HYDRAULIC SYSTEM AND MOTOR WITH VARIABLE SPEED AND TORQUE

BY

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HYDRAULIC SYSTEM AND MOTOR WITH VARIABLE SPEED AND TORQUE

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ADRIAN MARICA

A Thesis/Practicum submitted to the Faculty of Graduate Studies of the University of Manitoba in partial fulfillment of the requirements for the degree of

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ABSTRACT

The aim of the present work is to demonstrate the validity of the transmission of energy over a distance by longitudinal vibrations propagated in an elastic fluid contained in a pipe. A new transmission system that incorporates a pump, transmission lines and a special "hydraulic" motor has been designed. It offers high torque over a broad range of operating speeds. The motor incorporates a mechanism for converting reciprocating to a rotating motion that can also be used to control the output power of a shaft. Moreover, the motor has a variable speed and torque.

Efficiency characteristics are considered of simple controls (involving constant pressure or constant power) that can be applied to a fixed displacement and variable speed pump connected to a variable displacement motor. The most efficient way to implement a specific control is identified in terms of the internal flow resistance inside a rotor. The effects of the controls on fuel consumption and sizing are also discussed.

LIST OF FIGURES

TABLE OF CONTENTS

NOTATION

W	- work done by the receiver;
W_0	- work done by the generator;
ω	- angular speed of the generator;
D	- diameter of the generator's piston;
l	- length of pipe;
a	- interior cross sectional area of pipe;
A	- cross sectional area of generator's cylinder;
I	- intensity at the receptor end;
I_{max}	- maximum intensity at the generator;
P	- pressure at the receiver;
P_0	- maximum pressure at the generator;
β	- damping coefficient of the sound waves;
R	- resistance of the pipe;
λ	- wavelength of the sound waves traveling along a pipe;
C	- capacity;
ρ	- density;
g	- gravitational acceleration;
L	- inertia;

CHAPTER 1

1.1 INTRODUCTION

It is nearly three centuries since the French scientist Pascal demonstrated that pressure exerted on a confined fluid is transmitted unabated in all directions and acts with equal force on equal areas. Modern fluid power engineering is the practical application of this principle in linear moving devices, such as power cylinders or rotary drive motors.

Industrial jobs requiring the precise control of speed and power can be performed more easily hydraulically than with the gears, chains, belts or shafts of a mechanical transmission. The main advantage of a hydraulic system is its simplicity because fewer parts are required. Moreover, the output power is located closer to the point of use. A pump, fluid, controls, piping, hose or tubing and a cylinder are all components of a fluid power system. Valve levers and push buttons are all that are required to start, stop and control a fluid power system. A smooth motion, easily achieved speed control, low cost and versatility are the prime characteristics of a hydraulic system.

A fluid power system is one that generates, transmits and controls the application of power through the use of pressurized and moving fluids within an enclosed circuit. In commonly used hydraulic systems, water or oil (under a pressure of about 1000 lb. per square inch) is supplied from a pumping station and it is transmitted through a pipe to a motor or a simple cylinder. The hydraulic pump is driven by an electrical motor or engine. In most systems, a pump produces a flow of oil at the rate needed for a desired cylinder speed. Hydraulic oil, under pressure, is piped to the point of use and the oil flows back to a reservoir after its energy is expended. However, excess energy, i.e. which is not absorbed by the end effector, is dissipated in the return line when the oil flows to the reservoir so that the overall system has a low efficiency. This method involves a continuous flow of the working fluid that, in effect, serves as a flexible coupling between the pump and the motor.

Electrical and hydrostatic transmissions have attracted increasing attention in the last decade, principally in automobiles. Electricity can be used as a primary source of power for prototypes, or even for final products, where mobility is not a primary requirement. However, electricity is also beginning to appear as a secondary source of power on mobile construction machinery, like trucks and scrapers, that use diesel or gas engines to turn generators and provide power for electrical wheels. Lai and Grigg [7], for instance, showed that a Load-Haul-Dump Vehicle (LDV) equipped with hydrostatic drives achieved greater efficiency than standard vehicles. Preliminary studies [8] also demonstrated improved ergonomic design, fuel efficiency as well as lower noise and exhaust pollution.

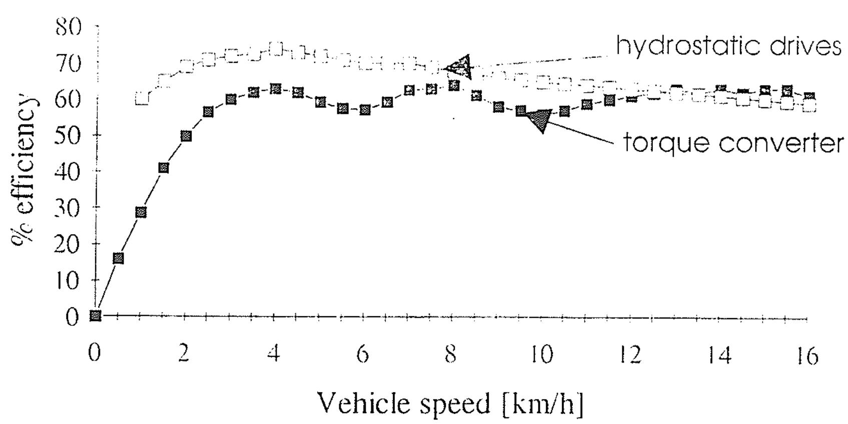

Fig. 1. Variation of drive efficiency with vehicle speed for a 78 kW constant power demand.

These characteristics are the result of a higher overall efficiency of the hydrostatic drive compared with that of a torque converter - an efficiency that peaks at 74% compared with the torque converter's 63%. On the other hand, **Figure 1** indicates that this difference narrows appreciably as a vehicle's speed increases.

A totally new transmission system was created by Constantinesco during the First World War [10]. The invention for which he is best known is the interrupter gear for synchronizing the firing of machine-gun bullets between the propeller blades of an airplane. It consists of a cam (on the propeller shaft) that drives a plunger in a small cylinder. The cylinder is mounted at one end of a flexible pipe that directs the waves generated by the rapidly reciprocating plunger. A trigger release gear is mounted at the other end of the pipe and it checks the gun's firing rate. A continuous pull of a control lever by the pilot operates the trigger release gear so that the gun fires only when the propeller blades do not restrict the bullets' passage. Although the gear is capable of firing 3000 bullets per minute, the gun limited the number of rounds fired to about 600 per minute in the early days of the war and to about 1200 per minute towards the end. Hence, the gear does not restrict the firing rate of the gun but checks it at the instants at which, if fired, a bullet would strike a propeller blade. The flexible pipe through which the waves travel gives the device a great advantage over similar, purely mechanical actions because the operation is unaffected by the airframe flexing in flight. This independence arises because the connecting member between the propeller shaft and the gun trigger (in the interrupter gear) is a tube containing liquid whose distortion does not affect the synchronization.

The novelty of a wave power transmission system is that there is no fluid flowing along transmission pipes and, consequently, there is no energy wasted in the return circuits. Pressure waves (illustrated in **Figure 2**) are generated in a fluid at one end of a finite pipe by means of a reciprocating plunger. The plunger is operated by a crank so that its motion is simple harmonic. Therefore, compression and rarefaction zones occur along the fluid because the fluid's particles are put in a state of reciprocating vibration. At the receiving end of the pipe, power is transmitted to another plunger by the pressure waves in the fluid. This power can be used to perform mechanical work.

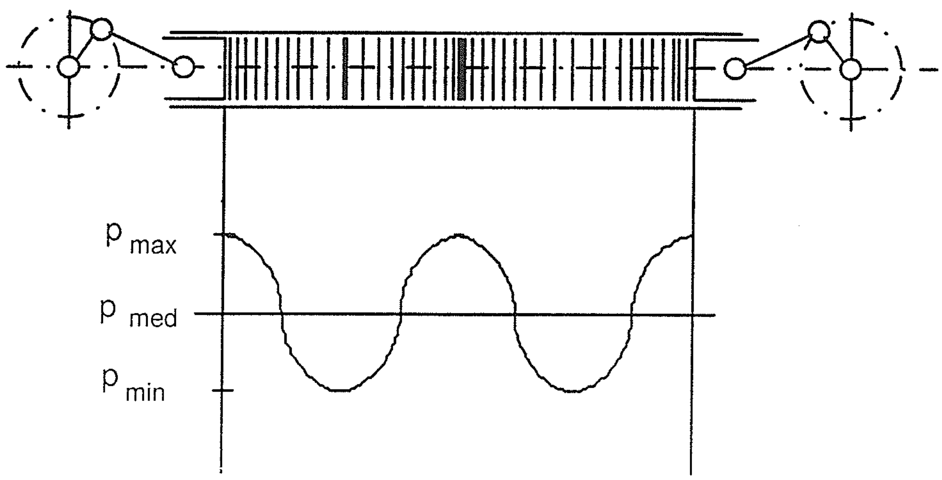

Fig. 2. Principle of power transmission through pipes filled with a fluid.

Obvious applications of wave power transmission are fabricating or assembling machine parts. For example, sonic hammers can be used for riveting, cutting, boring or fast acting pressing machines. Small, diamond hammers that operate at around 500 Hz are useful in cutting glass, quartz or hard rocks. Sonic tools, coupled with freezing methods, permit the recycling of stratified insulating materials. At present, the method of recycling these materials is by grounding and burning them in steam generating plants because the wastes contain high quantities of phenols. Laboratory research in Germany and Romania showed that wear resistance is increased greatly if insulating materials are added as a powder in asphalt. On the other hand, the cost of the asphalt mixture is reduced to about one third of the initial cost of the classical recipe that uses only sand.

1.2 Statement of the problem

The current production of hydrostatic transmission systems suggests that all problems have been solved. Nevertheless, a close examination suggests unresolved drawbacks for any hydrostatic transmission. First, the efficiency of a hydrostatic system is not at its greatest over the whole operating speed range. Moreover, the efficiency of a specific control device affects the entire system. Detailed studies [7;8] demonstrate a close relationship between the overall efficiency of a hydrostatic transmission and the operating speed. It has been established that the overall efficiency grows with increasing speed up to an 'optimum' speed, after which the efficiency decreases. The efficiency peaks at about 74% but it can vary by more than 10% between the minimum and optimum speeds [9]. Second, a system's efficiency depends upon the pressure and flow rate of the hydraulic fluid. An efficiency curve typically shows a just discernible maximum for constant pressure and flow rates whilst, conversely, the efficiency has a well-defined maximum for variable control parameters.

CHAPTER 2

State of the art

2.1 Hydrostatic systems

With the growing concern for energy conservation, transmission efficiency is one of the most critical factors. "Efficiency" and "control" are the cornerstones of hydrostatics. Transmission efficiency and power controllability are the main frontiers that have to be improved continuously and the gains promise more advantages than expensive technological improvements.

2.1.1 Transmission efficiency and controls

A "fluid power system" is one that generates, transmits and controls the application of power through the use of pressurized as well as moving fluids within an enclosed circuit. The main disadvantage of fluid power is a relatively low efficiency over the whole operating speed range. Much energy is wasted in return circuits when an end effector does not absorb the energy supplied from the pumping station. On the other hand, the pressure and flow rate delivered by a pump, say, greatly affect the efficiency of the entire hydrostatic system.

The form of a control can be absolutely arbitrary; the law relating flow and pressure is a matter of the designer's choice. This freedom is one of the most appealing advantages of hydrostatics. Whatever control scheme is adopted, the question to be answered is: "Once the operating pressure and maximum flow rate are assigned, is the overall efficiency maximized over the complete flow rate?" This question has not been answered satisfactorily

A "control" is a device that regulates the operation of a machine. It can also be defined as a relationship between the hydraulic power factors (the flow rate and pressure) and automatically actuated equipment by any combination of a hydraulic, mechanical or electronic system. The aim of using a control device is to seek substantial improvements in the safety and efficiency of workers and machines and, consequently, minimum external operator influences are desired.

Generally speaking, four procedures can be adopted to introduce controls into a hydrostatic transmission [1,2]. They can be associated with: (1) generators; (2) transmitters; (3) actuators; (4) or a combination of these devices. The first procedure implies the use of a variable displacement pump that is actuated automatically. The implemented control is independent of the circuit following the pump because such a pump regulates the operation of a machine by using a flexible correlation between the flow rate and fluid pressure.

A hydrostatic circuit is controlled by using valves or servovalves. Despite their use and abuse, these components do not usually fall into the valid definition of "control". This remark pertains primarily to directional valves that are designed to transmit basic operator demands like steering which is difficult to automate. In other cases, even if acting automatically on power parameters, transmitters (e.g. control valves) do the job with relatively low efficiency.

The third procedure implies "controlled actuators" that are synonymous with variable displacement motors. Such motors, however, exclude an important class of hydraulic motors, the high torque and low speed, radial piston type that has a fixed or discontinuously stepped displacement. The relationship between the hydraulic power factors requires a smooth variation of these factors whilst, in this class of motors, at least one hydraulic parameter does not vary continuously.

"Multiple controls" form the last procedure and they are exemplified by a variable displacement pump matched to a variable displacement motor. Such a scheme is very difficult to achieve and it seems that there are no major benefits to justify its application [3]. However, simple optimization studies were performed [4] on a transmission system consisting of an engine, a variable displacement pump and a fixed displacement motor. Despite encouraging results, the studies are not representative because only the simplest case of constant pressure control was considered. This type of control implies that the pressure in the hydrostatic circuit is kept constant for any flow rate, as suggested in **Figure 3**. Hence, merely the pressure level and flow rate have to be monitored and controlled.

Efficiency might be controlled easily by knowing, at any instant, the input and output power of a system. This situation represents a more relevant but complex case of constant power control in which a maximum pressure as well as a maximum flow and power level need to be held constant. For example, **Figure 4** indicates that the flow should be lower than F_{cm}

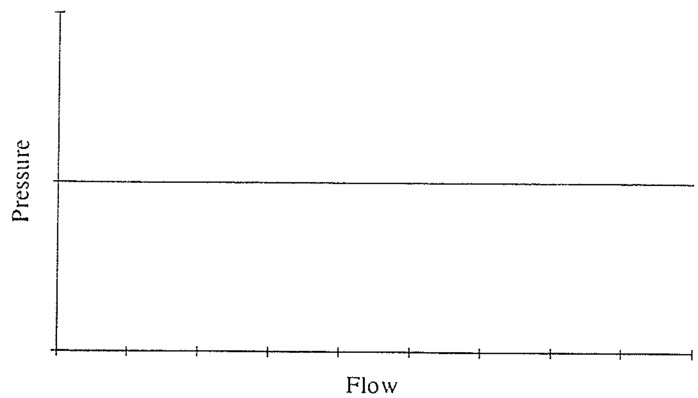

Fig. 3. Constant pressure control.

for a constant pressure. On the other hand, the output power of a machine running with constant efficiency has to be controlled continuously because the power level is affected directly by the pressure and flow rate. Therefore, an automated control device has to compensate one of these parameters when variations occur in the other parameter.

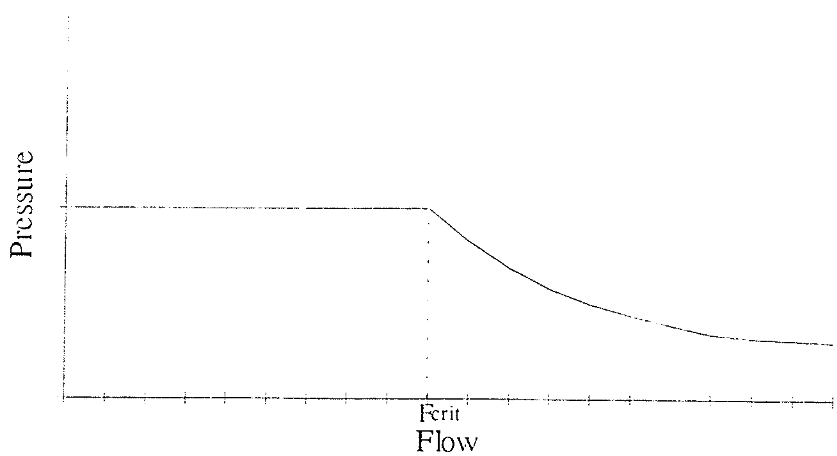

Fig. 4. Constant power control.

It was concluded in references 4 and 9 that a hydraulic pump offers better performance by varying the speed of the engine or primary motor. Thisgaard [4], for example, arrived at this conclusion by varying the rotational speed of an engine whilst measuring the output power of the hydraulic equipment driven by the engine. Running the engine with a variable speed avoids intermediate dissipative elements such as sliding clutches or fluid couplings that would reduce efficiency. It is noticeable that a completely hydraulic control system does not appear difficult to devise for an engine. Moreover, there is a potential fuel economy because the rotational speed of an engine decreases at approximately constant torque and this occurs without affecting the engine's sizing.

Zarotty [9] also performed the same type of experiment but with an electrical motor driving the hydraulic pump. His results are replotted in **Figure 5** where the relative flow is defined as the ratio between the actual flow and the maximum possible value at full

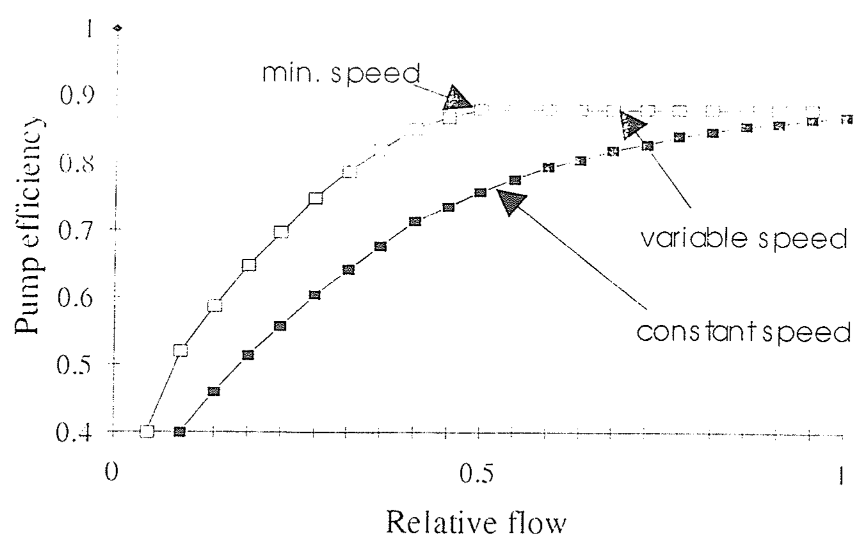

Fig. 5. Pump efficiency for constant pressure control.

displacement and maximum speed of the pump. The constant speed efficiency can be seen to grow in this figure (with increasing relative flow) up to a maximum value that corresponds to the pump's full displacement. Conversely, the variable speed control curve can be separated into two regions. In the first region, the displacement increases at a constant speed of the pump and produces a larger efficiency. The second region corresponds to an increasing speed at full displacement of the pump which produces an almost constant efficiency over a wide range of relative flows. This latter region coincides with the minimum flow supplied at full displacement of the pump.

2.2 Wave power transmission

Most work in the field of wave transmission through pipes filled with liquids was pioneered during and immediately after the First World War. Moss [5],for example, described efficiency experiments that were based upon Constantinesco's three-phase equipment. Two series of tests were performed by using the arrangements shown in **Figures** 6 and 7. In the first series, Set A, the hydraulic generator and motor each had three cylinders placed in line. Each cylinder and corresponding piston had an approximate bore and stroke of 25 mm and 30 mm, respectively. A hydraulic generator and a motor, each consisting of three cylinders but arranged radially at 120^{0}, were built for the second test, Set B. (See **Figure** 7.) The capacity was slightly larger than that for set A, the bore being 27 mm and the stroke 35 mm. However, vital components of the system were made from white metal so

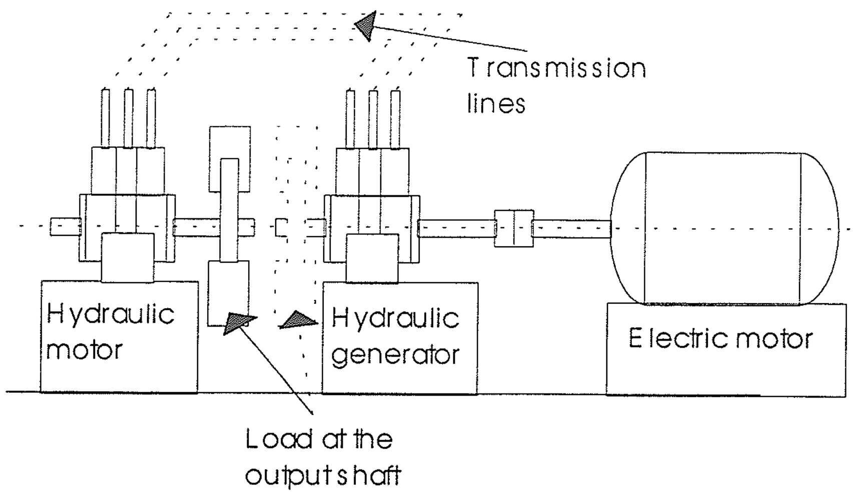

Fig. 6. Arrangement for transmission Set A.

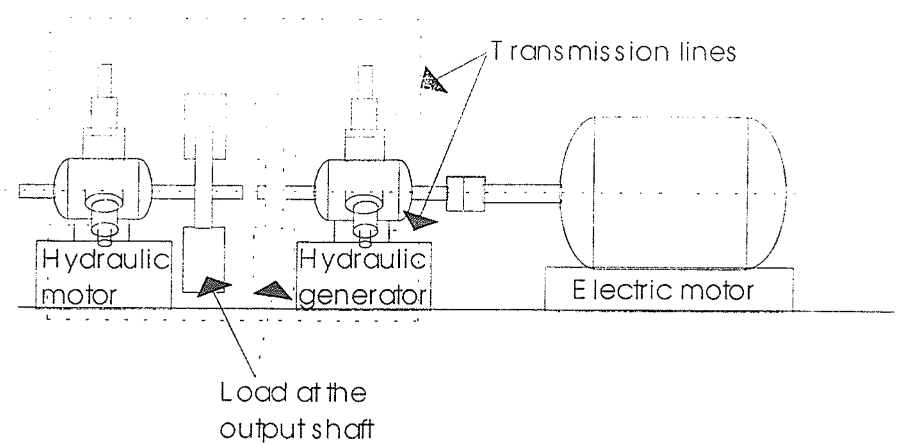

Fig. 7. Arrangement for transmission Set B.

that they generated great heat, wore quickly and produced appreciable leakage. Therefore, the light mineral oil was replaced by sperm oil and, later, by an equal mixture of sperm oil and paraffin oil. These changes were required to accommodate the increasing clearances between the cylinders and pistons. The solution of changing the fluid's viscosity was preferred to the cost and delay of manufacturing new components. Despite these imperfections, Moss reported an overall efficiency of around 80%.

Foster and Parker [6] demonstrated, in 1964, that the efficiency of a single-phase, wave power transmission system is reduced if the load does not equal the characteristic impedance of the pipeline. The characteristic impedance is defined as the ratio of the mechanical force to the velocity of the resulting vibration at points along the pipe filled fluid that have the greatest speed variation. Conversely, any receiver mounted at the end of the pipe, opposite to the generator, is considered a load. The experimental arrangement is presented in **Figure 8**.

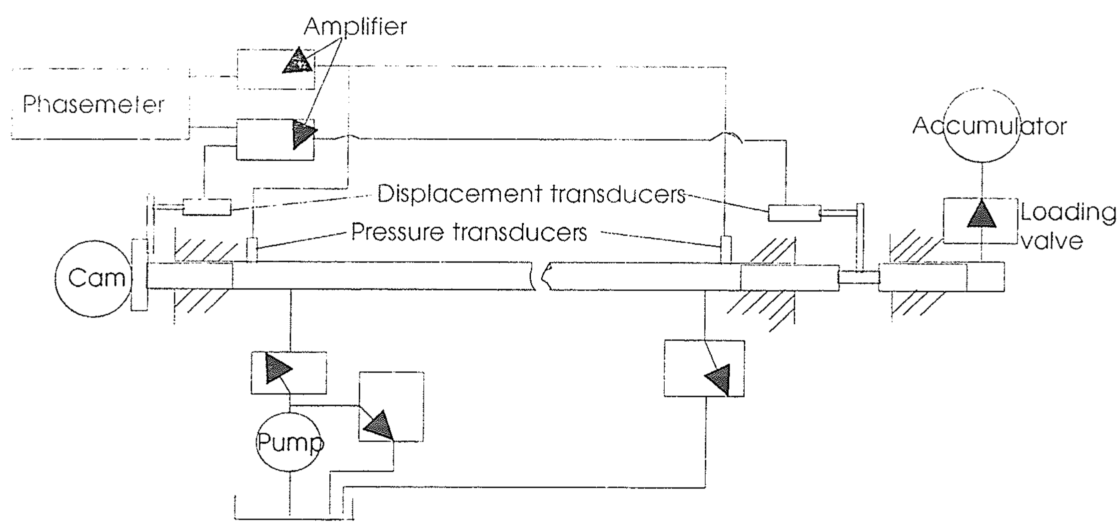

Fig. 8. Diagram of hydraulic, single-phase transmission line.

A variable amplitude piston oscillator was used to harmonically excite the oil inside an 80ft transmission pipe filled with Tellus 27 hydraulic oil. The pipe had a bore of 22.2mm and 9.5mm thick walls. To avoid cavitation and air penetration during the negative part of the cycle, provision was made to apply a mean pressure to the oil so that the dynamic pressures were always positive. Experiments were performed with a nitrogen bag accumulator and a piston fitted to the output end of the pipe. The accumulator and second piston were used merely to balance the mean pressure in the pipe. One interpretation of Foster and Parker's demonstration is that the efficiency of a single - phase transmission system is greatest if the generator and receiver are identical. To obtain the maximum power output from a given transmission pipe, the generator must produce the maximum permissible pressure or velocity at the pipe's input. This is true at all frequencies so that tuning of the pipe is not required.

CHAPTER 3

PROTOTYPE TRANSMISSION SYSTEM

Present fluid power systems are not highly efficient over their whole range of operating speeds and their efficiency is affected by any control device. Moreover, a hydrostatic motor's efficiency depends upon the basic parameters (pressure and flow rate) of the controls and, in addition, the output characteristics of torque and speed cannot be varied independently.

A highly efficient wave power transmission system has been designed in the present work based on the principles of power transmission through pipes filled with a liquid. The new transmission system aims to replace transmissions such as hydrostatic motors and torque converters. It incorporates power control and a motion conversion device that is also applicable to a new type of variable compression ratio, internal combustion engine. In addition to simplicity, the new mechanism offers power modulation of a rotating shaft so that the output power can be varied independently by the rotational speed of the engine itself.

Variable compression ratio, poly-fuel engines run with different fuels without major modifications but they require an automatic or very easily adjustable compression ratio. All variable compression ratio engines can be classified in the three major families of: mobile combustion head, mobile piston bottom or free piston generators. (See **Figure 9** for details.) Regardless of design, the first two families use oil, under pressure, to change the volume of the combustion chamber. The oil is delivered by a hydraulic pump (driven by the engine itself) to a hydraulic cylinder attached to the combustion head or to the engine's piston. Hence, for the mobile head combustion family, the position of the combustion head is changed whilst, for the mobile piston bottom family, the position of the piston's bottom is varied. In both cases, the piston is connected rigidly to a rod and crank mechanism and the piston has a constant stroke. The third family of free piston engines was developed originally as a gas generator for gas turbines. In this case, the piston is not connected to a mechanism that converts translational motion into a rotation of an output shaft. Consequently, the piston is free to oscillate in two opposed cylinders. This type of engine has the advantage of an automatically adjusted compression ratio. The compression ratio is adjusted by the combustion process and by the delivered gas pressure. Regardless of family, the primary characteristic of conventional mechanisms is reliability but they are complicated, bulky and require a crankcase having a large volume.

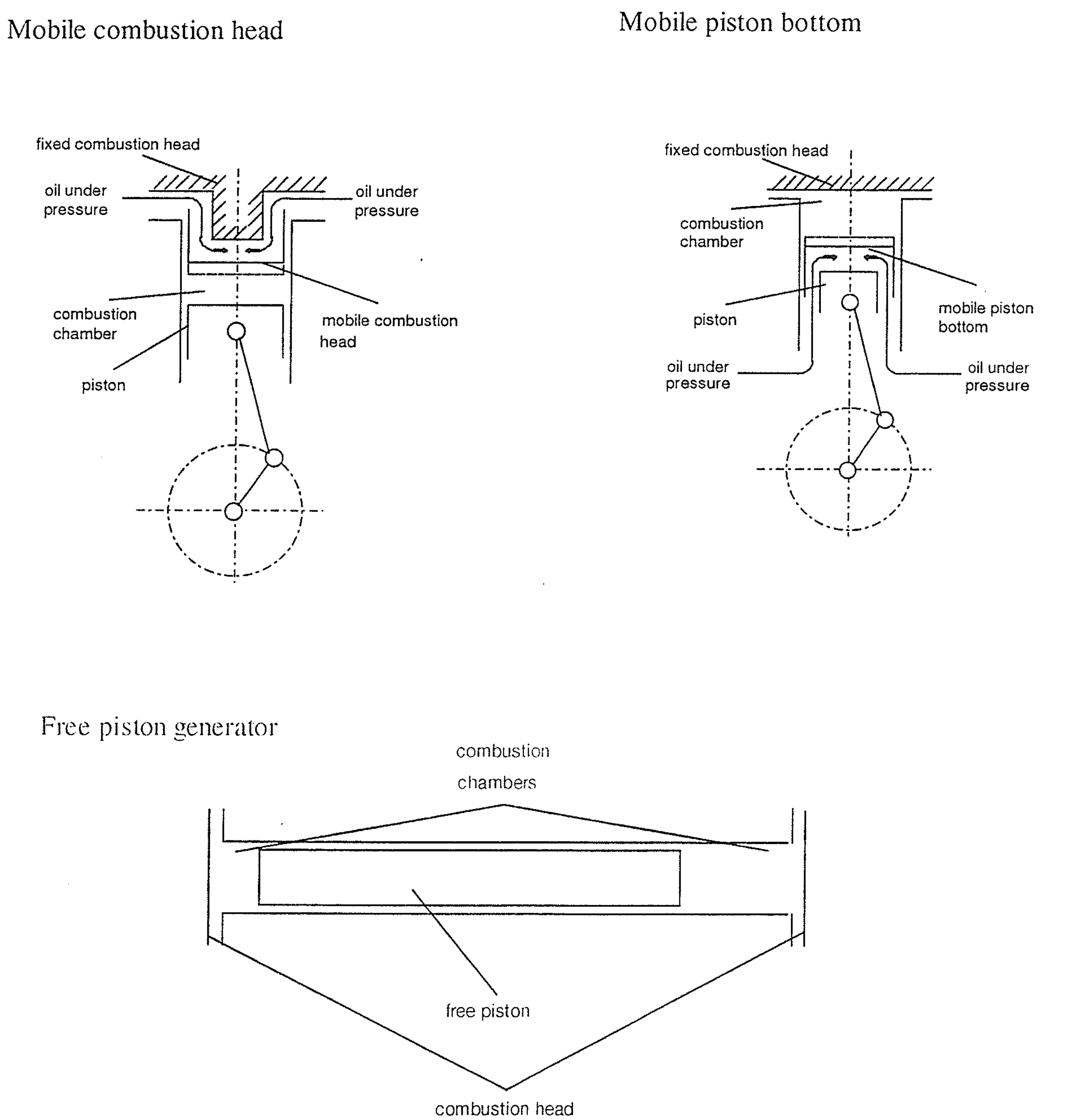

Fig. 9. Major families of the variable compression ratio engines.

The main objective of the new transmission system is to provide a highly efficient, power transmission system over a broad range of operating speeds. One subsidiary objective is to produce a mechanism for converting reciprocating motion to a rotating motion that also incorporates a power control device for smoothly modulating the output power. Another supporting objective is to provide a mechanism for converting a reciprocating motion into a rotating motion that can also be used to control the output power of a shaft.

A wave power transmission system may have a single line (or phase) with reciprocating pistons located at both ends or it may have three lines (phases) with a phase shift of 120^0 between the corresponding oscillating pressures. Details of the prototype system are given in **Figure 10** which shows a complete three phase system. A cross section through

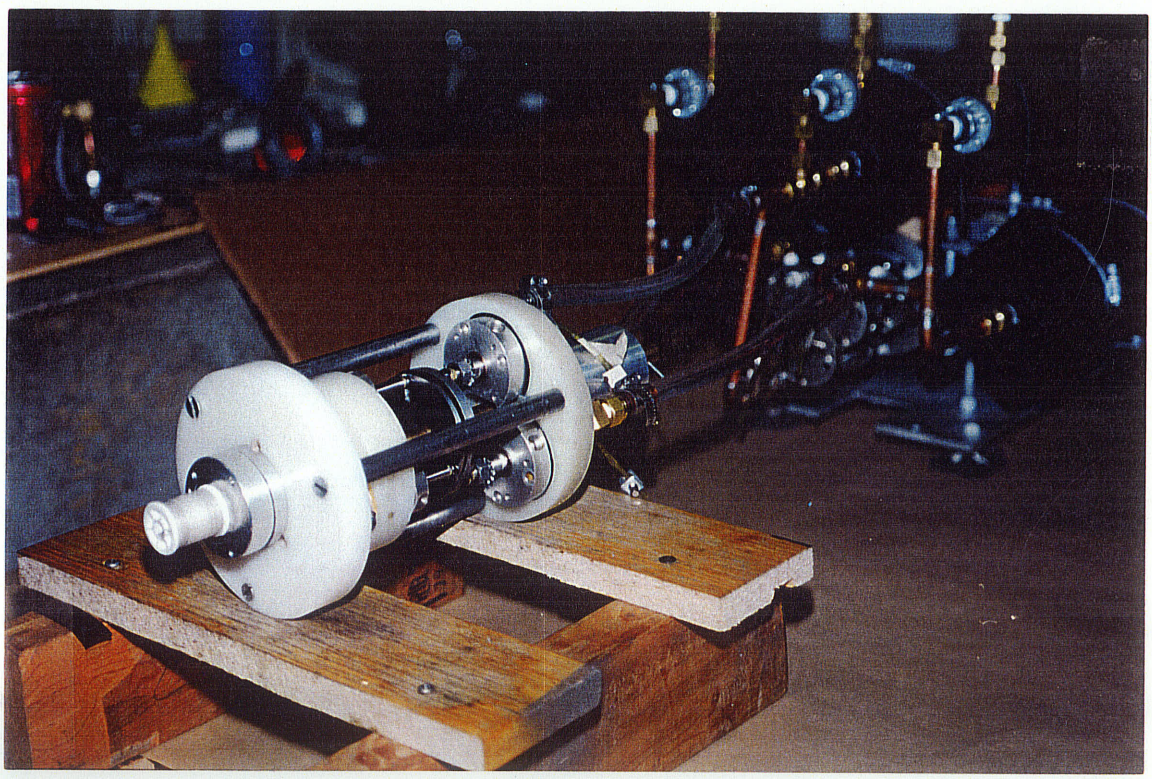

Fig. 10. Prototype transmission system.

one phase of the system is displayed schematically in **Figure 11** with numbered components. This figure shows a reciprocating plunger (13) that fits into a finite pipe (3) filled with liquid. The plunger is operated by a crank (1) so that the plunger's motion varies harmonically over time. The crank is turned by a primary motor that can be an electrical motor or an engine.

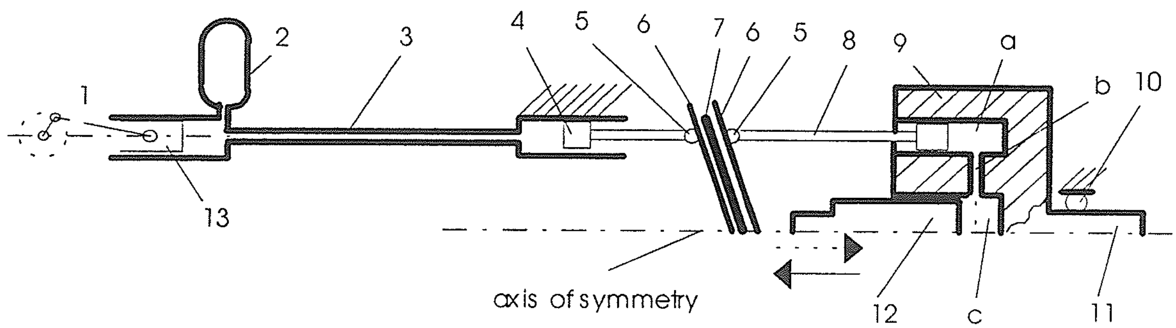

Fig. 11. Section through one phase of the new wave power transmission system.

Accumulator (2) is filled completely with liquid so that it can "communicate" with pipe (3) in the neighborhood of the generator. The accumulator has rigid walls and its volume is 3000 times greater than the generator's stroke volume. Pistons (4) simply rest on one of the plates (6). Three radial channels, spaced at 120°, are cut in one face of both plates, as shown in **Figure 12**. To minimize friction, a set of hard steel balls (5) are interposed between the piston and plate such that they are able to roll along the radial channels. An axial needle bearing (7) is inserted between two identical plates (6) so that they can rotate independently. The second of these plates rests on another set of steel balls and on pistons (8). Three

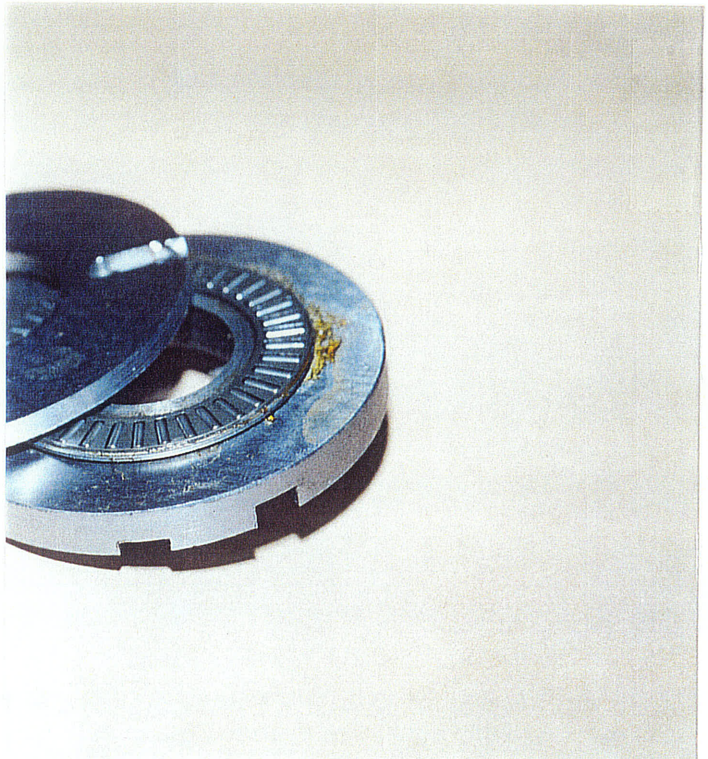

Fig. 12. Twin plates and axial bearing assembly.

cylinders, corresponding to each piston, are cut into body (9) which is filled with water. This body is connected rigidly to an output shaft (11) and it is supported radially and axially by a set of axial-radial bearings (10). Each piston defines a variable volume cavity 'a' in body (9) that communicates, through channel 'b', with central cavity 'c'. The common orifice between channel 'b' and cavity 'c' is controlled by a piston type part (12). The movement of this piston inside cavity 'c' is controlled by an operator by employing a screw-nut mechanism. (However, an automatic device could replace the operator.)

The behavior of the three phase power transmission system and the conversion mechanism, with its power control device, are described next. The three phase system incorporates three systems, each one identical to that presented in **Figure 11**. Hence, there are three piston generators, three pipes, three accumulators and three receiver pistons with their corresponding cylinders distributed at 120^0 around the axis of symmetry. All three piston generators are driven by the same crank and, therefore, they are connected mechanically. As the crank turns, the plungers (13) generate pressure waves in the liquid columns but with a 120^0 phase shift at the end of each pipe. Compression and rarefaction pressure zones occur along the elastic liquid columns. At the receiving end of each pipe, power is transmitted to the plungers (4) by the pressure waves. If the generators and receivers are identical, the piston at each receiving end moves synchronously with the corresponding generator and it is able to absorb all the energy of the waves traveling along the pipe. The capacities of the accumulators act effectively as springs that store the energy of the direct and reflected waves when the pressure is high and return this energy to the liquid when the pressure falls. By using container (2), pipe (3) can be closed partially or completely even though the crank is still rotating. Consequently, if the receiver is not in use and, thus, does not require power, the plungers (4) might be locked. Therefore, the rotating crank is required to perform work only when energy is actually utilized so that less energy is wasted.

Plungers (4) reciprocate in fixed cylinders and the entire assembly constitutes the stator of the hydraulic motor. (See **Figure 13.**) Further, the translational motion of the generator pistons is transmitted to the package of freely oscillating plates (6) and needle bearing (7). Body (9), together with both plates (6), as well as the needle bearing (7), piston type part (12), pistons (8) and second set of steel balls (5), constitute a conversion mechanism that incorporates a power control device and the rotor of the hydraulic motor. Because the

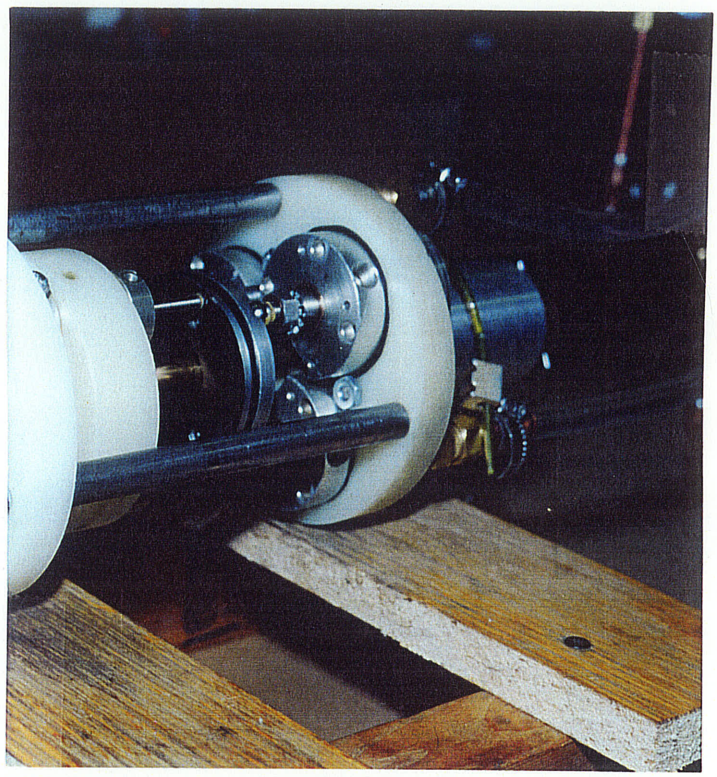

Fig. 13. New hydraulic motor assembly.

twin plates (6) and needle bearing (7) are kept between the pistons of the stator(4) and rotor (8), the entire package of plates transmits the motion from one set of pistons to the other. Hence, pistons (8) reciprocate in body (9). As one piston (8) is pushed in, there is a flow of liquid from cavity 'a', through channel 'b', into the central cavity 'c' and the other two cavities 'a'. Consequently, the other two pistons (8) are pushed out, which makes the plate package retain contact with all three pistons (4). Because there is a 120^0 phase shift between the pressure fluctuations in the liquid filled pipes (3), the stator's pistons impart an oscillatory motion to the plate package so that the line with the highest inclination of the plate's plane

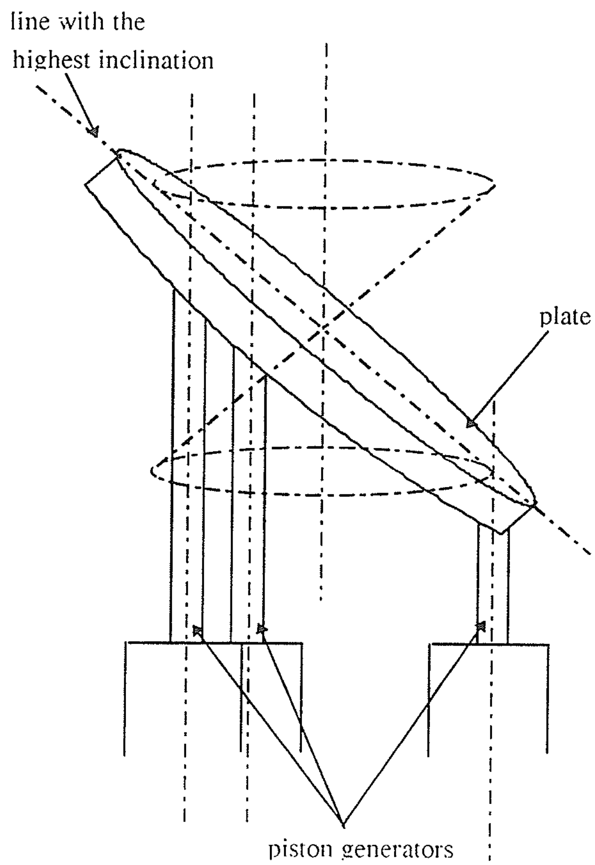

Fig. 14. Conical motion described by the plate package.

face describes a cone. (See **Figure 14.**) Consequently, the plates make the rotor rotate.

The flow resistance of the liquid inside the rotor is controlled by piston (12). It increases as this piston is pushed in so that the flow through cavity 'c' and between cavities 'a' is restricted. Besides, the flow resistance is reduced if the piston is pulled out and the fluid flows freely between all cavities. A resistant momentum at the output shaft (11) is removed only if the flow resistance inside the rotor is sufficiently high. If the fluid flows freely inside, the rotor is unable to remove the resistant momentum and it will stall. The package of plates oscillates and pistons (8) continue to reciprocate whilst the slippage between the stator and rotor is maximum. The slippage occurs between the twin plates (6) so that the friction between them is compensated by the needle bearing (7). As piston (12) is pushed in, thereby increasing the flow resistance, the rotor starts to rotate which reduces the resistant momentum. Slippage between the rotor and stator persists until the flow resistance is sufficiently high and the resistant momentum is removed. Only in this case does the rotor run synchronously with the stator and driving crank (1). The hydraulic motor develops its highest torque when piston (12) obstructs channels 'b' so that the flow resistance is greatest. For the highest flow resistance, the hydraulic motor removes the resistant momentum to the extent of the power delivered by the primary motor.

CHAPTER 4

EFFICIENCY OF WAVE-POWER TRANSMISSION

4.1 Mathematical model

The efficiency of the new wave-power transmission system is approximated next by assuming that leakage, i.e. the loss of liquid through joints or small apertures, is negligible. Also, the pipe is considered straight and perfectly continuous as well as without interior obstructions so that there is a constant flow resistance. Two identical pistons are presumed to be positioned at the generator and receiving ends of the fluid filled pipe. The generator piston is driven by a crank and rod mechanism turning at a constant speed whilst the other piston is free to oscillate at the same frequency.

The efficiency, η, is defined as the ratio of the useful work performed by a receiver, W, to the input work, W_0, done by the generator i.e.:

$$\eta = \frac{W}{W_0} . \tag{4.1}$$

Constantinesco [11] demonstrated that the efficiency of a straight transmission line has the general formula:

$$\eta = \frac{1}{\cosh 2\beta l + \frac{1}{2}\left(\frac{I}{P}\sqrt{\frac{L}{C}} + \frac{P}{I}\sqrt{\frac{C}{L}}\right)\sinh 2\beta l} \tag{4.2}$$

where β is the damping coefficient of the sound waves and l is the line's length. The line's capacity, C, and inertia, L, for example, are design parameters that are unaffected by the running conditions. Furthermore, the amplitude of the oscillating pressure wave occuring in the fluid, P, is determined by the power of the primary motor so that it is also a design parameter. Conversely, the intensity, I, of the traveling sound wave changes with the crank's angular velocity so that it changes with the running conditions. Now η has the maximum value:

$$\eta_{max} = \frac{1}{\cosh 2\beta l + \sinh 2\beta l} = e^{-2\beta l} \tag{4.3}$$

when the denominator of equation (4.2) is a minimum. It is shown in **Appendix C** that this occurs when:

$$I^2L = P^2C \,. \tag{4.4}$$

Equation (4.4) indicates that the occurrence of η_{max} is influenced by I and C. Now the capacity, C, can be increased by inserting an additional accumulator along the line. On the other hand, the intensity, I, varies with the generator plunger's frequency. Consider for example **Figure 15**. It can be seen that the generator's plunger displaces a volume of fluid during the in stroke (when the crank turns through half a revolution) and absorbs the same volume during the out stroke when the crank complete the rotation. Therefore, the volume

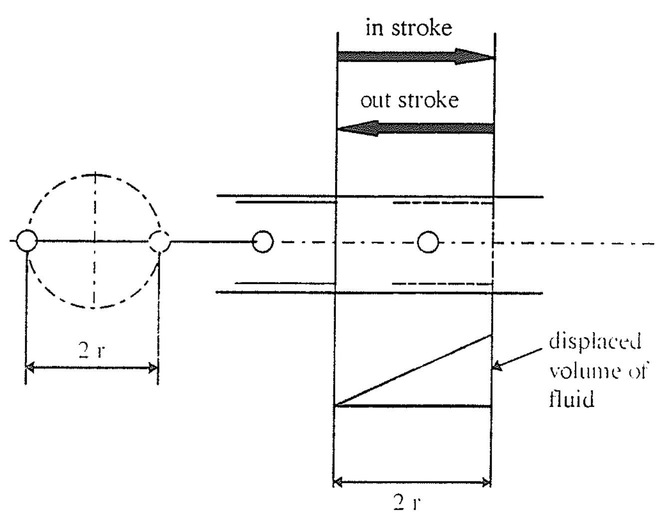

Fig. 15. Displaced volume of fluid during the in and out strokes.

of fluid, δ, displaced by a single stroke of the plunger varies directly with the double length, r, of the driving crank and with the cylinder's cross sectional area, A, i.e.:

$$\delta = 2rA \ . \tag{4.5}$$

The pressure oscillations generated by the plunger are passed from one part of the fluid to the next without a net motion of the fluid's mass. In a similar fashion to an alternating current in an electrical circuit, the oscillation can be characterized, at any instant, by an intensity and a pressure. Indeed, the intensity is identical to that of an electrical AC current whilst the pressure is analogous to an electrical voltage. The intensity of the oscillating fluid column is in phase with the piston's velocity near the generator's piston. For a crank and rod mechanism, the piston has a maximum velocity at the middle of its stroke when the crank has rotated $\pi / 2$ rads and the piston has travelled a distance r. Therefore, the maximum intensity, I_{max}, is:

$$I_{max} = r\omega A \ . \tag{4.6}$$

By combining equations (4.5) and (4.6), it can be seen that:

$$\delta = \frac{2I_{max}}{\omega} \tag{4.7}$$

or:

$$I_{max} = \frac{\delta\omega}{2}. \tag{4.8}$$

Equation (4.8) shows that, for a given stroke volume δ, the maximum intensity, I_{max}, increases with a greater angular speed, ω, of the plunger's crank.

The wave motion within the fluid is generated by successive compressions and expansions that happen at the plunger's frequency of oscillation. Consequently, the frequency of the oscillating fluid, ω, can be considered constant and equal to the crank's number of revolutions per second. Therefore, the intensity, I, and pressure, P, at any point of the fluid vary sinusoidally in time so that:

$$I = I_{\max} \sin(\omega t + \phi) \tag{4.9}$$

and

$$P = P_0 \sin \omega t \tag{4.10}$$

where ϕ is the phase angle by which the intensity leads the pressure. The last two expressions can be rewritten as:

$$I = \sqrt{2} I_{rms} \sin(\omega t + \phi) \tag{4.11}$$

and

$$P = \sqrt{2} P_{rms} \sin \omega t \tag{4.12}$$

in terms of the root - mean - square (rms) values. $I_{rms} = \dfrac{I_{\max}}{\sqrt{2}}$ and $P_{rms} = \dfrac{P_0}{\sqrt{2}}$.

In addition to the intensity and pressure, the oscillations can be characterized by a speed of propagation. The speed of the propagating sound wave, v, is determined by the (constant) cross sectional area of the pipe, a, containing the fluid. By considering the intensity of the oscillating fluid column as a flow capacity through a pipe, the speed of the liquid's oscillations, v, is:

$$v = \frac{I}{a} . \tag{4.13}$$

Therefore, the rms or effective velocity, v_{rms}, is:

$$v_{rms} = \frac{I_{rms}}{a} . \tag{4.14}$$

The efficiency, η, is a function of the damping coefficient, β, which is a design parameter affected by the running conditions. If β increases for a constant pipe length due to a very high intensity or a sudden insertion of a resistance along the line, the loss of power becomes considerable and the efficiency drops. Consequently, it is advisable to keep β as small as possible to obtain maximum efficiency. But β equals $\frac{k}{2c}$ so that it varies proportionally with the coefficient of friction, k, and inversely with the speed of sound, c. Therefore, a high efficiency requires a low coefficient of friction, k. In a liquid that oscillates in a pipe, viscous friction will occur at the surface of the pipe and in the body of the liquid itself. The coefficient of friction is related to the rms velocity, v_{rms}, and the pipe's inside diameter, d, by [11]:

$$k = \frac{v_{rms}}{100d}\left(1 + \frac{9}{\sqrt{v_{rms}d}}\right) . \tag{4.15}$$

Consequently, for a given liquid, generator and pipe, the efficiency of a transmission line can be calculated from equation (4.3) by using the expressions for k and β.

4.2 Experimental model

The efficiency of a transmission line and the relationships between the design parameters and the parameters affected by the running conditions are established through an experimental system in which a generator's plungers are operated by a swash plate. The experimental model is described in **Chapter 3.** The plungers' motion varies simple harmonically in time because the swash plate is turning uniformly so that it is equivalent to the crank considered theoretically. A 1/4 HP (i. e. $1/4*764.5=191.1W$) DC. electrical motor turns the plate at $n=600$ *rpm.* Each of the generator's pistons has a stroke of $h=8$ *mm* and each piston fits into a cylinder whose interior diameter is $D=25.4$ *mm.* By inserting numerical data for the stroke's volume and angular velocity in equation (4.8), the maximum intensity of the oscillating fluid column at the generator end is:

$$I_{ms} = \frac{2*4*506.7*\dfrac{2\pi 600}{60}}{2} = 127.4*10^{-6}\ m^3/s \tag{4.16}$$

for the particular experiment undertaken. In practice, distances of several centimeters to several meters are common between a pumping station and hydraulic motor. The experimental model used a copper pipe with a length of $l=450$ *mm* and an inside diameter of $d=7.5$ *mm* to connect the generator to the receiver. Substituting these particular values into equation (4.14) as well as the $127.4*10^{-6}\ m^3/s$ intensity given by equation (4.16) leads to a *rms* velocity of:

$$v_{rms} = \frac{\frac{127.4 * 10^{-6}}{\sqrt{2}}}{44.2 * 10^{-6}} = 2038.3\ mm/s = 7330\ m/h. \tag{4.17}$$

There is a wavelength, λ, associated with the wave motion because the disturbance produced by the generator's plunger is periodic in time. The inertia of the liquid opposes the piston's movement at any instant and, consequently, the nearby liquid is compressed on the in-stroke of the piston. The pressure at each point along the pipe oscillates between $\pm P$. The particle velocity is zero at the points of maximum and minimum pressure and it varies from $+v$ to $-v$ at the nodal points where the pressure variation is zero. A receiver connected at any of the points having a maximum pressure variation, that is at a multiple of a half or full wavelength, receives all the energy given to the line. At these points the speed of the propagating wave is always zero while the pressure alternates between its maximum and minimum values. This observation suggests that additional branches can be connected to a (long) main line at points corresponding to multiples of half and full wavelengths. On the other hand, to simultaneously distribute power to several machines, the main pipe may have several branches with a plunger at the end of each one. In the event that one of these machines is not in use, and, hence, not absorbing power, the corresponding branch may be disconnected from its plunger or the plunger may be locked. At multiples of the quarter and three-quarter wavelength points, conversely, the speed alternates between its maximum and minimum values but the pressure variation remains zero. Therefore, a receiver mounted at such positions is unable to run. If a receiver is connected at any intermediate point, it will utilize only part of the energy transmitted along the line. The remaining part is reflected back to the generator to increase the pressure along the line. As the pressure increases, the capacities that are mounted in the neighborhood of the generator (see **Figure 12**) act as springs that accumulate the "surplus" energy and return it when the pressure drops. In this way, the pressure does not exceed the high values at which either the pipe, generator or receiver might burst or be damaged.

To see how the efficiency is affected by the pipe's interior diameter, d, the particular values of $d = 7.5mm$ and the previously determined $v_{rms} = 2038.3mm/s$ were substituted into equation (4.15) to find the corresponding coefficient of friction, k. It was calculated to be

$$k = \frac{2038.3}{7.5 * 10^2}(1 + \frac{9}{\sqrt{2038.3 * 7.5}}) = 2.915s^{-1}. \tag{4.18}$$

Now the coefficient of absorption, β, is given by:

$$\beta = \frac{k}{2c}; \tag{4.19}$$

in which, as a first approximation, the speed of sound through the liquid employed (viz water) can be considered to be about $c = 1440m/s$. Consequently, by employing equations (4.18) and (4.19), it can be shown that:

$$\beta = \frac{2.915}{2 * 1440 * 10^3} = 10^{-7}cm^{-1}.$$

Then, the efficiency of the designed transmission line can be determined from equation (4.3) to be:

$$\eta = e^{-2*10^{-7}*450} = 0.99.$$

Thus, with a proper design of the hydraulic components, a transmission line can work with a very high efficiency.

4.3 Influence of various parameters on efficiency

4.3.1 Influence of friction

Friction occurs during the oscillation of the fluid and it is a function of the coefficient of friction k, the liquid's density, ρ, and the length of pipe l. One way of decreasing the friction is to increase the pipe's diameter, thereby lowering the *rms* velocity. Equation (4.15) demonstrates that, for the same value of k, greater *rms* velocities may be employed in pipes having correspondingly greater diameters. (Refer to **Appendix A** for a further development.) It follows that, in order to obtain the maximum efficiency for a given pipe diameter, the greater is the pressure difference that produces the flow then the greater should be the intensity and effective velocity.

Another way of decreasing the friction is to change the liquid's density. As the viscosity of many liquids, particularly hydraulic oils, is very sensitive to temperature (see **Figure 16**), considerable changes in k occur with changes in the ambient temperature of the transmitting liquid. A previous study [6] confirms this observation and concludes that improvements in the power output could be achieved by selecting a fluid with a low density

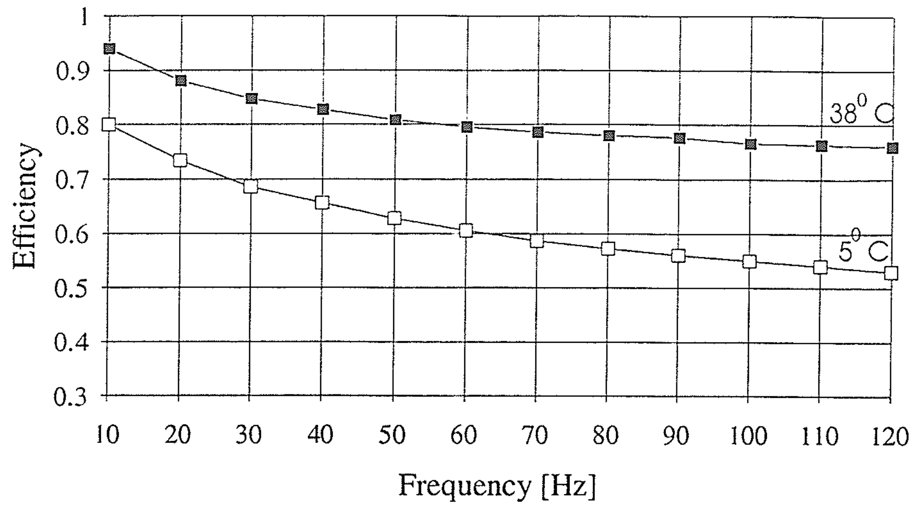

Fig. 16. Maximum theoretical efficiency for Tellus 27 [6].

P=3.4 N/mm^2, d=22.2 mm and l=24.5 m.

like water.

It should be noted from equation (4.15) that the coefficient of friction, k, is affected by the *rms* speed of the propagating waves and by the diameter of the pipe. The *rms* speed varies directly with the intensity of the oscillating fluid column. Therefore, the *rms* velocity becomes a function of the crank's angular speed, ω. A high angular speed imposes an intense alternating pressure that would produce a high friction inside the pipe and, consequently, a high coefficient of friction. **Figure 17**, for example, shows that the transmission efficiency decreases almost linearly with an increasing rotational speed of the crank for a given set of generator and pipe dimensions. The curve was obtained by calculating the maximum efficiency for various rotational speeds whilst the dimensions of the generator and pipe were kept constant. On the other hand, the graph shows that the efficiency decreases by only 5% for an increase of 1000 rpm so that the crank's rotational speed is hardly influential.

The inside pipe diameter is the second parameter that could affect the coefficient of friction and, therefore, the maximum efficiency. **Figure 18**, for instance, illustrates the effect on the efficiency of changing the pipe's diameter, d, whilst the *rms* velocity v_{rms} is constant. Various pipe diameters ranging from 2.5 mm to 30 mm were used in equation (4.15) for a given constant *rms* velocity. The efficiency was then calculated by using equation (4.3). The curves show that, for a given *rms* velocity, the efficiency decreases with a reduction in the pipe's diameter due to a higher coefficient of friction. The same graph also leads to the

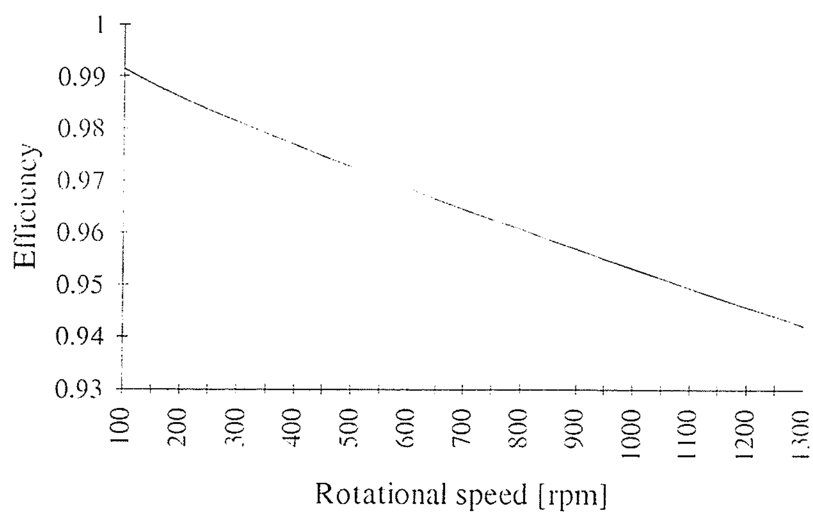

Fig. 17. Effect of the crank's speed on the transmission efficiency.

D=25.4 mm, h=8 mm, d=7.5 mm, l=11000 mm.

conclusion that a larger effective velocity reduces the transmission efficiency, particularly for a pipe diameter smaller than about 10 mm.

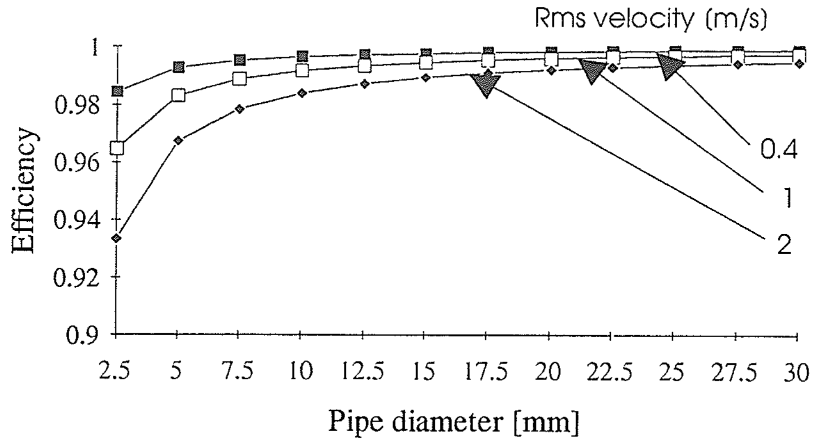

Fig. 18. Effect of changing the pipe diameter on the transmission efficiency.

l=11,000 mm.

4.3.2 *Influence of stroke's volume*

In addition to the frequency of the plunger's harmonic motion and the pipe's diameter, the stroke's volume is another factor that influences the efficiency, η. A high volume of fluid passing through a given cross section of a pipe means an intense, alternating flow which corresponds to high internal friction losses that drastically reduce η. **Figure 19** shows that η decreases by more than 10% when large increases in the stroke's volume occur. The curves were obtained for a given pipe diameter and length by using equations (4.3) and (4.15) for different stroke volumes and rotational speeds. A comparison of the curves given in this figure shows that the efficiency is affected more by a combination of the stroke volume and rotational speed of the crank because an intense oscillating motion of the fluid raises the friction coefficient.

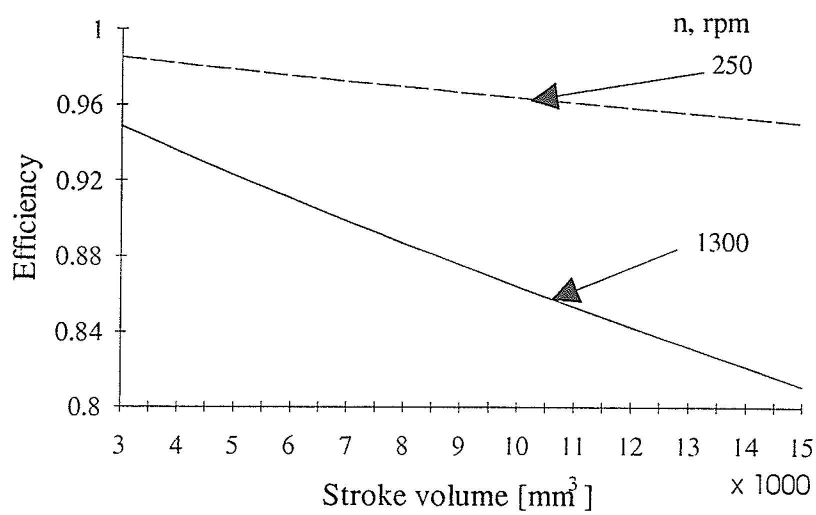

Fig. 19. Influence of stroke volume on efficiency.

d=7.5 mm. l=1100 mm.

CHAPTER 5

Experimental Results

The principle of the new transmission system and a functional description of it were given in Chapter 3. A set of experiments was conducted to determine the efficiency and mechanical characteristics at the output shaft of the prototype. The arrangement for the tests is shown in **Figure 19**. The results indicate that the overall efficiency of the hydraulic transmission involves two components: 1) the efficiency of the pump and hydraulic motor; and 2) the efficiency of the hydraulic system itself.

The hydraulic generator and motor each consisted of three cylinders distributed equally at 120^0 increments around the axis of symmetry. The cylinders had a bore and stroke of 25.4 mm and 8 mm, respectively. The hydraulic generator was rotated by a 190 W electrical motor to produce oscillations of the water columns filling the three pipe lines. These oscillations differ in phase by 120^0 and they were received by the hydraulic motor causing it to rotate. The output power was absorbed by the Prony brake. The test consisted of, first, determining the efficiency of the hydraulic motor for a constant speed and then finding the torque developed by the hydraulic motor at various loads and rotational speeds. In performing the test, the efficiency of the hydraulic motor was obtained by comparing the power required by the electrical motor to drive the hydraulic generator at a constant speed with that needed to drive the generator and motor at the same speed. The hydraulic motor was removed initially and the pipe lines were opened and placed vertically to avoid water losses. Therefore, when the electrical motor drove the hydraulic generator, the columns of water were free to oscillate inside the pipe lines. It was only possible to run the generator below 120 rpm because even a small loss of water from one column would upset the entire system and the resulting readings were inconsistent. The same conclusion was corroborated later when small quantities of entrapped air or water losses in one of the columns resulted in a malfunction of the hydraulic transmission.

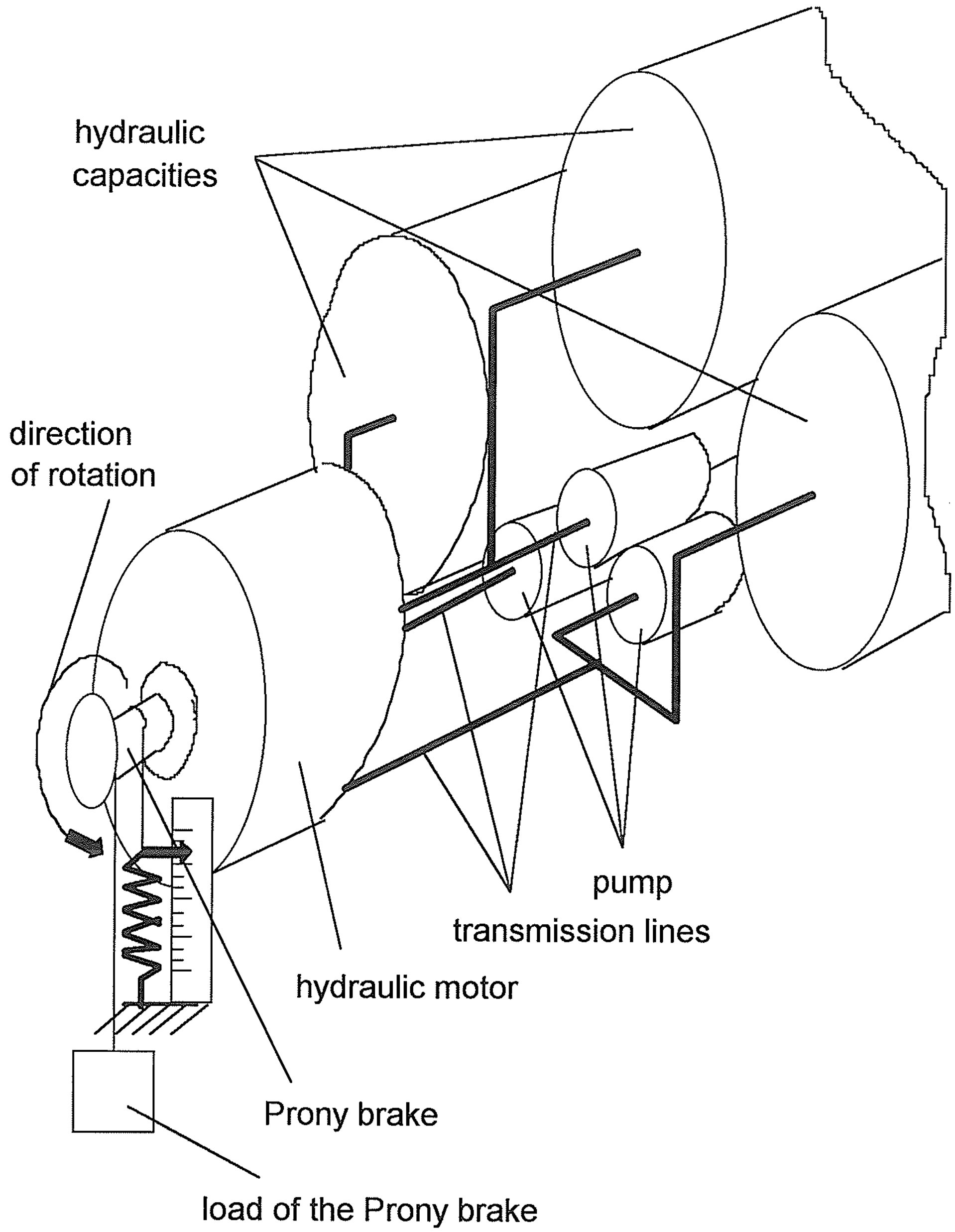

Fig. 20. Test arrangement.

prolonging the pipes and by taking more readings, the system was ran over the entire speed range of 90 rpm to 250 rpm yielding predictable and relatively consistent data.

The apparent power, which is simply the product of the electrical voltage and current required by the DC electrical motor to drive the hydraulic motor, was obtained by simultaneously measuring the voltage and current by using a standard voltmeter and ammeter. After the absorbed power was first measured at 90, 100, 110 and 120 rpm, and then over the entire speed range, the stator and then the rotor of the hydraulic motor were mounted successively and the measurements were repeated. The efficiencies of the stator and hydraulic motor were established by comparing the power required by the electric motor to drive the transmission with and without the hydraulic motor or its components connected. The efficiency of the hydraulic motor appeared to be constant at 92%. (Refer to **Appendix D** for a complete set of data.) However, the figure is credible because the efficiency of a transmission line was calculated to be around 99% and, therefore, the efficiency of all three lines would be 99% * 99% * 99% or about 97%. Moreover, the stator's efficiency was 95% for the same rotational speeds of the hydraulic generator. Consequently, the overall efficiency of the hydraulic transmission should be around 92% * 92% * 97% or about 80%. This figure is based on the assumption that the hydraulic motor and generator worked with about the same efficiency because they have a similar design.

The torque developed by the hydraulic motor was determined by loading the Prony brake with weights varying from 200 grams to 1200 grams (in increments of 100 grams) while the rotational speed of the hydraulic motor was varied separately from 90 rpm to 250 rpm in increments of 10 rpm for each load. Either electric or hydraulic motor did not stall under any of these loads so that they do not represent maximum or characteristic values. The speed was found by using a tachometer at the end of the hydraulic motor's shaft. For each combination of weight and rotational speed, five readings were taken and these results are shown in **Figures 21** and **22**. It can be noticed from these figures that the hydraulic motor

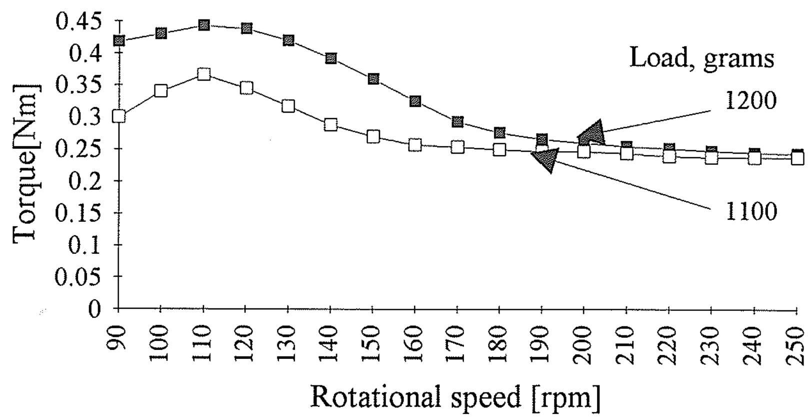

Fig. 21. Torque variations for a constant load.

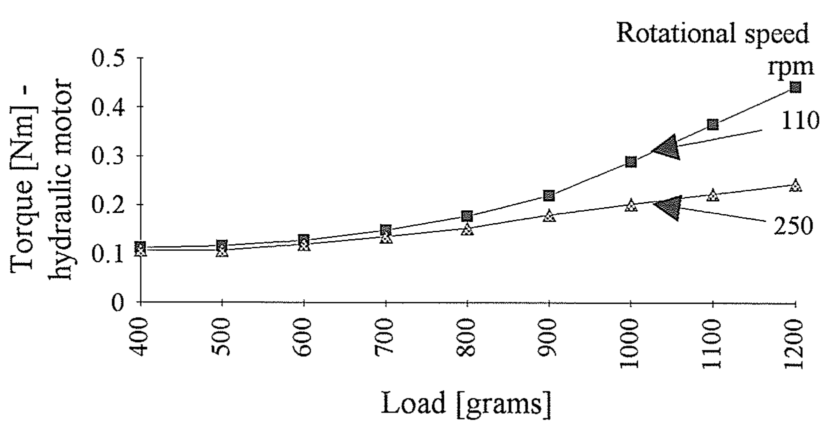

Fig. 22. Torque variations for a constant rotational speed.

developed a high torque under the larger loads as well as over a fair range of rotational speeds. This characteristic would make the motor suitable for traction applications.

During the tests, the configuration of the transmission system was changed in order to establish the effect of the hydraulic accumulators. Three main configurations were analyzed. In the first one, the accumulators were inserted along the pipe lines so that the pressure waves produced at the generator traversed the accumulators. In the second configuration, the capacities were mounted near the generator just communicating with the pipe lines. (See **Figure 20**.) In the third configuration, the accumulators were disconnected by obstructing the link between each vessel and its corresponding pipe line. The effect of the hydraulic capacities was established by comparing the power required by the electrical motor to drive the transmission system with and without the accumulators. Small variations of about ±0.5 W in the power absorbed by the electrical motor for the two configurations were attributed to experimental errors *. The results are presented in **Figure 23**. It is noticeable that the electrical motor absorbs more power when it drives a system without accumulators.

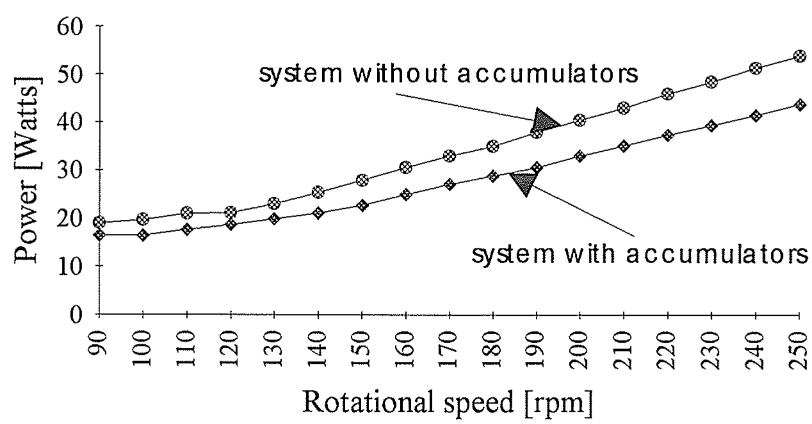

Fig. 23. Comparison of power variations for a system with and without accumulators.

* See Appendix D.

this case, the power increases by about 16% at 90 rpm and by 23% or so at 250 rpm. This phenomenon can be explained by an increasing pressure inside the pipe lines due to reflected pressure waves from the hydraulic motor. If a high resistance occurs at the output shaft, the rotor of the hydraulic motor slows down and, for an instant, the stator would not absorb the entire energy transmitted through the water. The pressure waves coming from the generator are returning from the stator pistons' surfaces to increase the pressure inside the pipe-lines until the resistance is removed. A higher pressure inside the pipe lines turns into a higher resistance at the electrical motor's output shaft. Consequently, the electrical motor absorbs more power to continue rotating against the increased resistance.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

It can be concluded that the advantages of the newly designed power transmission system are high torque and efficiency, simplicity, reliability and low cost. Moreover, the mechanism that transforms the reciprocating and rotating motions also incorporates a power control device for smoothly modulating the output power of a rotating shaft. The system runs with above 80% efficiency over operating speeds between 90 rpm and 120 rpm. With a proper sizing of the hydraulic components, the transmission system yields an improved drive efficiency, particularly at low speeds when maximum tractive effort is demanded. The 80% peak overall efficiency exceeds by more than 6% the 74% peak achieved by hydrostatic drives [7].

The system incorporates simple components like cylinders, pistons, bearings and plates. None of the parts require an elaborate manufacturing process. Moreover, a dusty and rugged environment cannot contaminate the working fluid that fills the pipes. Such an environment has less corrosive impact because most components are not exposed directly, they are sealed or protected very easily. Consequently, the motor's expected life should be extended and safe even for work in a highly corrosive or explosive environment. Also, a very low price (about $ 100 for the prototype) is achievable because most components are inexpensive and commercially available. The issue of reliability, however, is based only upon about 700 hours running time of the prototype. On the other hand, the maintenance cost should be low because the parts are cheap and they all can be replaced independently without affecting the entire system. The job of replacing a part would merely require an average skilled worker.

The power control device is essentially a piston that controls the internal flow resistance inside the rotor while the primary motor runs at a constant speed. This resistance can be increased gradually, either by an operator or automatically until the motor develops the highest torque. The torque varies smoothly because the flow resistance is controlled precisely by a uniform positioning of the piston.

Another important conclusion that relates to conventional hydraulic systems can be drawn from the experiments. The efficiency of such a system can be raised by adding a vessel filled completely with fluid in the neighborhood of the pump. A fixed displacement pump, which is common in hydraulics, delivers hydraulic fluid, under a preset pressure, at the rate needed for a desired cylinder speed. The flow rate is varied by an operator through a control device. If the cylinder is a component of a hydraulic motor, the piston will slow down when the resistant momentum at the output shaft is too high, but the flow rate will remain constant. Consequently, the flow will return from the piston and the pressure will increase along the pipes for a short time. This results in a peak power absorbed by the primary motor and, very possibly, in its overload. A vessel similar to that used in the tests will absorb the reflected energy when the pressure increases and it will return the energy when the pressure decreases. Such a hydraulic capacity will act as a pressure regulator that smoothes the pressure variations and avoids the power peaks absorbed by the primary motor.

A second conclusion that also relates to hydrostatic systems is that the efficiency of a transmission system can be raised through the reduction of the friction coefficient inside the pipe. Two ways are available to achieve this goal. The first way is to design a new pipe for the transmission line while the second one is to modify the speed of the fluid along the pipes. A new design of transmission pipe has to incorporate something similar to a dolphin's skin. They are well known for their very efficient propulsion system and their special body construction. On the other hand, the speed of the fluid can be modified by introducing different devices along the transmission line. The simplest device is a resistance mounted at the receiver end of the pipe. The resistance can be a pipe with a small diameter and a certain length or even a simple faucet.

6.2 Recommendations

The features of the hydraulic motor indicate that the new transmission system is very suitable for driving or even replacing systems working in a harsh environment. In addition, the motor does not have hot parts and there is no danger of generating sparks that would ignite an explosive medium. The motor is able to run without special protection under water, in places with dust or highly explosive materials and under extreme atmospheric conditions.

The real possibility of connecting the outputs from one rotor to the stator of a second motor makes a mechanical and hydraulic connection feasible between two or more motors. Coupling two or more motors on the same shaft creates values of torque and angular speed similar to those of a gear box. Any gear box can be replaced with such a compound motor that would preserve the same output characteristics. However, in comparison with a gear box, the compound motor is more flexible, even when it is set for a particular output torque. Through the hydraulic motor's control devices, the torque and angular speed can be independently varied, within limits, by the primary motor. Consequently, the compound motor might offer fuel savings. Moreover, the manufacturing cost of the compound motor is anticipated to be smaller than that of a gear box.

The motor's characteristics are also exploitable by variable speed air compressors (VSD). A conventional load/unload machine has a minimum 0.5 bar and an average 0.8 bar pressure fluctuation while field experience shows that the new type of machine keeps the pressure stable with a maximum +/-0.1 bar fluctuation [12]. VSD compressors allow a lower pressure setting in the compressed air net to guarantee a required minimum pressure.

This means an energy saving because a 1 bar higher pressure requires 6% more power. The new compressors ensure a true "soft" start so that there are no current peaks at start-up. VSD compressors also allow smaller buffer air receivers in the customer's airnet which is a cost saving at installation. Variable speed compressors have converters that rapidly adapt the motor's speed and power to new running conditions which would result in energy savings. Due to its torque and speed variability, the new hydraulic motor can drive air compressors on mobile equipment and yet preserve all the advantages of VSD compressors.

In addition to all these advances, the mechanism that transforms the linear into a rotary motion can be used independently to create a new family of poly-fuel engines. Variable compression ratio, poly-fuel engines require an automatic or very easily adjustable compression ratio. The primary characteristic of conventional mechanisms is reliability but they are complicated, bulky and require a crankcase having a large volume. In addition to reliability and simplicity, the new mechanism offers power modulation of a rotating output shaft. The new mechanism fulfills the condition of an easily adjustable compression ratio if a stator's cylinders are replaced by the cylinders of an engine. Internal flow resistance and pressure inside the rotor can easily adjust the stroke and, therefore, the compression ratio of the engine. A high pressure inside the rotor would push out the rotor's pistons which would increase the compression ratio. A reverse process would take place for a low pressure. The mechanism is elegant and very simple, yet it has only a few mechanical parts and offers the possibility of automation.

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APPENDIX A

Relationship between speeds of alternating pressures for two pipe diameters

The following discourse demonstrates that , for the same k, greater *rms* velocities, v_{rms}, may be employed in pipes having correspondingly greater diameters, d.

For a given pipe diameter d_1, the coefficient of friction is given by equation (4.15) of the main text, viz:

$$k_1 = \frac{1}{100 * d} v_{rms1} (1 + \frac{9}{\sqrt{v_{rms1} d_1}}). \tag{A.1}$$

Equation (A.1) can be rewritten as:

$$k_1 = \frac{v_1}{100 * d_1} (1 + \frac{9}{\sqrt{v_1 d_1}} \sqrt{\frac{v_1}{v_1}}) = \frac{v_1}{100 * d_1} (1 + \frac{9}{v_1} \sqrt{\frac{v_1}{d_1}}) \tag{A.2}$$

where $v_{rms1} = v_1$.

By using the notation:

$$u_1 = \sqrt{\frac{v_1}{d_1}}, \tag{A.3}$$

the coefficient of friction becomes:

$$k_1 = \frac{1}{10^2} u_1^2 (1 + \frac{9}{v_1} u_1) \tag{A.4}$$

Subscript 1 is used to designate variables corresponding to the first diameter.

Similarly,

$$k_2 = \frac{1}{10^2} u_2^2 (1 + \frac{9}{v_2} u_2) \text{ where } u_2 = \sqrt{\frac{v_2}{d_2}} = nu_1 \qquad \text{(A.5)}$$

for a pipe having an interior diameter, d_2.

But the coefficients of friction in the two cases have to be equal because the pipe lines have to work with the same efficiency. Hence $k_1 = k_2$

or

$$\frac{1}{10^2} u_1^2 (1 + \frac{9}{v_1} u_1) = \frac{1}{10^2} u_2^2 (1 + \frac{9}{v_2} u_2) . \qquad \text{(A.6)}$$

Combining equations (A.5) and (A.6) leads to:

$$u_1^2 (1 + \frac{9}{v_1} u_1) = nu_1^2 (1 + \frac{9}{v_2} nu_1) \qquad \text{(A.7)}$$

or, by rearranging:

$$v_2 = \frac{9n^2 u_1 v_1}{(1-n)v_1 + 9} \Rightarrow v_2 > v_1 \qquad \text{(A.8)}$$

Equation (A.8) shows that higher velocities may be employed in pipes with greater diameters. But higher diameter and *rms* velocities mean a higher intensity of the oscillating liquid column. Consequently, for maximum efficiency, the greater is the intensity, the greater should be the pressure along the lines. (See equation (4.4) of the main text.)

APPENDIX B

Calculating the design parameters of a transmission line and generator for a specified efficiency

The parameters of a transmission line and generator are calculated for a specified efficiency. It is assumed that the efficiency is imposed and that the transmitting fluid and power of the primary motor are known.

The maximum power transmitted through a wave-power transmission system is [11]:

$$W = \frac{RI^2}{2} \tag{B.1}$$

where R is the resistance of the transmission line; and I is the intensity of the oscillating pressure. But:

$$R = \frac{k\rho l}{ga} \quad \text{and} \quad a = \frac{\pi d^2}{4} \tag{B.2}$$

so that:

$$R = \frac{4k\rho l}{\pi g d^2} \ . \tag{B.3}$$

On the other hand, the *rms* intensity is:

$$I_{rms} = v_{rms} a \quad \text{where} \quad I_{rms} = \frac{I}{\sqrt{2}} \ . \tag{B.4}$$

Consequently, the intensity at the receiving end of the pipe may be written as:

$$I = \frac{\pi v_{rms} d^2}{2\sqrt{2}}. \tag{B.5}$$

Substituting equations (B.3) and (B.5) into equation (B.1) leads to:

$$W = \frac{k\rho l \pi v_{rms}^2 d^2}{4g} \tag{B.4}$$

or

$$v_{rms} d = \sqrt{\frac{4gW}{k\pi\rho l}}\ . \tag{B.5}$$

The coefficient of friction in the transmission line is

$$k = \frac{v_{rms}}{d} * \frac{1}{100}(1 + \frac{9}{\sqrt{v_{rms} d}}) \tag{B.6}$$

so that:

$$k = \frac{v_{rms}}{d} * \frac{1}{100}(1 + \frac{9}{\sqrt[4]{\frac{4gW}{\pi k\rho l}}}) \tag{B.7}$$

or:

$$\frac{v_{rms}}{d} = \frac{100k}{1 + \frac{9}{\sqrt[4]{\frac{4gW}{\pi k\rho l}}}}\ . \tag{B.8}$$

A rms velocity and the required pipe's inside diameter are obtained by solving equations (B.5) and (B.8). But:

$$\beta = \frac{k}{2c} \tag{B.9}$$

or

$$k = 2\beta c \tag{B.10}$$

and:

$$\eta = e^{-2\beta l} \tag{B.11}$$

which implies that:

$$k = -\frac{\ln \eta}{l} c \ . \tag{B.12}$$

Consequently, equations (B.5) and (B.8), respectively, become:

$$v_{rms} d = \sqrt{\frac{4gW}{\pi(-\ln \eta) c\rho}} \tag{B.13}$$

and

$$\frac{v_{rms}}{d} = \frac{100(-\ln \eta)\frac{c}{l}}{1 + \frac{9}{\sqrt[4]{\frac{4gW}{\pi(-\ln \eta) c\rho}}}} . \tag{B.14}$$

The pipe's inside diameter, d, and, the rms velocity, v_{rms}, are found by solving equations (B.13) and (B.14). The velocity of the oscillating pressure is:

$$v = \sqrt{2} v_{rms} \qquad \text{or} \qquad v = \frac{4I}{\pi d^2} \ . \tag{B.15}$$

Therefore:

$$I = \sqrt{2}\frac{\pi}{4}v_{rms}d^2 \qquad \text{or} \qquad I = 1.11v_{rms}d^2. \tag{B.16}$$

Finally, the stroke volume of the plunger, δ, is:

$$\delta = \frac{2I}{\omega} \qquad \text{or} \qquad \delta = 2.22\frac{v_{rms}d^2}{\omega}. \tag{B.17}$$

The final design parameters of the generator are found by solving equation (B.17) and imposing different construction constraints involving the pipe's diameter or the crank's angular speed.

APPENDIX C

The condition for maximum efficiency

The condition for maximum efficiency of a transmission line is calculated from:

$$\eta = \frac{1}{\cosh 2\beta l + \frac{1}{2}(\frac{I}{P}\sqrt{\frac{L}{C}} + \frac{P}{I}\sqrt{\frac{C}{L}})\sinh 2\beta l} \, . \tag{C. 1}$$

The maximum happens when the denominator of equation (C.1) is minimum, that is when:

$$\frac{1}{2}(\frac{I}{P}\sqrt{\frac{L}{C}} + \frac{P}{I}\sqrt{\frac{C}{L}}) = 1 \tag{C. 2}$$

or:

$$\frac{I}{P}\sqrt{\frac{L}{C}} + \frac{P}{I}\sqrt{\frac{C}{L}} = 2 \, . \tag{C. 3}$$

From equation (C.3) it is noticeable that:

$$\frac{P}{I}\sqrt{\frac{C}{L}} = \frac{1}{\frac{I}{P}\sqrt{\frac{L}{C}}} \, . \tag{C. 4}$$

Consequently, equation (C. 3) can be rewritten as:

$$\frac{I}{P}\sqrt{\frac{L}{C}} + \frac{1}{\frac{I}{P}\sqrt{\frac{L}{C}}} = 2 \, . \tag{C. 5}$$

It can be concluded from equation (C. 5) that the condition for maximum efficiency is produced when:

$$\frac{I}{P}\sqrt{\frac{L}{C}} = 1 \ . \tag{C. 6}$$

Hence, by squaring and rearranging:

$$I^2 L = P^2 C \ . \tag{C. 7}$$

APPENDIX D

Measured data

Table 1

Electric power absorbed by an unloaded system and its calculated efficiencies

RPM electric motor	power in (W_0) - without stator [W]	power in [W_1] - with stator [W]	% stator eff. (W_0/W_1 *100)	power in (W_2) - with stator and rotor [W] (unloaded system)	% rotor eff. (W_1/W_2*100)	% eff hyd motor (W_0/W_2*100)
180	15.0	15.7	95	16.4	95	90
200	15.3	15.8	97	16.5	95	92
220	16.4	16.9	97	17.6	96	93
240	17.3	17.9	96	18.7	95	91
260	18.6	19.0	98	20.0	95	93
280	19.7	20.3	97	21.2	95	92
300	21.0	21.9	96	22.8	96	92
320	23.0	24.0	95	25.0	96	91
340	25.2	25.6	98	27.0	95	93
360	26.5	27.6	96	29.0	95	91
380	28.4	29.3	97	30.6	95	92
400	30.8	32.0	96	33.1	96	92
420	32.7	33.7	97	35.2	95	92
440	34.8	35.8	97	37.4	95	92
460	36.7	37.8	97	39.5	95	92
480	38.7	40.0	96	41.6	96	92
500	40.8	42.0	97	44.0	95	92

Table 2

Electric power absorbed by a loaded system and its overall efficiency

RPM hydraulic motor	torque [Nm] - system with 1200g load	torque [Nm] - system with 1100g load	power in [W_3] - system with a 1200g load [W]	% overall efficiency hyd. system (W_0/W_3) - loaded system [W]	power in - system without vessels,unloaded [W]
90	0.418	0.300	20.3	74	19.0
100	0.430	0.340	20.1	76	19.7
110	0.443	0.366	22.6	72	21.0
120	0.438	0.345	24.0	72	21.2
130	0.420	0.317	25.7	72	23.0
140	0.392	0.288	27.0	73	25.4
150	0.360	0.270	28.5	73	28.0
160	0.325	0.257	30.4	75	30.7
170	0.293	0.254	32.3	78	33.0

(Table 2 cont.)

RPM hydraulic motor	torque [Nm] - system with 1200g load	torque [Nm] - system with 1100g load	power in [W_3] - system with a 1200g load [W]	% overall efficiency hyd. system (W_0/W_3) - loaded system	power in - system without vessels,unloaded [W]
180	0.276	0.250	34.4	77	35.1
190	0.266	0.247	36.1	78	38.0
200	0.260	0.247	38.5	80	40.5
210	0.255	0.244	40.8	80	43.1
220	0.252	0.240	43.2	80	46.0
230	0.248	0.238	45.5	76	48.6
240	0.245	0.238	47.8	80	51.5
250	0.244	0.238	50.3	81	54.0

Note:

The maximum variations in the voltage measurements were ±0.8 V. Similarly, they were ±0.6 A for the current. The power absorbed by the DC motors equals W=I*V so that the maximum variation in the power is 0.8V*0.6A=0.5W.