chamber. Tests are then repeated and all experimental results are obtained with an oil layer on the cylinder wall. We observed that we are more near the isothermal zone under these circumstances.

Some applications of accumulators are mentioned.

ACKNOWLEDGEMENTS

I wish to acknowledge my gratitude to Dr. S. Katz, my advisor, for his help and guidance in completing my work.

A special word of thanks is also due to Mr. J. Elliott for his assistance in carrying out the experimental phase of my work in the Fluid Control Center.

I also wish to extend my thanks to the willing cooperation and assistance received from many individuals, Thereby, many thanks are dedicated to Mrs. Nicole Trudeau for her kindness by typing the content.

NOMENCLATURE

A = Surface area of walls and piston

A_e = Effective flow area of pipe

ABE = Area of flask of the accumulator

AUD = Area of outlet of accumulator

B = Bore of pump

C = Constant

C_v = Constant specific heat

DPA(I) = Differential pressure in psia

F = Frequency radius parameter

F_1 = Constant for a particular type of pump

F_2 = Constant for a particular type of pump

g = Acceleration of gravity

h = Heat transfer coefficient

J_o = Bessel function of the first kind of zeroth order

J_1 = Bessel function of the first kind of first order

K = Force

L = Length of fluid chamber

LUD = Length at outlet of accumulator

m = Mass

N = Rotation speed of pump

P = Gas pressure

PE = Potention energy

PF' = Fluid pressure at outlet of accumulator

PF" = Fluid pressure at inlet of accumulator

PFBL = Fluid pressure at load line

P_g = Gas pressure

P_m = Maximum surge pressure

P_s = System pressure, or maximum accumulator charging pressure

P_x = Precharge pressure

P = Minimum working pressure

P_1 = Initial system pressure and accumulator precharge pressure, psia

P_2 = Maximum allowable surge pressure

P_3 = Minimum system pressure at which additional volume of fluid is needed, psi

$P_1(I)$ = Pressure at the lower point

$P_2(I)$ = Pressure at the higher point

PIA = Absolute pressure

PIW = Pressure in inch of water

P_v = Fluid pressure in pipe at valve, psi

Q = Fluid flow rate

q = Rate of heat transfer between the gas and the surrounding

R = Rate of flow

R_c = Constant

τ = Thermal time constant

T = Average gas temperature

T_o = Tempurature of the surroundings

T_1 = Temperature at time t

Δt = Change in time

U = Velocity

V = Volume

$\dot{V}$ = Rate of change of gas volume

dV = Change in volume

V_f = Fluid volume

V_1 = Accumulator capacity

V_2 = Compressed volume of gas at minimum pressure

V_3 = Expanded volume of gas at minimum system pressure

V_x = Volume of fluid discharged from accumulator, accumulator capacity

V_t = Accumulator total volume, or size

v = Velocity of flow in pipe before valve closure

ω = Specific weight of fluid

ρ = RHO mass density of fluid

Z_n = Polytropic exponent

TABLE OF CONTENTS

1. INTRODUCTION

Accumulators, are essentially vessels for storing hydraulic fluid under pressure. They were originally developed as a mean of providing the total system requirements (e.g. with water hydraulics) when pumps were slow running and of relatively low efficiency. Although still employed in this manner (e.g. to operate a batch of machines on a water hydraulic system from a single pumping station), their main use today is to supply peak demands of power. This offers considerable economy in pumping since a smaller pump may be used to charge the accumulator with fluid at a low rate, compared with the demand which would be required from a directly coupled pump. An accumulator also possesses the inherent capacity of holding a high pressure in a system for a given period of time, even without recharging. It is also capable of giving a smooth, even flow on demand at an appreciably constant pressure.

Other useful characteristics of an accumulator are:

1. its ability to attenuate shock pressures in a system
2. its ability to act as an emergency source of power in the event of pump failure
3. its suitability as a holding device for fluid make up, etc., and for the synchronisation of movements

4. It may be introduced into some systems normally supplied by a directly coupled pump purely for shock absorption
5. it may be employed as the power source in a "pumpless" emergency system.

2. THEORY ON ACCUMULATORS

Accumulators may be classified according to type and construction.

1. Weight-loaded accumulators
   a) simple
   b) differential
2. Spring-loaded accumulators
3. Gas-loaded non separated accumulators
4. Gas-loaded separated accumulators
   a) diaphragm separator
   b) bag separator
   c) free piston separator

2.1. WEIGHT-LOADED ACCUMULATORS [1] Figures 1 and 2

The weight-loaded accumulator is inherently bulky and is normally erected outdoors to act as the central station feeding a number of separate machines or hydraulic circuits, or heavy industrial requirements. Particular advantages offered are

(i) the extremely high capacity possible
(ii) the fact that it can be erected anywhere
  (e.g. out in the open on spare ground)
(iii) relatively low cost per volume

The basic design consists of a fixed, vertical cylinder into which fits a ram (or a fixed vertical ram surrounded by a sliding vertical cylinder). A crosshead attached to the upper part of the ram (surrounded) carries a weight case which is loaded with ballast-concrete slab weights or cast iron weights in the smaller sizes, or slag or similar ballast in larger sizes. The total weight of the ballast plus the ram assembly then effectively acts on the cross sectional area of the ram bearing on the fluid contained in the cylinder.

The fluid pressure available is thus determinable by the formula

$$\text{Pressure} = \frac{\text{loaded weight of moving assembly}}{\text{cross sectional area of ram}}$$

The capacity or ultimate working fluid volume available is determined by the cylinder bore and length of stroke. It is normal to use a long stroke since this produces more efficient operation. A stroke of between 10 and 15 times the cylinder bore is common practice for weight loaded accumulators working up to 1,500 psi (10,342.5 KPA) and higher stroke bore ratios for higher pressures. The total headroom required is over twice the stroke.

The two main type of weight-loaded accumulators are the self-guided and externally guided, referring to the arrangement of constraining the weight case. In the self-guided type the weight case is provided with internal guides. On externally guided types external guide channels are mounted around the weight case to constraint any lateral movement during its vertical displacement. These guide channels are usually mounted on a steel structure, although on earlier types brick or stone housings were often used. Externally guided types are usually preferred for high pressure accumulators to minimize bending stresses.

For working pressures up to 1,500 psi (10,342.5 KPA) cast iron cylinders are normally used but for higher pressures cast steel or forged steel cylinders are preferred. Rams may be of cast iron although on many modern designs are fabricated from steel or alloy steel (stainless), or chrome plated.

The size of the accumulator and the stroke bore ratio should be selected so that at maximum demand the falling speed will not exceed 3 feet per sec (.915 m/S) and should preferably be kept to a lower figure (e.g. not exceeding 1 feet per sec (.305 m/S). This will ensure that there is no excessive hydraulic shock when the weight stop moving.

The practical pressure available is reduced from the theoretical amount through fractional losses at the gland and ram seals. These may be variable with the condition of the packings and the speed of movement but should not be significant if the packings are properly maintained. It is also important that the gland packing be kept in good condition in order to minimize leakage.

The actual pressure will also be variable with ram movement due to the inertia of the system. From a given static pressure the fluid pressure will tend to drop slightly as the weight starts to move and then increase again above the static pressure as the weight slows to a stop. In general the likely pressure variation can be estimated as 5 per cent maximum (2½ per cent above and below the average) although momentary peak pressures may be lower and higher at the beginning and ending of the stroke, respectively, of a single movement.

DIFFERENTIAL WEIGHT-LOADED ACCUMULATOR [1] Figure 3

The differential weight-loaded accumulator employs a fixed ram with the lower part fitted with a sleeve to increase its diameter. The cylinder is weight-loaded and slides over the ram. The difference is diameters D and d being quite small, very high pressures can be produced for a comparatively small loaded weight. The volume of fluid which can be accomodated is strictly limited. Hence this form of weight-loaded accumulator is essentially a high-pressure, low-capacity type and has limited application.

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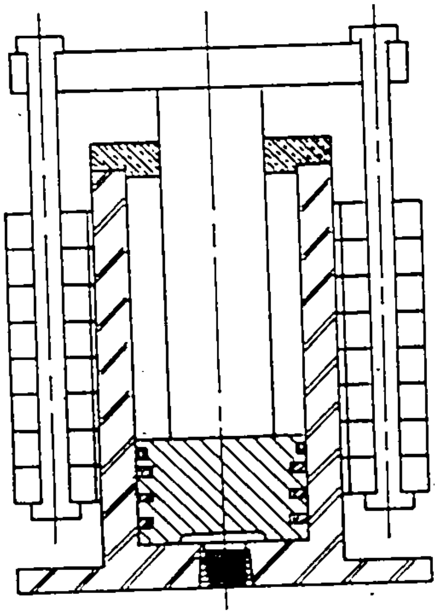


Fig.1. Variable Load

RAM TYPE

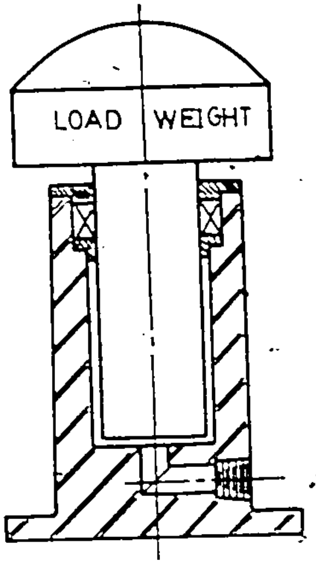
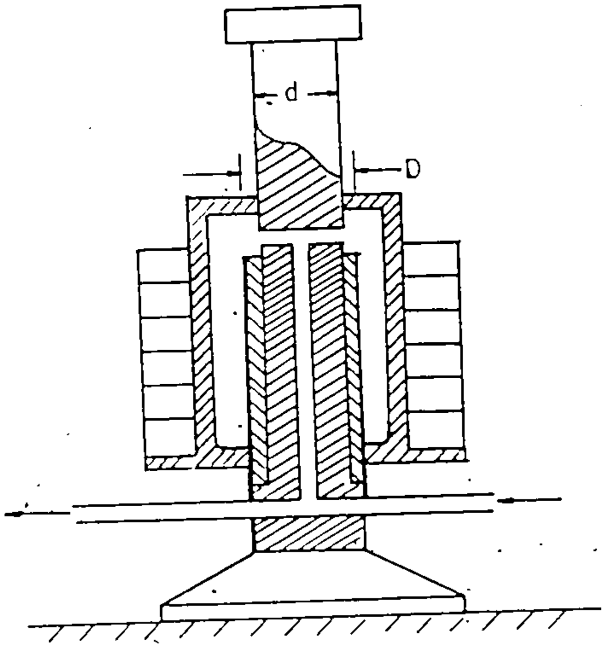


Fig.2. Constant Load



Weight-loaded Accumulator

Fig 3. Differential

2.2 SPRING-LOADED ACCUMULATORS [1]   Figure 4

The spring-loaded accumulator in its basic form consists of a free piston in a cylinder with a compression spring applying loading to the piston as fluid is pumped in via the oil port. The pressure is equal to the intataneous spring force divided by the piston area

$$\text{Pressure} = \frac{\text{spring force}}{\text{area}}$$

Where spring force = spring constant x compression distance

As liquid is pumped into a spring-loaded accumulator, the stored fluid pressure is determined by the spring constant, compression rate of the spring. The design is essentially limited in application to low pressures and small volumes and also suffers from the fact that the pressure loading is not constant due to the varying force of the spring (unless specially constructed springs are employed). Pressure is normally a maximum when the spring is fully compressed, falling to a minimum when fully extended,

To avoid accumulation of leakage fluid, the spring chamber of a spring-loaded accumulator is vented. Leakage fluid will eventually discharge from the vent hole.

Spring-loaded accumulators are not externally drained back to tank because they can cause oil foaming. With an external drain terminating either above or below fluid level, leakage accumulated above the piston will tend to foam during accumulator operation. As the accumulator discharges

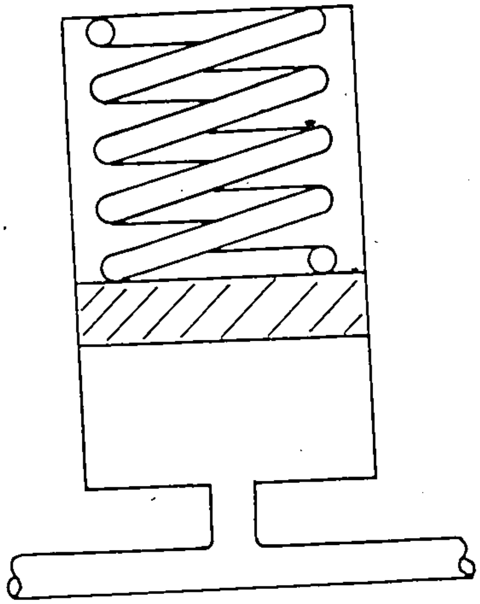


Fig.4. Spring-Loaded Accumulator

Non Separated Gas loaded Accumulator

Fig.5. Pistonless Type

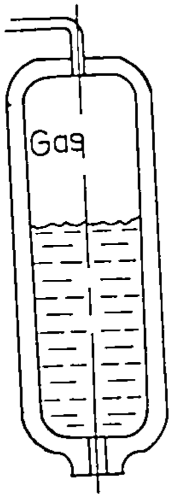
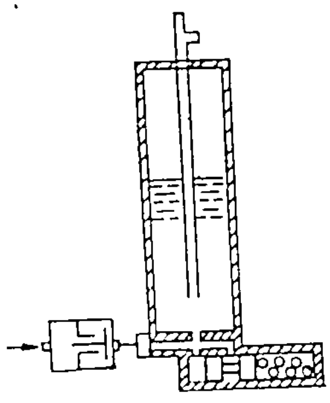


Fig.6. With Pressure Cut Off



rapidely, fluid above the piston will be unable to keep up with piston movement. A less than atmospheric pressure will be generated in the spring chamber resulting in dissolved air coming out of the liquid. When the accumulator is recharged, the piston moves up pushing the aerated oil to tank. Since air bubbles in a reservoir are undesirable, spring-loaded accumulators are not generally externally drained.

W[illegible] spring chamber vented, spring-loaded accumulators demand immediate attention once their piston seal wears. If maintenance is not performed on a spring-loaded accumulator with a poor seal, large oil spills could arise.

The particular advantage of this type of accumulator is that it can be mounted in any position and may be built directly into the power unit.

2.3 GAS CHARGED ACCUMULATORS [1]

Gas charged accumulator use compressed air or nitrogen to force pressurized fluid into the system. These accumulators may be a non-separator type in which the compressed gas acts directly on the pressurized fluid, or a separator type, where some barrier separates the gas from the pressurized fluid. The barrier may be a piston, a bladder, or a diaphragm.

In gas-loaded accumulator,air is used as a spring. The spring constant could be determined from the thermodynamic properties of the air. It is possible to express a relation between the pressure and density of a gas during a polytropic process which is valid for a perfect gas.

In this case

$$\frac{Pa}{\rho^n} = C$$

Where Pa = absolute gas pressure

ρ = gas density

n = constant which depends on the process and how heat is transferred to the cylinder

C = constant

If the gas remains at a constant temperature, then n equals one and the equation is the same as the state equation for a perfect gas. This assumes that the piston moves slowly and all heat generated inside the cylinder is transferred to the outside. If the piston moves quickly in a frictionless cylinder which is well insulated so that no heat flows in or out, n equals 1.4 for air and the process is called isentropic.

The gas pressure and density are related to the cylinder dimensions as follows:

$$AP_a = F$$

Where A is the piston area, F is the force exerted on the piston. The density of the gas in the cylinder can be related to cylinder dimensions

$$\rho = \frac{M}{AL}$$

Where M is the mass of gas inside the cylinder. Since no leakage is assumed, M is a constant. L is the lenght of the gas column and A is its area.

In order to establish the relationship between the change in gas column length ΔL and force F, it is necessary to find an expression for ΔL . This can be done by substituting the relation for pressure and density into the process equation and differentiating. Hence:

$$\frac{F/A}{(M/AL)^n} = C$$

Differentiating this expression yields

$$\frac{dF/A}{(M/AL)^n} + n \left(\frac{F}{A}\right) \left(\frac{A}{M}\right)^n L^{n-1}dl = 0$$

Then

$$dF = \frac{An(F/A) - dl}{L}$$

F can be replaced by P_SA

If dF is identified as ΔF and considering small changes in L, dl = $-\Delta L$. So, $\Delta F = An\ (P_SA/A)\ x\ \ \Delta L/L$

In this case, the spring constant ($\Delta F/\Delta L$) is

$$K = \frac{AnP_s}{L}$$

If the air spring is uninsulated and moves slowly, the temperature will remain constant so that n = 1.

Then

$$K = \frac{A\ P_s}{L}$$

For isentropic case, n = 1.4, the value of K will be

$$K = 1.4\ \frac{A\ P_S}{L}$$

NON-SEPARATED GAS-LOADED ACCUMULATORS Figures 5 and 6

In non-separated types the gas acts directly on the fluid surface. The pressure vessel is invariably tall and narrow and mounted in a vertical position so that the contact area is relatively small. Gas absorption by the fluid is limited to a relatively shollow depth of fluid which acts in the nature of a "boundary layer" to inhibit further penetration of gas into the fluid.

The simplest accumulator of this type thus consists of a pressure vessel partly filled with hydraulic fluid, the space above the fluid being charged with gas at a pressure corresponding to the hydraulic working pressure required. As fluid is withdrawn the gas expands, causing a slight drop in pressure.When fluid is pumped into the vessel the gas is compressed, causing a slight increase in pressure. The percentage variation in pressure is determined by the particular application and is largely controlled by the ratio of the gas volume to the fluid volume in the vessel. At the same time, it will be appreciated that no gas is actually used up during cycling, other than the small amount lost due to absorption.

Where the variations in pressure due to expansion and contraction of the gas are too great, then the gas volume may be increased (and pressure fluctuations thus reduced) by connecting an auxiliary gas bottle to the accumulator. Equally, capacity can be increased by adding further pressure vessels in parallel.

Thus, the type is particularly flexible both as regards capacity and oontrol of pressure variation.

A reserve of liquid volume is essential to maintain a safety margin to prevent any possibility of gas being drawn out of the cylinder into the system. Thus a low-level control device cuts in while there is still a reserve capacity of fluid, and normally before the actual low level point is reached. This allows a time margin for the switch gear to shut down the accumulator, or start the pump and bring it up to maximum output before the design low level point is reached. A stop-valve control is also necessary to prevent complete emptying of the accumulator due to an exceptional heavy demand, or failure of the pump.

Another important feature is that by maintaining a reserve fluid capacity none of gas-rich boundery layer of fluid is ever drawn into the system, thus eliminating the possibility of free gas being evolved at low pressure points in the system. Correct positioning of the fluid draw off point also ensures that it cannot be exposed by a vortex developing in the fluid at low level and thus opening the way for gas exit into the system.

Working pressures vary considerable, according to system requirements and typically range from about 500 psi (3,447.50 KPA) to 6,000 psi (41,370. 00 KPA) in standard industrial productions. Controls are normally fully automatic, together with relevant safety devices, and may also include "monitoring" of the actual fluid level inside the vessel. This can be done quite simply by pressure measurement (the actual pressure of the fluid being related to the actual gas volume); directly by floats (indirectly coupled to external indicators); or electronically. The latter system is usually the most accurate.

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SEPARATED TYPE GAS-LOADED ACCUMULATORS

Diaphragm type accumulators are usually spherical in shape and used for specialized applications e.g. aircraft hydraulic accumulators. Bag-type accumulators have numerous industrial applications, particularly where relatively small capacities are required. The bulk of industrial requirements, however, is usually met by one or other of the piston-type accumulators.

DIAPHRAGM SEPARATOR ACCUMULATOR [20] Figure 7

A diaphragm type accumulator consists of two metal hemispheres which are bolted together, but whose interior volume is separated by a synthetic rubber diaphragm. The diaphragm divides the vessel into two separate compartiments - a gas chamber and a fluid chamber. When the vessel has been precharged with air or gas to the required pressure, the diaphragm is normally fully flexed so that the gas occupies the complete volume of the accumulator. Fluid is then pumped into the accumulator, displacing the diaphragm and compressing the gas. Gas and fluid pressure are always equal, consequently there is no pressure stress induced on the diaphragm.

During normal operation the pump will build up pressure in the accumulator to a predetermined level. At this point the unloading valve will switch the pump delivery directly to the reservoir. Thus the pump operates under pressure only for relatively short periods at a time, i.e., as necessary to recharge the accumulator. A certain minimum volume of fluid

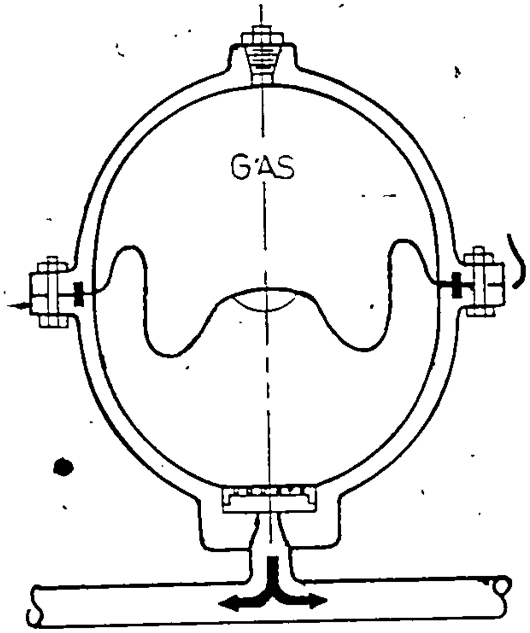

Fig.7. DIAPHRAGM TYPE

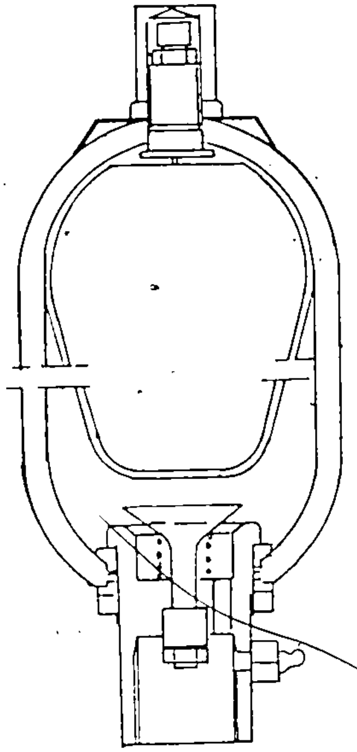

Fig. 8.

BLADDER TYPE

is always maintained in the reservoir but a safety device would normally be incorporated to ensure that in the event of fluid being completely exhausted (e.g. through excessive demand or pump failure) the diaphragm cannot extrude into the fluid outlet and be damaged.

Such an accumulator is best suited to systems where demand is intermittent and fluid volumes required not very large, otherwise the size of accumulator required would become prohibitive. The pump is normally run continuously as noted, with the accumulator providing a useful reserve to meet peak power demands. Isothermal conditions applying where cycling is intermittent, or approaching fully adiabatic conditions when there is varying accumulator operation between given pressure limits.

BAG TYPE ACCUMULATORS[1]

Figure 8

The bag type accumulator consists essentially of a cylindrical pressure vessel with hemispherical ends inside which is damped a flexible bag. The shell is usually made from alloy steel without welds, seams or joints. Bag and shell are assembled by means of a high pressure valve moulded into the bag and clamped with an external nut. The other end of the shell has a relatively large hole into which is assembled the oil part. This joint is normally closed with an o-ring and the design made such that the lower mouth of the shell will spread at a pressure below the design bursting pressure of the shell as a precaution. The wide hole is necessary both to accomodate a large oil port for unrestricted flow, and also to allow the bag to be removed from that end (after removal of the oil port) for inspection or replacement.

Built-in safety factors normally include a provision that the oil discharge plug cannot be removed from the shell until all the pressure has been exhausted from the bag. The oil port is also fitted with a poppet valve to prevent extrusion of the bag through the port should all the fluid be drawn off and at the same time helps to maintain a high volumetric efficiency.

In use the bag is charged with air or gas to the required precharge pressure and fluid then pumped into the vessel to compress the bag. Compression ratios as high as 5:1 may be achieved according to service requirements with nominal working pressures up to 3,000 psi (20,685.00 KPA) in standard sizes, or up to 5,000 psi (34,475.00 KPA) for special duties.

Where additional gas volume is required (i.e. to reduce the pressure variation) this type of accumulator may be connected to an external gas bottle. In certain circumstances, however, this can lead to troubles as an overnight temperature drop could lead to an appreciable fall in gas pressure to the extent that the bag is damaged by crushing.

The bag itself is normally tapered or pearshaped with the largest section at the top, since this gives optimum pressure distribution. The shape of the bag may, however, be dictated by the material used for its construction and most suitable method of prefabrication. Bag failures, with modern designs, are usually produced as the result of improper charging or lack of appreciation of temperature drop when using external bottles, rather than material faults.

Bag-type accumulators are particularly suitable for use with oil fluids, but can also accommodate other types of fluids provided that the bag material is selected on the basis of compatibility. Used with water or water-fluids, it is generally necessary to preheat the steel shell to eliminate corrosion.

A typical treatment in such cases is a stove enamelled interior finish for the shell.

PISTON TYPE ACCUMULATORS[1] Figure 9

The simple piston-type accumulator merely employs a free piston within a cylinder as the separator, the gas charge being introduced into one end of the cylinder and the fluid into the other. The particular advantages of this type are that it is simple, essentially safe and robust and should require little or no maintenance. The question of providing a complete one hundred per cent seal at the piston is not important although a high degree of seal is usually achieved. The seal material can be selected to suit the fluid concerned and thus piston-type accumulator can be used with all types of hydraulic fluids.

Accumulators of this type are normally intended to be mounted vertically with the gas head uppermost, although most can be mounted horizontally, if necessary, with no loss of performance.

Where such an accumulator is working continuously, the accumulator should have an initial inflation pressure (P_1) as near as possible to the lower or cut-in valve of the system working pressure (P_2). This gives a greater swept volume over the working pressure and thus reduces the number of pressure cycles. Where the accumulator is only being used intermittently for peak demands, or as an emergency power source after pressure generation has ceased, lower inflation pressures and consequently higher compression ratios can be used. Compression

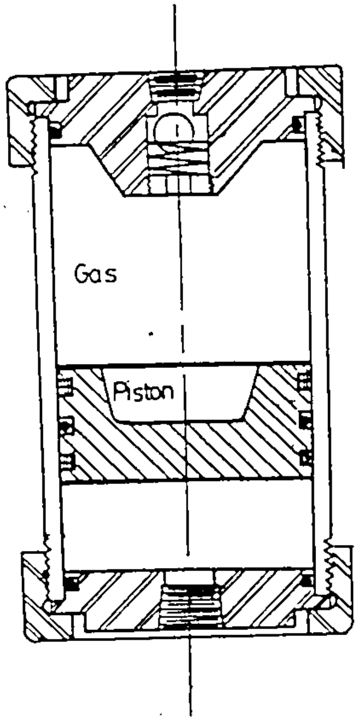

Fig.9. Piston Type

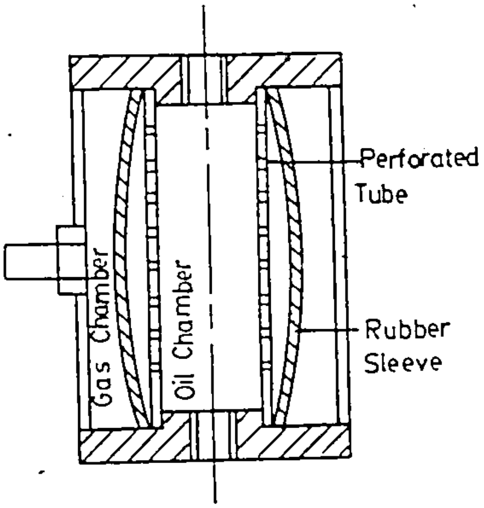

Fig12. Tubular Type

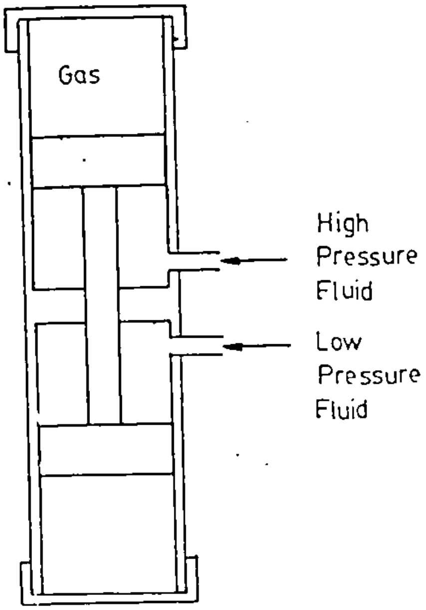

Fig.10. Tandem Piston Type

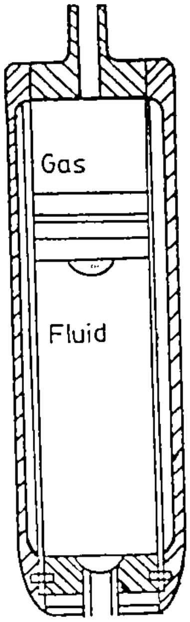

Fig.11. Annular Piston Type

ratios are generally in the range 1.5:1 to 3:1, depending upon the particular application. A ratio of 2:1 being a typical average figure.

The pressure difference over the working range ($P_3 - P_2$) can, of course, be reduced by increasing the gas volume (i.e. by coupling the accumulator to an auxiliary gas bottle). In this case a large size port is usually provided in the gas head, in addition to the charging valve, for connection to a gas bottle.

A simple variation in piston design can be utilised to provide "dashpot" damping at the end of the stroke, thus eliminating hydraulic shock should the liquid content be exhausted. The liquid side of the piston is simply reduced in section to enter a corresponding "damper" section in the liquid head.

An alternative method of increasing the gas volume is to employ an inner steel liner inside the cylinder casing providing an annular gas volume around the outside of the "working" cylinder (Figure 11). The separator is a free piston as before. The liner can be relatively thin since it is pressure balanced (gas pressure on the outside equal to liquid pressure on the inside) and considerable economy in weight is possible for an equivalent liquid capacity and compression ratio. This has the advantage of providing a substantially larger gas volume without increasing the length of the cylinder, or employing gas bottle.

An other variation in piston type accumulator design is the liquid seal piston assembly designed to shut off the oil supply before the piston contacts the end cover and also provide a reservoir of pressurised oil to act as a liquid seal to prevent gas leaking into the hydraulic system when the system is shut down. This takes the form of a "probe" attached to the piston which enters the head at the extreme limit of travel to seal the outlet and trap a quantity of fluid in the space between the piston and the end cover. No gas leakage can take place into the fluid outlet. If any leakage does occur, e.g. due to the deterioration of the piston seal, it would be of fluid into the gas side of the cylinder.

A further variation on the piston-accumulator is a conventional piston-accumulator with the addition of a small diaphragm in the piston. This, it is claimed, overcomes the inertia effects of a normal piston-type accumulator to give it the same instant response to pressure changes as a flexible separator type accumulator.

A tandem type piston accumulator is shown in Figure 10 and is used for special duties. It is also called a self-displacing accumulator and basically comprises an accumulator combined with a pressurized reservoir, and thus being capable of maintaining a constant volume of active fluid in the hydraulic circuit. The gas precharge displaces the tandem piston to fill the low pressure cylinder with liquid. When the system is pressurised, the high pressure (hydraulic) side of the accumulator is filled with liquid and the gas compressed. The liquid to fill the high pressure side is drawn from the low pressure side, thus maintaining a constant volume of liquid both in the system and the accumulator.

TUBULAR ACCUMULATORS [1]

Figure 12

Tubular accumulators are intended primarily to act as pump pulsation dampers or shock absorbers, rather than energy storers. They comprise inner and outer cylinders, the inner cylinder tube being perforated and covered with a rubber sleeve. The unit is precharged with gas in the annular space between the two cylinders, fluid flow being through the inner cylinder, but with access to the inner side of the rubber tube through the perforations. Advantages are compact design, and excellent performance in reducing shock, noise and pulsations.

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2.4 GAS LIQUID ACCUMULATOR PERFORMANCE [4]

As a hydro-pneumatic accumulator develops pressurized flow or maintains pressure, usable volume is an important consideration. While developing a flow, a certain amount of fluid must discharge between two pressures in order for system demands to be met. While maintaining pressure, an accumulator must have available within a certain pressure range sufficient fluid to compensate for leakage. Consequently, three pressures are involved in accumulator operation. Precharging pressure is the pressure of the gas before liquid is introduced into the accumulator. In most commercial accumulators, the gas volume at this pressure is equal to the volumetric rating of the accumulator. Minimum working pressure is the lowest pressure at which the device operated by the accumulator will still work. This pressure could equal precharge pressure if the accumulator ran down completely. Generally this minimum pressure is higher than the precharge pressure, so that some oil will be left in the accumulator after the minimum working pressure is reached. Maximum pressure is determined by the working liquid volume stored in the accumulator. This volume is the sum of piston area times stroke for both forward and return stroke of all cylinders, or hydraulic motor displacements multiplied by the total number of revolutions that the motor must operate. All displacements must be based on the required system energy or torque at the minimum working pressure.

Sizing the accumulator for this working volume is not sufficient because the accumulator should not be emptied during the work stroke. Liquid volume may also be decreased by leakage losses and voids created by the momentum of the liquid at the tank port of the control valve. These losses occur even if the tank port has a counterbalance valve. Altogether a reserve capacity of 2½ to 3 times the demand volume is recommended. In addition, a safety factor of 1½ to 5½ should be applied, depending on the type of accumulator and operating conditions.

For a piston type accumulator in which wear of the piston packing and leakage between oil and gas are minimized, a factor of 1.5 is suggested. Thus the total working volume should be from 4 to 5 times the component displacement. This storage of working volume raises the minimum working pressure.

Maximum working pressure is limited by the pressure rating of the accumulator or of the hydraulic system.

Since hydro-pneumatic accumulators use compressed gas to maintain pressure on a liquid, gas properties affect accumulator operation.

As a hydro-pneumatic accumulator is filled with liquid, gas is compressed. We know that as a gas is compressed it heats up.

"Isothermal" describes the operation of an accumulator as the gas is maintained at a constant temperature. While an

[POOR COPY]

accumulator is being filled, isothermal operation indicates that the gas is being compressed slowly enough for the heat of compression to dissipate.

"Adiabatic" describes the operation of an accumulator as gas temperature changes. While an accumulator is being filled, adiabatic operation indicates that the gas is being compressed rapidly so that all heat of compression is retained.

A hydro-pneumatic accumulator which is being charged with liquid up to a certain pressure, will hold more liquid if it is charged isothermally rather than adiabatically.

As a hydro-pneumatic accumulator discharges liquid, gas expands. We know that as a gas expands it cools. With pressure remaining constant a cooler gas occupies less space than a gas at an elevated temperature.

An accumulator discharging liquid under isothermal condition, indicates that discharge occurs slowly as gas expands; it is capable of acquiring heat from the ambient through accumulator walls or from the fluid. Adiabatic operation indicates that discharging occurs rapidly with no heat gain; as gas expands it cools.

A hydro-pneumatic accumulator which is discharging liquid until a lower pressure is reached, will discharge more liquid if it is discharged isothermally rather than adiabatically.

Hence more fluid enters and exits a hydro-pneumatic accumulator as it is operated isothermally. But, this is usually an ideal situation. Ordinarily, accumulators are charged and discharged either isothermal or polytropic. For processes taking less than one minute use adiabatic and for processes taking more than 3 minutes assume isothermal. For times between one and three minutes, n varies between 1 and 1.4 and it is polytropic; for example 1.1 for low to moderate rates of cycling and some heating effect, 1.3 for rapid cycling where heating effects are very apparent.

The biggest concern is not how much fluid the accumulator holds, but how much fluid is discharged before a lower pressure reached. This is largely effected by gas precharge.

The precharge pressure of an accumulator should be chosen so that most efficient use can be made of the available oil. All of the oil pumped in should be made available as use-able oil for work.

Walter Ernst[11;pt.2] suggests a way to select a most economical size of accumulator. Determination of accumulator size is accomplished with the help of the five following concepts.

1. Determination of Maximum System Pressure p_3

This is the maximum pressure that the system will stand. This pressure should be as high as the accumulator rating, or pump or system rating permit.

Maximum rating of most commercial accumulator is 3,000 psi (20,685.00 KPA), standard pump ratingsare 1,000 psi (6,895.00 KPA), 2,000 psi (13,790.00 KPA) and 3,000 psi (20,685.00 KPA). System rating may be given as in an existing system to which an accumulator is to be added, or it may be determined to fit pump or accumulator rating.

2. Determination of Precharge Pressure p_1

For straight isothermal charge and discharge, the precharge pressure should be made equal to the minimum oil pressure required to do work.

Or for best utilization makes $p_1 = 2/3p_3$. If this is not possible, it should be made as high as feasible, but not more than 3/4 nor less than 1/3 of maximum system pressure.

3. Minimum Pressure p_2

This is the minimum pressure at which the system will operate. This pressure should be made equal to the precharge pressure, but never less than one half of the maximum pressure.

4. Working Volume

The working volume V_w of the accumulator should be four to five times the displacement of the system for one cycle including forward and return stroke. System displacement is based on operation at minimum pressure p_2.

5. Accumulator Size

The required accumulator volume (size rating) for isothermal change of state equals

$$V_1 = \frac{V_w}{1 - p_2/p_3} \quad (p_2/\ p_1)$$

If precharge pressure is equal to minimum working pressure, this simplies to

$$V_1 = \frac{V_w}{1 - p_2/p_3}$$

For polytropic changes of state, the accumulator volume equals:

$$V_1 = \frac{(P_3/P_1)^{1/n} V_w}{(P_3/P_2)^{1/n} - 1}$$

ALPHONSE A. JACOBELLIS [9] observes that usually a system is charged slowly and discharged fast. The optimum precharge to use in this instance may be calculated by

$$P_1 = P_2^{(1 - 1/n)} \times P_3^{1/n}$$

If discharge is pure adiabatic, that is $n = k = 1.4$ then the equation becomes:

$$P_x = P_w^{0.285} \quad P_s^{0.715}$$

Where P_x = Precharge pressure

P_w = Minimum working pressure

P_s = System pressure or maximum accumulator charging pressure, psi.

If compression and expansion are both adiabatic, and expansion immediately follows compression with no appreciable time lag, then the precharge should equal the minimum pressure.

3. ANALYTIC METHODS [8]

3.1 VOLUME CONTROL APPROACH

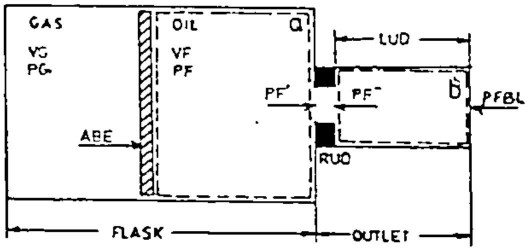


Figure 13

CONVENTIONALIZED RUBBER BAG ACCUMULATOR

1. The control volume a

using the momentum equation to the volume a:

$$Kdt = d\,(m.u) = mdu + udm \quad (1)$$

and substituting

$$u = \frac{Q}{ABE} \quad (2) \rightarrow du = \frac{dQ}{ABE} \quad (3)$$

$$m = \rho \cdot V_f \quad (4)$$

$$Q = \frac{dV}{dt}$$

$$dm = \rho \cdot \frac{dV}{dt} = \rho \cdot Q \cdot dt \quad (5)$$

Then

$$K = \frac{\rho . V_f}{ABE} \times \frac{dQ}{dt} + \frac{\rho}{ABE} Q^2 \quad (6)$$

The net force K acting on the fluid volume a is supposed to arise solely from the differential pressure across the control volume.

Consequently, a force balance equation gives

$$K = (PG - PF') \cdot ABE \qquad (7)$$

A flow equation for control volume a is now composed of equations (6) and (7).

$$\frac{dQ}{dt} = \frac{ABE}{\rho V_f} \left((PG-PF') \, ABE - \frac{\rho}{ABE} Q^2 \right) \qquad (8)$$

2. Control volume b (oil contained in the neck)

The momentum equation and force balance equation give, in combination:

$$PF'' - PFBL = \rho \cdot \frac{LUD}{AUD} \cdot \frac{dQ}{dt} \qquad (9)$$

3. Flow resistance of the outlet

The pressure loss at the outlet, comprising the bottom valve, is described by a quadratic flow equation.

$$PF' - PF'' = RUD \cdot Q^2 \qquad (10)$$

Wherein the ' quadratic resistance number 'RUD is a characteristic of the accumulator.

The flow equation of the A cc. is now obtained, combining equations (8), (9) and (10).

$$\frac{dQ}{dt} = \frac{(PG - PFBL) - \left(RUD - \frac{\rho}{ABE^2}\right) Q^2}{\rho \left(\frac{V_f}{ABE^2} + \frac{LUD}{AUD} \right)} \qquad (11)$$

Equation (11) defines the dynamic relationship between the load pressure PFBL at the outlet of the accumulator and the outlet flow rate Q.

3.2 HEAT TRANSFER APPROACH [12]

The conservation of energy equation for the charge gas gives:

$$mc_v \frac{dT}{dt} = q - P\dot{V} \qquad (1)$$

The oil flow rate out of the accumulator, Q, is equal to the rate of change of gas volume, that is:-

$$Q = \dot{V}$$

So we have

$$mc_v \frac{dT}{dt} = q - PQ \qquad (2)$$

The cooling of the compression-heated charge gas is by free convection and we assume the rate of heat transfer is proprotional to the temperature difference and the surface area, that is,

$$q = hA\ (T_o - T) \qquad (3)$$

Where h is a constant called the heat transfer coefficient, A is the surface area of the walls and piston, and T_o is the temperature of the surroundings.

To relate these data to the model, Equation (1) is solved for the constant volume cooling process. Substituting Equation (3) into Equation (1) and noting that $\dot{V} = 0$ for a constant volume process.

$$\frac{dT}{dt} = \frac{hA}{mc_v} (T_o - T) = -\frac{1}{\tau} (T - T_o)$$

and by integration

$$\frac{T - T_o}{T_1 - T_o} = e^{-t/\tau} = \frac{P - P_2}{P_1 - P_2}$$

Where

$$\tau \equiv \left(\frac{h\ A}{mc_v} \right)^{-1} .$$

The parameter τ is called the thermal time constant, and is assumed a constant in the integration. The procedure recommended for selecting τ, is as follows. Find the time at which the experimental curve for the temperature (or pressure) has dropped to a value of .37. That time is τ

The time constant, τ , is now the key parameter that characterizes the thermal behaviour of the accumulator. Once its value is known, it can be used to-calculate the gas process for any work cycle.

COMPUTER SIMULATION OF A WORK CYCLE

Whence, the thermal time constant, τ , is introduced in the energy equation, we have:

$$\frac{dT}{dt} = \frac{T_o - T}{\tau} - \frac{P}{mc_v} Q$$

Replacing the derivatives by finite elements one obtains

$$\Delta T = \left(\frac{T_o - T}{\tau} - \frac{P}{mc_v} Q \right) \cdot \Delta t$$

The change in gas volume is simply

$$\Delta V = Q \cdot \Delta t$$

The pressure is obtained from the ideal gas equation of state

$$P = \frac{mRT}{V}$$

We could write the basic algorithm as:

```
VOL  = VOL + FLOW * DELTI
TEMP = TEMP + ((TZERO-TEMP)/TAU
         - (PRES/(MASS * CV)) * FLOW) * DELTI
PRES = MASS * R * TEMP/VOL
```

The algorithm allows the calculation of volume, temperature, temperature and pressure at the time $t + \Delta t$ given values for all properties at time t.

The analytic methods are complicated and required a computer to visualize the process in which the accumulator behaves during a work cycle. This, we need only in complex problems requiring precision demand of an accumulator that actuates a mechanism to provide the desired performance. However, usually we have to furnish the system an amount of fluid without great precision and we wish to have the accumulator behaves as close as possible to the ideal condition, that is the isothermal one. This may be accomplished by placing fexible, open-celled foam into the charge gas volume. (figure 14).

As the gas is compressed, it passes energy (in the form of heat) to the foam, through a small temperature difference, and the charge-gas experiences nearly is othermal compression. During gas decompression, the heat transfer is reversed and the foam heats the expanding gas charge.

Energy flows back and forth between the gas and the foam during cycling. Heat losses to the accumulator and the surrounding area are reduced because during the cycle, gas temperatures are lower.

Professor D.R. Otis has conducted experiences with accumulator enveloped with and without foam.

A 150 cu. in. (2,458.05 cm^3) piston accumulator is subjected to the following work cycle, with and without foam: a 5 second filling period, a 44 second holding period and a

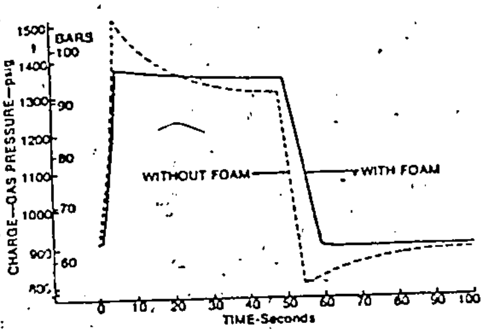


Fig. 15. Pressure-time history for a single work cycle compares accumulator performance with and without foam.

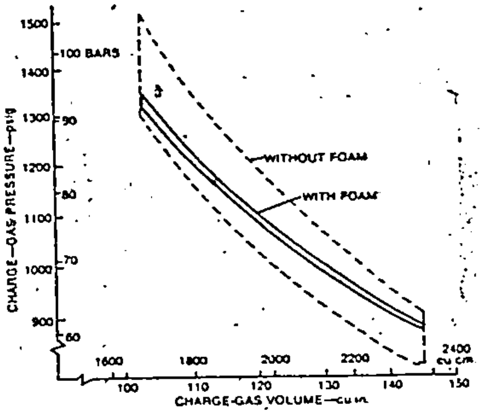


Fig. 16. Pressure-volume history for a single work cycle compares accumulator performance with and without foam.

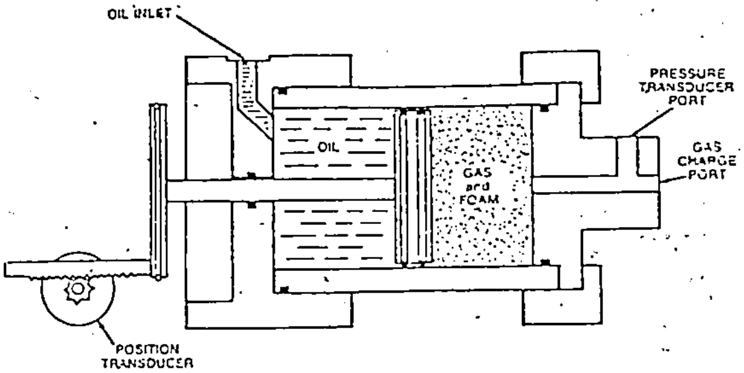


Fig. 14. Foam absorbs and releases heat from and to the charge gas in the accumulator. Transducers are used in the tests.

Figs. 14,15,16 from Ref.[13]

6 second emptying period.

In each test the charge gas is nitrogen at a precharge pressure of 900 psig (6,205.50 KPA). The foam fill is Scott Industrial Foam (120 pores per inch) impregnated with wax (Union Carbide Carbowax No. 750) to enhance the foam's heat capacity. (Foam weight is 62.6 grams, and increased to 191 grams after being treated with carbowax). The results are shown in Figures 15 and 16 as pressure time and pressure volume histories for a single cycle.

In the design without foam, Figure 15, the gas is compressed almost adiabatically with a rise in gas temperature. During the holding period, the gas cools with a resulting loss in pressure. This represents an irreversible loss in the potential to do useful work. This emptying process is almost adiabatic, and the pressure undershoots due to cooling of the charge-gas below ambient temperature during expansion. The enclosed hysteresis loop, Figure 16, represents a work loss of 15%. This energy loss heats the accumulator.

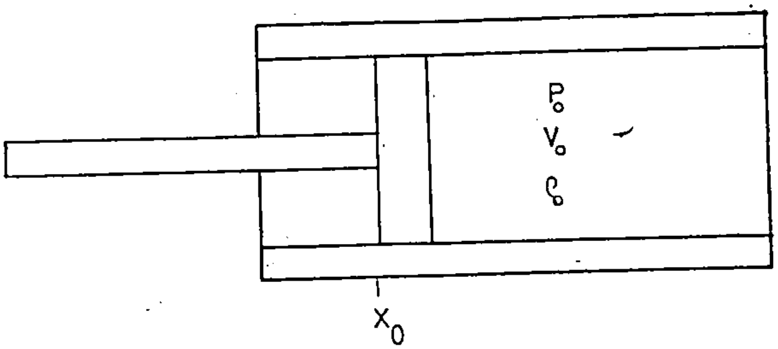
With foam, the process is considerably different, as shown in Figure 14. During compression, gas temperature remains almost constant as energy flows from the gas into the foam. During the holding period, pressure drops only slightly. The expansion process is nearly isothermal, and the enclosed hysteris loop, Figure 16, represents only a 1.4% loss.

The table summarizes the numerical results for the two tests. Flow energy input is 7.5% less and the output 7.3% greater with foam in the charge-gas volume; the heating load is reduced by a factor of 11.6 (o.8/0.069). Similar results have been confirmed for other foam fill materials and for a variety of test conditions.

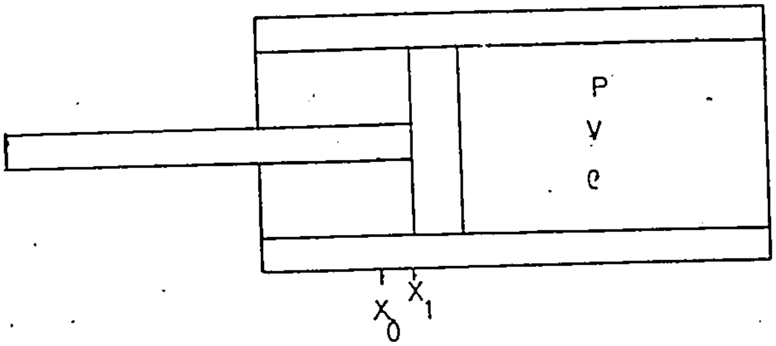
ACCUMULATOR TEST RESULTS

	WITHOUT FOAM	WITH FOAM
Flow energy input ft-lb/cycle	4135	3825
Flow energy output, ft-lb/cycle	3514	3771
Loss per cycle %	15	1.4
Heating Load Btu/cycle	0.8	0.069

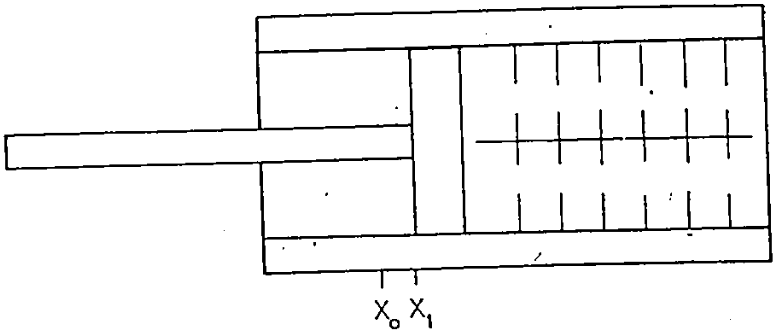
Fig.17.



a. Piston in Initial Position



b. Piston in Final Position



c. Copper Wool Placed in Air Chamber

4. EXPERIMENTS[18]

The purpose of our work is to study the effect of frequency on the polytropic exponent in a gas-charged piston type accumulator.

An experiment was set up to obtain the polytropic exponent of an cylindrical air-filled chamber as a function of-frequency. The air chamber of the cylinder was then partially filled with cooper wires creating smaller air chambers and the result of this addition on the polytropic exponent was examined.

A theoretical value for the polytropic exponent as a function of the signal frequency was obtained by studying Daniels' theory for the acoustic impedance of cylindrical enclosures. Daniels provides an analytical solution for the capacitance of infinitely long circular cylinders as a function of frequency and dimensions. The solution is valid throughout the entire range from isothermal to adiabatic conditions. The theory requires the derivation of the temperature distribution in the enclosure. Having the temperature distribution we can then obtain the polytropic exponent as a function of the signal frequency.

$$n = \frac{\gamma}{1 + \frac{2\ (\gamma - 1)}{j^{1.5}\ F} \cdot \frac{J_1\ (j^{1.5}\ F)}{J_o\ (j^{1.5}\ F)}}$$

Based on the theory of Daniels, we have the theoretical curve plotted for guidance with n, the polytropic exponent in function of F, the signal frequency.

EXPERIMENTAL PROCEDURE (Figure 17)

The inside rod was set at two inches (50.80 mm.) from back closure. The eccentricity of the crank of the driving mechanism was set at 0.230 inches (5.84 mm.). The disk driving mechanism was rotated by a motor whose frequency changed by the use of the motor speed control unit. As the piston within the cylinder reciprocated, the variation of pressure was sensed through a pressure transducer and was recorded on a strip chart recorder.

The pressure differential values at various frequencies were thereby obtained. The experiment was repeated with different amounts of copper wire stuffed inside the air chamber of the cylinder.

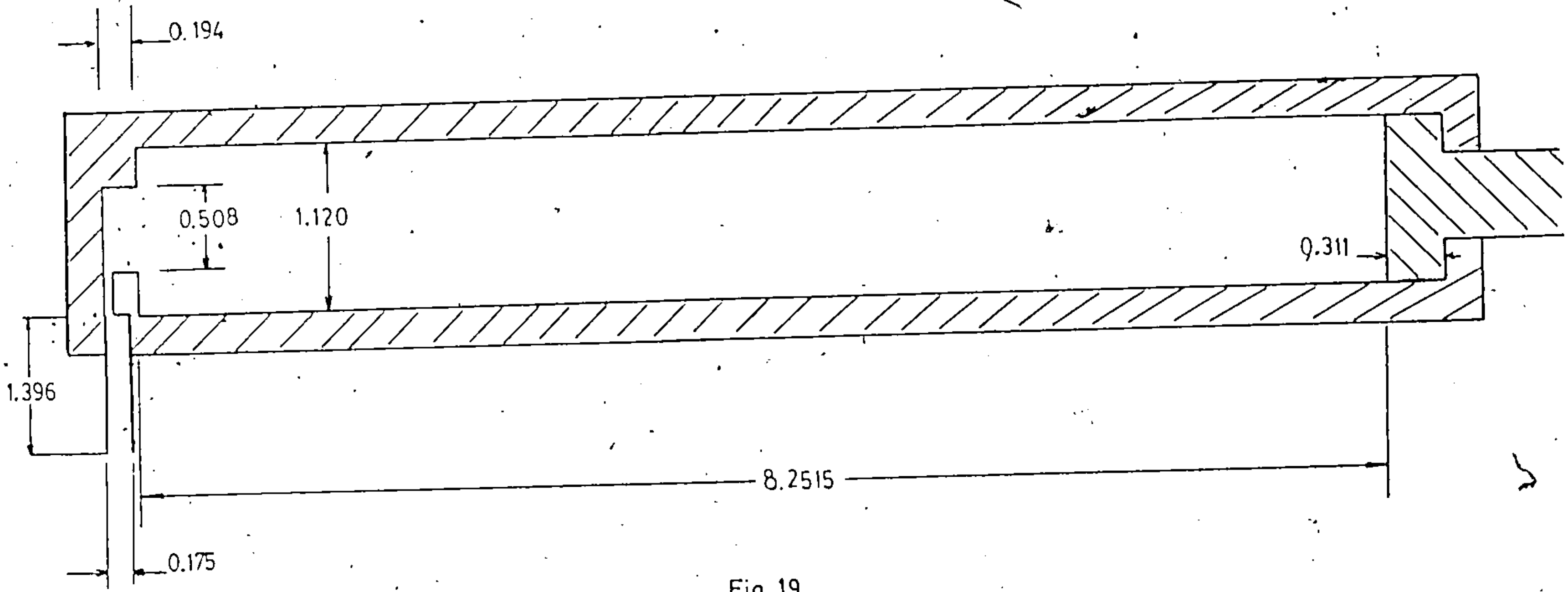


Fig. 19.

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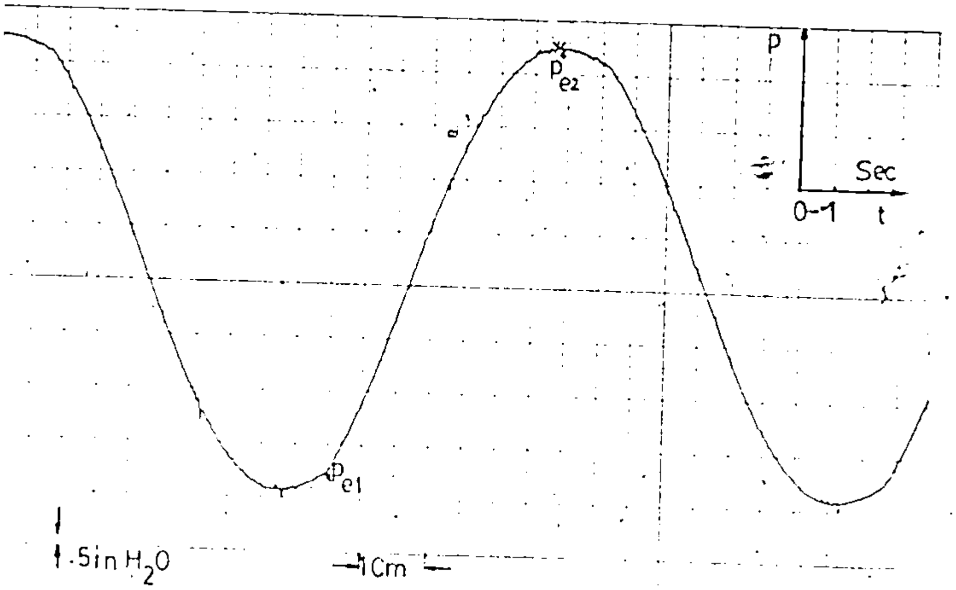
EXPERIMENTAL INTERPRETATION [21]

The sinusoidal shape of the pressure developed on the strip chart recorder represents the resultant or "effective" pressure pe; it is thus the algebraic difference i.e. $p_e = p_n \pm pi$. The net pressure p_n is the pressure difference between the two sides of the piston. pi, the virtual pressure is similar to that of the piston acceleration and such "pressure" opposes the net pressure p_n of the working medium during the early part of the stroke and assists it during the remainder. The value of $p_e = p_n \pm pi$ is valid for a cylinder in horizontal position.

On the graph there are two values that was important for us, the beginning of instroke, i.e. P_{e1} and the end of instroke i.e. p_{e2}. Then the pressure differential $\Delta P_e = P_{e2} - P_{e1}$ is what we need for graphic representation.



EXPERIMENTAL TEST EQUIPMENT SET UP (figure 18)

(1) Motor speed controller, G.K. Heller Corp., Model S-12

(2) Variable speed, reversible motor, G.K. Heller Corp., Model 6T60-20.

(3) Disk driving mechanism

(4) Cylinder with rod, Scoville Corp., Model 200-0080

(5) Pressure transducer, validyne Engineering Corp., Model DP15TL

(6) Transducer indicator, Validyne Engineering Corp., Model CD12

(7) Strip chart recorder, Brush Corp., Model Mark 280

CALIBRATION EQUIPMENT

(a) Air pressure regulator, Moore Instrument Co. Ltd., Model Nullmatix 40-2

(b) Inclined Manometer, Air flow Developments Ltd., Model Mark 4

(c) Dial indicator.

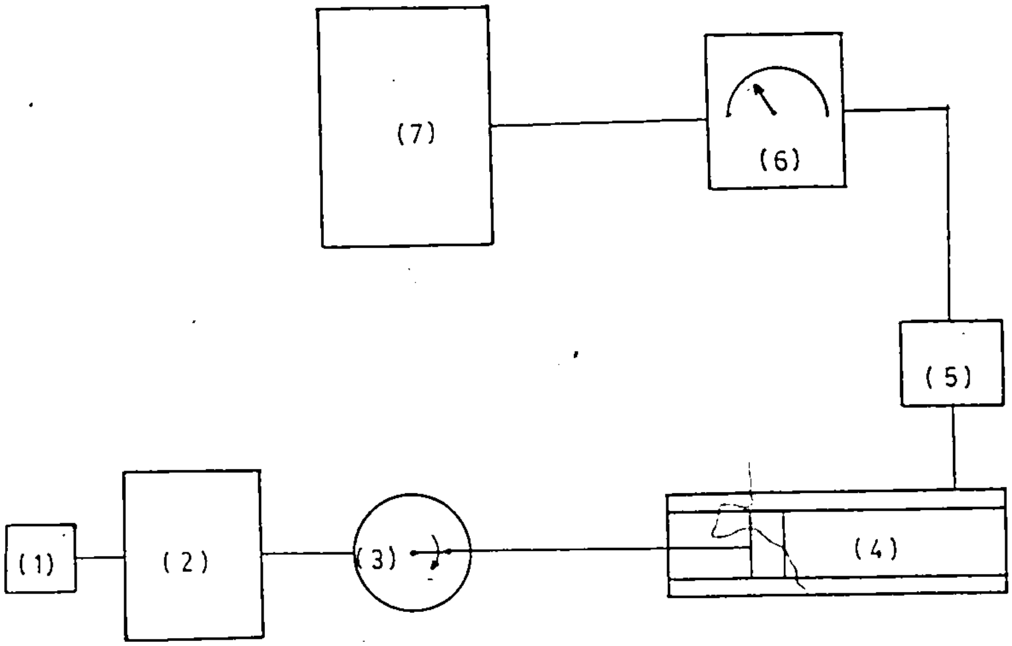


Fig. 18 .Test Equipement Set Up Representation

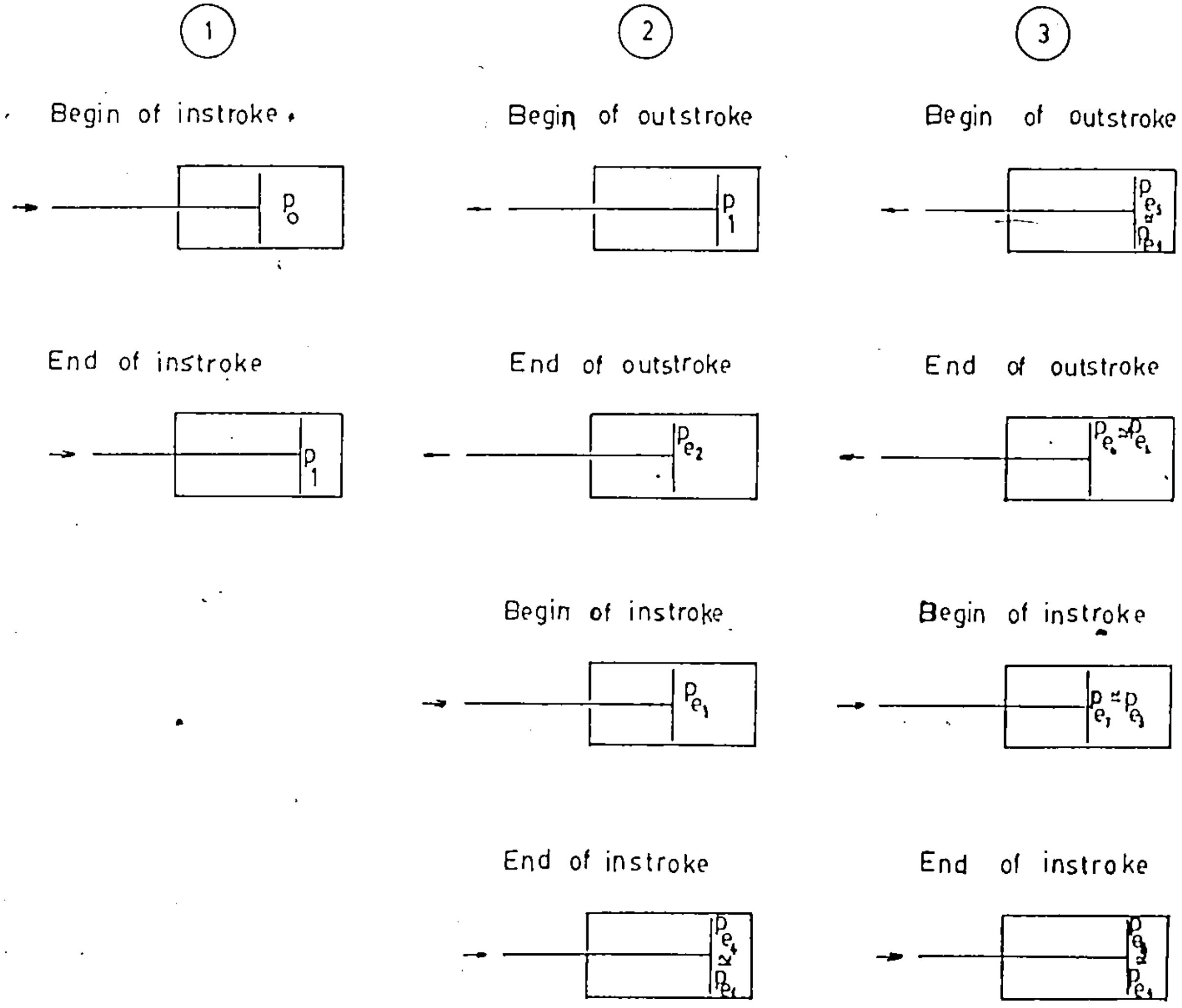
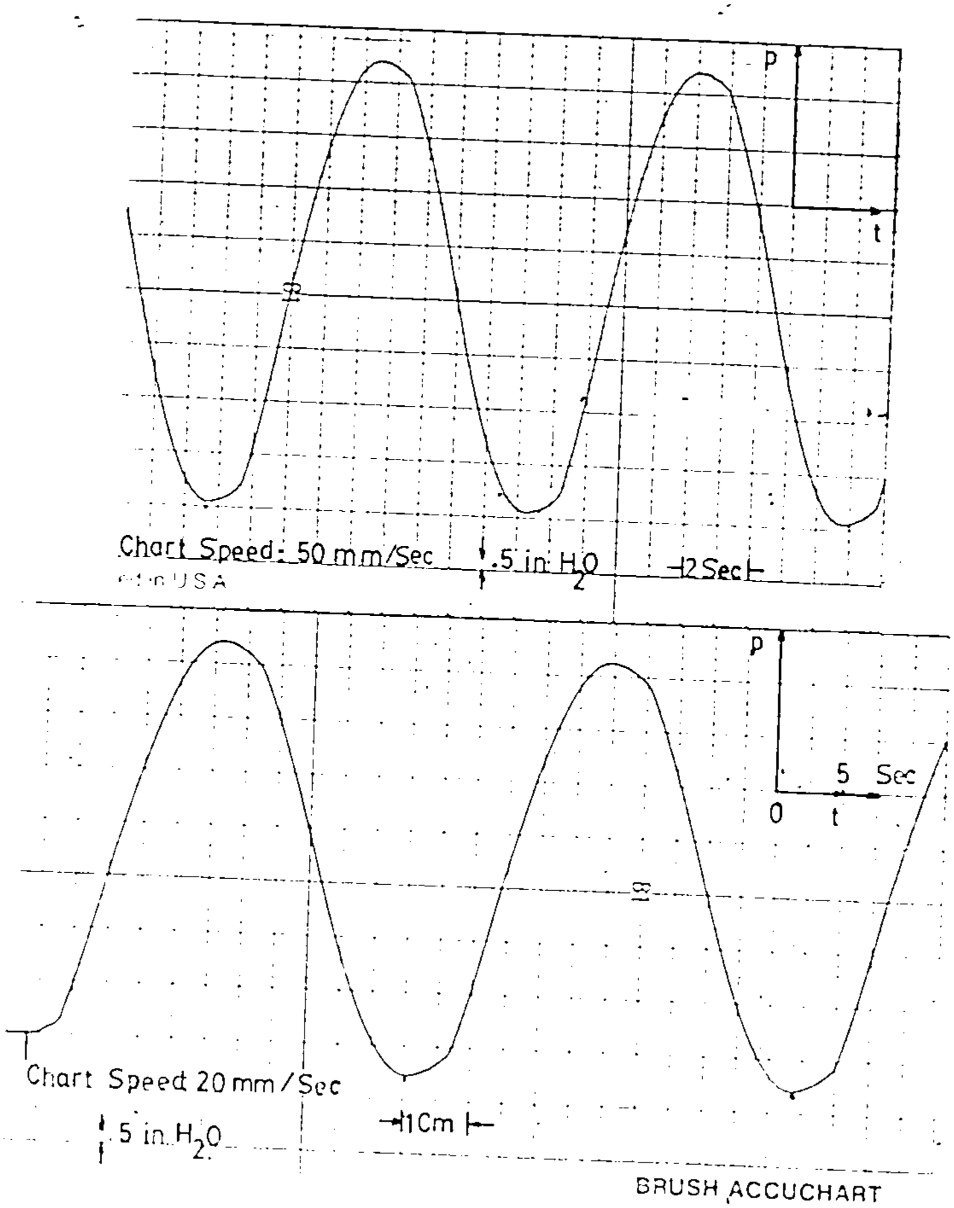


Fig.I. History of the reciprocated mechanism

EXPERIMENTAL RESULTS

1. WITH OIL

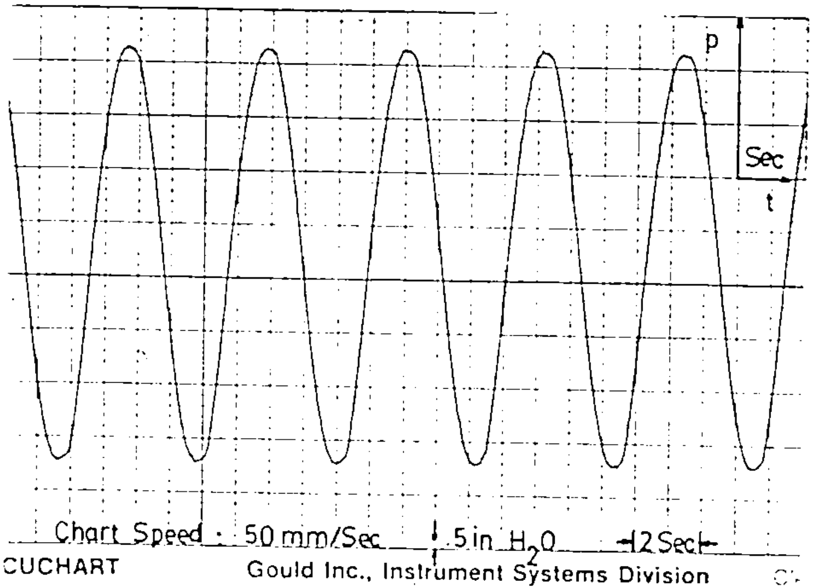
Section A - 0 gram of Copper Mesh in Cylinder

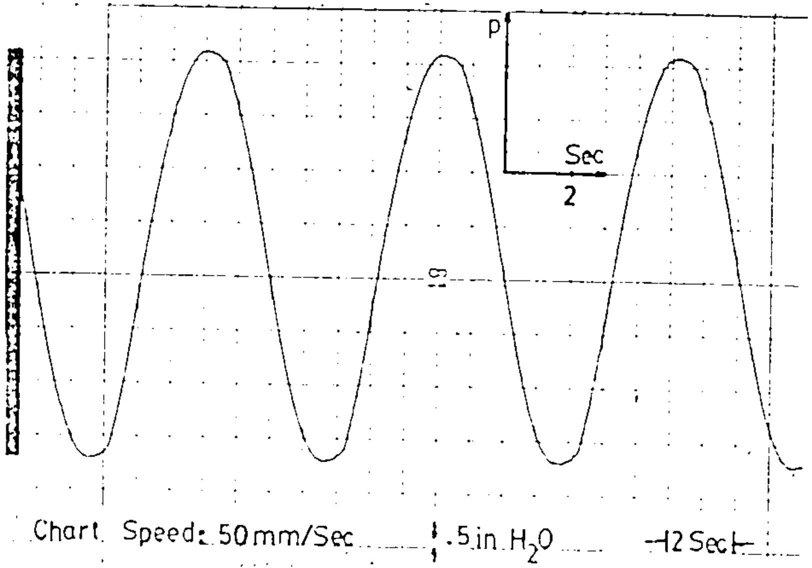


EXPERIMENTAL RESULTS

1. WITH OIL

Section B - 17.05 grams of Copper Mesh in Cylinder

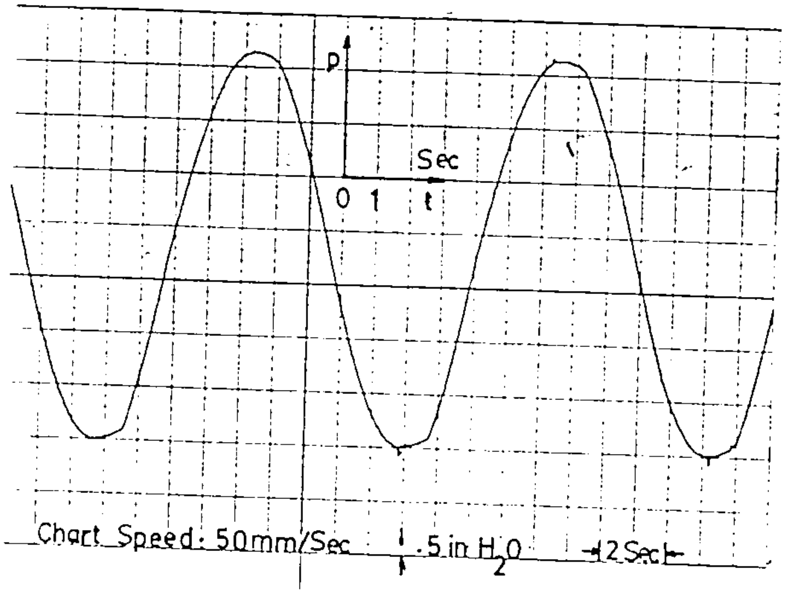


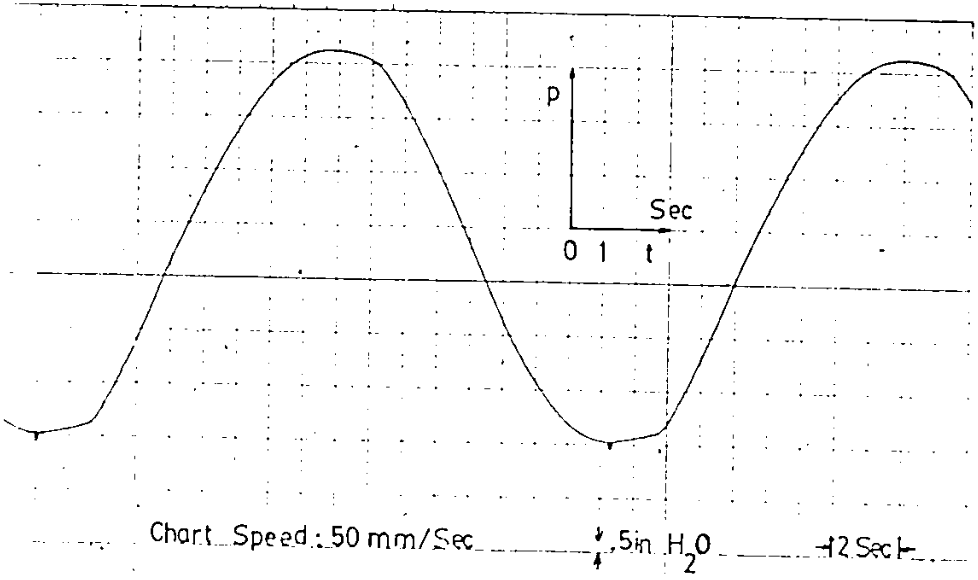


EXPERIMENTAL RESULTS

1. WITH OIL

Section C - 34.57 grams of Copper Mesh in Cylinder





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PIW	PIA	CYCLE	DPA(I)	P1(I)	P2(I)	ZN	F
17.600000	.635219	.125000	.615369	14.678345	15.315369	1.090244	2.762049
18.900000	.682139	.250000	.635219	14.663908	15.335219	1.148738	3.906127
19.000000	.685748	.238095	.635219	14.663908	15.335219	1.148738	3.811990
19.400000	.700185	.322581	.649656	14.654885	15.349656	1.188681	4.437062
19.400000	.700185	.333333	.644242	14.653080	15.344242	1.182789	4.510407
19.400000	.700185	.333333	.642438	14.654885	15.342438	1.176610	4.510407
19.600000	.707403	.338983	.653265	14.656690	15.353265	1.191555	4.548470
19.700000	.711012	.416667	.647851	14.645862	15.347851	1.201469	5.042788
19.800000	.714622	.416667	.649656	14.642253	15.349656	1.210811	5.042788
20.000000	.721840	.512821	.649656	14.642253	15.349656	1.210811	5.594471
20.000000	.721840	.531915	.656874	14.638644	15.356874	1.229203	5.697672
20.100000	.725449	.625000	.653265	14.642253	15.353265	1.216845	6.176129
20.300000	.732668	.666667	.664093	14.656670	15.364093	1.209646	6.378679
20.350000	.734472	.769231	.658679	14.638644	15.358679	1.232219	6.851800
20.400000	.736277	.769231	.664093	14.631425	15.364093	1.253921	6.851800
20.600000	.743495	.888889	.667702	14.631425	15.367702	1.259948	7.365464
20.750000	.748909	1.041667	.667702	14.624207	15.367702	1.272612	7.973348
21.000000	.757932	1.290323	.674920	14.627816	15.374920	1.278331	8.874124
21.000000	.757932	1.333333	.671311	14.616988	15.371311	1.291309	9.020814
21.100000	.761541	1.428571	.680334	14.627816	15.380334	1.287366	9.337429
21.200000	.765150	1.666667	.674920	14.618793	15.374920	1.294165	10.085577
21.300000	.768760	1.739130	.682139	14.624207	15.382139	1.296709	10.302495
21.300000	.768760	1.886792	.678530	14.615184	15.378530	1.306526	10.730956
21.400000	.772369	2.222222	.685748	14.624207	15.385748	1.302730	11.645021
21.400000	.772369	2.000000	.685748	14.620598	15.385748	1.309064	11.048196
21.400000	.772369	2.173913	.678530	14.618793	15.378530	1.300189	11.518540
21.500000	.775978	2.380752	.682139	14.609770	15.382139	1.322055	12.054570

Table A - NO COPPER MESH - WITH OIL

POOR COPY

PIW	PIA	CYCLE	DPA(I)	P1(I)	P2(I)	ZN	F
17.300000	.624392	.111111	.618978	14.698195	15.318978	1.034917	2.604085
17.350000	.626196	.120482	.624392	14.703609	15.324392	1.034544	2.711673
17.750000	.640633	.243902	.628001	14.692782	15.328001	1.058865	3.858197
17.900000	.646047	.303030	.624392	14.681954	15.324392	1.071417	4.300504
17.900000	.646047	.217391	.631610	14.692782	15.331610	1.064755	3.642482
17.950000	.647851	.628931	.637024	14.671126	15.337024	1.110487	6.195520
18.000000	.649656	.208333	.631610	14.687368	15.331610	1.073975	3.565790
18.200000	.656874	.429185	.631610	14.674736	15.331610	1.095501	5.117978
18.200000	.656874	.454545	.635219	14.680149	15.335219	1.092162	5.267021
18.250000	.658679	.526316	.640633	14.685563	15.340633	1.091768	5.667605
18.300000	.660484	.537634	.631610	14.674736	15.331610	1.095501	5.728223
18.300000	.660484	.333333	.637024	14.680149	15.337024	1.095106	4.510407
18.350000	.662288	.322581	.638828	14.680149	15.338828	1.098049	4.437062
18.400000	.664093	.645161	.638828	14.676540	15.338828	1.104201	6.274953
18.500000	.667702	.754717	.644242	14.680149	15.344242	1.106877	6.786853
18.550000	.669507	.434783	.674920	14.676540	15.374920	1.162997	5.151248
18.600000	.671311	.434783	.640633	14.674736	15.340633	1.110220	5.151248
18.600000	.671311	.833333	.642438	14.674736	15.342438	1.113163	7.131580
18.600000	.671311	.746269	.638828	14.672931	15.338828	1.110353	6.748760
18.650000	.673116	.877193	.640633	14.669322	15.340633	1.119451	7.316846
18.650000	.673116	.930233	.644242	14.674736	15.344242	1.116105	7.534807
18.650000	.673116	1.052632	.646047	14.674736	15.346047	1.119047	8.015203
18.700000	.674920	.540541	.642438	14.671126	15.342438	1.119316	5.743683
18.800000	.678530	1.000000	.642438	14.671126	15.342438	1.119316	7.812254
18.800000	.678530	1.081081	.646047	14.674736	15.346047	1.119047	8.122795
18.850000	.680334	.666667	.644242	14.671126	15.344242	1.122259	6.378679
18.850000	.680334	1.290323	.649656	14.674736	15.349656	1.124930	8.874124
18.850000	.680334	1.111111	.644242	14.672931	15.344242	1.119182	8.234839
18.900000	.682139	1.282051	.646047	14.667517	15.346047	1.131356	8.845636
18.900000	.682139	1.428571	.646047	14.671126	15.346047	1.125201	9.337429
18.900000	.682139	1.250000	.649656	14.672931	15.349656	1.128007	8.734366
18.900000	.682139	1.481481	.649656	14.674736	15.349656	1.124930	9.508773
19.000000	.685748	1.724138	.649656	14.667517	15.349656	1.137239	10.257992
19.000000	.685748	1.851852	.649656	14.667517	15.349656	1.137239	10.631131
19.000000	.685748	1.739130	.651461	14.672931	15.351461	1.130948	10.302495
19.000000	.685748	1.600000	.651461	14.674736	15.351461	1.127871	9.881807
19.000000	.685748	1.612903	.647851	14.671126	15.347851	1.128143	9.921573
19.000000	.685748	1.600000	.653265	14.671126	15.353265	1.136966	9.881807
19.000000	.685748	1.538462	.649656	14.671126	15.349656	1.131084	9.689909
19.000000	.685748	1.428571	.649656	14.671126	15.349656	1.131084	9.337429
19.050000	.687553	1.739130	.651461	14.672931	15.351461	1.130948	10.302495
19.050000	.687553	2.000000	.653265	14.667517	15.353265	1.143121	11.048196
19.100000	.689357	1.818182	.653265	14.671126	15.353265	1.136966	10.534041
19.100000	.689357	2.083333	.653265	14.667517	15.353265	1.143121	11.276017
19.100000	.689357	2.272727	.653265	14.671126	15.353265	1.136966	11.777416
19.100000	.689357	1.818182	.653265	14.672931	15.353265	1.133889	10.534041
19.100000	.689357	2.000000	.653265	14.700000	15.353265	1.087778	11.048196
19.150000	.691162	2.105263	.653265	14.667517	15.353265	1.143121	11.335209
19.150000	.691162	2.000000	.655070	14.671126	15.355070	1.139906	11.048196
19.200000	.692766	2.380952	.656874	14.671126	15.356874	1.142846	12.054570

Table B - 1705 g COPPER MESH - WITH OIL

POOR COPY

PIW	PIA	CYCLE	DPA(I)	P1(I)	P2(I)	ZN	F
17.700000	.638828	.142857	.644242	14.714437	15.344242	1.069109	2.952754
17.750000	.640633	.243902	.640633	14.703609	15.340633	1.081885	3.858197
17.800000	.642438	.131579	.638828	14.710828	15.338828	1.066365	2.833802
17.800000	.642438	.136986	.638828	14.710828	15.338828	1.066365	2.891445
17.800000	.642438	.240964	.642438	14.707218	15.342438	1.078625	3.834885
17.850000	.644242	.344828	.644242	14.707218	15.344242	1.081625	4.587514
17.850000	.644242	.327869	.644242	14.707218	15.344242	1.081625	4.473284
17.850000	.644242	.243902	.644242	14.709023	15.344242	1.078496	3.858197
17.900000	.646047	.454545	.644242	14.703609	15.344242	1.087886	5.267021
17.900000	.646047	.350877	.640633	14.703609	15.340633	1.081885	4.627580
17.950000	.647851	.434783	.646047	14.700000	15.346047	1.097148	5.151248
18.000000	.649656	.666667	.653265	14.714437	15.353265	1.084104	6.378679
18.000000	.649656	.563380	.646047	14.700000	15.346047	1.097148	5.863773
18.000000	.649656	.555556	.646047	14.707218	15.346047	1.084625	5.822910
18.000000	.649656	.540541	.644242	14.700000	15.344242	1.094148	5.743683
18.050000	.651461	.467290	.646047	14.714437	15.346047	1.072108	5.340347
18.050000	.651461	.362319	.640633	14.716241	15.340633	1.059980	4.702424
18.100000	.653265	.578035	.649656	14.712632	15.349656	1.081236	5.939547
18.100000	.653265	.781250	.656874	14.710828	15.356874	1.096358	6.905122
18.100000	.653265	.943396	.655070	14.710828	15.355070	1.093360	7.587932
18.100000	.653265	.781250	.656874	14.710828	15.356874	1.096358	6.905122
18.100000	.653265	.634921	.647851	14.698195	15.347851	1.103280	6.224953
18.150000	.655070	1.063830	.655070	14.709023	15.355070	1.096490	8.057725
18.200000	.656874	1.388889	.656874	14.710828	15.356874	1.096358	9.206830
18.200000	.656874	1.081081	.649656	14.698195	15.349656	1.106279	8.122795
18.200000	.656874	.909091	.649656	14.698195	15.349656	1.106279	7.448692
18.200000	.656874	.769231	.649656	14.700000	15.349656	1.103147	6.851800
18.200000	.656874	1.176471	.649656	14.696391	15.349656	1.109411	8.473579
18.200000	.656874	1.204819	.656874	14.710828	15.356874	1.096358	8.575063
18.250000	.658679	1.470588	.658679	14.710828	15.358679	1.099355	9.473750
18.250000	.658679	1.666667	.658679	14.710828	15.358679	1.099355	10.085577
18.250000	.658679	1.818182	.651461	14.696391	15.351461	1.112410	10.534041
18.250000	.658679	1.333333	.649656	14.696391	15.349656	1.109411	9.020814
18.300000	.660484	1.481481	.649656	14.694586	15.349656	1.112543	9.508773
18.350000	.662288	2.500000	.653265	14.694586	15.353265	1.118541	12.352258
18.350000	.662288	2.105263	.651461	14.694586	15.351461	1.115542	11.335209
18.350000	.662288	1.666667	.649656	14.694586	15.349656	1.112543	10.085577
18.350000	.662288	1.666667	.649656	14.692782	15.349656	1.115676	10.085577
18.350000	.662288	2.040816	.662288	14.709023	15.362288	1.108478	11.160363
18.400000	.664093	1.851852	.664093	14.712632	15.364093	1.105216	10.631131
18.400000	.664093	2.325581	.664093	14.707218	15.364093	1.114605	11.913576
18.400000	.664093	2.127660	.664093	14.710828	15.364093	1.108345	11.395344
18.500000	.667702	2.631579	.667702	14.703609	15.367702	1.126857	12.673149

Table C – 34.57 g COPPER MESH – WITH OIL

POOR COPY

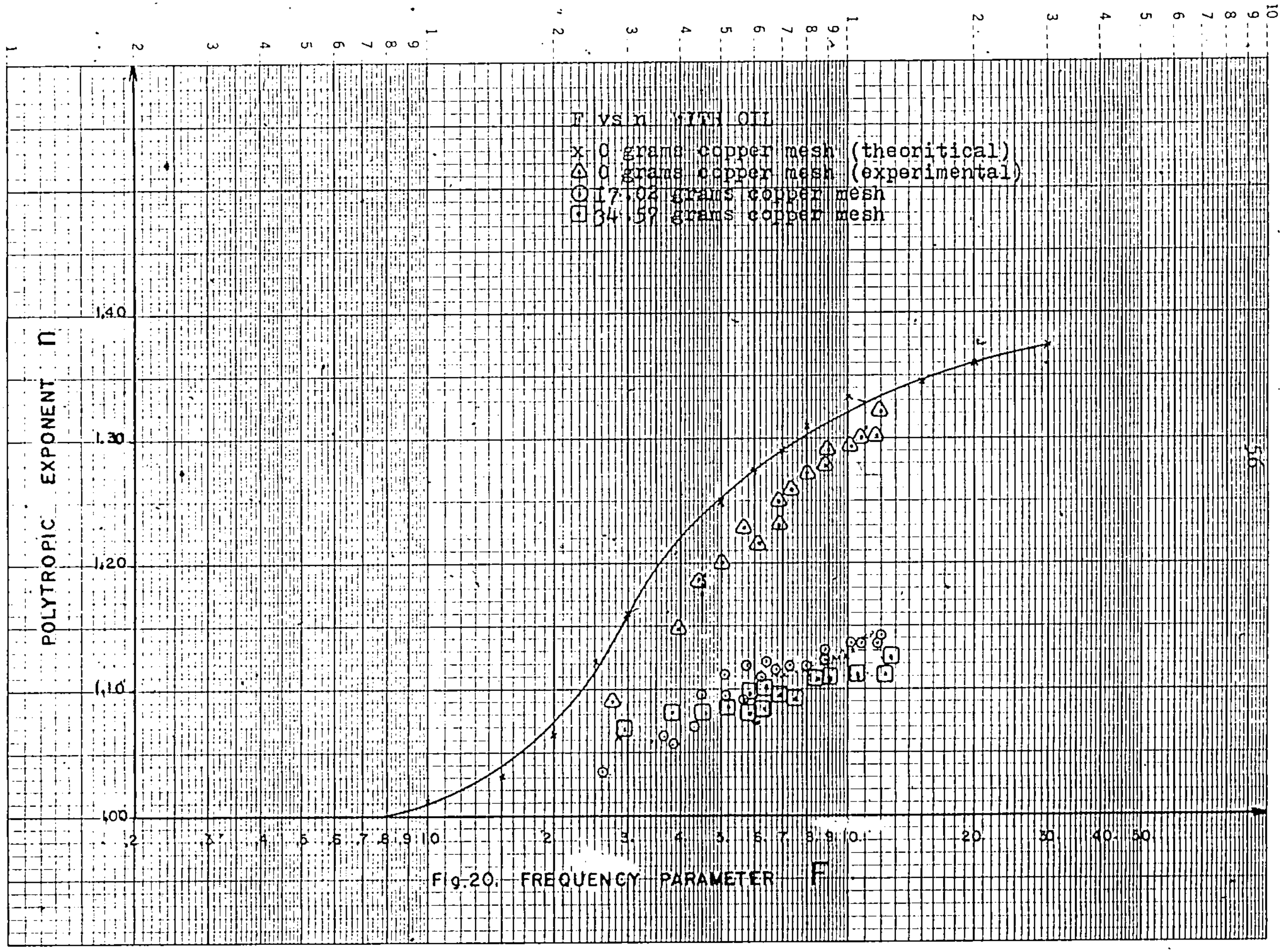


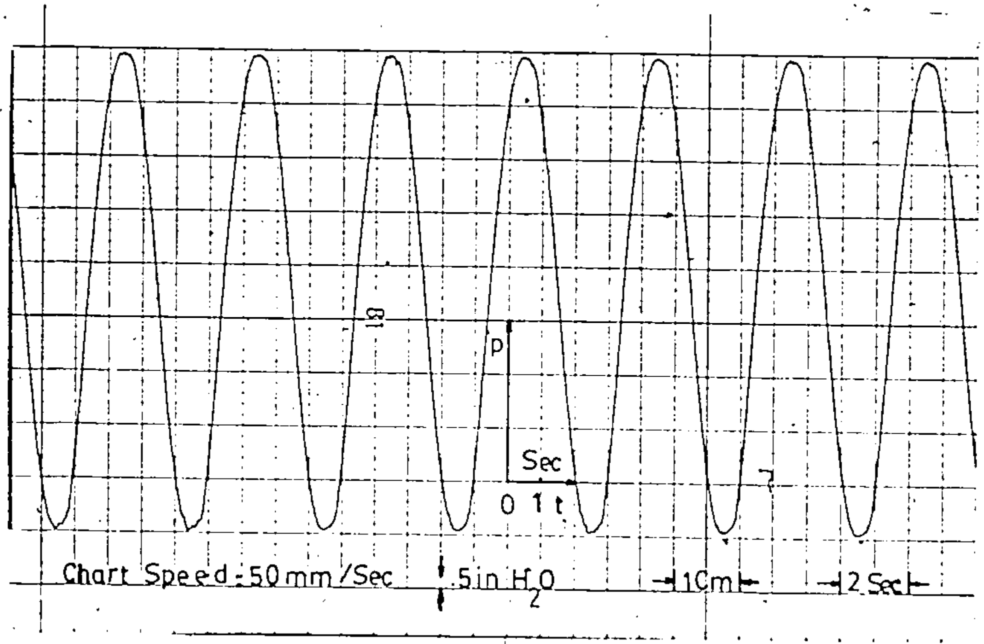
Fig.20. FREQUENCY PARAMETER F

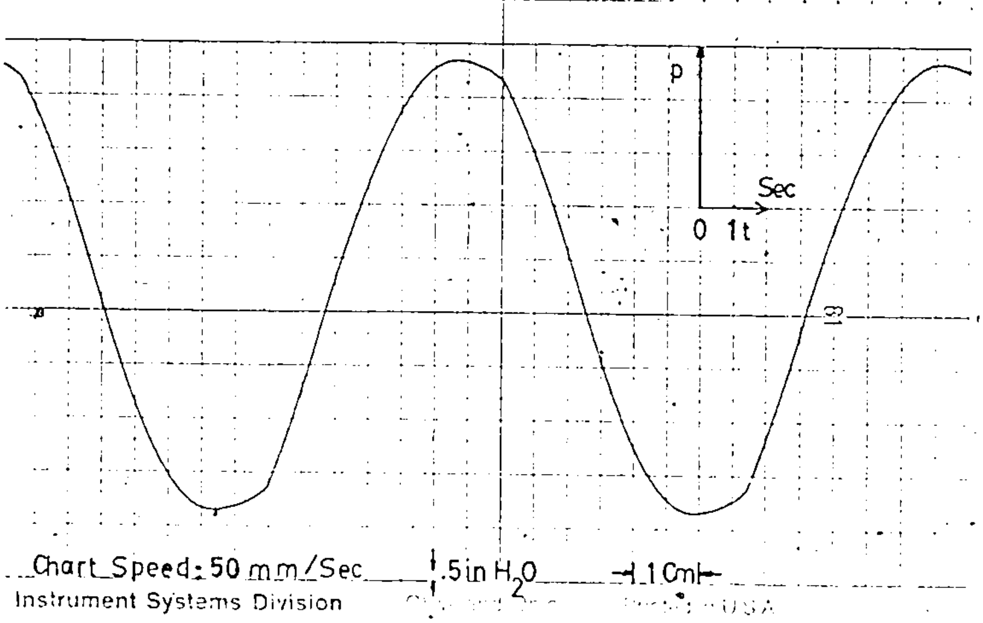
POOR COPY

EXPERIMENTAL RESULTS

1. WITHOUT OIL

Section A - 0 grams of Copper Mesh in Cylinder

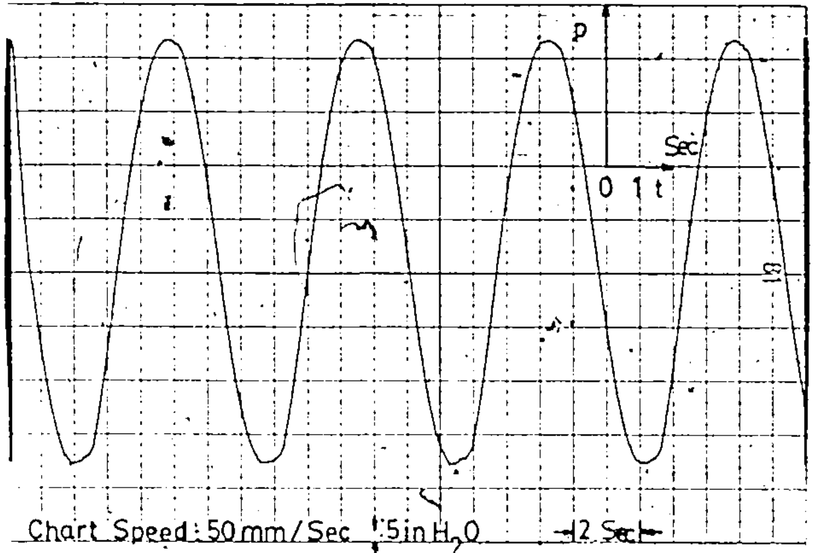


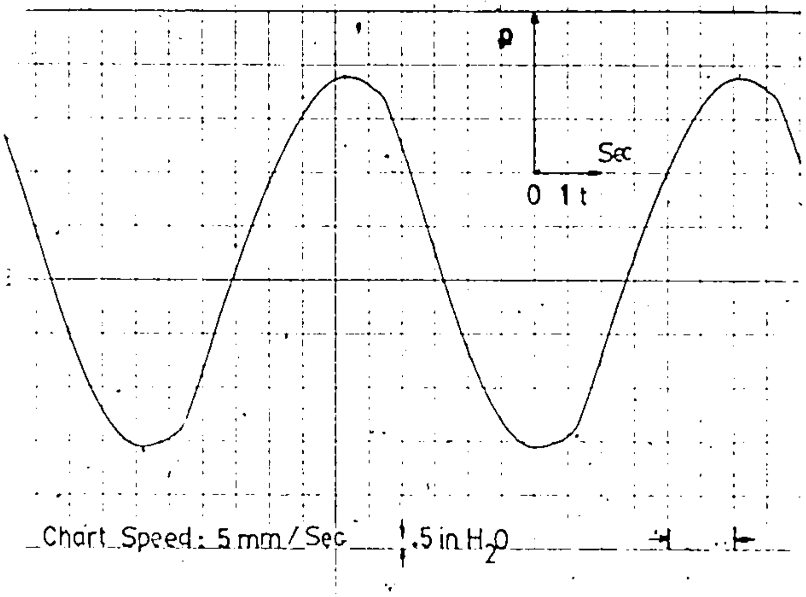


EXPERIMENTAL RESULTS

1. WITHOUT OIL

Section B - 17.05 grams of Copper Mesh in Cylinder

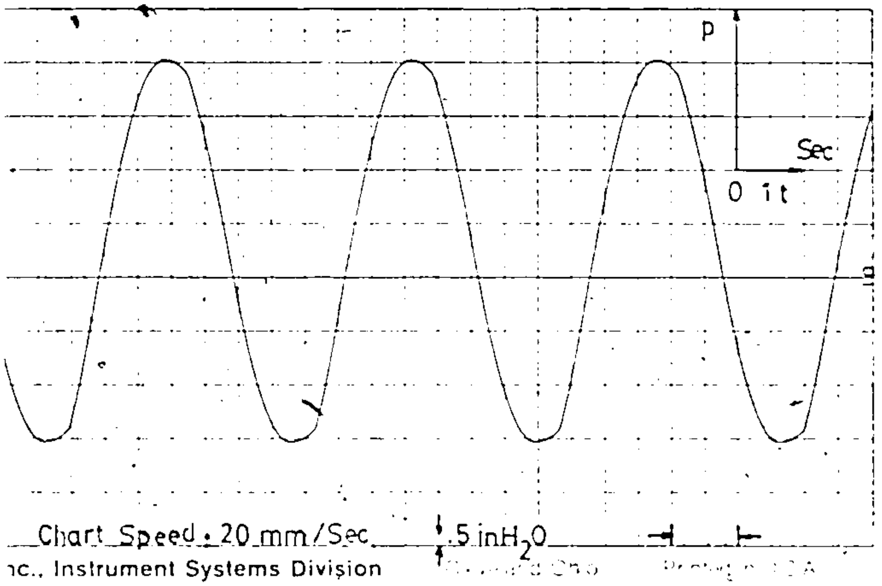


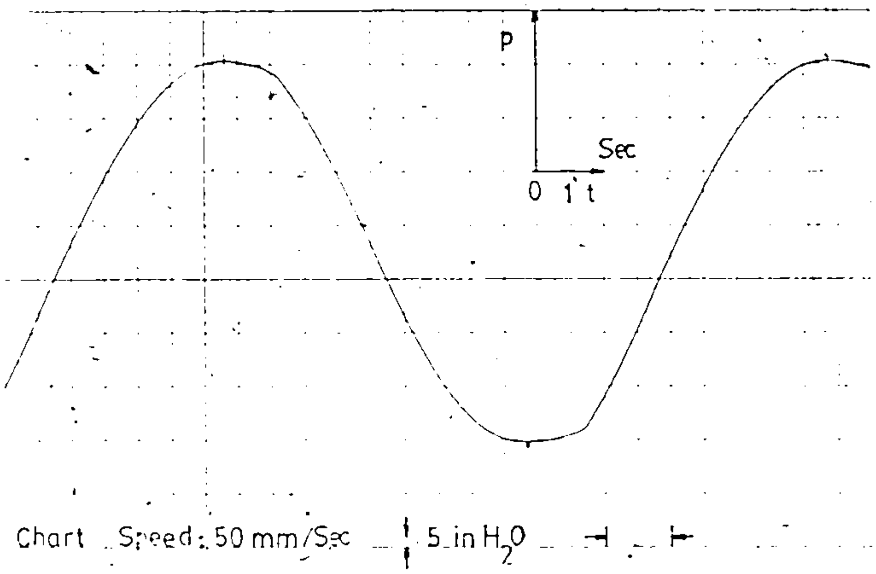


EXPERIMENTAL RESULTS

1. WITHOUT OIL

Section C - 34.57 grams of Copper Mesh in Cylinder





POOR COPY!

FIW	FIA	CYCLE	DPA(I)	P1(I)	P2(I)	ZN	F
18.200000	.656874	.108696	.633415	14.680149	15.333415	1.102819	2.575624
18.200000	.656874	.105263	.638828	14.685563	15.338828	1.102422	2.534630
18.750000	.676725	.142857	.649656	14.674736	15.349656	1.138978	2.952754
19.650000	.709208	.222222	.669507	14.663908	15.369507	1.190411	3.682732
19.650000	.709208	.224719	.664093	14.660299	15.364093	1.187722	3.703364
19.750000	.712817	.236686	.664093	14.658494	15.364093	1.190840	3.800695
20.050000	.723645	.316456	.680334	14.660299	15.380334	1.214484	4.394736
20.050000	.723645	.327869	.669507	14.645862	15.369507	1.221602	4.473284
20.100000	.725449	.307692	.671311	14.649471	15.371311	1.218334	4.333459
20.250000	.730863	.338983	.673116	14.645862	15.373116	1.227549	4.548470
20.350000	.734472	.408163	.676725	14.642253	15.376725	1.239738	4.991066
20.450000	.738081	.416667	.680334	14.645862	15.380334	1.239440	5.042788
20.450000	.738081	.425532	.676725	14.640448	15.376725	1.242860	5.096153
20.500000	.739886	.413223	.685748	14.649471	15.385748	1.242113	5.021907
20.550000	.741691	.434783	.676725	14.642253	15.376725	1.239738	5.151248
20.700000	.747104	.561798	.689357	14.645862	15.389357	1.254296	5.855532
20.900000	.754323	.526316	.683943	14.627816	15.383943	1.276613	5.667605
20.950000	.756127	.645161	.685748	14.633230	15.385748	1.270211	6.274953
20.950000	.756127	.694444	.694771	14.640448	15.394771	1.272570	6.510212
21.000000	.757932	.740741	.685748	14.627816	15.385748	1.279584	6.723718
21.000000	.757932	.625000	.685748	14.636839	15.385748	1.263965	6.176129
21.050000	.759737	.769231	.687553	14.633230	15.387553	1.273182	6.851800
21.100000	.761541	.625000	.685748	14.635034	15.385748	[illegible]	6.176129
21.200000	.765150	.769231	.685748	14.624207	15.385748	1.265835	6.851800
21.250000	.766955	.869565	.691162	14.627816	15.391162	1.288496	7.284964
21.400000	.772369	.909091	.687553	14.624207	15.387553	1.288806	7.448692
21.400000	.772369	1.052632	.692966	14.624207	15.392966	1.297716	8.015203
21.400000	.772369	1.041667	.700185	14.629621	15.400185	1.300216	7.973348
21.400000	.772369	1.052632	.692966	14.624207	15.392966	1.297716	8.015203
21.500000	.775978	1.176471	.696576	14.624207	15.396576	1.303654	8.473579
21.600000	.779587	1.052632	.691162	14.620598	15.391162	1.300998	8.015203
21.650000	.781392	1.298701	.703794	14.627816	15.403794	1.309276	8.902890
21.700000	.783196	.540541	.682139	14.640448	15.382139	1.251777	5.743683
21.700000	.783196	1.333333	.700185	14.622402	15.400185	1.312717	9.020814
21.750000	.785001	1.612903	.707403	14.627816	15.407403	1.315211	9.921573
21.800000	.786806	1.481481	.700185	14.624207	15.400185	1.309591	9.508773
21.850000	.788610	1.666667	.701989	14.618793	15.401989	1.321938	10.085577
21.850000	.788610	1.666667	.701989	14.616988	15.401989	1.325065	10.085577
21.900000	.790415	1.960784	.711012	14.627816	15.411012	1.321143	10.939344
21.900000	.790415	1.851852	.711012	14.624207	15.411012	1.327394	10.631131
21.950000	.792219	.540541	.683943	14.635034	15.383943	1.264117	5.743683
21.950000	.792219	1.739130	.703794	14.620598	15.403794	1.321779	10.302495
22.000000	.794024	2.173913	.711012	14.627816	15.411012	1.321143	11.518540
22.000000	.794024	2.000000	.703794	14.615184	15.403794	1.331160	11.048196
22.000000	.794024	2.272727	.712817	14.627816	15.412817	1.324109	11.777416
22.000000	.794024	1.818182	.703794	14.613379	15.403794	1.334283	10.534041
22.000000	.794024	2.000000	.703794	14.615184	15.403794	1.331160	11.048196
22.000000	.794024	2.439024	.712817	14.624207	15.412817	1.330360	12.200691
22.050000	.795829	2.222222	.705599	14.615184	15.405599	1.334127	11.645821
22.100000	.797633	2,105263	.707403	14.615184	15.407403	1.337094	11.335209
22.100000	.797633	2.352941	.707403	14.613379	15.407403	1.340222	11.983451
22.100000	.797633	2.222222	.707403	14.615184	15.407403	1.337094	11.645821
22.100000	.797633	2.500000	.707403	14.615184	15.407403	1.337094	12.352258
22.100000	.797633	2.631579	.714622	14.624207	15.414622	1.333326	12.673149
22.150000	.799438	2.666667	.709208	14.618793	15.409208	1.333807	12.757357

Table I - NO COPPER MESH - WITHOUT OIL

POOR COPY

FIW	FIA	CYCLE	DPA(I)	F1(I)	F2(I)	ZN	F
17.250000	.622587	.083333	.613564	14.696391	15.313564	1.049735	2.255203
17.400000	.628001	.070909	.624392	14.703609	15.324392	1.055241	2.355483
17.400000	.628001	.103093	.620782	14.696391	15.320782	1.061761	2.508364
17.850000	.644242	.178571	.620782	14.678345	15.320782	1.093114	3.301280
17.950000	.647851	.178571	.629805	14.683759	15.329805	1.098729	3.301280
18.000000	.649656	.350877	.638828	14.674736	15.338828	1.129429	4.627580
18.050000	.651461	.198020	.635219	14.689172	15.335219	1.098332	3.476407
18.150000	.655070	.263158	.633415	14.681954	15.333415	1.107372	4.007602
18.250000	.658679	.239521	.640633	14.681954	15.340633	1.119882	3.823386
18.300000	.660484	.268456	.642438	14.681954	15.342438	1.122884	4.047746
18.500000	.667702	.333333	.629805	14.663908	15.329805	1.133249	4.510407
18.550000	.669507	.338983	.646047	14.680149	15.346047	1.132023	4.548470
18.700000	.674920	.454545	.631610	14.656690	15.331610	1.148817	5.267021
18.800000	.678530	.476190	.649656	14.672931	15.349656	1.150574	5.390967
18.850000	.680334	.454545	.646047	14.672931	15.346047	1.144573	5.267021
19.000000	.685748	.666667	.647851	14.667517	15.347851	1.156991	6.378677
19.000000	.685748	.666667	.651461	14.671126	15.351461	1.156713	6.378679
19.000000	.685748	.655738	.649656	14.671126	15.349656	1.153713	6.326178
19.100000	.689357	.769231	.653265	14.667517	15.353265	1.165990	6.851800
19.100000	.689357	.769231	.649656	14.663908	15.349656	1.166271	6.851800
19.100000	.689357	.869565	.649656	14.663908	15.349656	1.166271	7.284964
19.100000	.689357	.869565	.653265	14.667517	15.353265	1.165990	7.284964
19.150000	.691162	.930233	.655070	14.665713	15.355070	1.172129	7.534807
19.250000	.694771	.877193	.656874	14.662103	15.356874	1.181409	7.316846
19.300000	.696576	1.052632	.658679	14.663908	15.358679	1.181267	8.015203
19.350000	.698380	1.176471	.658679	14.663908	15.358679	1.181267	8.473579
19.350000	.698380	1.111111	.658679	14.662103	15.358679	1.184407	8.234839
19.400000	.700185	1.212121	.660484	14.663903	15.360484	1.184265	8.601009
19.400000	.700185	1.333333	.660484	14.662103	15.360484	1.187405	9.020814
19.450000	.701989	1.290323	.660484	14.662103	15.360484	1.187405	8.874124
19.450000	.701989	.655738	.660484	14.663908	15.360484	1.184265	6.326178
19.450000	.701989	1.149425	.658679	14.660299	15.358679	1.187548	8.375615
19.500000	.703794	.256410	.628001	14.674736	15.328001	1.111410	3.955889
19.500000	.703794	1.369863	.658679	14.660299	15.358679	1.187548	9.143552
19.500000	.703794	1.481481	.662288	14.662103	15.362288	1.190403	9.508773
19.600000	.707403	1.600000	.665897	14.663908	15.365897	1.193257	9.881807
19.600000	.707403	1.739130	.664093	14.660299	15.364093	1.196542	10.302495
19.600000	.707403	1.515152	.662288	14.658494	15.362288	1.196686	9.616270
19.600000	.707403	1.600000	.664093	14.660299	15.364093	1.196542	9.881807
19.700000	.711012	1.904762	.665897	14.662103	15.365897	1.196398	10.781935
19.750000	.712817	1.724138	.662288	14.658494	15.362288	1.196686	10.257992
19.800000	.714622	.465116	.649656	14.674736	15.349656	1.147436	5.327913
19.800000	.714622	2.000000	.664093	14.656690	15.364093	1.202825	11.048196
19.900000	.718231	.555556	.649656	14.671126	15.349656	1.153713	5.822910

Table 2 - 17.05 g COPPER MESH - WITHOUT OIL

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PIW	PIA	CYCLE	DPA(I)	P1(I)	P2(I)	ZN	F
17.350000	.626196	.476190	.631610	14.716241	15.331610	1.036039	5.390967
17.450000	.629805	.116279	.631610	14.707218	15.331610	1.051550	2.663957
17.450000	.629805	.101523	.629805	14.712632	15.329805	1.039265	2.489191
17.500000	.631610	.250000	.631610	14.700000	15.331610	1.063966	3.906127
17.500000	.631610	.232558	.633415	14.710828	15.333415	1.048321	3.767403
17.550000	.633415	.333333	.631610	14.700000	15.331610	1.063966	4.510407
17.550000	.633415	.238095	.635219	14.705414	15.335219	1.060606	3.811990
17.600000	.635219	.308642	.635219	14.707218	15.335219	1.057503	4.340141
17.600000	.635219	.322581	.635219	14.709023	15.335219	1.054400	4.437062
17.600000	.635219	.416667	.637024	14.710828	15.337024	1.054273	5.042788
17.650000	.637024	.444444	.635219	14.700000	15.335219	1.069919	5.208169
17.700000	.638828	.333333	.638828	14.703609	15.338828	1.069662	4.510407
17.700000	.638828	.416667	.637024	14.703609	15.337024	1.066686	5.042788
17.750000	.640633	.540541	.635219	14.696391	15.335219	1.076129	5.743683
17.750000	.640633	.512821	.638828	14.707218	15.338828	1.063455	5.594471
17.750000	.640633	.543478	.637024	14.700000	15.337024	1.072895	5.759270
17.800000	.642438	.526316	.642438	14.700000	15.342438	1.081821	5.667605
17.800000	.642438	.416667	.642438	14.703609	15.342438	1.075612	5.042788
17.800000	.642438	.657895	.638828	14.698195	15.338828	1.078976	6.335575
17.800000	.642438	.625000	.640633	14.707218	15.340633	1.066430	6.176129
17.800000	.642438	.615385	.638828	14.698195	15.338828	1.078976	6.128436
17.850000	.644242	.769231	.640633	14.707218	15.340633	1.066430	6.851800
17.850000	.644242	.547945	.637024	14.694586	15.337024	1.082211	5.782890
17.850000	.644242	.701250	.638828	14.696391	15.338828	1.082081	6.905122
17.900000	.646047	.888889	.640633	14.696391	15.340633	1.085056	7.365464
17.950000	.647851	1.010101	.640633	14.694586	15.340633	1.088162	7.851611
17.950000	.647851	.877193	.640633	14.696391	15.340633	1.085056	7.316946
18.000000	.649656	1.333333	.638828	14.694586	15.338828	1.085187	9.020814
18.000000	.649656	1.176471	.642430	14.696391	15.342438	1.088031	8.473579
18.000000	.649656	1.250000	.642438	14.692782	15.342438	1.094243	8.734366
18.000000	.649656	1.052632	.638828	14.694586	15.338828	1.085187	8.015203
18.050000	.651461	1.538462	.642438	14.696391	15.342438	1.088031	9.680909
18.100000	.653265	1.666667	.644242	14.692782	15.344242	1.097217	10.095577
18.100000	.653265	1.369863	.644242	14.692782	15.344242	1.097217	9.143552
18.100000	.653265	1.515152	.644242	14.694586	15.344242	1.094111	9.616220
18.100000	.653265	1.818182	.642438	14.692782	15.342438	1.094243	10.534041
18.150000	.655070	1.785714	.646047	14.692782	15.346047	1.100192	10.437564
18.200000	.656874	1.666667	.642438	14.692782	15.342438	1.094243	10.095577
18.200000	.656874	1.818182	.642438	14.692782	15.342438	1.094243	10.534041
18.200000	.656874	2.222222	.644242	14.694586	15.344242	1.094111	11.645821
18.200000	.656874	2.000000	.646047	14.692782	15.346047	1.100192	11.048196
18.200000	.656874	1.923077	.646047	14.692782	15.346047	1.100192	10.833647
18.250000	.658679	2.173913	.647851	14.694586	15.347851	1.100059	11.519540
18.300000	.660484	2.352941	.646047	14.689172	15.346047	1.106405	11.803451
18.350000	.662288	2.325581	.647851	14.692782	15.347851	1.103165	11.913576
18.350000	.662288	2.631579	.649656	14.692782	15.349656	1.106129	12.673140

STOP

Table 3 - 34.57 g COPPER MESH - WITHOUT OIL

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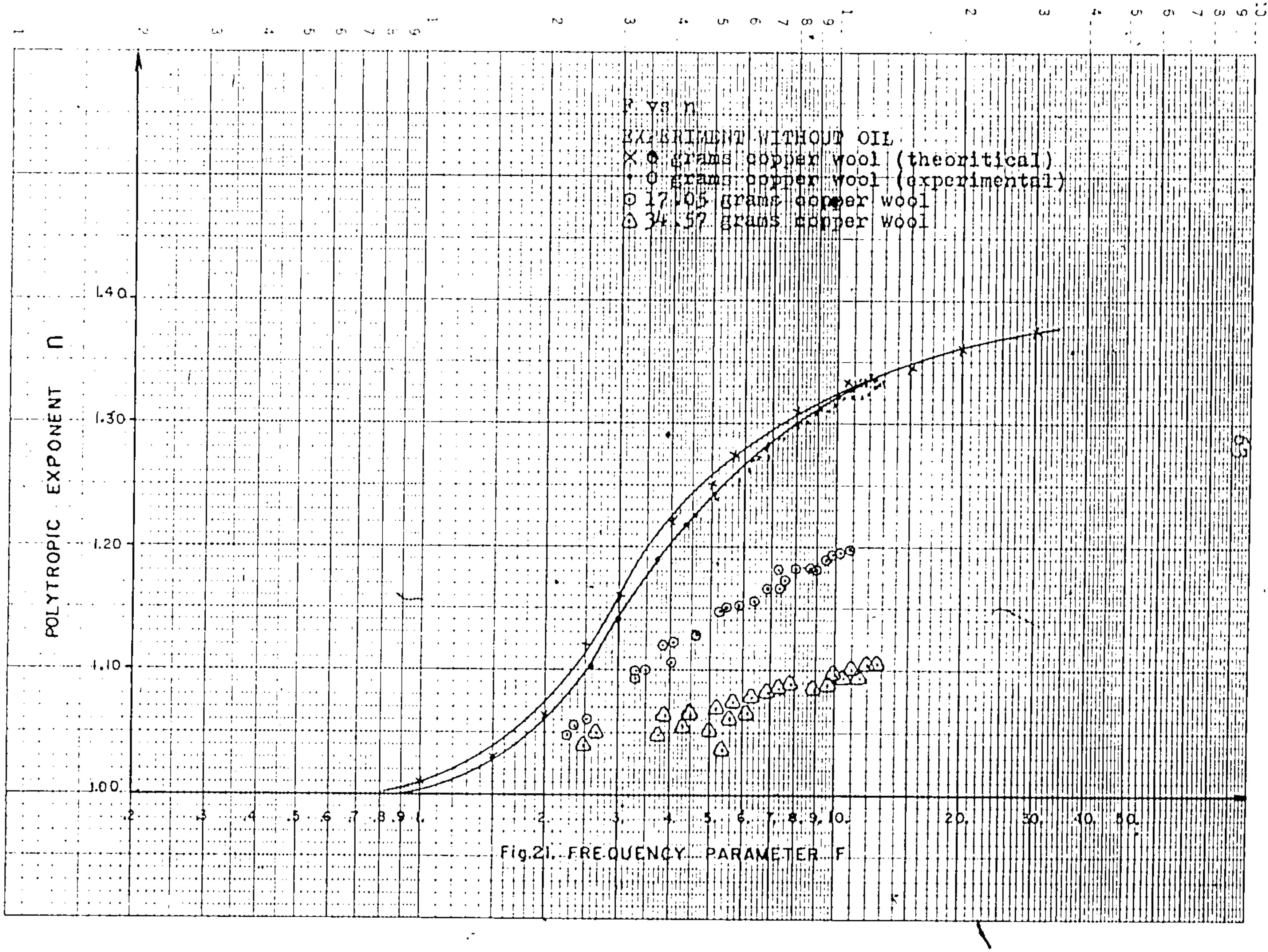

Fig.21. FREQUENCY PARAMETER F

CONCLUSION:

The experiment was set up to obtain the polytropic exponent of an cylindrical air-filled chamber as a function of frequency. The correlation between Daniels' theory and the experimental data on the polytropic exponent n as a function of the signal frequency F were examined. The two curves have the tendency to be closed to each other at values of the high frequency parameter F when the cylinder chamber contains only air. When filled with copper mesh and air, we observe that the polytropic exponent approached the isothermal conditions as the amount of copper increased. When the inner cylinder wall is wetted with a thin film of oil, the polytropic exponent n value is less than in the dry condition and we have a better simulation of our study.

5. APPLICATIONS

5.1 ENERGY STORAGE [7]

The accumulator is used to reduce the pump size. The hydraulic force in a machine is used to move four cylinders. From the time the first cylinder extends until all four have retracted, 10 seconds will have passed. Figure 22 shows the timing diagram for this machine. In the first two seconds cylinder one extends. In the second 2 seconds, the cylinder 2 extends. In the third 2 seconds, cylinder 3 extends. In the fourth 2 seconds, cylinder 4 extends and in the fifth 2 seconds, all four cylinders retract. To move the cylinder in 2 seconds, a certain flow rate (gpm) will be required. Cylinder 1 requires 3 gpm (11,356.24 cm^3/min) to extend, 2 gpm (7,570.82 cm^3/min) to retract; cylinder 2, 7 gpm (26,597.88 cm^3/min) to extend, 6 gpm (22,762.47 cm^3/min) to retract; cylinder 3, 5 gpm (13,248.94 cm^3/min) to extend, 4 gpm (15,141.65 cm^3/min) to retract; cylinder 4, 4 gpm (16,655.21 cm^3/min) to extend, 3 gpm (11,356.24 cm^3/min) to retract.

When these flow rates are projected down to a flow-demand chart (Figure 23), we find the first 2 second in the cycle will require 3 gpm (11,356.14 cm^3/min); the next 2 seconds, 7 gpm (26,479.88 cm^3/min); the third 2 second, 5 gpm (18,927.06 cm^3/min) ; the fourth 2 seconds, 4 gpm

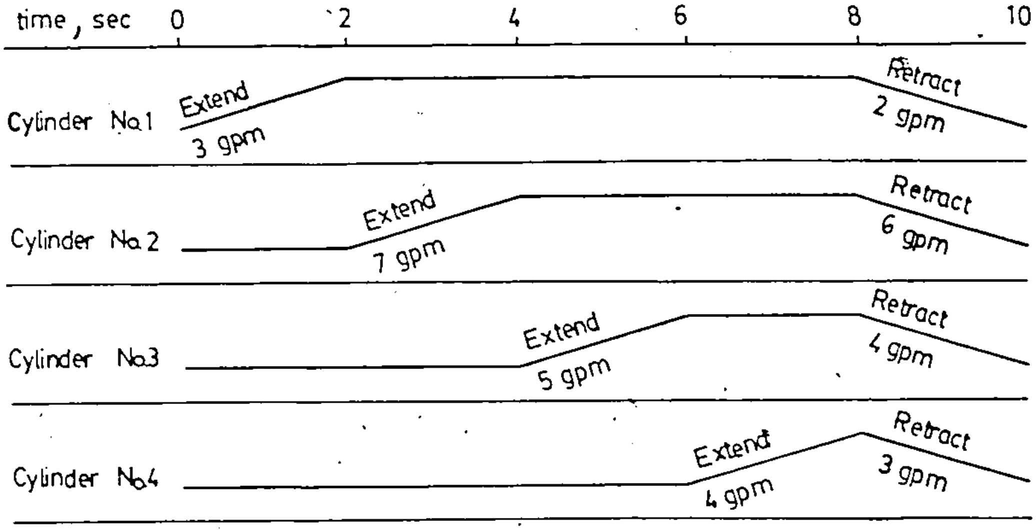

Fig. 22. Timing Diagram For Hydraulic Machine

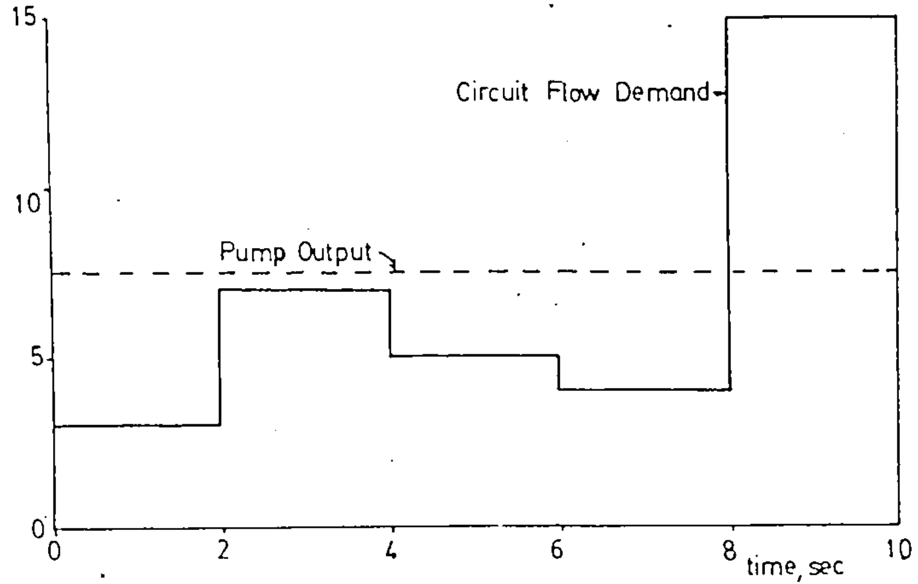

Fig. 23. Flow. Demand Chart

(15,141.65cm^3/min) and the last 2 seconds will demand 15 gpm (56,781.18 cm^3/min). Without an accumulator, the pump capacity will be determined by the peak demand. In this case it is 15 gpm. But this demand will only be for 20 per cent of the cycle. So if the pump was a fixed-displacement type, the relief valve will be discharging most of the hydraulic liquid for 80 per cent of the time. That represents a lot of horse power going into heat. Adding the gpm rates: 3 + 7 + 5 + 4 + 15, and dividing the sum by the five 2 seconds time periods, we find the average circuit demand is 6.8 gpm (25,740.80 cm^3/min).

In a particular pump catalog, the next size pump above the 6.8 gpm average is a 7.5 gpm pump (pumping against 1,000 psi (6,895.00 KPA)). This pump rate is designated with a dotted line across the demand chart, Figure 23 . When the circuit demand is below the pump rate line, the excess pump flow goes into the accumulator. But when the circuit demand rate is above the pump rate line, the pump and the accumulator supply the needed flow.

In this described circuit, only the last 2 seconds of the cycle are above the pump rate line, the pump and the accumulator supply 15 gpm. Thus during these 2 seconds, the accumulator must supply 15 gpm minus the pump rate of 7.5 gpm (28,390.59 cm^3/min) or 7.5 gpm (28,390.59 cm^3/min) to satisfy the circuit demand. Accumulator are generally sized in cubic inches. So the demand from the accumulator will be 7.5 gpm

(28,390.59 cm^3/min) multiplied by 231 (cubic inches per gallon), and divided by 30 (30 x 2 seconds periods in one minute), which is about 58 cu. in (950.45 cm^3/min). Now let us double check and make sure that 58 cu. in (950.45 cm^3/min) of hydraulic liquid will be put into the accumulator during the balance of the cycle. By a coincidence, the pump rate and the demand rate from the accumulator are the same, 7.5 gpm (28,390.59 cm^3/min) or about 58 cu. in (950.45 (cm^3/min) in 2 seconds.

In the first 2 seconds, while the pump is moving 58 cu. in.(950.45 cm^3/min) cylinder 1 is using 23 cu. in (376.90 (cm^3/min) (3 x 231/30). This means that 58-23 = 35 cu.in (573.54 cm^3/min) available for storage in the accumulator. When cylinder 2 moves in the second 2 seconds, it will require 54 cu. in. from the pump (7 x 231/30). This means that 58-54 = 4 cu. in. (950.45 - 884.90 = 65.55 cm^3/min) are available for storage in the accumulator. When cylinder 3 moves in the third 2 seconds, it will require about 38½ cu.in. (630.90 cm^3/min) (5 x 231/30). This means that 58 - 38½ = 19½ cu. in. (950.45 - 630.90 = 319.55 cm^3/min) are available for storage in the accumulator. When cylinder 4 moves in the fourth 2 seconds period, it will require about 31 cu. in. (508.00 cm^3/min) (4 x 231/30). This means that 58 - 31 = 27 cu.in. (950.45 - 508.00 = 442.75 cm^3/min) are available for storage in the accumulator.

When we add the four periods that have excess liquid for storage in the accumulator, we find the total to be around 85 cu. in. (1,392.89 cm^3/min) well above the circuit demand from the accumulator of 58 cu. in (950.45 cm^3/min). Of course, the pump will put no more flow into the accumulator than the accumulator puts out into the circuit. Thus the difference between what the pump has available and what the accumulator puts out is discharged through the relief valve.

5.2 COMPENSATION FOR EXPANSION [4]

In a closed hydraulic circuit, expanded fluid volume due to thermal expansion can readily increase system pressure beyond safety limits. To prevent this, a properly sized accumulator can be located in the system to absorb the increased volume of fluid and then return it into the line as system temperature goes down.

The factors which must be considered for calculation of accumulator sizing under the conditions described are shown in the following formula:

$$V_1 = \frac{V_a\ (t_2 - t_1)\ (\beta - 3a)\ (P_2 / P_1)^{1/n}}{1 - (P_2 / P_3)^{1/n}}$$

Where

V_1 = size of accumulator required, cu. in. This maximum volume occupied by the gas at precharge pressure.

P_1 = gas precharge pressure of accumulator, psi. This pressure must be less than or equal to minimum system pressure (P_2).

V_a = Total volume of fluid in the pipeline (area of pipe, sq.in. x pipe length, in.)

t_1 = initial temperature of the system, °F

t_2 = final temperature of the system, °F

P_2 = minimum system pressure at temperature t_1, psi.

P_3 = maximum system pressure at temperature t_1, spi

a = coefficient of linear expansion of pipe material, per °F

β = coefficient of cubical expansion of the fluid per °F

n = 1.4 for nitrogen

5.3 MAINTAINING CONSTANT PRESSURE [20] Figure 24

This can be required in one leg of a circuit while pump/ electric motor is delivering flow to another portion of the system.

In the circuit illustrated, two clamp cylinders are required to hold a part in place. As the directional valves are shifted, both cylinders extend and clamp at the pump's compensator setting. During this time, the accumulator is charged to the setting also.

System demands require that cylinder B maintain pressure while cylinder A retracts. As directional valve A is shifted, pressure at the pump as well as in line A drops quite low. Pressure at cylinder B is maintained because the accumulator has stored sufficient fluid under pressure to make up for any leakage in line B

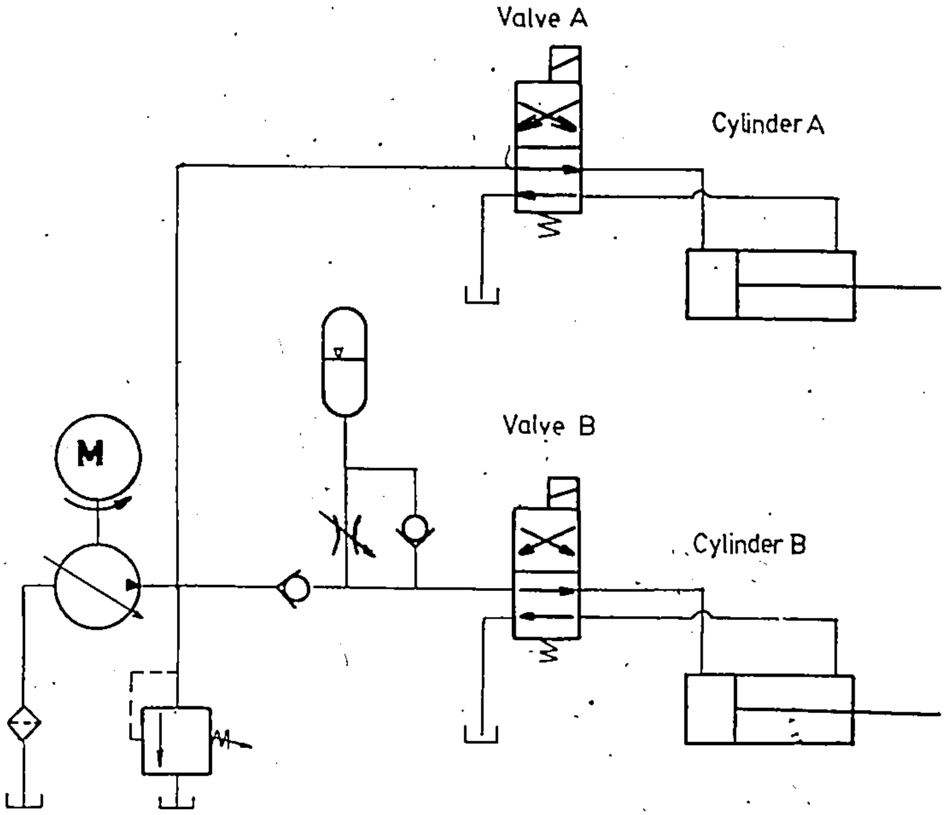

Fig.24. Maintaining Pressure

5.4. PRESSURE HOLDING AND LEAKAGE COMPENSATION [1] Figure 25

In a closed system where pressure must be held against the work by a holding ram for long periods while further duties in the operating cycle call for pump capacity, use of an accumulator to replenish lost oil through leakage is advantageous.

The accumulator in a blocked circuit eliminates the problems of holding pressure variations created by the varying demands of branch circuits on the pump in open-center systems. In addition, system leakages, which are normally present or which develop over a period of time are automotically taken care of.

When lengthy holding times are required, two or more hydraulically operated presses can be run economically with the use of accumulators.External or internal leakage through ram packings, valves or seals, results in piston creep and variation of the load on the work.

The accumulator compensates for such leakage, maintaining the correct loading for the required period of time. Providing each press and accumulator is isolated during the holding cycle, the system pump is free to meet the volumetric requirements of the other presses.

5.5 SHOCK SUPPRESSION [15]

This method of setting the capacity of the accumulator for a particular system assumes that friction losses are small enough to disregard and that the energy absorbed by compressing

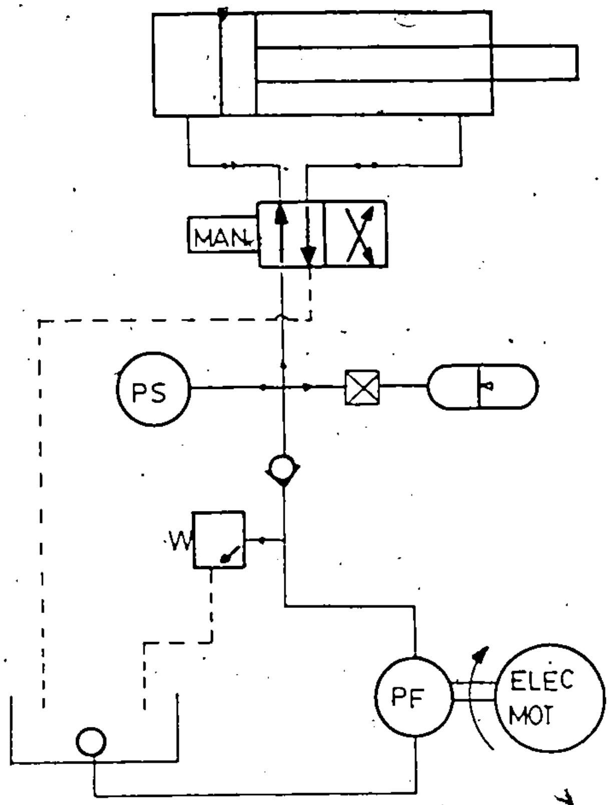

Fig.25. System For Holding Or Leakage Compensation

the column of fluid and expanding the pipe line is very small compared to the energy stored and absorbed by the accumulator. It too may be disregarded.

A fluid flowing through a pipe line has a definite amount of kinetic energy

$$K\,E = \frac{WaLv^2}{2g}$$

If fluid flow stops abruptly, the kinetic energy in the fluid must be transformed into another form of energy. This new form of energy is used up in doing work on the pipe by expanding it, on the equipment by straining it, or is dissipated through heat transfer.

If an accumulator with a capacity V_1 is installed upstream as close as possible to the rapidly closing valve, the kinetic energy present in the fluid before the valve closes can be transformed into potential energy, PE, stored in the compressible precharged gas of the accumulator. This occurs when the surge pressure exceeds the precharge gas pressure (also the normal system pressure) P_1 of the accumulator.

Suppose the allowable maximum surge pressure is P_2. Therefore, the kinetic energy present in the system before the valve closes must equal the energy stored in the accumulator gas volume V_1 between the pressure limits P_1 and P_2.

If the compression of the gas in the accumulator follows an adiabatic process (no heat added to or taken from the system) so that PV^n is a constant, then the energy stored in the accumulator may be expressed by

$$PE = -\int_1^2 Pdv \qquad (1)$$

We know

$$P_1V_1{}^n = P_2V_2{}^n = C \qquad (2)$$

substituting $P = C/V^n$ in $\int_1^2 Pdv$

$$PE = -C\int_1^2 dV/V^n$$

$$PE = -\left[\frac{CV^{1-n}}{1-n}\right]_1^2$$

$$PE = -C \ (V_2{}^{1-n} - V_1{}^{1-n}) \ / \ 1-n$$

$$PE = \frac{CV_1{}^{1-n} \ (1 - (V_2/V_1)^{1-n})}{1 - n} \qquad (3)$$

From equation (2)

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n} = \left(\frac{P_2}{P_1}\right)^{-1/n} \qquad (4)$$

substituting equation (4) into equation (3)

$$PE = \frac{P_1V_1{}^nV_1{}^{1-n} \ \ (1 - (P_2/P_1)^{\frac{n-1}{n}}}{1 - n}$$

$$PE = \frac{P_1V_1 \ (P_2/P_1)^{n-1/n} - 1}{n - 1} \qquad (5)$$

This energy is equal to the kinetic energy of the system before the valve closes, or

$$V_1 = \frac{12wALv^2 \ \ (n - 1)}{2gP_1 \ (P_2/P_1)^{\frac{n-1}{n}}} - 1 \qquad (6)$$

V_1 = Accumulator capacity, in^3

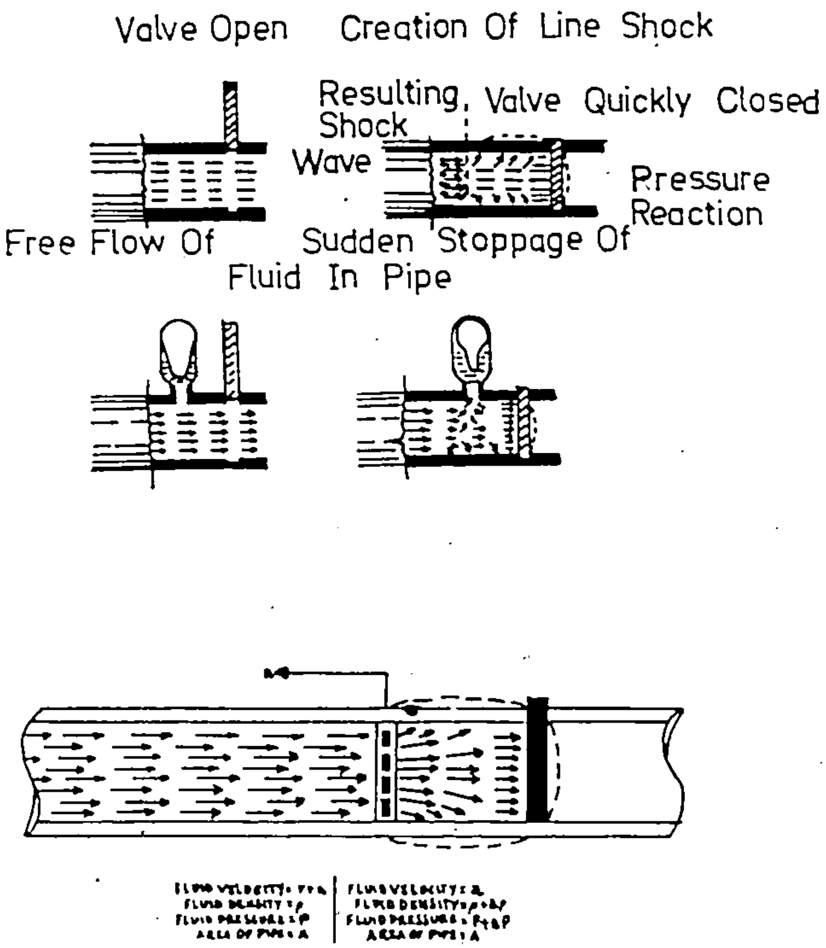

Fig.26. Line Shock and its absorption

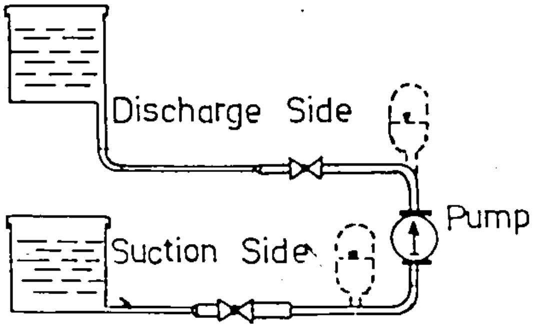

Fig.27. Pump Pulsation Dampening

For accumulators precharged to pressures P_x other than system pressure P_1, the required accumulator capacity V_x can be calculated from the following equation provided V_1 is already calculated

$$V_x = V_1 \left[\frac{P_1}{P_x} \right]^{1/n}$$

Figure 26

5.6 SURGE DAMPING [1]

An accumulator has the inherent property of eliminating pulsations of a frequency greater than the cut-off frequency of the device. The cut-off frequency of an accumulator can be calculated from the geometry fluid density and pressure.

$$\text{Cut-off frequency (radians per second)} = \frac{2}{\sqrt{\frac{\rho L}{A} \cdot \frac{dV}{dP}}}$$

Where

ρ = mass density of the fluid

L = length of fluid chamber

A = cross sectional area of fluid chamber

V = volume of fluid chamber

P = pressure

All gas-loaded accumulators will have a low cut-off frequency. Thus, they will pass all moderate to high frequency pulsations but tend to absorb all frequency pulsations below the cut-off frequency.

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This inherent characteristic is put to advantage to eliminate pump pulsations or pump ripple. In general, however, for satisfactory performance in this respect it is necessary to "size" the accumulator to provide a cut-off frequency higher than the known frequency of the pump. If necessary, two accumulators may be used, one on the inlet side and the other on the outlet side of the pump.

5.7 PUMP PULSATION ABSORPTION [9]

In the normal development of hydraulic power, various types of pumps are used. The modern pump itself takes many familiar forms, in each case delivering a pulsating flow, the degree of which varies with each type of pump. The higher the pressure, the greater is the effect of pulsations.

In piston-type pumps, which are by far the best of high pressure pulsations are most prevalent. The use of several piston pumps discharging simultaneously into the pipeline may create very severe pressure surges liable to cause damage or failure. Pulsations and pressure surges such as these can be minimized or even eliminated only by the use of a bag-type accumulator whose absence of inertia or friction permits the extremely rapid response essential for effective pulsation dampening.

To calculate accumulator size for specific pressure fluctuations, use:-

$$V_t = \frac{0.785 \; S F_1 B^2 \; (P_s/P_x)^{0.715}}{1 - (P_s/P_m)^{0.715}}$$

Where F_1 = constant for a particular type of pump, as follows:-

2

Pump type	Constant F_1
Simplex single	0.60
Simplex double	0.25
Duplex single	0.25
Duplex double	0.15
Triplex single	0.13
Triplex double	0.06

A simplified, more conservative formula may be used to reduce pump pulsations to within approximately ± 10 per cent of operating pressure.

$$V_t \text{ gal} = \frac{F_2 R}{N}$$

Where F_2 is a constant depending on pump type.

Values are:

Pump type	Constant F_2
Simplex single	5
Simplex double	2.5
Duplex single and double	1.3
Triplex single and double	0.45

The above equation is used, as is, for speeds up to 100 rpm. For speeds above 100 rpm, use $N = 100$ rpm (Figure 27).

5. 8 PULSATING HYDRAULIC SYSTEMS [5]

The most elemental form of a pulsating system is shown in Figure 28. A piston, driven by a scotch yoke, supplies oil to the system. Because of the form of drive, the oil flow is

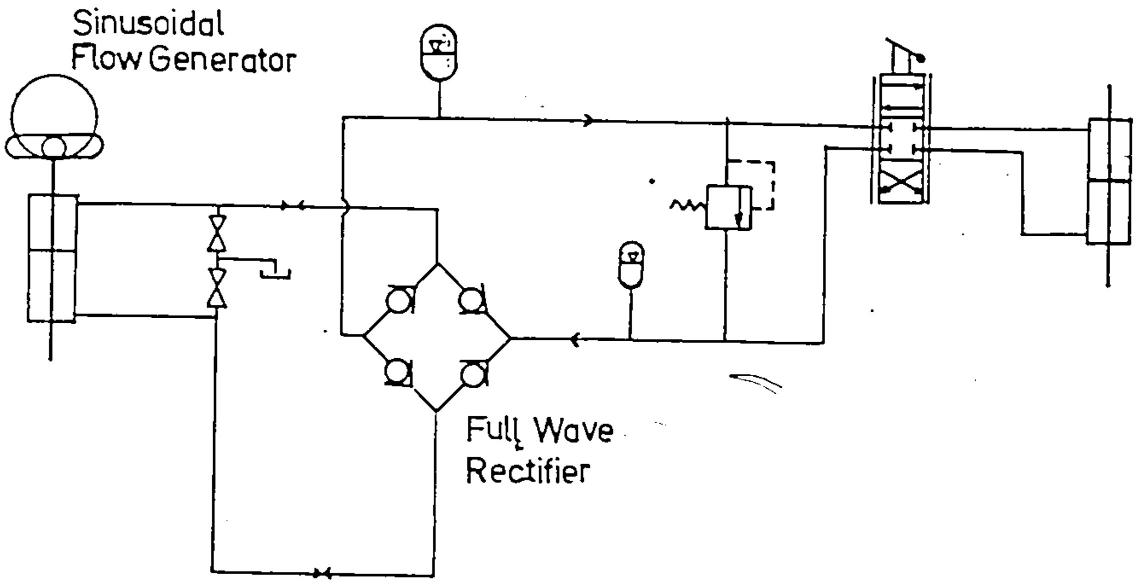

Fig.28. Basic Pulsating System

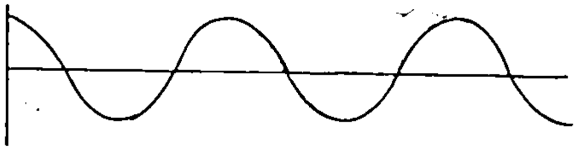

Fig.29.

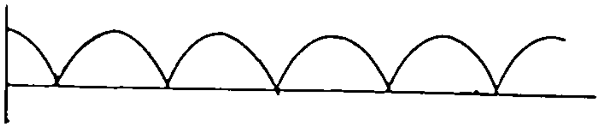

Fig. 30

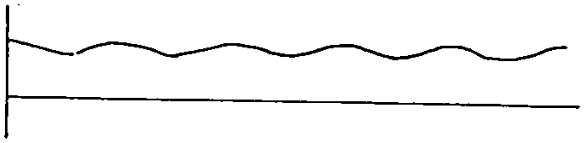

Fig.31

sinusoidal and alternating in direction as shown in Figure 29. The rate of drive determines the frequency of pulsation. The piston stroke determines the pulsation amptitude. With two transmission lines, the phase difference between the lines is 180°.

In order to use this pulsating flow in conventional out put actuators, the flow must be converted to continuous flow. This conversion is done in a ring demodulator which consists of a continuous series of half sinusoids, Figure 30. A smoothing fitter which consists of the fluid inertia and the capacitance of an accumulator changes the half waves to continuous rippled flow. Figure 31.

5.9 SELF-STARTING SYSTEM [5]

Hydraulic self-starting systems take advantage of the fact that most pumps can be operated as motors if the energy inputs were reversed. Instead of using an electrical battery to store energy for starting as does an automobile engine, the hydraulic system can store energy in an accumulator. In its simplest form, Figure 32, an accumulator can be charged initially by a hand pump and subsequently by a starter motor pump. The high pressure oil in the accumulator can be used to start the engine directly by being ported to the engine starter motor. If the engine is large, the accumulator oil is often use to start an auxiliary power unit (APU). The starter motor-pump on the APU is then operated as a pump to start the main engine and to recharge the accumulator.

This sort of starting system is particularly adaptable to large marine and aircraft jet engines. Such engines often have, in addition to an APU, an auxiliary gear box. Power from the engine shaft, operating through a clutch, then drives hydraulic pumps, electrical generators, etc. The accumulator supplies hydraulic power to start the APU by means of a pump/motor acting as a motor. The APU then drives the pump/motor as a pump to supply energy enough to the main engine motor to start the engine. System hydraulic and electrical power is then provided by pumps, pump/motors, and generators mounted on an accessory gear box. When the main engine is not running, the APU pump can supply oil directly or can drive other accessories through the pump/motor mounted on the gear box.

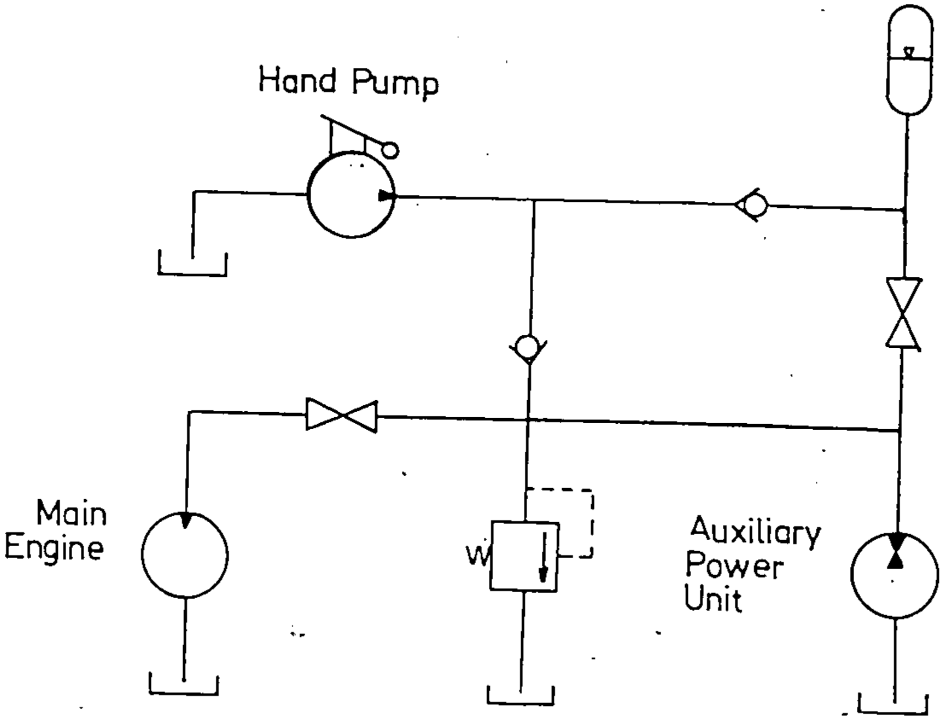

Fig.32. Self Starting System

5.10 HYDRO-PNEUMATIC SPRING [1] (Figure 33)

The installation of an accumulator in a rigid hydraulic system introduces hydropneumatic springing which can be used to advantage in many applications.

For instance, steel mill rolls or sugar mill cylinders are required to exert a constant pressure as material passes between them. If foreign matter, or oversize material, is introduced the rolls must move apart to prevent damage and automatically resume their normal positions at the required pressure. This springing action is accomplished by an accumulator or a series of accumulator if necessary, of sufficient capacity to absorb and release displaced fluid at almost constant pressure.

The rapid action of bag type accumulators due to lack of inertia, friction and "stiction" is advantageous, particularly where movements are small and even more important where pressure are low.

5.11 LIQUID-VAPOR ACCUMULATOR [16]

In some systems, it is necessary to develop pressure in one side of the circuit, and transfer the pressure as developed into another fluid without the possibility of having them intermix.

Figure 34 shows Greer-Mercier accumulators employed to ensure an emergency gland-sealing oil supply to a compressor or fan. In the event of pump failure, gas pressure forces

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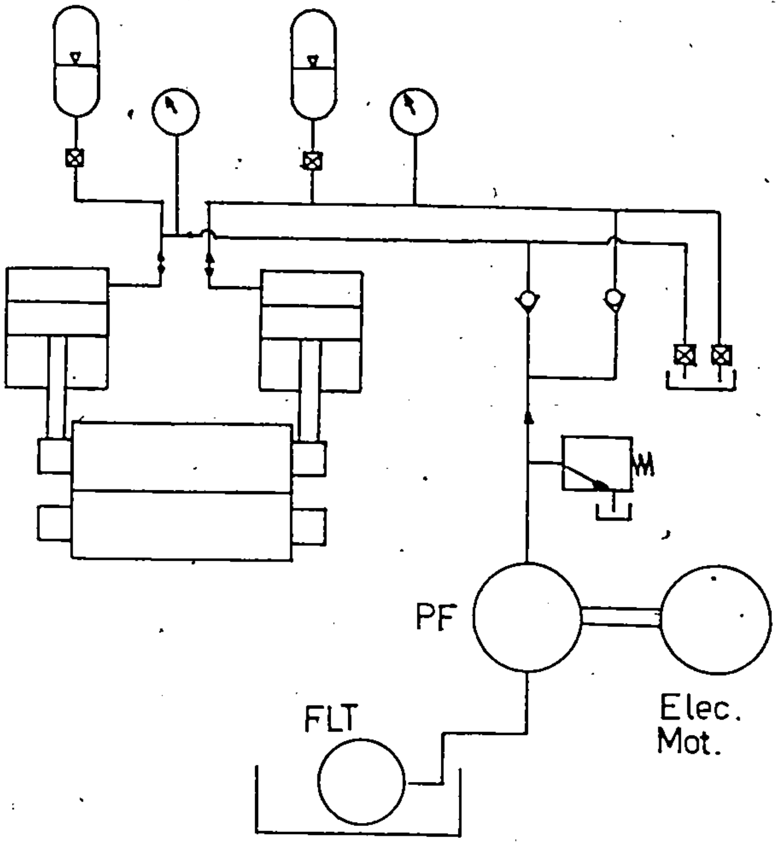


Fig.33. Loaded Mill Roll

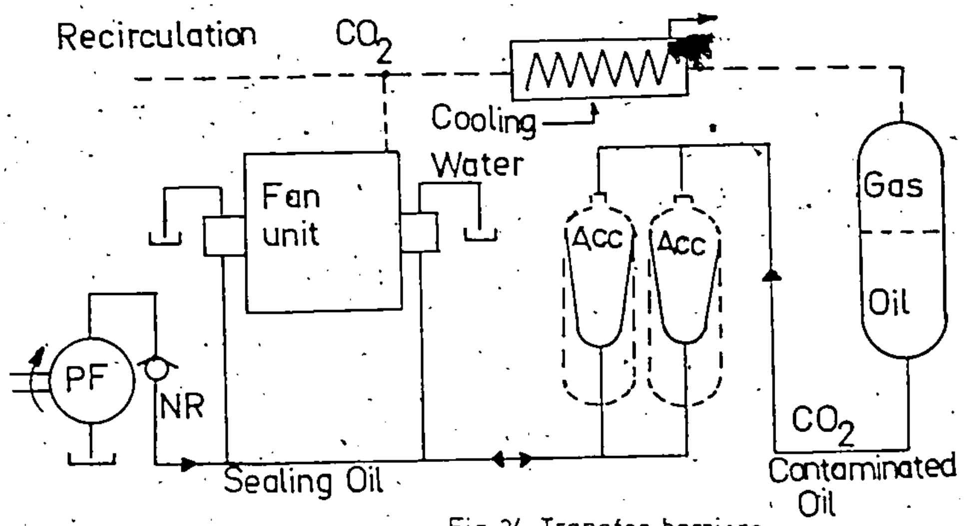


Fig.34. Transfer barriers

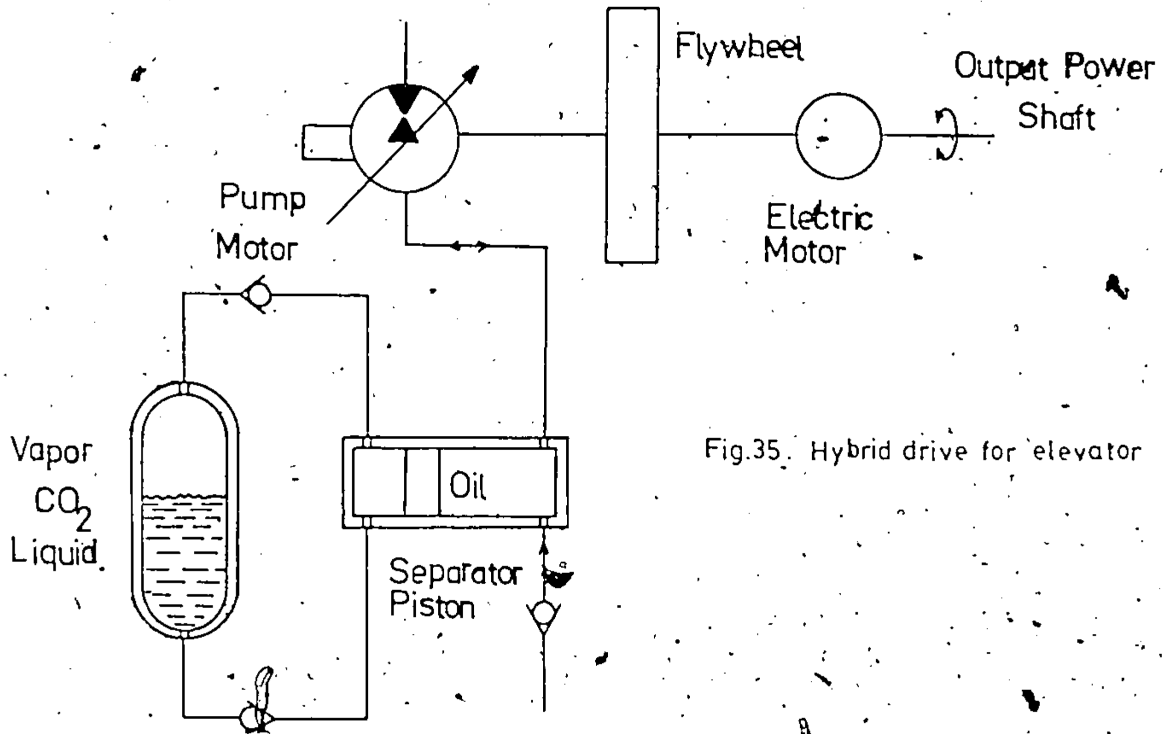


Fig.35. Hybrid drive for elevator

contaminated oil to expand the separator bladder, thus expelling clean oil into the system to seal the glands and prevent gas escaping into the plant.

The Hybrid CO_2 electric drive consists of an induction motor, flywheel, hydraulic motor, floating piston, and two-phase accumulator within a single drive package for an elevator pulley. The stored energy in the accumulator working through the separator piston drives the pump motor as a motor to accelerate the flywheel. Once up to speed the induction motor takes over and powers the elevator pulley drive shaft.

On slowdown, the flywheel energy is transferred to the pump motor, driving it as a pump and repressurizing the accumulator. The CO_2 vapor converts back to liquid when sufficient pressure is applied. Figure 35.

Another interesting design is the ocean wave absorber. As the float rises, it pressurizes the CO_2 vapor and condenses it in the accumulator. At the same time, the center piston drawn in oil from the reservoir.

When the ocean wave falls, the stored energy in the accumulator forces the center piston downwards and drives the hydraulic motor. A group of these drives produce continuous power so long as there were waves. Figure 36.

The hydraulic power stored in an accumulator can be used to actuate an artificial arm similar to the pneumatic

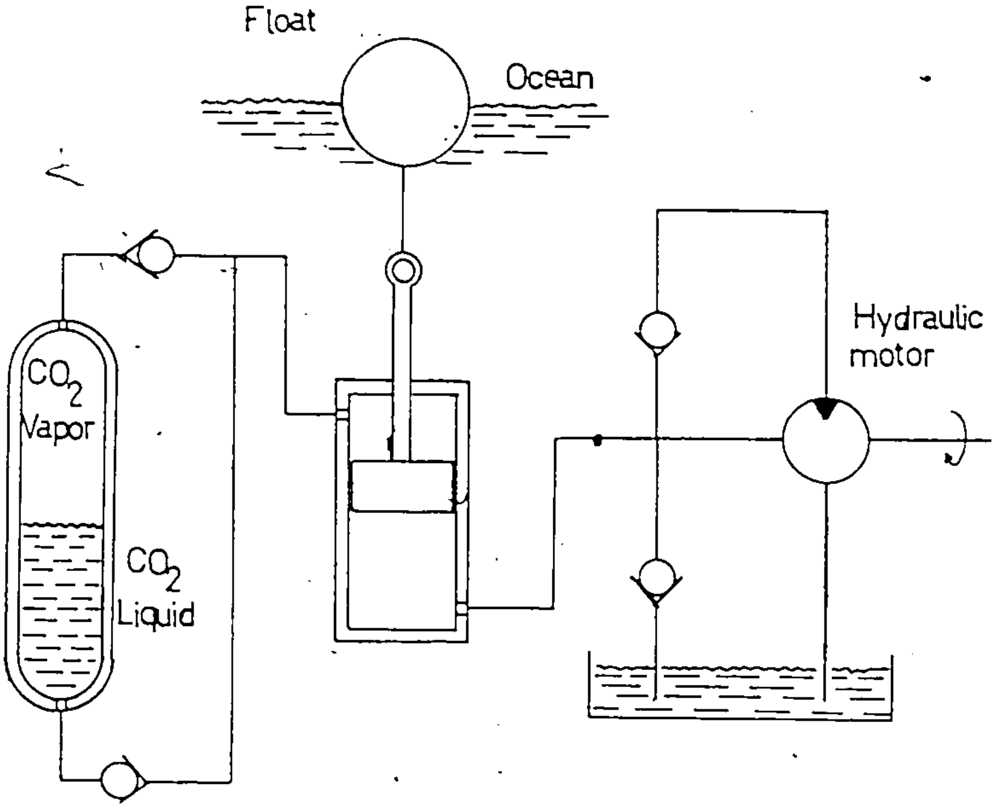
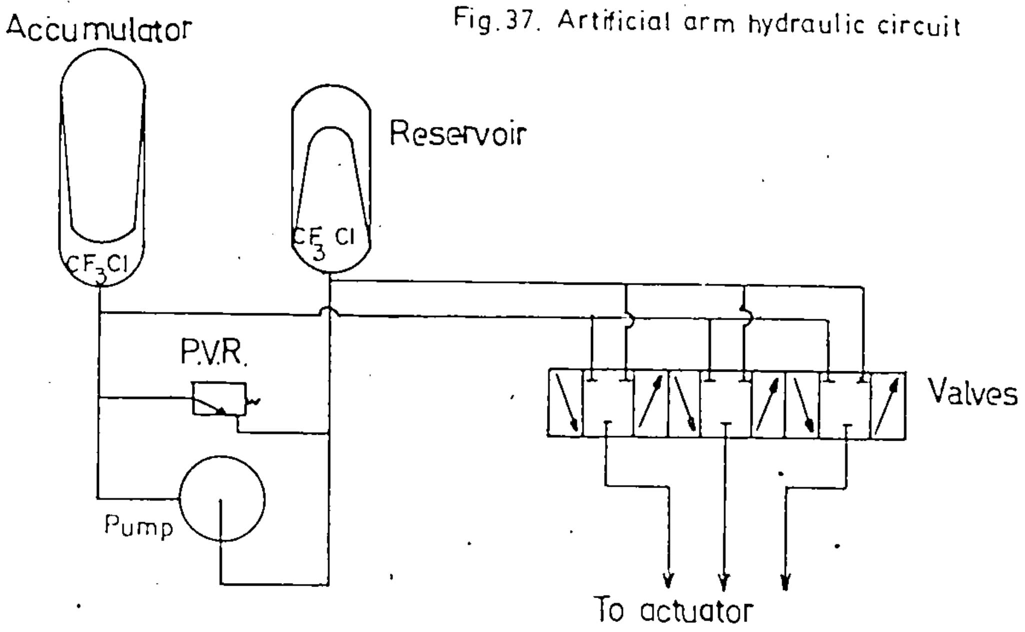


Fig .36. Ocean Wave ABSORBER

Fig.37. Artificial arm hydraulic circuit



arms currently in use, and provide the added advantages of increased stiffness, improved stability and faster response.

Fluid is drawn from the reservoir and pumped at high pressure through small reciprocating pumps in the heel of the shoe into an accumulator. When the maximum working pressure has been reached, excess fluid is discharged through a relief valve to the reservoir. The stored energy is distributed by the valve block to the arm actuators as required.

A liquifiable gas accumulator offer variations in delivery pressure. A volatile liquid will change its state within an accumulator depending on the temperature variation and rate of charge and discharge. The latent heat required by the fluid together with the heat available in the hydraulic fluid and container is therefore important in ensuring rapid phase change with little pressure variation. Very few fluids are available offering vapor pressures of up to 34 bar at normal room temperature and the only suitable one is trifluorochloromethane (CF_3Cl) which has a vapor pressure of 34 bar at 23.8°C. It is envisaged that this type of accumulator would comprise of an outer shell enclosing the liquifiable gas sealed inside a flexible membrane, ensuring a positive separation from the hydraulic fluid. Figure 37.

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accumulator sizing table

	OPERATING PRESSURE – PSI (gauge)																													
GAS PRECHARGE PRESSURE – PSI (gauge)	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000
100		95.6 124.0	126.0 169.0	159.0 192.0	174.0 207.0	185.0 215.0	192.0 222.0	200.0 228.0	205.0	209.0	212.0	215.0	219.0	221.0	223.0	225.0	226.0	228.0	230.0											
200			60.2 84.5	99.0 128.0	120.0 155.0	140.0 173.0	152.0 186.0	162.0 196.0	171.0 203.0	178.0 208.0	182.0 213.0	188.0 219.0	193.0 222.0	195.0 225.0	199.0 229.0	203.0 231.0	206.0	208.0	210.0	212.0	214.0	215.0	217.0	218.0	219.0	221.0	222.0	223.0	224.0	225.0
300				47.3 63.9	78.6 104.0	101.0 132.0	116.0 147.0	131.0 163.0	141.0 175.0	150.0 182.0	158.0 191.0	164.0 197.0	170.0 203.0	175.0 207.0	179.0 211.0	183.0 214.0	186.0 217.0	190.0 220.0	192.0 222.0	195.0 224.0	197.0 226.0	199.0 228.0	201.0 230.0	203.0	205.0	207.0	208.0	210.0	211.0	212.0
400					37.9 51.6	64.5 86.6	85.4 112.0	101.0 131.0	114.0 146.0	125.0 157.0	134.0 167.0	142.0 175.0	149.0 182.0	155.0 188.0	160.0 194.0	164.0 198.0	169.0 202.0	173.0 205.0	177.0 208.0	179.0 211.0	182.0 213.0	185.0 216.0	187.0 218.0	190.0 220.0	192.0 222.0	194.0 224.0	195.0 225.0	198.0 227.0	200.0 228.0	201.0 229.0
500						31.7 42.4	55.4 74.4	74.0 97.9	88.8 116.0	101.0 131.0	112.0 142.0	121.0 152.0	129.0 162.0	136.0 169.0	142.0 175.0	147.0 181.0	152.0 186.0	155.0 190.0	161.0 194.0	164.0 198.0	168.0 201.0	172.0 204.0	175.0 207.0	177.0 209.0	180.0 212.0	182.0 212.0	184.0 215.0	186.0 217.0	188.0 219.0	192.0 220.0
600							27.1 37.2	48.1 65.1	64.8 87.0	79.5 105.0	91.6 119.0	102.0 131.0	111.0 142.0	119.0 150.0	126.0 158.0	132.0 165.0	138.0 171.0	142.0 176.0	147.0 180.0	151.0 185.0	155.0 188.0	159.0 192.0	162.0 195.0	164.0 199.0	168.0 201.0	171.0 202.0	173.0 206.0	175.0 208.0	178.0 210.0	180.0 212.0
700								23.5 33.0	42.6 58.0	58.5 78.5	71.8 97.9	83.4 109.0	93.5 121.0	102.0 131.0	110.0 140.0	117.0 148.0	123.0 155.0	129.0 161.0	134.0 167.0	138.0 171.0	142.0 175.0	146.0 180.0	150.0 184.0	154.0 187.0	157.0 190.0	160.0 193.0	163.0 196.0	165.0 198.0	168.0 201.0	170.0 203.0
800									21.0 29.0	38.6 52.5	53.0 71.3	65.5 87.5	76.6 101.0	86.2 113.0	94.6 123.0	103.0 131.0	109.0 139.0	115.0 147.0	121.0 153.0	125.0 158.0	131.0 162.0	135.0 166.0	139.0 172.0	142.0 175.0	145.0 180.0	149.0 183.0	152.0 186.0	155.0 189.0	158.0 192.0	160.0 194.0
900										19.1 26.2	34.8 47.6	48.2 65.4	60.3 80.9	70.5 94.1	80.1 105.0	88.3 115.0	95.7 124.0	103.0 132.0	109.0 139.0	114.0 145.0	119.0 151.0	124.0 155.0	129.0 161.0	132.0 165.0	135.0 169.0	140.0 172.0	142.0 175.0	146.0 179.0	149.0 182.0	152.0 185.0
1000											17.3 23.5	32.6 42.6	44.7 62.6	55.0 75.4	65.0 87.8	74.2 99.9	82.6 109.0	89.9 117.0	95.5 125.0	103.0 132.0	108.0 138.0	113.0 144.0	119.0 149.0	122.0 154.0	125.0 159.0	129.0 163.0	134.0 167.0	136.0 170.0	140.0 172.0	143.0 176.0
1100												15.7 22.5	29.5 42.4	41.2 55.4	52.2 72.5	61.7 82.4	72.0 93.1	77.6 102.0	84.8 111.0	91.2 119.0	97.4 125.0	103.0 132.0	108.0 138.0	112.0 143.0	115.0 148.0	120.0 153.0	124.0 157.0	128.0 161.0	131.0 164.0	135.0 167.0
1200													14.6 20.5	27.4 37.8	38.6 52.6	48.9 66.3	57.5 75.5	65.9 88.0	72.4 97.4	80.7 106.0	86.6 112.0	92.2 120.0	97.4 127.0	103.0 132.0	107.0 138.0	112.0 142.0	115.0 147.0	120.0 152.0	122.0 156.0	126.0 159.0
1300														13.6 18.9	25.5 35.1	35.9 49.2	45.2 61.7	54.3 72.1	62.2 82.2	69.5 92.5	76.2 101.0	82.1 109.0	86.0 115.0	93.0 121.0	99.0 127.0	103.0 132.0	107.0 137.0	111.0 142.0	114.0 146.0	118.0 150.0
1400															12.8 17.8	23.9 33.0	34.0 44.6	42.1 58.5	51.1 67.2	59.0 79.2	65.3 83.3	72.5 93.2	78.5 103.0	83.6 110.0	89.0 116.0	93.8 122.0	98.0 127.0	103.0 130.0	107.0 135.0	110.0 141.0
1500																11.7 16.5	22.4 27.9	31.9 42.5	40.7 55.3	48.9 68.0	56.0 75.5	62.8 81.4	69.2 92.6	75.0 98.9	80.3 105.0	85.6 112.0	89.8 117.0	94.6 123.0	97.5 129.0	102.0 132.0
1600																	11.2 15.7	21.3 29.5	30.3 41.7	38.6 52.6	40.2 62.7	50.2 71.8	57.8 80.3	65.0 83.0	71.5 94.9	75.6 101.0	82.1 108.0	85.7 112.0	91.1 119.0	95.2 124.0
1700																		10.4 14.6	20.0 27.7	28.7 29.6	30.4 52.0	44.2 60.1	50.8 68.6	57.5 77.1	62.4 84.5	68.6 93.2	73.9 97.8	79.0 104.0	83.0 109.0	87.7 114.0
1800																			9.9 13.8	19.1 25.6	27.4 37.7	34.8 47.8	47.3 57.7	48.9 65.9	54.8 72.8	60.6 81.7	65.0 84.2	71.0 94.4	75.6 100.0	80.0 106.0
1900																				9.2 12.0	18.3 25.3	26.3 35.2	33.5 46.0	40.4 55.0	45.6 62.4	52.7 71.3	58.0 78.0	62.3 84.9	68.4 91.2	72.9 98.8
2000																					9.1 12.5	17.5 24.2	25.0 34.6	32.5 44.4	38.8 52.9	45.2 61.2	50.8 68.6	56.0 75.2	61.2 81.9	66.0 87.2
2100																						8.5 12.0	16.5 22.9	23.9 33.0	30.6 42.0	37.2 50.8	43.1 58.7	48.3 66.0	54.0 72.3	59.1 79.2
2200																							8.3 11.7	16.0 22.1	22.7 31.9	29.5 42.7	35.9 49.0	41.9 56.6	47.1 63.8	52.4 70.7
2300																								8.0 11.2	15.4 21.3	22.1 32.6	28.5 37.2	34.1 47.1	40.2 54.2	45.2 61.7
2400																									7.7 10.6	14.6 20.5	21.3 29.2	27.4 37.8	33.0 45.9	38.8 52.3
2500																										7.2 10.2	14.1 19.7	20.5 29.5	26.6 35.7	32.5 41.6
2600																											6.9 9.8	13.6 18.9	19.7 27.2	25.5 35.4
2700																												6.7 9.6	13.3 18.2	19.9 25.7

1 GALLON SIZE
(231 Cubic Inch Capacity)
PERFORMANCE TABLE (Adiabatic and Isothermal)

Max. Gas Capacity (piston against hydraulic cap) = 256 Cu. In.
Max. Oil Capacity (piston against gas cap) = 231 Cu. In.
Adiabatic = Bold Type
Isothermal = Italic Type

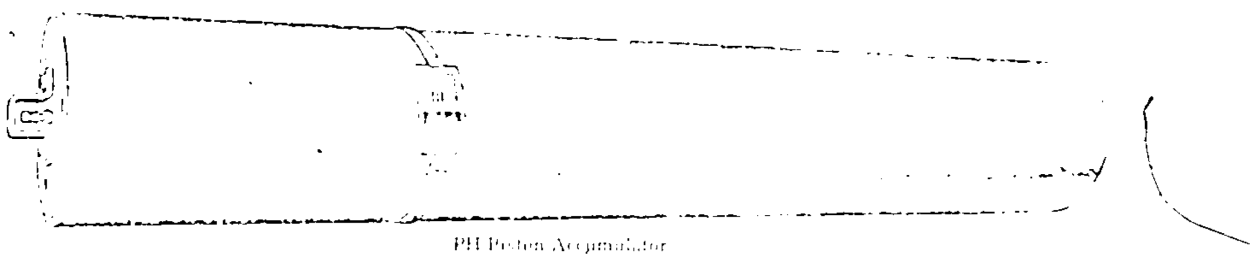
ISOTHERMAL vs. ADIABATIC

Pressure change of the gas is inversely proportional to its change in volume. When 100 cu. in. of gas originally at 1000 psia is compressed to 50 cu. in. volume, pressure will be 2000 psia if gas temperature is kept constant. This is ISOTHERMAL performance.

Compression and expansion of the gas cause heating and cooling which increase and decrease pressure in addition to the effect of volume change. If gas were perfectly insulated to prevent any [illegible] of this [illegible] heat to or thru the metal in which it is contained, or picking up heat [illegible] performance would be ADIABATIC. Here 100 cu. in. of gas originally at 1000 psia [illegible] compressed to 50 cu. in. [illegible] 2000 psia. There [illegible] the accumulator. The [illegible] construction of the Parker accumulator [illegible] this effect.

Actual performance [illegible] between isothermal and adiabatic [illegible] figures [illegible] isothermal. The isothermal figures [illegible] allowance [illegible] the effect of temperature changes resulting from [illegible]

table 6-2



PH Piston Accumulator

illustration 6-1

6. SELECTION

6.1 CHARTED METHOD

a) Piston accumulator

Parker-Hannifin gives us a hint how to select the size of the accumulators through tables.

Amount of oil contained in the accumulator in the table is based on the absolute pressure - psia (gage + atmospheric) - related to the gage pressure - used in the column heading. For example (1 gallon capacity) (3,785.41 cm^3), at 1,000 psig (6,895.00 KPA) gas precharge, 2,000 psig (13,790.00 KPA) operating pressure, the accumulator contains 132 cubic inches (2,163.08 cm^3) of oil (isothermally). If purpose of accumulator is to store oil for requiring 1,500 psig (10,342.50 KPA), the amount of oil available is the difference between that contained at 2,000 psig (13,790.00 KPA) and at 1,500 psig (10,342.50 KPA). (At 2,000 psig, (13,790.00 KPA), 132 cu. in. (2,163.08 cm^3)) - (at 1,500 psig, (10,342.50 PKA), 87.8 cu. in (1,438.78 cm^3)) = 44.2 cu. in. (724.30 cm^3) oil available (isothermally). It means we are dealing with an accumulator which has a maximum working pressure 2,000 psig and a minimum working pressure 1,500 psig (P_2). The volume of oil available is what need the system demand.

b) bladder accumulator Figure 38

The graph shown in Figure 38 is from a supplier's catalog. It is for a gas type bladder accumulator and has a gas-oil capacity of 200 cu. in (3,277.40 cm^3). When

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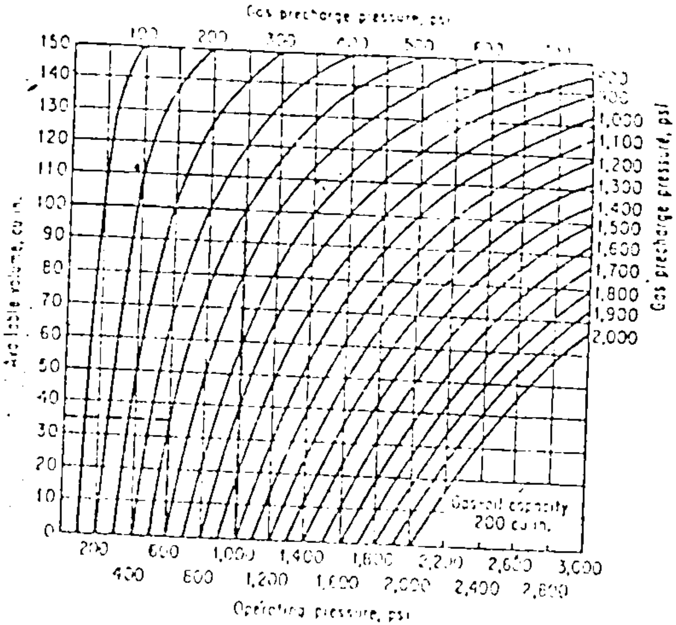


Chart for sizing bladder type hydraulic accumulator.

Figure 38

using the graph as example, two conditions are known:- Working pressure is 600 psi (4,137.00 KPA) and the accumulator size must be adequate to supply 58 cu. in ($950.45\ cm^3$) above the working pressure. We must determine from the chart how high the pressure in the accumulator must be to deliver 58 cu. in ($950.45\ cm^3$) of liquid above the working pressure.

We should remember that a bladder accumulator may not be completely discharged on every cycle because of possible damage to the bladder.

So choosing a precharge gas pressure of 500 psi (3,447.50 KPA) project the 600 psi (4,137.00 KPA) vertical pressure line upward until it intersects the 500 psi (3,447.50 KPA) curved precharge line. From the intersection, project a line to the left and read the number of cubic inches, which, in this case, is 36 ($589.93\ cm^3$). So add 58 cu. in. ($950.45\ cm^3$) in to 36 ($589.93\ cm^3$) and the answer is 94 cu. in. ($1,540.38\ cm^3$). The next horizontal line above 94 ($1,540.38\ cm^3$) is 100 cu. in. ($1,638.70\ cm^3$), now follow this line to the right until it intersects the 500 psi (3,447.50 KPA) curved gas precharge. From this point go down vertically and read 1,000 psi (6,895.00 KPA), supposed it falls in the range of the working pressure of the system.

The difference between these two cubic inch readings is the amount of liquid available from the accumulator as the pressure drops from p_3 = 1,000 psi (6,895.00 KPA) to p_2 = 600 psi (4,137.00 KPA). In this case, it will be 100 - 36 = 64 cu. in. ($1,048.77\ cm^3$). Since only 58 cu. in. ($950.45\ cm^3$)

are required, the 200 cu. in. (3,277.40 cm^3) accumulator with a precharged gas pressure of 500 psi (3,447.50 KPA) will do the job.

In fluid Power Handbook Engineering Data, we find a method to size the accumulator **through** equations and graph (Figure 39).

$$V_1 = \frac{V_x \quad (P_3/P_1)^{1/n}}{1 - (P_3/P_2)^{1/n}} \qquad (1)$$

$$R = P_2/P_1$$

$$V_1 = V_x \left\{ 1 + [1 /(R - 1)] \right\}$$

Where:

V_1 = size of accumulator necessary, in.^{3}

V_x = volume of fluid discharged from accumulator, in.^{3}

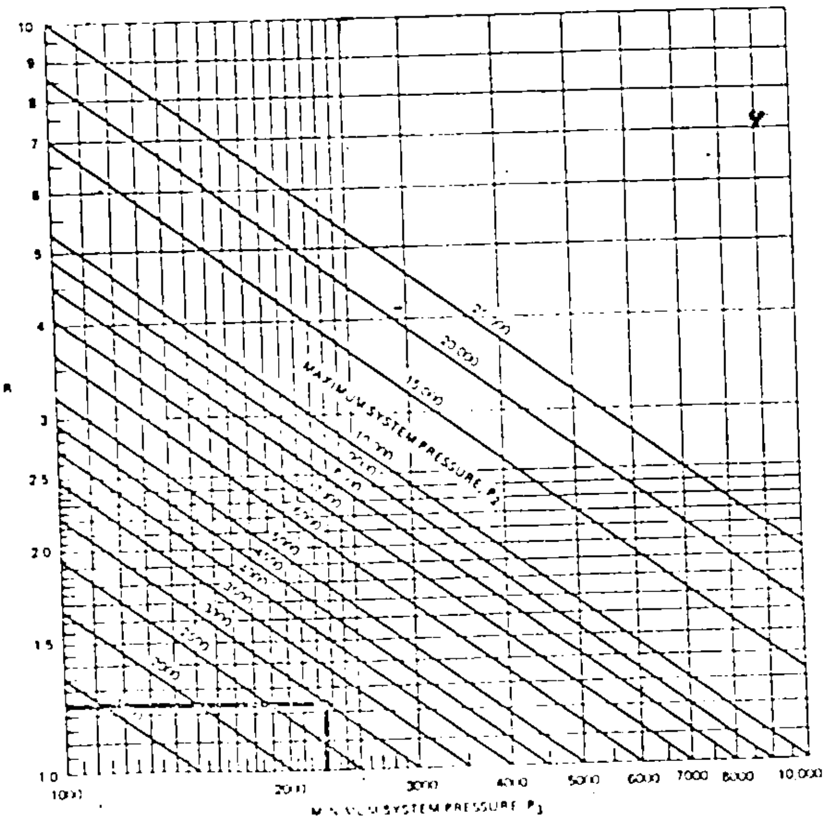
P_1 = gas precharge of accumulator, psi.

P_2 = maximum system operating pressure, psi.

V_2 = compressed volume of gas at minimum pressure, psi.

P_3 = minimum system pressure at which additional volume of fluid is needed, spi.

V_3 = expanded volume of gas at minimum system pressure, in.^{3}



1976-77 Fluid Power Handbook & Directory, A.67

Figure 39

These curves can be used to find any two of the parameters of equation 1, provided the other three are known. Assumptions are that conditions are adiabatic, and that P_1 and P_2 are at the same gas temperature.

For example, consider a system where maximum operating pressure required is 3,000 psi (20,685.00 KPA) minimum system pressure after discharge is 2,250 psi (15,513.75 KPA) and system fluid demand is 100 cu. in.(1,638.70 cm^3). What will be the required precharge pressure and accumulator size?

Find $P_3 = 2{,}250$ psi (15,513.75 KPA) on the x-axis and follow this line vertically until it reaches the line $P_2 =$ 3,000 psi (20,685.00 KPA). Project a line left to the y-axis and read $R = 1.24$. Precharge pressure P_1 then equals P_2/R, or 3,000 psi (20,685.00 KPA)/1.24 = 2,420 psi (16,685.90 KPA) Accumulator will be $V_1 = 100[1 + (1/.24)] = 580$ cu. in. (9,504.46 cm^3).

Christie Hydraulics Ltd. gives us a way to select accumulator through a chart as followed.

A simple graph as shown in Figure 40 has been devised which is universal for all gas loaded accumulators. The only calculation required is the division of the maximum pressure P_3 by the minimum pressure P_2 to find a pressure ratio. For example, if $P_3 = 3{,}000$ lb/in^2 (20,685.00 KPA) and $P_2 = 2{,}500$ lb/in^2 (17,237.50 KPA) then the gas precharge will be 0.9 x2,500= 2,250 lb/in^2 (15,513.75 KPA) and the pressure ratio will be 1.2.

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ACCUMULATOR SELECTION GRAPH
SHEET NO. 1

REGISTERED OFFICE & WORKS,
SANDYCROFT INDUSTRIAL ESTATE,
CHESTERROAD, SANDYCROFT, DEESIDE CLWYD,
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Using Figure 40 locate 1.2 on the vertical axis and take a horizontal line across the curve. Project this line vertically downwards and it will be seen to cut the horizontal axis at 10.5%. Check the manufacturers literature to find a column headed "Actual gas Volume". This is the volume of the accumulator at condition V_1. If this column is not given take the nominal volume of an accumulator, eg. 20 litres. Multiply this figure by 0.105, to find the actual volume of oil stored between the pressures of 3,000 lb/in^2 (20,685.00 KPA) and 2,500 lb/in^2 (17,230.50 KPA). A further refinement of the graph is that by following the line vertically to the top of the page the actual volume stored in cubic inches can be read off for all accumulators manufactured by Christie Hydraulics Ltd.

Conversely, if it is found that a system requires 200 cubic inches (3,277.40 cm^3) of oil at a maximum pressure of 3,000 lb/in^2 (20,685.00 KPA), it is desirable to know the minimum pressure the system will fall to. A glance at the top of the curve shown in Figure 40 will show that there are four places where 200 cubic inches (3,277.40 cm^3) appears on the horizontal scale. Starting at the top left there is the model AC54, then below to the right the AC37 and so on down to the AC10. If the acceptable pressure drop is very small then select figures well to the left, if a large pressure drop can be tolerated, then select figures more to the right. In our example we could choose a middle of the road size such as the AC20, which, when we drop the line vertically to the curve and then project horizontally to the pressure ratio scale, we have a figure of 1.4. Therefore, when operating at a pressure of 3,000 lb/in^2 (20,685.00 KPA) the accumulator pressure will drop to 3,000/1.4 = 2,143 lb/in^2 (14,775.98 KPA) when 200 cubic inches of fluid has been removed from it. The precharge pressure can now be fixed at 0.9 x 2143 (14,775.98 KPA) = 1,929 lb/in^2 (13,300.45 KPA).

Should it be found that it is unacceptable to allow the pressure to drop to 2,143 lb/in^2 (14,775.98 KPA) then the next larger size will have to be considered. The AC37 size would give a pressure ratio of 1.17 which in turn gives a minimum pressure of 2,564 lb/in^2 (17,678.78 KPA), when used with a precharge of 2,308 lb/in^2 (15,913,66 KPA).

7. SUMMARY

We have reviewed the notion of accumulators. Thereby accumulator means energy-storer; it is a tank or chamber built to hold reserve fluid at system pressure which is immediately available to provide or absorb momentary flow. It uses the spring effect to release the energy to the system.

We have described different kinds of accumulators:- Weight-loaded accumulators, Spring-loaded accumulators, Gas-loaded non-separated accumulators and Gas-loaded separated accumulators.

We mentioned some ways for analyzing and selecting a proper accumulator for a specific system of fluid distribution.

An experiment was held for showing the correlation between Daniel's theory and the experimental data on the polytropic exponent n as a function of the signal frequency F. We observed that the two curves have the tendency to be closest to each other at high frequency parameter F while the cylinder chamber is empty. When filled with copper mesh we observed that n approaches the isothermal conditions as the amount of copper increases. When the cylinder wall was wetted with a thin film of oil, the n value is lower compared to the dry condition. With the oil impregnated, we have a better simulation of our study. The same study has been made by J. Hulet with steel wool and we observed the same behaviour for roughly the same amount of stuffing materials.

Many applications of the accumulator have been discussed.

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9. APPENDIX A

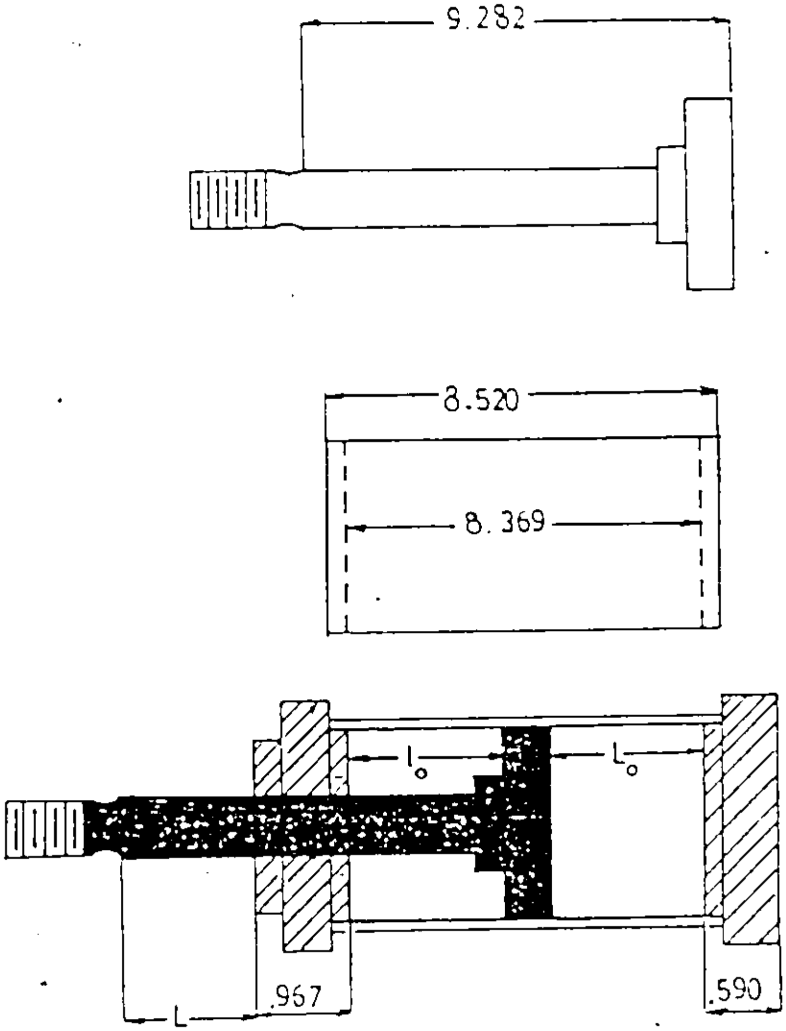

Fig. II. Piston in Initial Position

From Daniel's Theory

$$F = W^{.5}\left(\frac{r_w^{\,2} \times Np}{\gamma}\right)^{.5}$$

Where, F = frequency radius parameter

w = frequency rad/sec

r_w = radius of cylinder ft

N_p = Prandtl number

γ = kinematic viscosity ft^2/sec

for air, $N_p = .714$

$\gamma = 160 \times 10^{-6}$

$r_w = 1.120/2$ $.\frac{560}{12}$ ft

$$F = w^{.5}\left(\frac{(.560)^2\ (.714)}{(160 \times 10^{-6})(144)}\right)^{.5} = 9.7^{.5}W^{.5}$$

$$= 3.115\ W^{.5}$$

$$F^2 = 9.7 \times W$$

$$W = .1031\ F^2$$

$$f = \frac{.1031\ F^2}{6.28}$$

$$f = .01642F^2 \rightarrow F = \sqrt{f/.01642}$$

Where, f = frequency

EXPERIMENT I ONE STUFF - WITH OIL

$L = 5.769$ in.

$ex = .232$ in.

$l_o = 9.282 - (5.769 + .967) = 2.5460$ in.

$L_o = 8.369 - 2.5460 = 5.823$ in.

$V_o = .985203\ (5.823) + .09640$

$= 5.736837 + .09640 = 5.833237$ in.^{3}

$V = .985203\ (5.823 - .232) + .09640$

$= 5.508270 + .09640 = 5.604670$ in.^{3}

$$\frac{V}{V_o} = \frac{5.604670}{5.833237} = .960816$$

$$\ln \frac{V}{V_o} = -0.039972 \rightarrow \ln \left(\frac{Po}{P}\right) = n \ln \frac{V}{V_o} = - n \times .039972$$

EXPERIMENT II NO STUFF WITH OIL

$L = 5.814$ in.

$ex = .229$ in.

$l_o = 9.282 - (5.814 + .967) = 2.501$ in.

$L_o = 8.369 - (2.501) = 5.8680$ in.

$V_o = \pi \frac{(1.120)^2}{4} \times 5.868 = .09640$

$= .985203 \times 5.868 + .09640 = 5.877571$ in.^{3}

$V = .985203\ (5.868 - .229) + .09640$

$= 5.555712 + .09640 = 5.652112$ in.^{3}

$\frac{V}{V_o} = \frac{5.652112}{5.877571} = .961641$

$\text{Ln}\ \frac{V}{V_o} = -\ .039114$

$\text{Ln}\ \frac{P_o}{P} = n\text{Ln}\ \frac{V}{V_o} = -\ n \times .039114$

EXPERIMENT III TWO STUFF WITH OIL

$L = 5.805$ in.

$l_o = (9.282 - (5.805 + .967) = 2.5100$ in.

$L_o = 8.369 - 2.51 = 5.8590$ in.

$$V_o = \frac{\pi(1.120)^2}{4} \times 5.8590 + .09640$$

$$= .985203 \times 5.8590 + .09640$$

$$= 5.772304 + .09640 = 5.868704 \text{ in.}^3$$

$$V = .985203 (5.8590 - .229) + .09640$$

$$= 5.546693 + .09640 = 5.643093 \text{ in.}^3$$

$$\frac{V}{V_o} = \frac{5.643093}{5.868704} = .961557$$

$$\ln \frac{V}{V_o} = - 0.039202$$

EXPERIMENT IV TWO STUFF WITHOUT OIL

$L = 5.755$ in.

$l_o = 9.282 - (5.755 + .967) = 2.56$ in.

$L_o = 8.369 - 2.56 = 5.809$ in.

$V_o = .985203 \times 5.809 + .09640 = 5.819444$

$V = .985203\ (5.809 - .229) + .09640$

$= 5.497433 + .09640 = 5.593833\ \text{in}^3$

$$\frac{V}{V_o} = \frac{5.593833}{5.819444} = .961231$$

$$\text{Ln}\ \frac{V}{V_o} = -.039540$$

$$\text{Ln}\ \frac{P_o}{P} = n\text{Ln}\ \frac{V}{V_o} = -\ n \times .039540$$

EXPERIMENT V NO STUFF WITHOUT OIL

$L = 5.764$ in.

$l_o = 9.383 - (5.764 + .967) = 2.551$ in.

$L_o = 8.369 - 2.551 = 5.818$ in.

$V_o = .985203 (5.818) + .09640 = 5.6027$ in^3

$\frac{V}{V_o} = \frac{5.6027}{5.828311} = .961290$

$\text{Ln} \frac{V}{V_o} = -.039479$

$\text{Ln} \frac{P_o}{P} = n\text{Ln} \frac{V}{V_o} = -n \times .039479$

EXPERIMENT VI ONE STUFF WITHOUT OIL

$L = 5.755$ in.

$l_o = 19.282 - (5.755 + .967) = 2.56$ in.

$L_o = 8.369 - 2.56 = 5.809$ in.

$V_o = .985203 \times 5.809 + .09640 = 5.819444 \text{ in}^3$

$V = .985203\ (5.809 - .227) + .09640 = 5.595803 \text{ in}^3$

$$\frac{V}{V_o} = \frac{5.595803}{5.819444} = .061570$$

$$\text{Ln} \frac{V}{V_o} = - .039188$$

$$\text{Ln} \frac{P_o}{P} = n\text{Ln} \frac{V}{V_o} = - n \times .039188$$

F	f, hz	n	ln Po/P	Po/P	P psi	ΔP, psi
1.0	.0161	1.010	- .0415	.9595	15.32	.620
1.5	.0363	1.030	- .0422	.9537	15.34	.640
2.0	.0646	1.055	- .0437	.9574	15.37	.670
2.5	.1008	1.120	- .0459	.9552	15.40	.700
3.0	.1453	1.160	- .0476	.9536	15.42	.720
4.0	.2582	1.220	- .0500	.9514	15.45	.750
5.0	.4035	1.250	- .0513	.9501	15.48	.780
6.0	.5810	1.275	- .0523	.9492	15.50	.800
7.0	.7900	1.290	- .0529	.9486	15.51	.810
8.0	1.032	1.310	- .0537	.9478	15.52	.820
10.0	1.610	1.325	- .0544	.9472	15.53	.830
15.0	3.63	1.345	- .0552	.9464	15.54	.840
20.0	6.450	1.360	- .0558	.9458	15.55	.850
30.0	14.53	1.375	- .0564	.9453	15.56	.860

Table11: Theoretical Results

10. APPENDIX B

10.1. THERMODYNAMIC ASPECT OF THE GAS LOADED ACCUMULATOR [22]

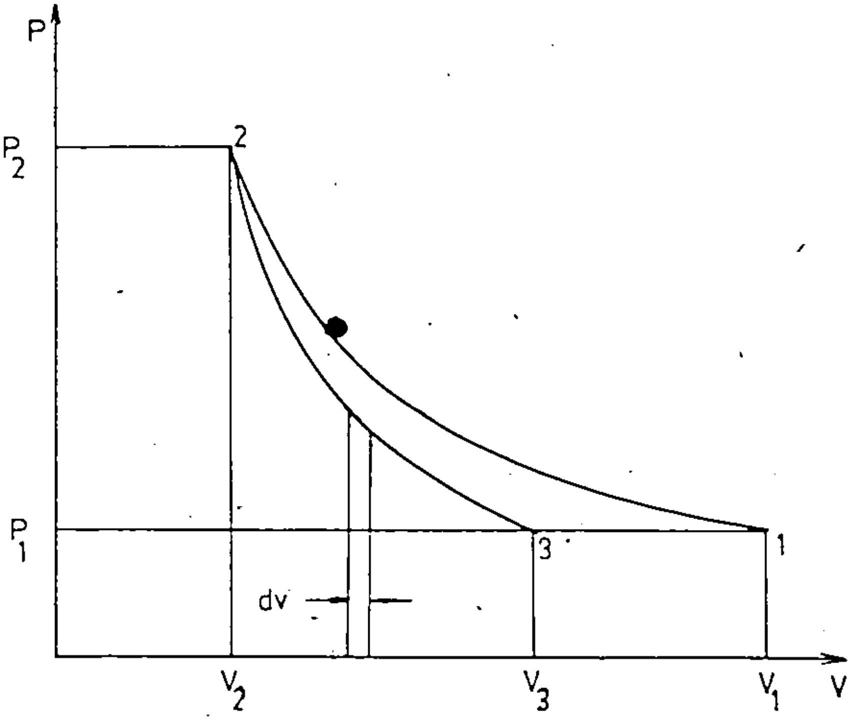

Fig. IV. P-V Diagram

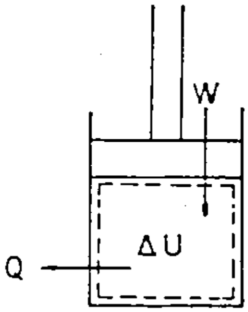

Fig.V.System Representation

The path of an isothermal process is illustrated by the line 1-2 in figure.IV. . And its work is formulated as:

$$W_{1-2} = \int_1^2 P\,dv = \int_1^2 \frac{P_1 v_1}{v}\,dv = \int_1^2 \frac{P_2 v_2 dv}{v}$$

$$W_{1-2} = P_1\ v_1\ \mathrm{Ln}\frac{v_2}{v_1} = P_1\ v_1\ \mathrm{Ln}\frac{P_1}{P_2}$$

The first law of thermodynamics gives us the balance equation

$$W = Q + \Delta U$$

Since for an isothermal process both the temperature and internal energy are constant, it follows that:

$$u_2 = u_1$$

and

$$W_{1-2} = \ - Q_{1-2}$$

During the process all of the work supplied is simultaneously delivered as heat since no additional internal energy can reside in the gas. While the temperature is constant, there is no heat gain and the gas occupies less space than a gas at an elevated temperature. therefore, with a same maximum pressure we could do more compression on the gas.

Path 3-2 represents an adiabatic reversible compression process. The area under curve 3-2 is the work of compression of the accumulator.

An energy balance gives:

$$W = Q \ + \Delta U$$

During this process, we have a sudden increase in temperature; the gas is heated up and it occupies more space. So with the same work the compression of the gas is less and the fluid storage is less. Therefore, the work during a reversible isothermal compression is more efficient than during a reversible adiabatic compression in terms of capacity of storing liquid.

During the discharge, the accumulator uses the spring effect of the gas to push back the liquid to the system. This is the expansion process of the gas. Since expansion is the reverse operation of compression, we could deduce that the work done by expansion during an isothermal process releases more fluid to the system than during a reversible adiabatic process.

Since work is not created at the expense of internal energy, all of it must be supplied by heat addition during an isothermal expansion process. How could the accumulator acquires this heat? It could be given up from the ambient through the accumulators walls or from the fluid. But we could absorb only a small amount of heat by such means. A way of enhancing this situation is by storing the heat dissipated during the compression process. Hence, we fill the air chamber with materials.

As the gas is compressed, it passes energy to the fill materials, through a small temperature difference. During gas decompression, the materials give back this energy and heats the expanding gas charge. Therefore, we did experiments with copper mesh as the fill materials.

We are attempting to realize the isothermal condition by allowing the energy to flow back and forth between the gas and the fill materials during cycling.

STORAGE OF ENERGY FOR HYDRAULIC SYSTEMS

In self-contained assemblies (aircraft, engines) the problem of storing energy for the hydraulic circuit arises. The following are generally used: accumulators with air or an inert gas; electric batteries (these require an electric motor which, in turn, needs a hydraulic pump, a reservoir and in most cases a small buffer accumulator); a source of chemical energy: solid fuel, hydrogen peroxide, kerosene, liquefied gas, etc with their associated equipment; a diversion of some of the propulsive energy from the engine, e g of mechanical power to a pump, bleeding air from the compressor of a jet engine to drive a turbopump, etc. The choice is mainly determined by the allowable weight of the assembly

If the total energy to be stored is E and the maximum power H, the weight of the complete generating assembly can be approximately expressed in the form

$$W = \alpha E + \beta H$$

where α is the ratio of the weight of the accumulator to the energy stored, which varies considerably with the type of accumulator, and β is the ratio of the weight of the supply unit to the power, which depends on the nature of the unit.

It is obviously impossible to give precise values of α and β, since in every case these coefficients depend on the technological quality of the different parts, the durability and degree of safety required, and the magnitude of the total energy to be stored, the maximum hydraulic pressure and the maximum rate of flow.

Table gives a comparison between the different types of accumulators and sources of energy. In view of the numerical values quoted, it will be appreciated that: (a) accumulators with springs are rarely used and then for an operation which need only be carried out once (e.g. releasing a safty device). (b) pneumatic accumulators which are not recharged while operating, are used continuously only in machines whose active life is limited to a time of about 10 sec. (c) electric accumulators which cannot be recharged are used only for a duration of a few minutes; for longer duration, an external source of energy must be employed, such as a mechanical pump to recharge a pneumatic accumulator, a generator to recharge the electric batteries, etc.

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1 *Type of energy*	2 *Type of accumulator*	3 *Energy per unit weight kg m/kg* *Stored energy* *Algebraic expression*	4 *Numerical value (see col. 7)*	5 *Hydraulic energy recovered* *Algebraic expression*	6 *Numerical value (see col. 7)*	7 *Definition of parameters, remarks (in brackets, numerical values used for calculation of columns 4 and 6)*
	compressed liquid	$\frac{10P_1^2}{2Bw}$	26	not used		P_1 = pressure, in kg/cm² (300) B = bulk modulus in kg/cm² (20,000) w = specific weight (0·86 g/cm³)
	steel in tension or compression	$\frac{1{,}000f_1^2}{2Ew} = \frac{f_1^2}{328}$	30	not used		f_1 = stress, in kg/mm² (100) E = Young's modulus, in kg/mm² (21,000) w = specific weight (7·8 g/cm³)
	coiled spring accumulator (steel)	$\frac{1{,}000f_1^2}{4Gw} = \frac{f_1^2}{250}$	40	$\frac{f_1^2-f_2^2}{250}$	22	f_1 = stress, in kg/mm² (100) G = shear modulus, in kg/mm² (8,000) w = specific weight (7·8 g/cm³) Col. 5 and 6: spring operating between f_1 (100) and f_2 (66·7)
Mechanical deformation energy	rubber spring accumulator	—	~ 400	—	~ 200	largely depends on nature of rubber, type of spring and mode of use
	compressed air	$JC_vT_1 = \frac{P_1V_1}{\gamma-1}T_1 = 72T_1$	20,700			adiabatic expansion to zero pressure (not to P_2) T_1 = absolute initial temperature, in °K (288 = 15°C)
	oil and air accumulator (cf. Chapter 10, Example 2)	$\frac{72T_1}{1+K_1+K_2}$	2,500	$\frac{72T_1}{1+K_1+K_2}\times\left[1-\left(\frac{V_1}{V_2}\right)^{0.4}\right]$	375	adiabatic expansion volumes V_1, V_2: $V_1/V_2 = \frac{1}{3}$ (volume of oil used = $V_2 - V_1$)
				$\frac{6740T_1}{1+K_1+K_2}\log\frac{V_2}{V_1}$	410	isothermal expansion T_1 = initial absolute temperature, in °K (288 = 15°C) K_1 = weight of container necessary for air only, per kg of air (4·1)* K_2 = weight of oil and container (per kg of air) (3·2)*

Mechanical deformation energy	air cylinder + pneumatic motor + hydraulic pump (cf. Chapter 10, Example 2)	$\frac{72T_1}{1+K_1}$	4,050	$\frac{100T_1}{1+K_1}\times\left[1-\left(\frac{P_2}{P_1'}\right)^{0.287}\right]\eta_m\eta_p$	1,090‡	pre-expansion without any work being done from P_1 (300 kg/cm²) to P_1' (7 kg/cm²) by means of a pressure-reducing valve adiabatic expansion from P_1' to P_2 (1 kg/cm²) in a motor of efficiency η_m (0·6)† efficiency of hydraulic pump, η_p (0·75) (equation valid only if $P_1' < P_1$ (see Section 8.2)
Kinetic	flywheel + hydraulic pump	$\frac{V_1^2}{2g}$	4,500	$\frac{V_{1_i}^2-V_{1_f}^2}{2g}\eta_p$	2,500‡	V_1 = tangential speed of flywheel, in m/sec V_{1_i} = initial speed (300): V_{1_f} = final speed (150) efficiency of hydraulic pump η_p (0·75)
Electrical	battery + motor and pump	E_e	~ 8,000	$E_e\eta_m\eta_p$	~ 4,000‡§	lead battery efficiency of electric motor, η_m
			~ 18,000		~ 9,000‡§	nickel-iron battery efficiency of pump, η_p
			~ 30,000		1,500‡§	zinc–silver oxide battery ($\eta_m\eta_p$ = 0·5)
Chemical	hydrogen peroxide + motor and pump	E_c	160,000	$E_c\eta_i\eta_p$	40,000‡	pure hydrogen peroxide catalytic decomposition efficiency of motor, η_i efficiency of pump, η_p ($\eta_i\eta_p$ = 0·25)
			110,000		27,500‡	industrial hydrogen peroxide
	solid fuel + motor and pump		200,000‖		50,000‡‖	combustion
	kerosene oil + motor and pump		4,500,000		1,125,000‡	combustion (weight of oxygen not included)
Potential	water power	H	~ 1,200			H = height of the fall, in m (La Bathie, Savoie)

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* Values for the Air Equipment accumulator No. 3066 (accumulator for engines). Weight empty, 1·200 kg; 530 cm³ of air at 300 kg/cm², 0·200 kg; 270 cm³ of oil, 0·230 kg. When greater endurance is required, it is necessary to increase the coefficients from 6 to 12 for K_1 and from 4 to 8 for K_2.

† Air Equipment pneumatic motor No. 28 [illegible] power 3·5 kW, weight 1 kg.

‡ These figures take into account the weight of the motor at 1 [illegible], which varies considerably according to type, size and endurance required. The following are a rough guide: hydraulic pumps [illegible] kg/kW, pneumatic motors [illegible] kg/kW, electric motors 0·70–2·0 kg/kW.

§ These values correspond to a discharge time of about 1 h; if shorter, they must be divided by a factor of ca. 3 for 6 min, ca. 12 for 1 min. For a comparison between different types of electric accumulators, see e.g. [illegible].

‖ Using the [illegible] value for [illegible] (1,400 K).