The University of Salford

Department of Civil Engineering

A historical survey of low-head hydropower generators and recent laboratory based work at the University of Salford

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Degree of Doctor of Philoscphy

by

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List of Contents

Page

Page

List of Figures

Page

Page

Chapter 4 - The AUR Water Engine

Page

Chapter 5 - The Salford Transverse Oscillator

Chapter 6 - The Underwater Motor

Page

Chapter 7 - A hydro-pneumatic generator

Appendix 1 - Prototype Water Engines

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A HISTORICAL SURVEY OF LOW-HEAD HYDROPOWER GENERATORS AND RECENT LABORATORY BASED WORK AT THE UNIVERSITY OF SALFORD

Abstract

All life depends upon water. For man, however, water can supply the power necessary to relieve him from the drudgeries of life and give the time and means to enrich his existence beyond the bounds set by the mere need to survive.

Large scale hydropower makes a significant contribution to the total generated power of the developed world. The means of harnessing water power for large flows and heads above three metres are efficient and well established. There remains however, the fact that many people, especially in developing countries live adjacent to water courses and tidal estuaries where heads of 2 metres or less exist but are not exploited. The reasons for this are often that practical machines do not exist or are insufficiently well developed to be used. Much effort has been made in recent years to develop equipment suitable for small, low-head sites in remote places.

The writer has worked on three such devices in both the laboratory and on site - these are the Salford Transverse Oscillator, the AUR Water Engine and the Underwater Motor.

This thesis describes these machines in detail together with other contemporary devices.

The initial chapter contains an assessment of machines used in the past to exploit low head sites. There is little doubt that some of these older machines, suitably brought up to date in design and materials, would be more suited to the needs of developing peoples than many of the esoteric devices described in later chapters.

The thesis is concerned mainly with the need to provide a means of generating power rather than the assessment or suitability of a site for power generation. The writer draws on experience gained in the laboratory, and as a Mechanical Engineering Consultant for small installations in the United Kingdom and overseas.

SMALL SCALE LOW HEAD HYDROPOWER

Introduction

Power is not essential to life, but power, especially electric power greatly improves the quality of life. It can provide light to extend the day; it can work refrigerators to preserve food and medicines improving health and prosperity. Television, radio and telephones bring entertainment, education and instant contact with the rest of the world. Yet, in spite of all these obvious benefits, many millions of people do not enjoy them. The cost of providing the electricity cannot be met by the society in which they live. Even if the capital cost of a diesel generator is met by others, the costs of fuel, maintenance and replacement are often an unbearable burden.

The wind can provide a free source of energy, but save for a few unfavoured locations the wind is fickle and the power output of wind machines is very variable. Although windmills have a long ancestory they still require considerable maintenance to remain reliable and are vulnerable to nature's whims. Furthermore efficient machines require high technology in terms of gearboxes designed to withstand widely fluctuating loads with mechanisms to control pitch, slew and runaway.

Most human settlements are near to moving water which can be harnessed to provide a reliable supply of power. Many devices have been developed over the centuries to do this, hence the technology is widely known and understood.

CHAPTER 1

Past Methods of Power Generation from Water

1.0 Water wheels

The machines used in the distant past were of necessity of simple construction made with natural materials. Until the sixteenth century they were made almost entirely of wood.

Two basic designs existed side by side - the vertical axle/horizontal wheel type generally known as the Norse wheel or sometimes the Greek Mill due to a description in the writings of Antipater C.60 BC and the horizontal axis water wheel first described in the writings of the Roman Vitruvius C.25 BC.

Both types were made in a variety of forms to suit local conditions or skills, and both types exist in various parts of the world to the present day.

The application of modern materials to improved versions of these old designs is one way of approaching the problem of providing equipment in remote locations that can be made and maintained by local craftsmen.

1.1 The Norse Wheel

In its simplest form this consists of a pole to which flat pieces of wood are fixed at the lower end as shown in figure 1.1a. A mill stone is affixed to the upper end and the lower end is set upon a rock in the stream near the bank so that only one side is in the flowing water. This causes the wheel to turn so driving the mill stone which is usually set above the stationary stone, the hole in the lower stone acting as a bearing for the pole.

This machine is most suitable for fast flowing streams in hilly regions - the power developed being small, typically less than 0.5kW.

A development of this machine raises the paddles above the water course - so reducing drag - and directs the feed water onto the paddles from a chute. This allows much better control of the wheel by enabling the water jet to be deflected easily when it is required to stop the mill wheels.

In the hands of the Moors this type of wheel reached a high degree of sophistication. Figure 1.1b shows the type of wheels used at Cordoba, Spain in the 10th Century AD. The vanes have now become curved to give a degree of reaction and an outer rim has been added. The rim has the double benefit of strengthening the wheel and containing the water jet.

Figure 1.1c shows the arrangement of traditional vertical axis mills, still in use today in parts of Nepal. The moving parts are of wood with the bottom bearing on a pivoted pole so that it can be raised or lowered to adjust the clearance between the mill stones.

Although the vertical axis wheel probably preceded the horizontal axis type it could not, in its original form, produce very much power and hence was rarely used for anything other than grinding corn.

The power for mechanised industry, mining, pumping and other applications came with the invention of the horizontal axis wheel.

1.2 The Horizontal Axis Water Wheel

This type of wheel was described by the Roman author Vitruvius (C.25 BC) in his work "Ten books on architecture" (reference 1). Hence is has been in use for at least 2000 years and new machines, of modern materials and design, are being installed to this day.

The wheel described by Vitruvius was of the undershot type where the lower part of the wheel is submerged in the stream and the motion of the water acts on paddles fixed to the rim of the wheel so driving it round.

The other basic configurations of the horizontal axis wheel are shown in figure 1.2. The overshot wheel uses the weight of the water, rather than the velocity to turn it whilst the breastshot wheel utilises both the velocity and weight of the falling water. A variety of modifications to these basic types are used. The breastshot wheel for example can be low-breast when the water enters below the axle line, medium-breast or high breast when the water strikes it well above the axle line.

1.2.1 Undershot Wheels

These wheels are the simplest type to construct requiring no civil works other than a support for the axle across the water course. Any stream of flowing water will suffice to drive one of these wheels although the faster the flow the more power it is possible to produce. In sluggish streams it is possible to speed up the flow by narrowing the channel at the wheel. Many large installations - for example the London Bridge Waterworks - were built under the arches of bridges to utilise the faster flow existing between the piers.

Mills powered by undershot wheels were built on estuaries to utilise the ebb and flow of the tides. An advantage of undershot wheels is that they will work equally well with water flow from either direction.

Floating mills, with wheels mounted on barges so as to maintain the optimum submergence regardless of the height of the water in the river, have been used in many countries. The best known probably being those on the Loire in France.

The major disadvantage of the traditional undershot wheel is its poor efficiency and hence low power output.

The Poncelet wheel described in section 1.3.3 has a much higher efficiency (over 60%) but it requires some extra civil works in the way of a dam and sluice and so loses the simplicity of the traditional wheel.

1.2.2 Overshot Wheels

This wheel requires more civil works than the undershot wheel in the way of a chute to carry the water above it. It can only operate efficiently when the head of water available is greater than the diameter of the wheel. The wheel itself is also more complex than the undershot wheel as it must be fitted with buckets to carry the water rather than simple flat paddles. In return for these complexities however a well designed overshot wheel operating at its design head and flow can give hydraulic efficiencies as high as 70% compared to less than 30% for an undershot wheel.

A disadvantage of the overshot wheel is that generally the direction of rotation is against the direction of flow of the tail race. Hence any raising of the tail race - in times of high water - could cause the lower part of the wheel to become submerged, the resultant drag causing loss of power.

This type of wheel was developed in the middle ages as a result of building dams to impound water for undershot wheels. Water could be taken from the top of the dam to the top of the wheel so increasing the power obtained from a given amount of water.

As it is generally necessary to make the wheel diameter only slightly less than the head available some very large wheels were constructed.

Agricola in his book "De Re Metallica" published in 1556 describes overshot wheels almost 12 metres in diameter used to power cranes for mining. These machines were extremely sophisticated being made with two sets of buckets, side by side but facing different ways. Thus the operator could reverse the direction of rotation by directing the water from one set of buckets to the other. These huge machines were constructed almost entirely of wood (reference 2).

Not until the nineteenth century were iron wheels made in any numbers.

1.2.3 Breastshot Wheels

By allowing the incoming water to strike the wheel above the level of the tail race some of the potential energy in the stream is recovered as well as the kinetic energy. Clearly the height at which the water strikes the wheel depends upon the head available.

An advantage of this type of wheel compared with the overshot wheel is that the rotation is in the direction of the tail race flow. Hence rise in tail water level has a less detrimental effect on the efficiency of the breastshot wheel than it has upon the overshot wheel.

This type of wheel reached a high degree of development in Britain during the 19th century as described in the next section.

1.3 19th Century Developments of the Water Wheel

The water wheel reached its peak of development during the 19th century. Even the increasing use of steam power did not prevent the continued use of water wheels where local conditions allowed. The simplicity, reliability and lack of fuel costs often offset the convenience and increased power that the new steam machines could offer.

1.3.3 Overshot and Breastshot Wheels

John Smeaton

The foundations for the success of the water wheel in Britain were laid in the mid 18th century by John Smeaton. In 1759 he published his paper to the Royal Society describing a series of lengthy tests he had carried out on model water wheels using apparatus he had designed and built himself (reference 3).

This work was all the more noteworthy as Smeaton, realising that relating model results to full size machines was not straightforward, had postponed publishing his results until he obtained further information from tests on full size machines.

Smeaton's work was widely acclaimed and formed the basis of water wheel design for the next seventy years.

The important points contained in his work are:

1. Overshot wheels are about twice as efficient as undershot ones in the ratio of 66% to 30%.

2. The weight of water alone drives an overshot wheel - the impact of the water contributes little.

3. Energy is lost if the water is allowed to impact onto a flat plate.

4. Undershot wheels give the best efficiency at water velocity ratios - wheel rim to stream - of between 2:5 and 1:2.

(J. C. Borda later obtained a value of 1:2 by analysis - see page 10).

5. Breastshot wheels are better than undershot wheels as some benefit is gained from the weight of the water.

Smeatons' work contributed to the virtual demise of the undershot wheel in Britain for industrial use.

The Use of Iron

Smeaton was also a pioneer of the use of iron wheels. His work resulted firstly in the use of cast iron axles for wheels and then in 1778 the use of iron for the rims.

Iron has several major advantages over wood:

1. It enables efficient gears to be cast at the rim allowing power to be taken off at the periphery instead of at the axle. This relieves the load on the spokes and so lightens the construction of the wheel.

2. Buckets can be made of a more complex and efficient shape than is possible with simple wooden buckets formerly used.

3. Buckets can be more numerous because thinner sections of material are used.

1.3.2 The Pitchback Wheel

This machine is the most flexible of the 19th century machines in terms of making the maximum use of available water and control of power output.

It is essentially a high breastshot wheel usually equipped with a variable height opening for the entry of water to the wheel. This "sliding hatch" was the invention of John Rennie and enabled an increase in head water level to be used to advantage as well as providing a means of controlling the power output of the wheels by reducing the water flow when necessary. Simple guide vanes in the hatch improve the flow onto the wheel thus avoiding the splashing so deplored of by Smeaton.

These are illustrated in figure 1.3.

The overshot wheel, whilst more efficient at design head than the breastshot, cannot benefit from any increase in head by use of a sliding hatch. The ventilated bucket, invented by William Fairbairn, also contributed to the success of the pitchback wheel by allowing a complete filling of each bucket.

Both overshot and pitchback wheels of this period exhibit a large number of elegantly curved buckets, some large wheels having eighty or more.

1.3.3 The Undershot Wheel

Although Smeaton's work was known to French engineers they did not abandon interest in undershot wheels.

Charles Bossut published "Traite theorique et experimental d'hydrodynamique" in 1771 in which he described practical tests on the undershot wheel.

The points he made were:

1. Performance in a broad open channel varies little from that in a narrow flume.

2. The number of blades on a given wheel affects the efficiency, generally the more blades the better.

3. Inclined blades, between 15° to 30° from the radial give a better power output.

Analytical studies carried out by Jean Charles Borda showed that for maximum power the peripheral speed of the wheel should be half that of the water. Borda also made two basic propositions to obtain the maximum power from a water wheel - these were:

a) The water should be applied to the wheel without percussion (shockless entry).

b) The water should leave the wheel with minimum velocity.

These ideas were put into practice by Poncelet who in 1825 published a paper describing his work and an undershot wheel of his own design (reference 4).

Figure 1.4 shows the Poncelet wheel as described by the inventor.

A large number of curved vanes receive water from a deep sluice. The entry angle of the vanes is such that the water impinges on them without appreciable shock. The water climbs up the blade losing velocity until at exit the water falls from the blade substantially without velocity. On a full size prototype Poncelet obtained efficiencies between 60% and 70% - almost as good as the best overshot wheels.

Using all iron construction many Poncelet machines were built in France and the rest of Europe.

1.3.4 Summary of Water Wheels

The design of water wheels reached its apogee in the mid nineteenth century. The demand in Britain for power for Colton and woollen mills gave engineers like Thomas Hewes and William Fairbairn the opportunity to design and install wheels of over 80kW power output.

These machines were invariably of the suspension type where the wheel rim and buckets are suspended from a massive iron axle on thin wrought iron spokes in the manner of a bicycle wheel. This gives a strong but light construction to these wheels. A rim gear, fixed to the wheel rim is used to transmit the power thus relieving the spokes of this load.

The rim gear, being of a much smaller diameter than the wheel, runs at a higher speed - hence allowing the use of lighter shafts to carry the power to the machines.

Many of these wheels were fitted with mechanical ball governors of the Watt type. These governors were connected to sliding hatches which controlled the water flow to the wheel and hence its speed and power output.

The reliability and simplicity of these machines is illustrated by the fact that many were in use for a hundred years or more. For example such a wheel installed at a factory in Bakewell, Derbyshire in 1827 was still in use in 1934, and was maintained until after 1945 as a standby for emergency use.

1.4 Novel 18th and 19th Century Machines

Concurrent with the development of the water wheel in the 18th and 19th centuries was the emergence of several machines of a novel character, some of which enjoyed a limited success but did not develop further. Other machines are clearly the forerunners of the turbines in use today.

As pressure pipes did not become readily available until the second quarter of the 19th century the majority of these machines used heads of less than 6 metres and so can be considered as low head machines - even though subsequent work proved some of them more suitable for higher heads.

1.4.1 Barkers Mill

This is the water powered version of Hero's steam turbine. Water from a high level is introduced into a tall drum which is able to turn about a vertical axis. The water is expelled at the base of the drum through two or more nozzles at the ends of hollow radial arms. The nozzles are set at right angles to the arms, the resultant tangential force drives the drum round.

Mill stones could be connected directly to the top of the vertical axle or driven by a belt wrapped around the drum.

The mathematician, Leonard Euler, produced an elaborate version with sixteen nozzles. A feature of this machine is the careful design of the top part of the drum which allows the water to enter smoothly without splash or shock. An elegant design by James Whitelaw, called the Scotch Turbine was produced in the 1840's. In this machine the rotating head is fed from below by a pipe upon which is fixed a bearing, this allows the head to rotate. The drive being taken off from above. The exit nozzles are rectangular boxes, carefully tapered and curved to maximise exit velocity. A description is given in reference 5. These mills were in wide use in Europe and America up to the mid nineteenth century.

1.4.2 Tub Wheels

Tub wheels are essentially vertical axis machines developed from the earlier Norse Mill.

To reduce splashing and contain the water Norse mills were fitted inside a cylindrical chamber or tub, the water being introduced through a tube or chute in the side of the tub.

In 17th century France wheels of this type were frequently used to grind corn. The wheel is supported in a masonry chamber similar to a well shaft. The water feed enters from a side shaft just above the wheel and strikes it at an angle thus driving it round.

By the early 18th century these wheels had become much more sophisticated hydraulically. The vanes were curved and the water jet struck the wheel tangentially exhibiting the shockless entry not achieved on horizontal axis wheels for almost another century.

Figure 1.5 shows one of the wheels at Basacle, Toulouse on the river Garonne. These are described by B F Belidor in "Architecture Hydraulique" and hence wheels of this type are often referred to as Belidor's wheels (reference 6).

The earlier wheels were essentially pressureless wheels driven by the kinetic energy of the water jet. The wheels described by Belidor derive energy from both the velocity of the jet and the pressure.

Typically these wheels were about one metre diameter supplied from a head of about 2 metres. They were a reliable, robust wheel but not efficient - even compared to contemporary water wheels.

Tests carried out on wheels of this type by Tardy and Piobert using a simple friction dynamometer gave efficiencies of between 12 and 15%.

An improved version of these wheels, attributed to Charles Borda in which the flow was largely axial could give higher efficiencies - up to 70% (reference 7).

Wheels of this type were used in the New World - where their inventors often gave them colourful names - for example - the Green Mountain Wheel, the Wry Fly Wheel and the Chase Special. As in Europe these wheels were used in rural locations to drive flour mills or saw mills at sites where water was plentiful but at a low head. The benefit of a low cost, easily manufactured machine greatly outweighed any advantages of efficiency in remote locations.

1.5 Outward Flow Turbines

1.5.1 The Fourneyon Turbine

The success of this machine is attributable to the engineering genius of the Frenchman, Benoit Fourneyon. Although the concept of water entering through the middle of a machine and exiting on the outside had been proposed by Claude Burdin it was Fourneyon who built the first successful wheel in 1827. This machine worked on a head of 1.4 metres and an efficiency of 80% was measured at a speed of 60rpm.

The machine is shown in figure 1.6 together with an enlarged view of the rotor and guide vanes.

The guide vanes direct the water almost tangentially onto the runner blades which are curved in the opposite direction to the guide vanes.

The flow is all in one plane - no axial flow occurs inside the guide vanes or the runner.

The early Fourneyon machines had large clearances and were essentially pressureless with the exit water discharging into atmosphere above the tail water. Later machines were made with smaller clearances and often fitted with a diffuser on the outlet anulus so enabling them to run submerged.

Although the full flow efficiency of these machines was very high by contemporary standards, the part load efficiency was poor. The expanding water passages of the turbine giving rise to flow instability.

This situation was improved at the expense of mechanical complexity by making the runner and guide vanes in three or four layers, one above the other. A cylindrical gate was then slid down inside the guide vanes so that at part load one or more layers could be shut down.

The ability of a water turbine to speed up and run away, sometimes with disastrous results was first recognised with this machine.

1.5.2 The Girard Turbine

This outward flow pressureless turbine was developed by the French enginer Girard. It was constructed in both vertical and horizontal axis forms.

In the horizontal axis machine the runner is very similar to a Foureyon turbine runner but the stator, instead of being a complete circle, is only a small arc containing four or five vanes.

This configuration makes the machine much less complex mechanically as the rotor is simply supported on an axle in the manner of a water wheel.

Part load control is also possible by shutting off a number of the vane passages. In this form the machine was used mainly for high heads when efficiency could be up to 80%.

A series of tests described in reference 7a on a wheel of 2.7m diameter under 122m (400ft) head gave an efficiency of 79% at full load (300kW) and 59% at quarter load (60kW).

The vertical axis Girard turbine is essentially an axial flow machine very similar in construction to the Jonval turbine (section 1.7) with a complete ring of stator vanes.

On the Girard machine some control at part load is possible by covering over a proportion of the stator so reducing the total flow but maintaining the same flow through the vanes in use.

Turbines of this type have commonly been used on heads down to 3m.

(The Jonval turbine is a pressure turbine with the stator and runner totally submerged. The axial flow Girard turbine is of the pressureless type and discharges into air. The runner is only partly filled with water on the concave sides of the vanes, there being a ventilation hole adjacent to the convex side of the vane to maintain this condition. This system is also used on outward flow Girard machines).

1.6 Inward Flow Turbines

J V Poncelet, the inventor of the improved undershot water-wheel, proposed an inward flow turbine in 1826 in the form of a Poncelet wheel mounted on a vertical axis and supplied with water all around the periphery. The water escaping through the centre of the wheel. Poncelet, himself, however did not build any full size wheels to demonstrate this principle.

1.6.1 The Thomson Turbine

James Thomson, elder brother of Lord Kelvin, and Professor of Engineering at Queens College, Belfast made important contributions to inward flow turbine design with his 'vortex' turbine.

These were first used in Ulster in 1852 and were the first turbines to use pivoted guide vanes to improve the part load efficiency. Figure 1.8 shows a view of these.

The guide vanes, though few in number by modern standards, enabled the flow passage to the runner to be reduced without any sudden changes in the section available to the flow. Movable guide vanes have become the standard means of controlling flow on inward-flow machines.

Another major contribution by Thomson was the use of a scroll casing to direct flow into the guide vanes. This spiral casing was designed to give a uniform velocity and optimum flow direction at all the guide vanes.

Reference to figure 1.9 illustrates this. Comparison with figure 1.8 which does not have a scroll casing shows that at the left hand side of the runner the vanes receive a much less favourable flow pattern than those at the right hand side.

Spiral casings are difficult items to fabricate and for this reason Thomson compromised on some turbines by offsetting the centre of the runner from the centre of the casing (as in figure 1.8) thus giving some reduction in flow area as the water passes to the further side of the runner.

Another of Thomsons adaptions to the technology available was the use of the double exit runner. The vortex turbine was double sided thus allowing water out from both sides of the runner. This virtually eliminated the end thrust on the shaft bearings which was a major problem on turbines until the invention of the Mitchell thrust bearing.

Thomson was also the first designer to provide a valve on his draft tube to allow air into the tube to improve the flow of water at part load when considerable turbulence could be present.

Although the Vortex machine relied entirely on radial flow through the runner it was successfully used on heads from 1m to 140m.

Typical examples are:

Site	Head	Power
Lowbridge Mill - Belfast	11.3m	21 kW
Ballyshannon	4.3m	112 kW
Reading	1.0m	12 kW

1.6.2 The Francis Turbine

James Francis carried out an impressive series of tests on inward flow turbines between 1850 and 1855 whilst working as Chief Engineer for the Lowell Manufacturing Company at Lowell Massachusetts. The results of these were first published in 1855 as "Lowell Hydraulic Experiments" (reference 8). Francis as a result of these tests was able to design a machine which was, with the possible exception of Thomson's vortex, the most thoroughly researched and engineered machine of its time.

Further work resulted in the mixed flow turbine, that is partly radial flow and partly axial flow within the runner. It is in this form that the Francis turbine is made today. By varying the degree of the radial and axial components on the water velocity within the runner - a function of the runner geometry - the same type of machine can be made to work efficiently at a wide range of heads.

The modern forms of this machine are described in section 2.2.1.

1.7 Axial Flow Turbines

Although the tub wheels and early American machines such as the Wry Fly wheel where essentially axial flow machines, the first successful pressure machine was that of the Frenchmen Jonval and Keochin.

The essential features of this are shown in figure 1.10.

The upper ring of stationary guide vanes directs the water onto the lower ring of rotating vanes. Early machines ran above the tail water with atmospheric outlet from the runner. Koechin, however, conceived the draught-tube as an essential part of the design, thus making it a true pressure turbine.

These machines, with a vertical axle were used mainly in the head range of three to five metres, where with the use of a draught-tube, a very neat arrangement could be made.

To improve part load performance machines were produced with two or more concentric rings of vanes. One of the rings could thus be shut off at times when water flow was low without greatly reducing the efficiency.

These machines enjoyed a long popularity being made from 1840 right up until the twentieth century when they were superseded by the propeller turbine.

1.8 The Pelton Wheel

This impulse turbine is basically best suited to high heads although in its early form, the hurdy-gurdy wheel, used by the goldminers of California and Nevada in the 1850's, it did operate on lower heads than would be thought useful today. Even so heads of below ten metres were rarely used and the concept of a free jet driving a wheel cannot be applied to low head hydro power unless extremely low speeds are acceptable. For heads below three metres a Pelton wheel would take on the proportions of a Poncelet wheel to produce any reasonable power output.

1.9 Low-Head Turbines

A great deal of effort was expended in the USA during the latter part of the nineteenth century to develop a turbine suitable for heads below ten feet. The most successful of these were similar to the Francis turbine being of the mixed flow type.

Several of these were made or imported into Britain. The names given to them - such as the Hercules, the Sampson and the Little Giant reflected the manufacturers' opinions of their performance. A number of solutions to obtaining good part load performance were tried.

The Hercules was equipped with fixed inlet guide vanes and used a cylinder gate which could be lowered into an annular space between runner and vanes thus reducing the flow area.

The Samson had guide vanes, similar to the Vortex turbine. The Little Giant was equipped with a horizontal plate which could be slid across the runner to divide it into two separate compartments, water thus only being admitted to one of them at part load.

None of these designs was essentially better than the Francis machine but they represent individual attempts to obtain good performances at low heads.

1.10 The hydraulic ram pump

No description of 19th Century hydraulic machines would be complete without a mention of the hydraulic ram pump. This ingenious device was used to pump water from a flowing stream at low level to a consumer at a higher level. Examples were made which could raise water up to 1300m or could deliver up to 95 cubic metres per hour.

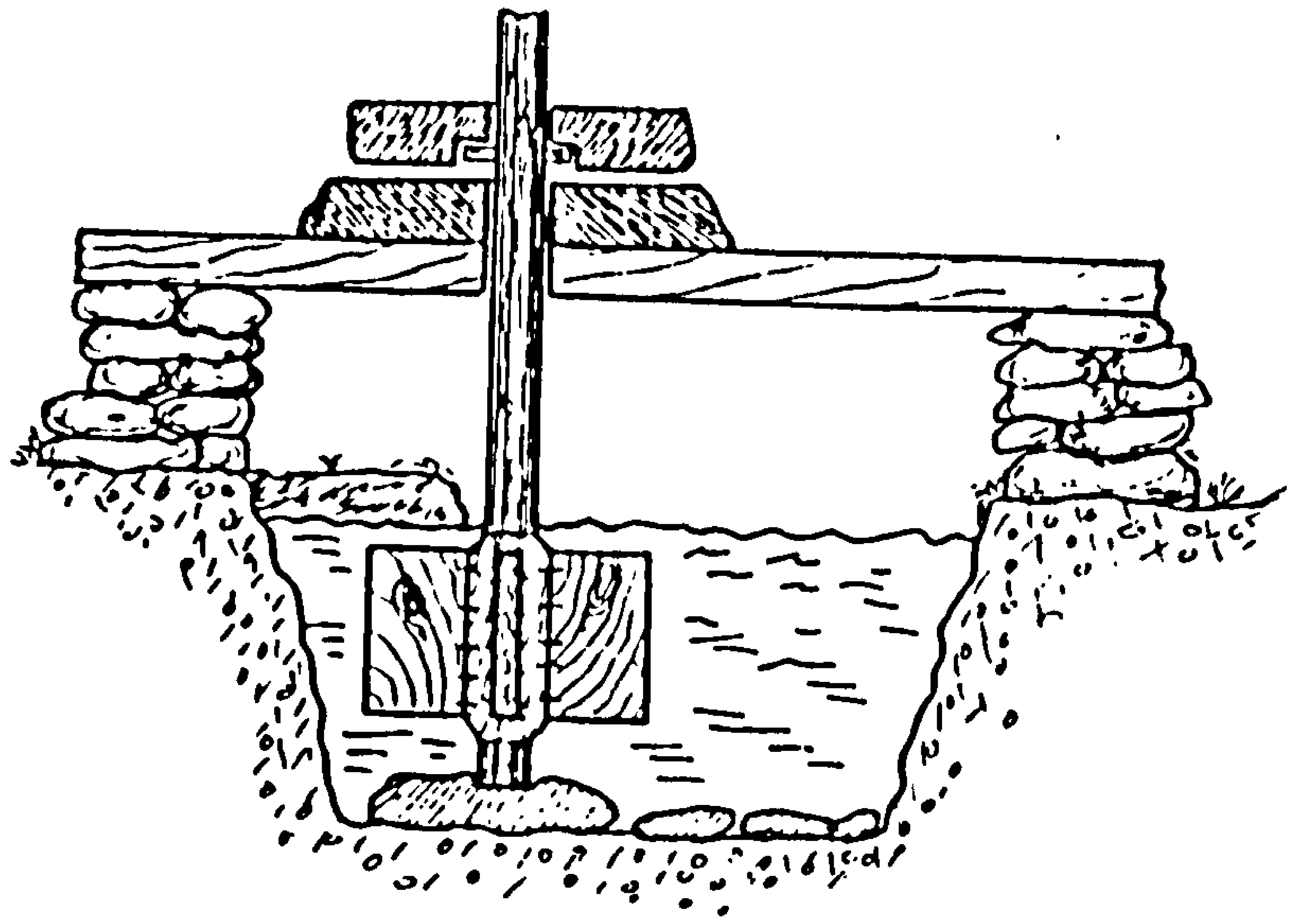
The original design is credited to J M Montgolfier but it was refined over a number of years, being mass produced at the end of the 19th century to provide water supplies to thousands of farms and isolated houses.

The device is shown in figure 1.11. It works by means of pressure waves set up in the feed pipe by the sudden closure of the automatic valve at the lower end of the pipe. This pressure wave forces a small quantity of water into the chamber at the base of the air vessel.

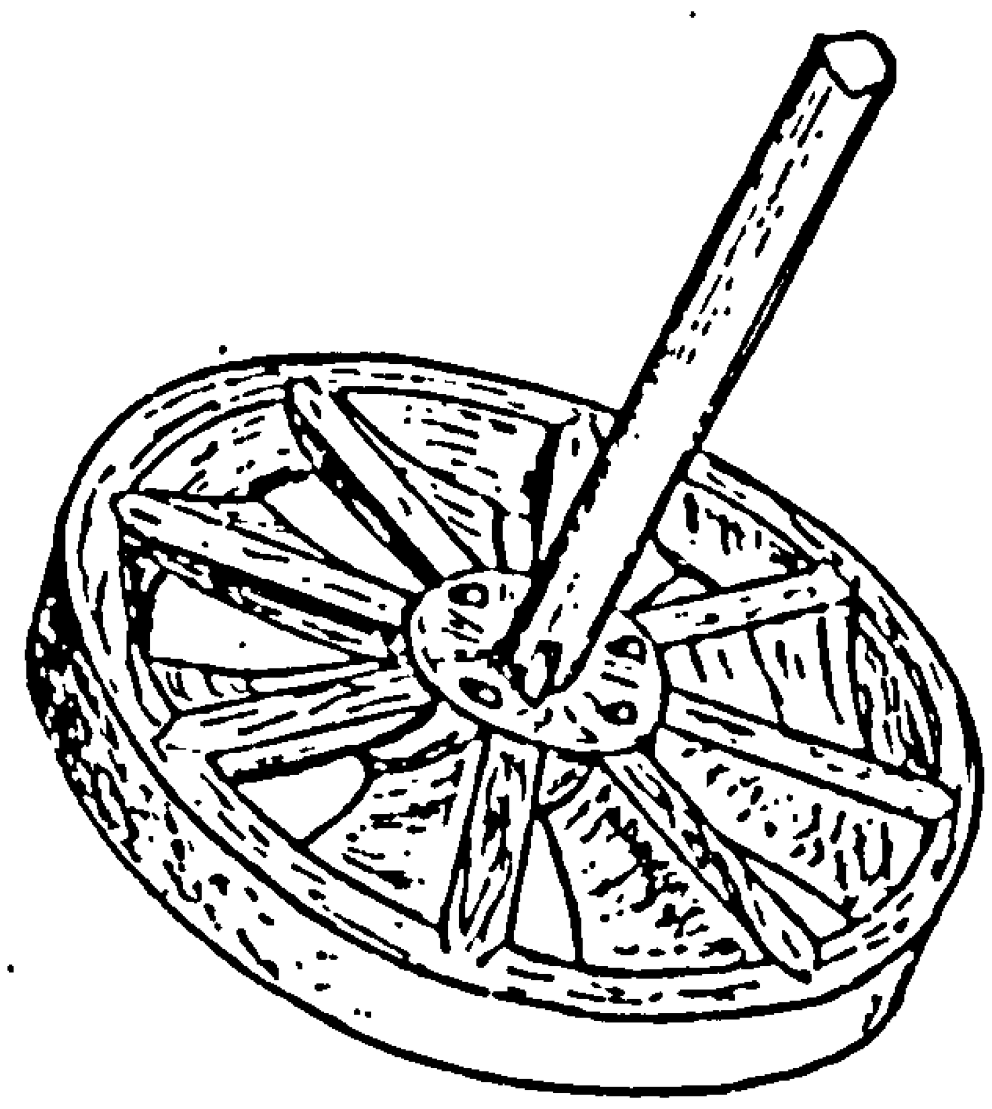
Water is then forced out of the air vessel to the higher level. A non return valve at the base of the vessel prevents back flow of water. The cycle continues automatically - the frequency being a function of the pipe lengths and the discharge pressure.

An excellent analysis is given in Gibson (reference 7) in which efficiencies as high as 70% are claimed for the device. This figure was obtained by comparing the energy gain of the water pumped to the high level with the energy loss of the water flowing into the feed pipe.

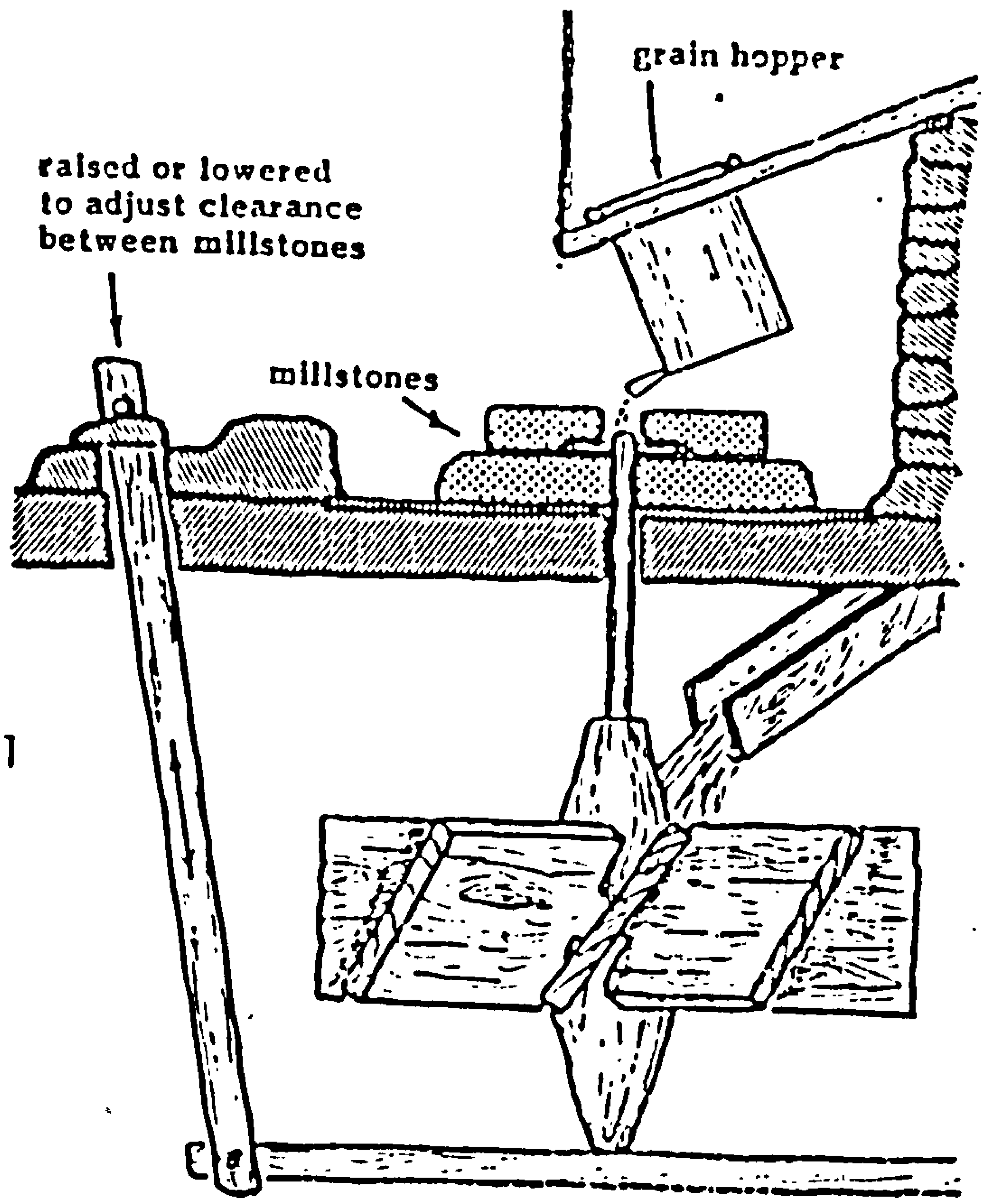
These rams often replaced water wheels or other more conventional pumps at sites where low maintenance and reliability were important. Figure 1.12 shows a small (2.4m diameter) breastshot water wheel, complete with Rennie shuttle once used for pumping water from the river Wye to a farm about 1km away. In front of the wheel can be seen the pressure vessel of the hydraulic ram which replaced the relatively cumbersome wheel and pump.



Norse Mill
Figure 1.1a

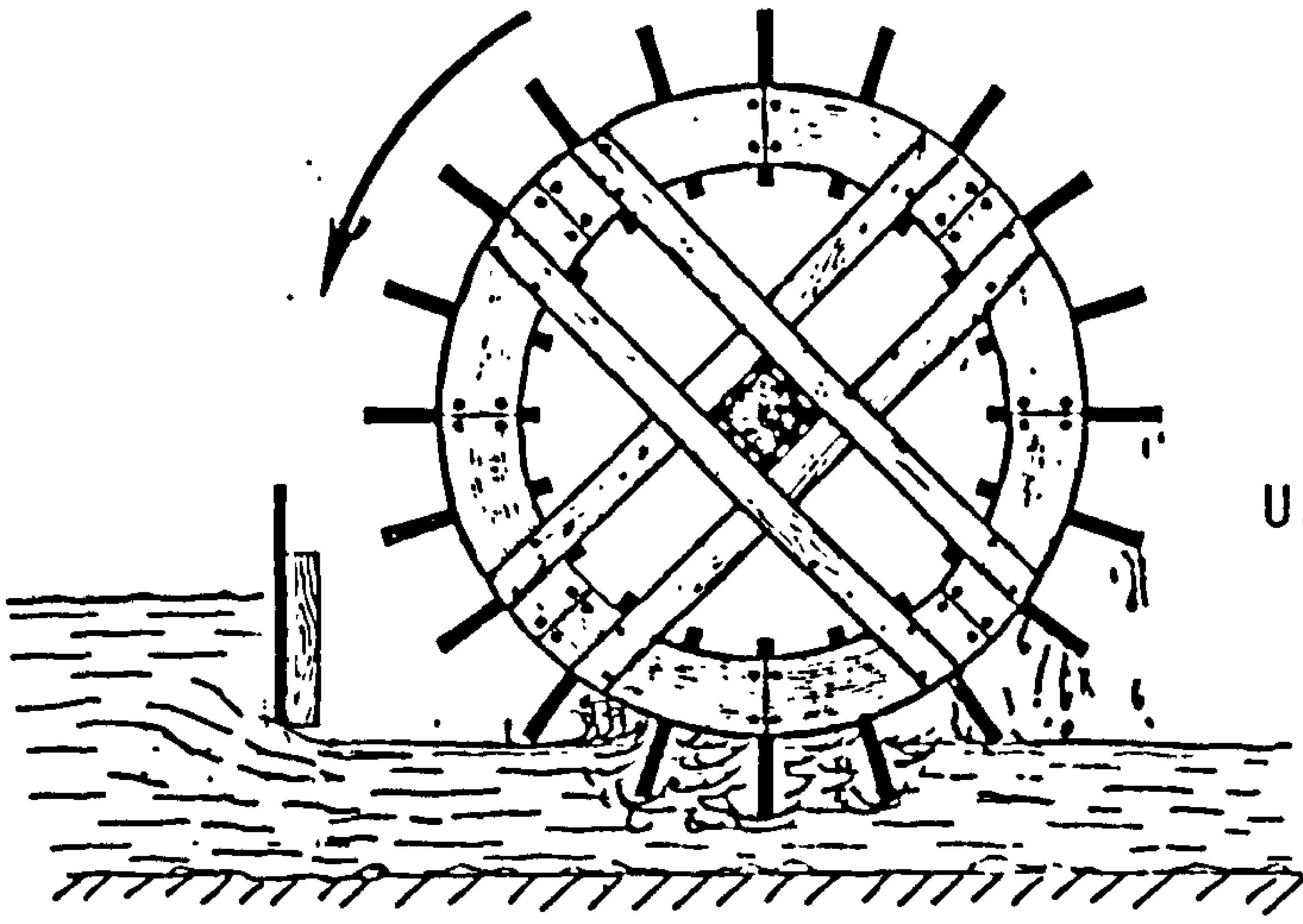


Moorish wheel - about 1000 AD
Figure 1.1b



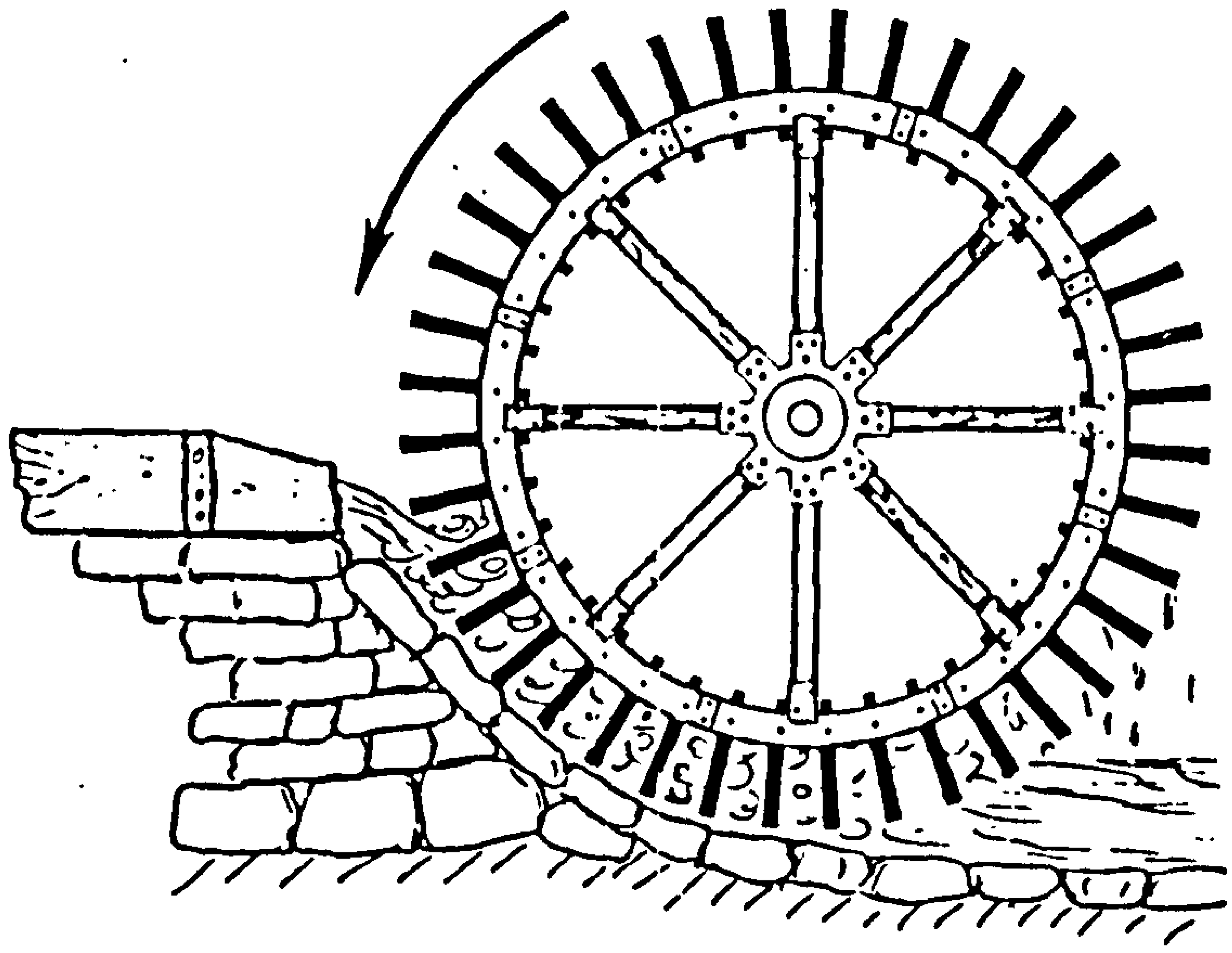
Traditional Nepalese Wheel
- in use today
Figure 1.1c

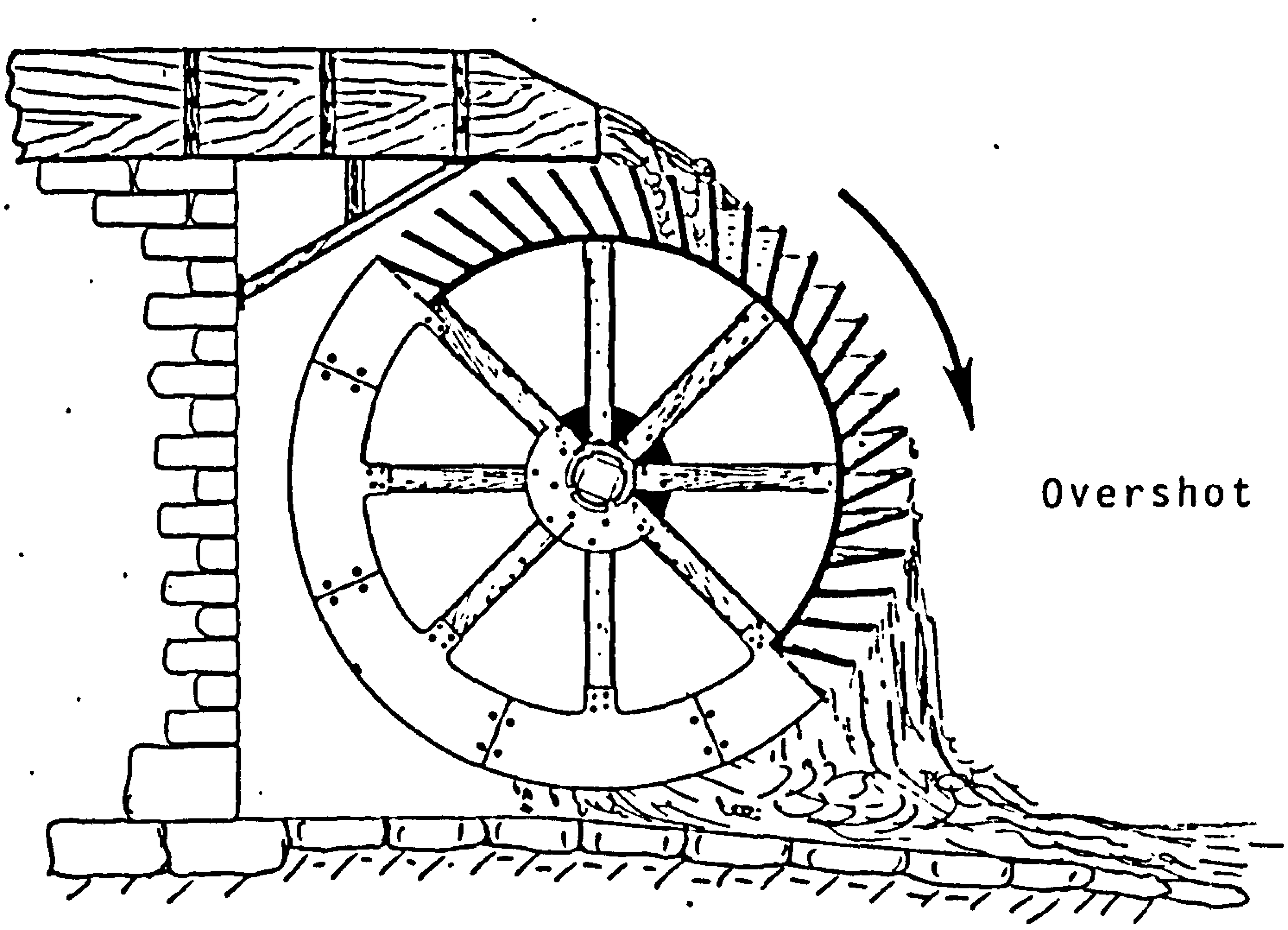
Figure 1.1



Undershot

Breast Shot

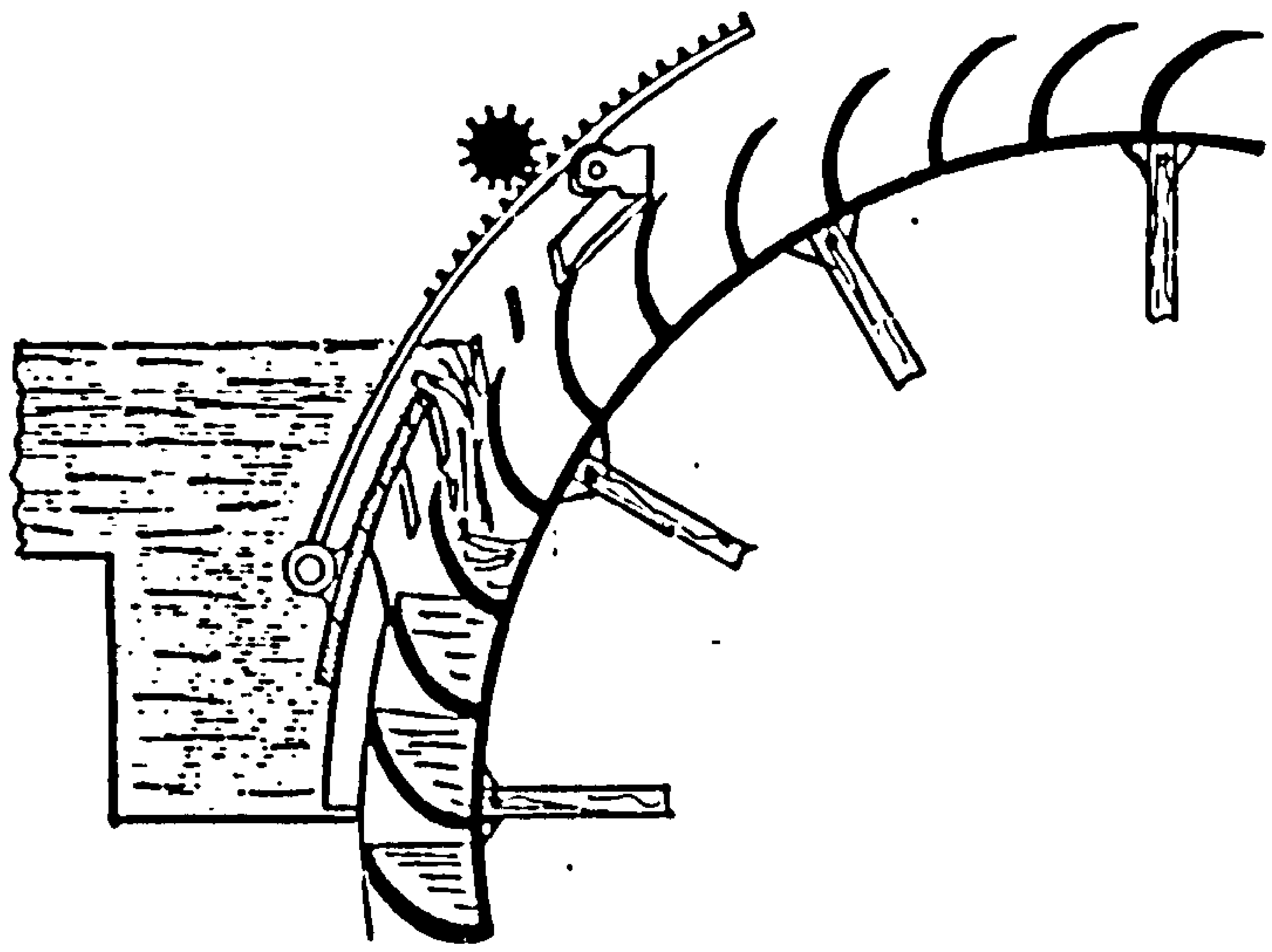




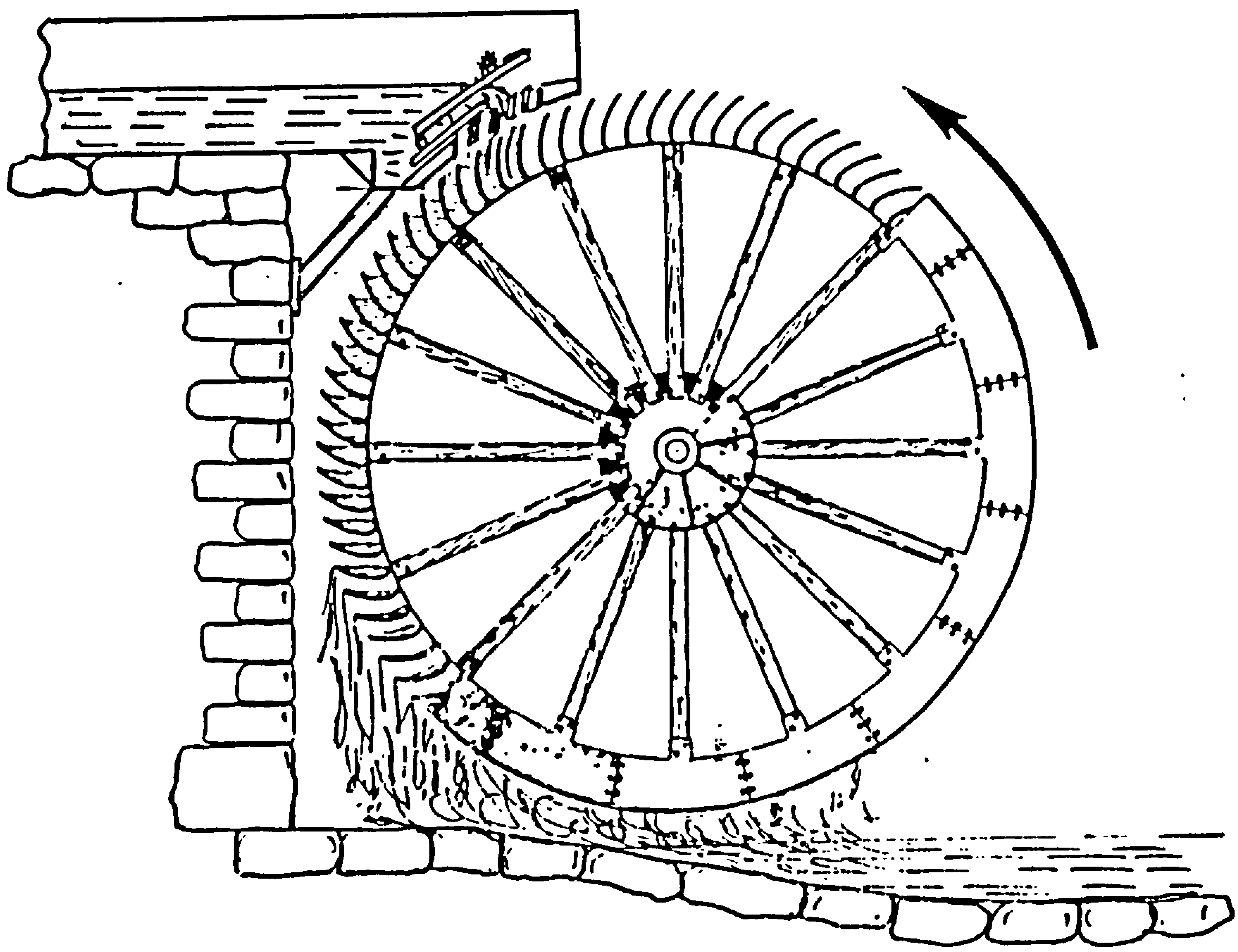
Overshot

Basic types of horizontal axis water wheels

Figure 1.2

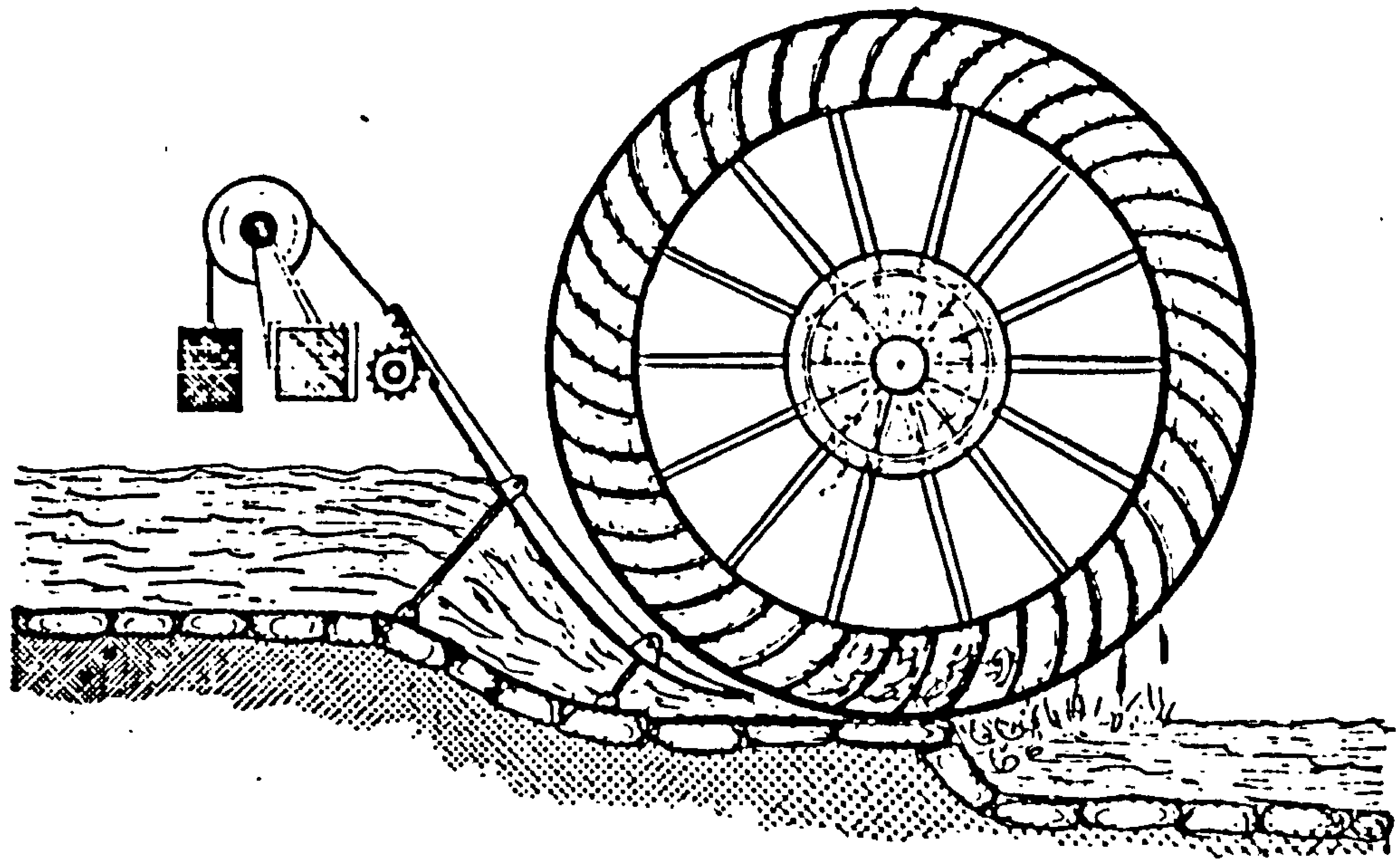


Detail of sliding hatch



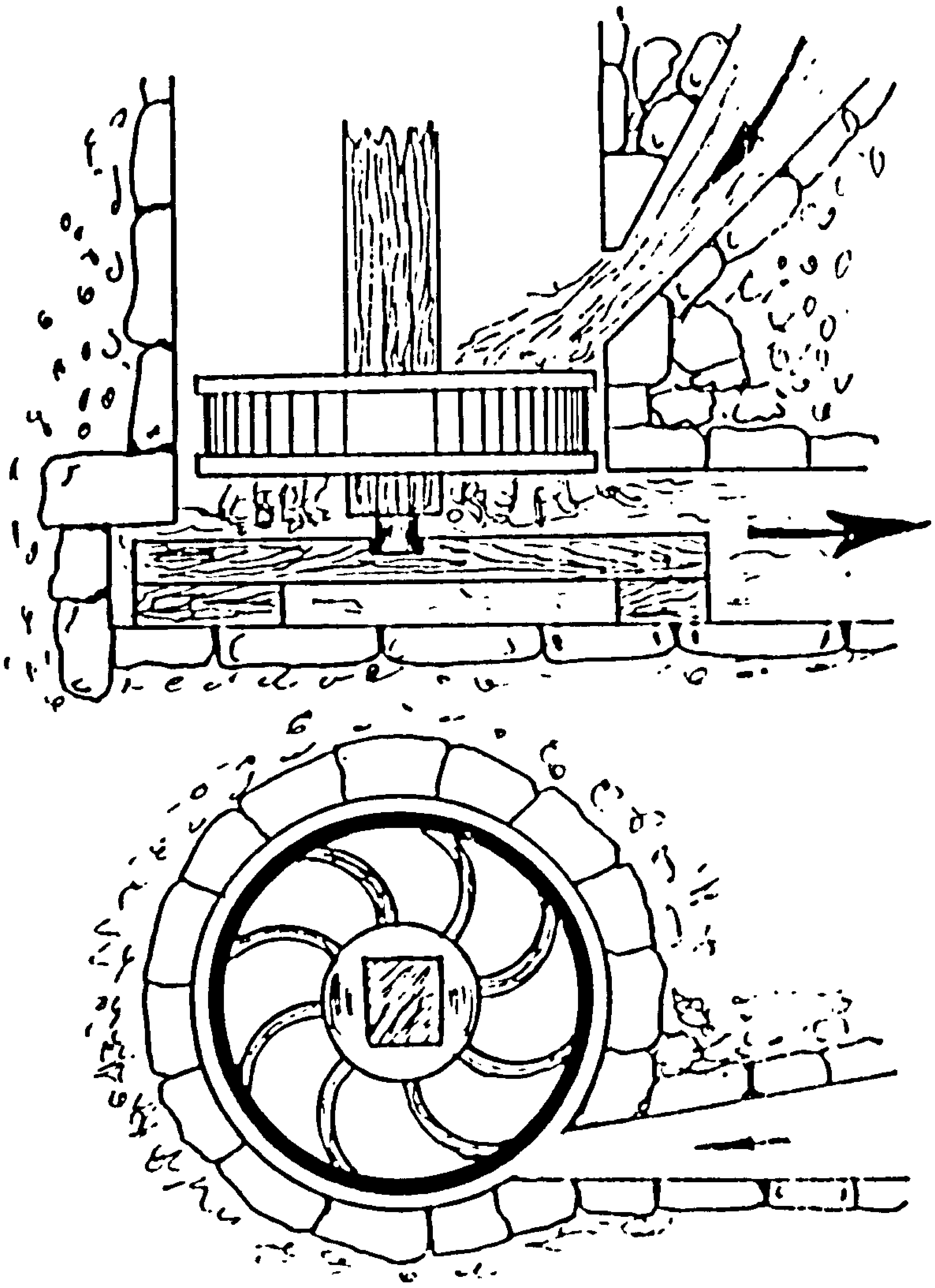
Pitchback wheel with sliding hatch

Figure 1.3



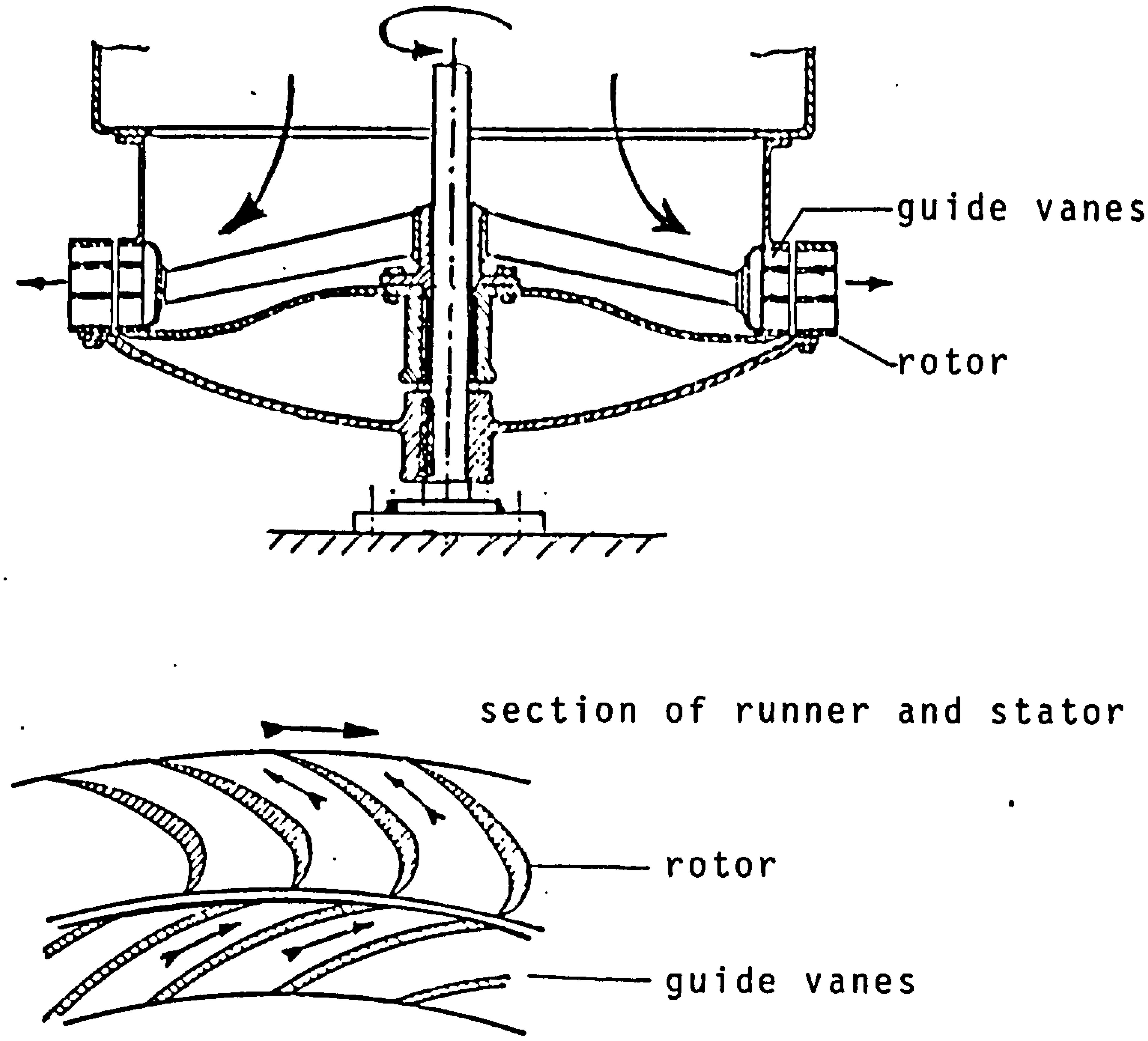
Poncelet Wheel

Figure 1.4



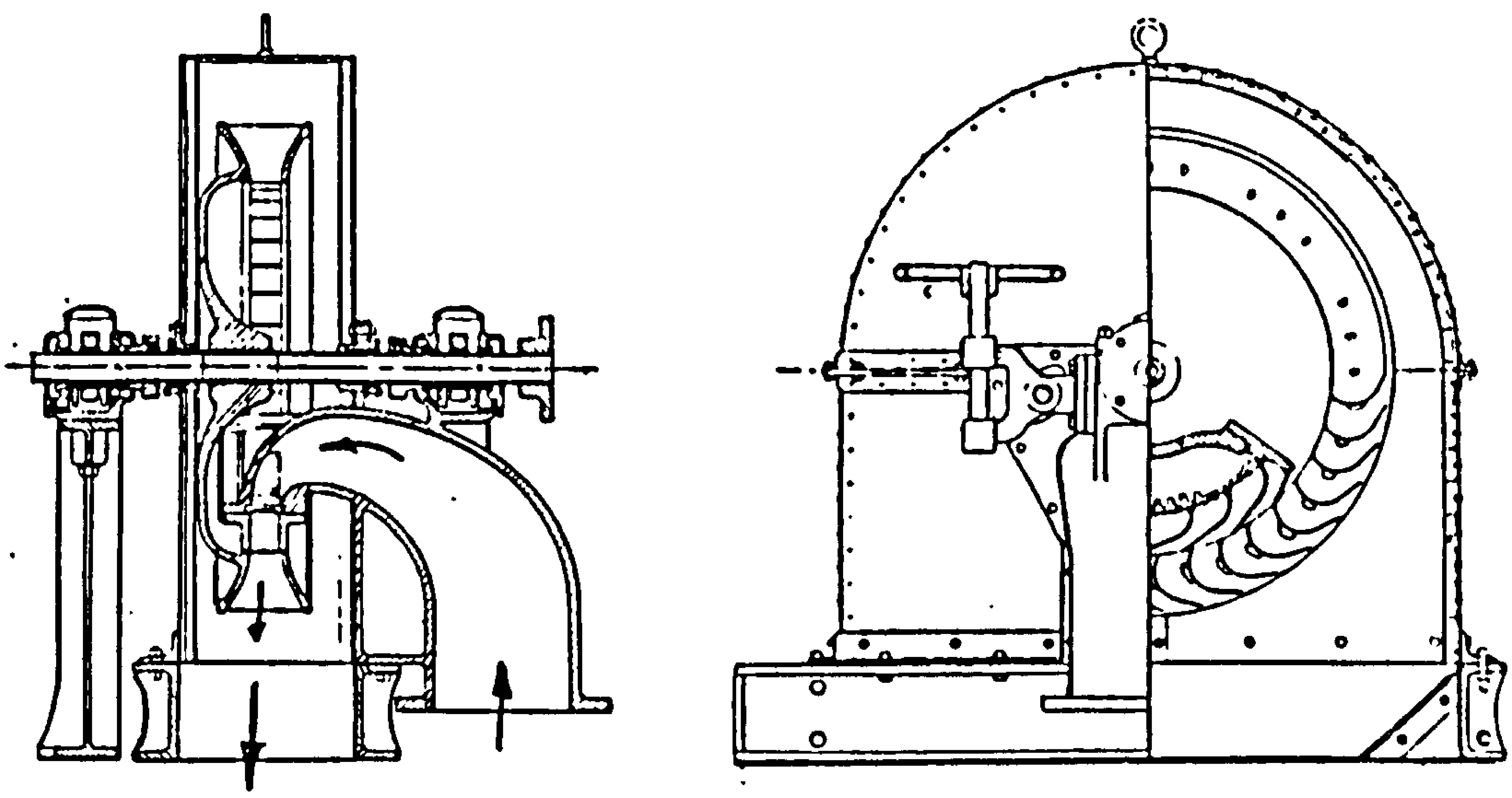
Tub wheel-circa 1700

Figure 1.5



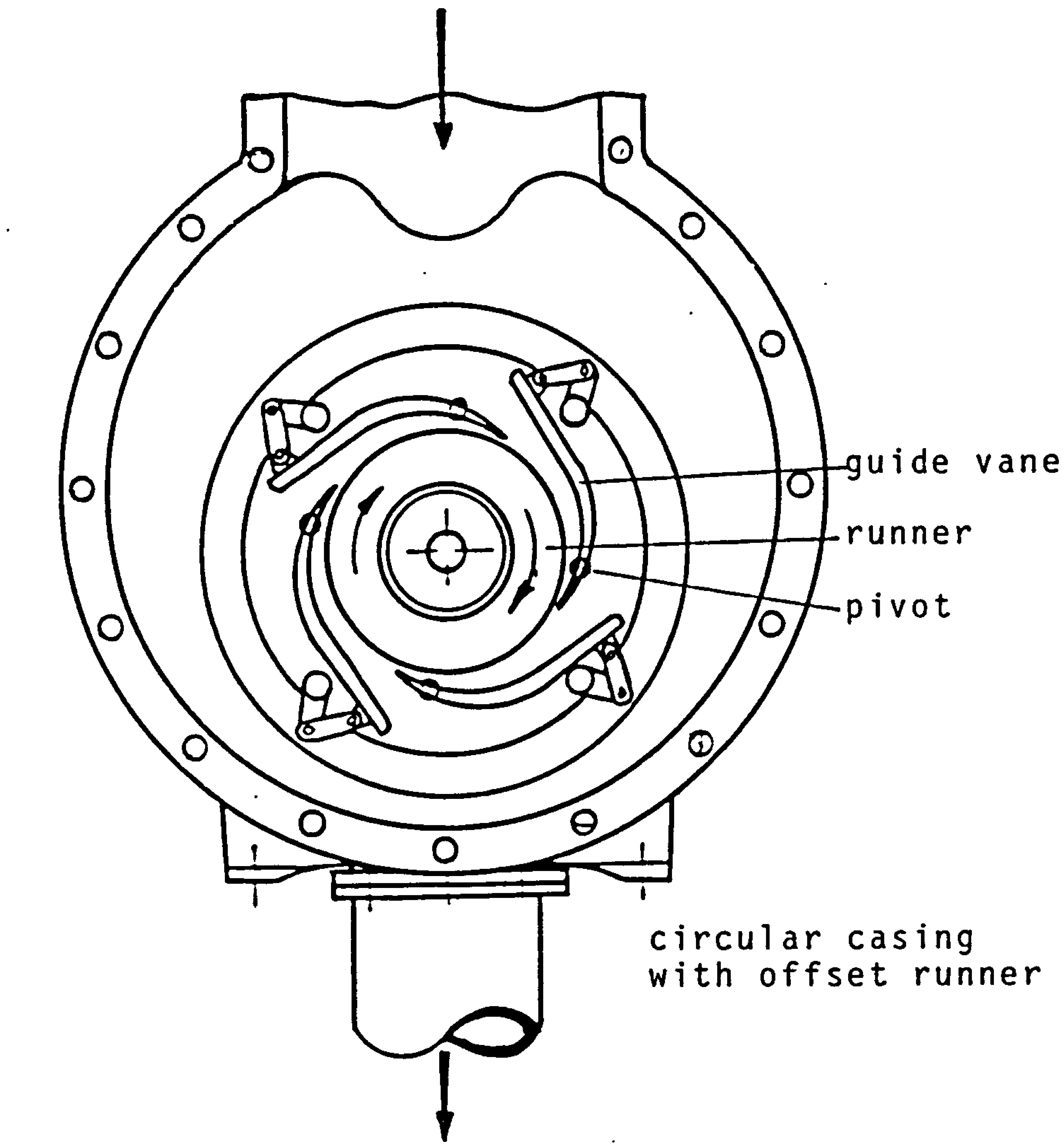
Fourneyon turbine

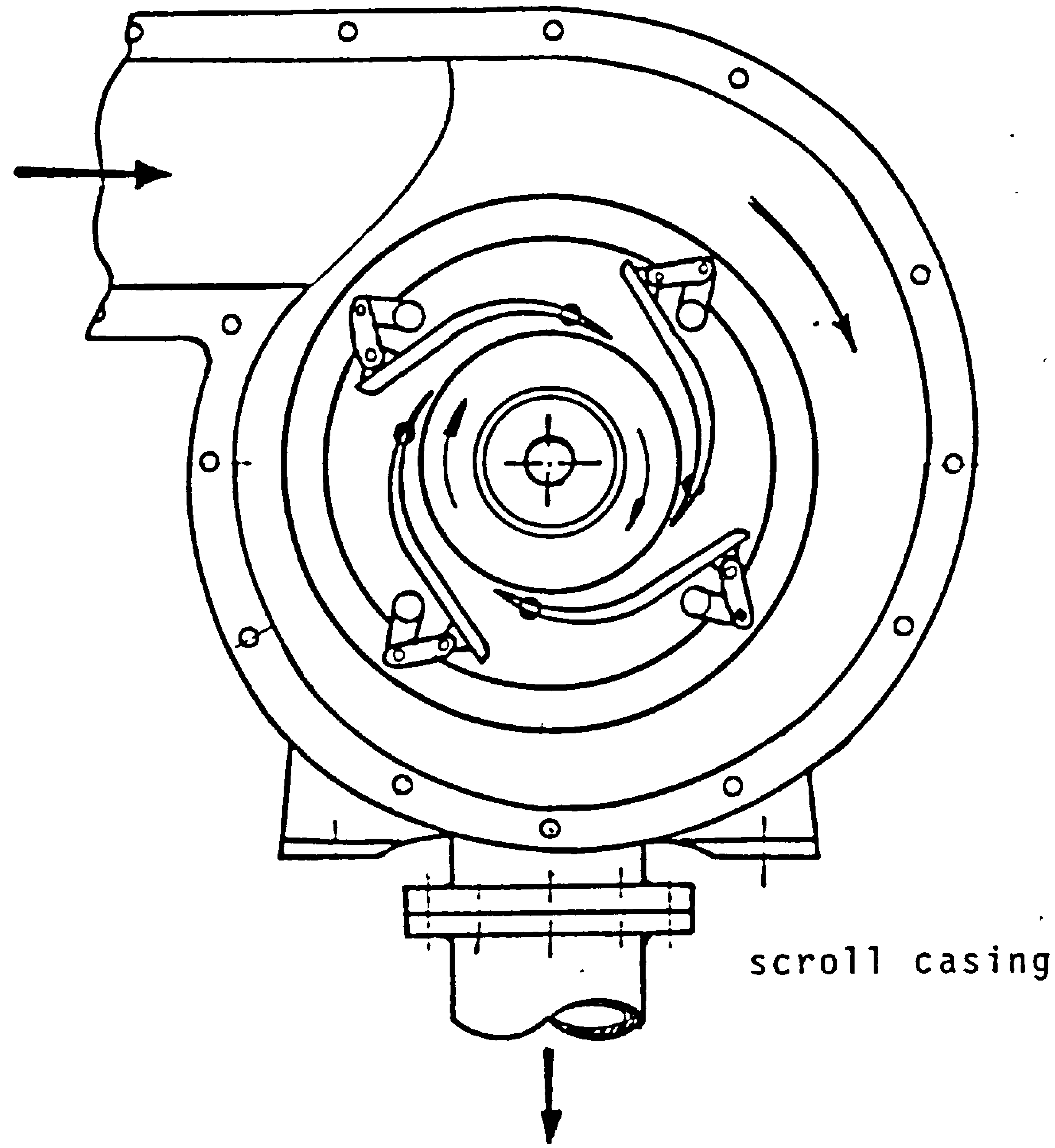
Figure 1.6



Horizontal axis Girard turbine

Figure 1.7

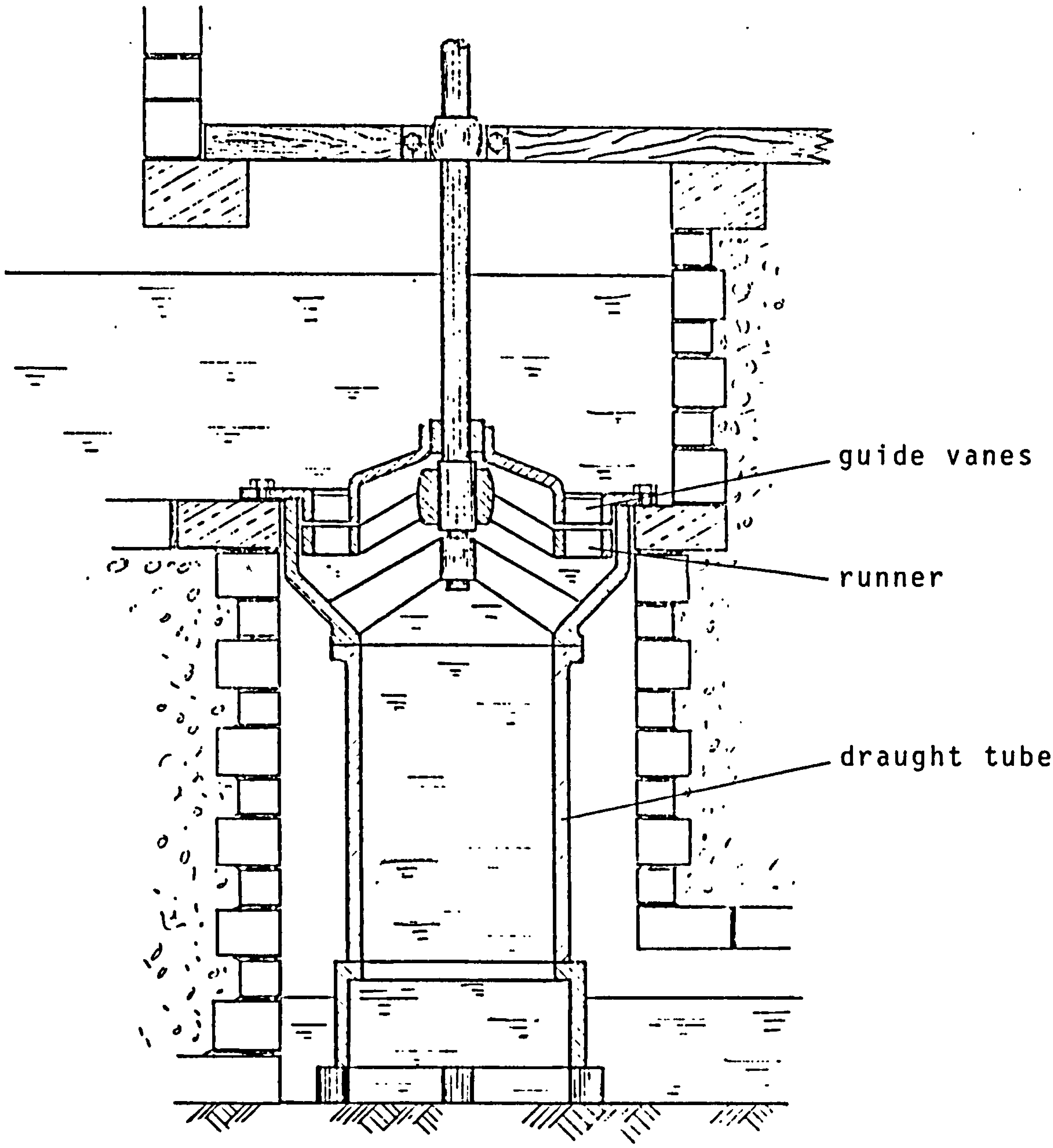




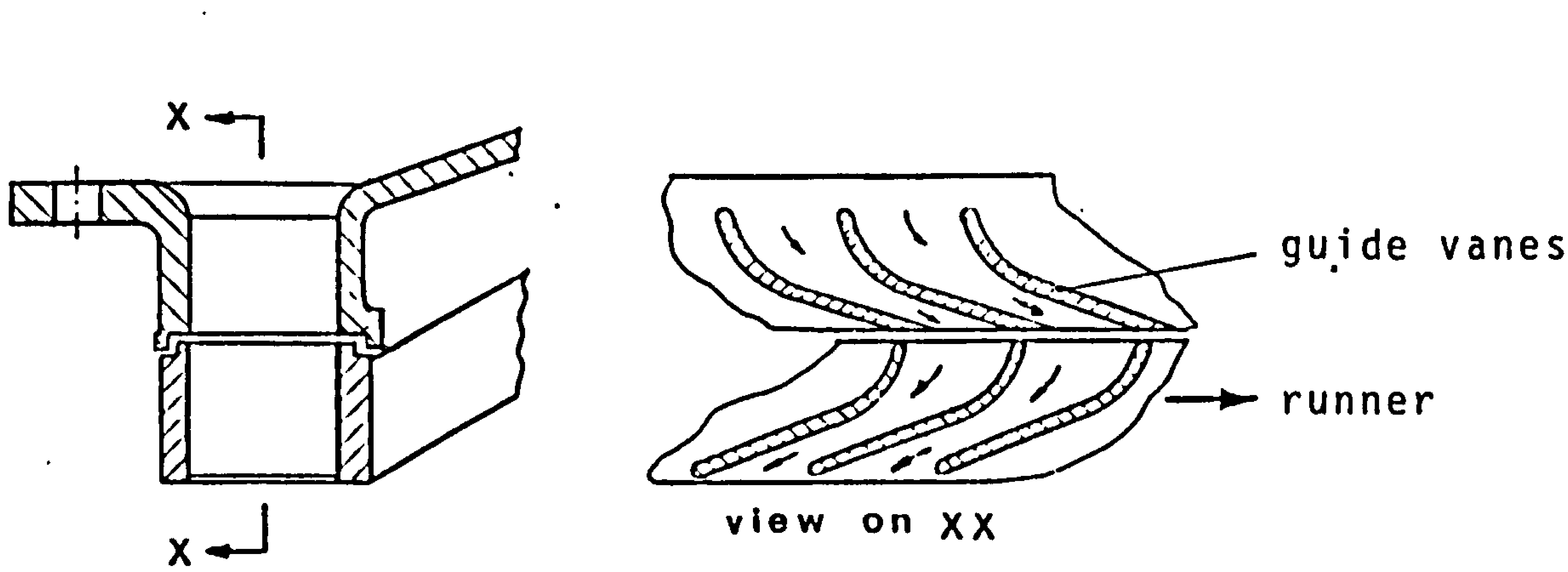
Thomson's vortex turbine

Figure 1.8

Figure 1.9



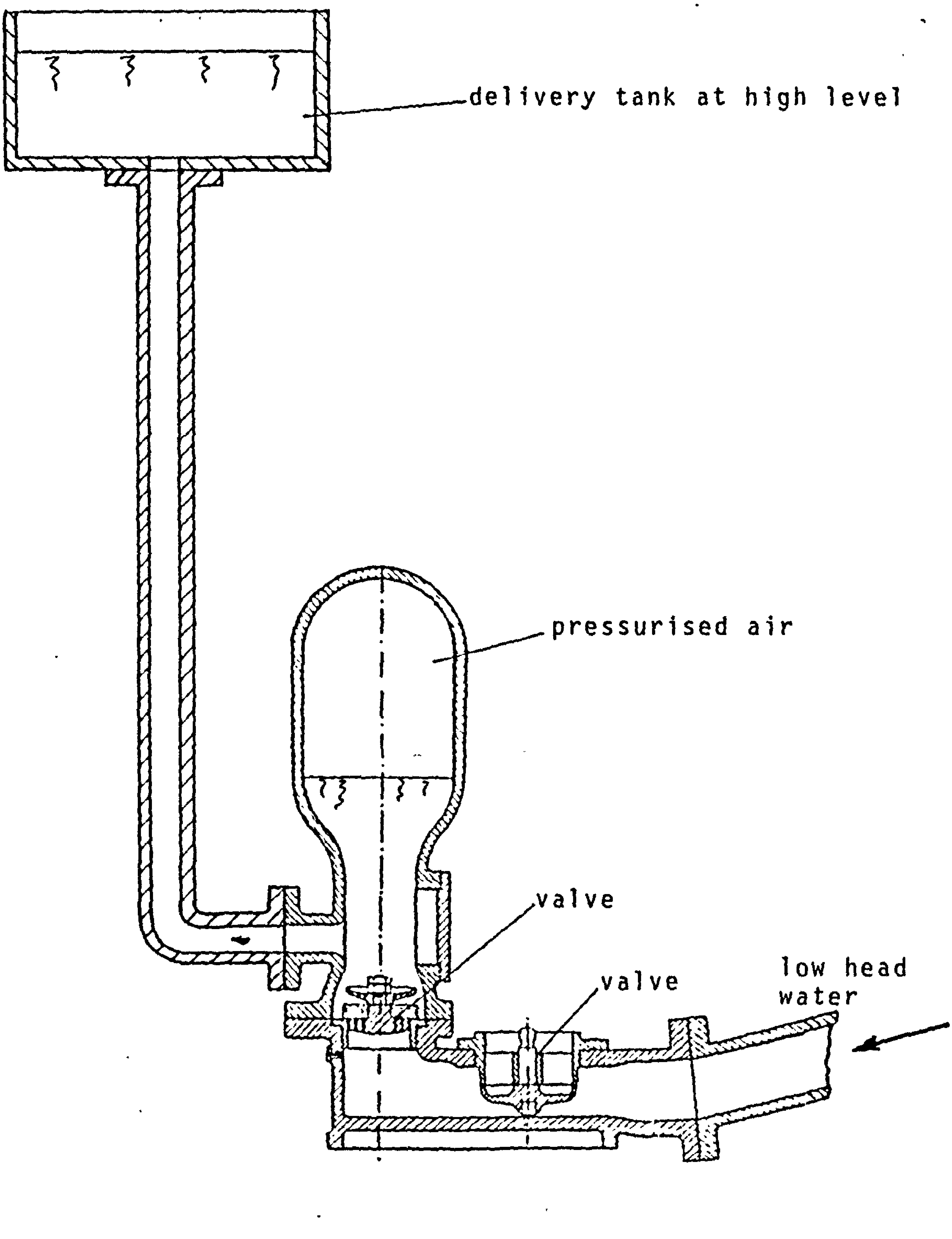
Turbine installation



Enlarged view of guide vanes and runner

The Jonval axial flow turbine

Figure 1.10



The hydraulic ram pump

Figure 1.11

Water wheel and hydraulic ram pump

Figure 1.12

CHAPTER 2

Twentieth Century - Conventional Machines and Small Scale Hydropower

2.0 Design Criteria

By the early twentieth century turbine design had passed from being an empirical science where trial and error methods produced the best machines, to a more exact science.

There existed a better understanding of the hydraulic processes that occur in a turbine system - in the pipework, the guide vanes, the runner and the diffuser. As a result it was possible to design a machine for a specific site with a reasonable certainty that the optimum performance would be obtained.

A significant contribution to this understanding was made by Professor Camerer of Munich who in 1903 introduced the concept of specific speed. Specific speed may be defined as the speed at which a turbine would operate if it was of such a size as to produce a power of one kilowatt at a head of one metre. (Specific speeds quoted in the ft. lb system of units are for one horse power under a head of one foot and are therefore 4.45 times smaller than the metric system equivalent). Usually specific speeds are quoted for the best efficiency point of a turbine. Different types of machines fit into quite distinct ranges of specific speed thus enabling machines to be classified as to their suitability for a given application.

Not only does specific speed enable a machine to be classified and compared with other machines under similar conditions but it provides a means of predicting the performance of different size versions of the same machine. Thus the interpretation of scale model tests is made easier and more certain.

A third benefit of specific speed is that it has made designers aware of the existence of gaps in the range of specific speeds. Thus areas where useful development can be carried out have been highlighted.

A full development of the concept can be found in standard fluid mechanics text books (reference 9). Simply specific speed Ns can be defined for a machine of shaft speed N(rpm) under unit flow Q and unit head H, as

$$Ns = \frac{N\sqrt{Q}}{H^{3/4}}$$

using unit power P instead of Q

$$Ns = \frac{N\sqrt{P}}{H^{5/4}}$$

It should be noted that specific speed does not have the units of speed - nor by convention does it have a consistent set of units, as N is generally in rpm, H in metres and Q in cubic metres per second.

2.0.1 Turbine Types

Twentieth century turbines can be divided into two broad classes, the impulse turbine and the reaction turbine. These have been referred to earlier by their historical labels as the pressureless turbine and the pressure turbine. Impulse and reaction, however, describe more accurately the mechanisms of their working and have become the standard nomenclature.

2.1 Impulse Turbines

In this type of turbine the whole head of the supply water is converted to kinetic energy in a nozzle before reaching the runner. The runner itself turns in air - usually at or about atmospheric pressure inside a casing designed to contain the water leaving the runner.

The water leaving the nozzle (or nozzles) impinges on vanes or buckets on the runner. There is a transfer of momentum at the vanes, hence work is done on the runner.

2.1.1 The Pelton and Turgo Wheels

The Pelton wheel figure 2.1 is the most widely known machine of this type. The Turgo wheel is suitable for a wider range of heads than the Pelton wheel.

Unlike the Pelton wheel which has buckets around the periphery, the Turgo uses a runner similar to the Girard axial flow wheel with radial vanes. The jet from the nozzles impinges on one side of the runner and emerges from the other. This system gives a better specific speed than the Pelton allowing a smaller wheel to be used for the same power and water flow. The penalties are a runner which is more difficult to manufacture with a considerable end thrust on the support bearing.

Both machines give excellent part load efficiences - losing less than 10% between 100% load and 25% load. Much of this is due to the considerable effort that has gone into the design and development of variable discharge nozzles and reactive control systems.

Pelton and Turgo machines are not suitable for low-head applications. Typically the head range of these machines lies between 10 metres at the lower end for a Turgo to 400 metres and above for a Pelton.

These types of machines have very low specific speeds, 10 or less in metric units, which for low head applications means that a physically large machine is necessary to pass the quantity of water needed to obtain a useful amount of power. Furthermore, as the runner turns in air a potential head, equal to at least the runner diameter, would be lost.

2.1.2 The Crossflow Turbine

This cross-flow turbine, sometimes called the Banki or the Michell turbine, is a mixed effects machine.

Depending upon the conditions, about 80% of the power developed is from an impulse effect and 20% from a reaction effect. A section through the machine is shown in figure 2.2.

The flow through the turbine is essentially two dimensional.

Water enters the turbine through a rectangular nozzle controlled by an inlet guide vane.

The shape of the nozzle is such that a considerable section of the circumference of the runner is covered by the jet - typically 120°.

The water jet strikes the vanes giving an impulse effect on the runner as the water is turned into the centre of the wheel. The water passes out of the centre of the wheel through the vanes on the opposite side to the nozzle when the reaction of the water leaving the wheel adds to the power produced.

The specific speed of this machine is much higher than for other impulse turbines. Figures as high as 150 (in metric units) are quoted by manufacturers. This makes the crossflow turbine suitable for low-head and medium head applications. The major manufacturer of this type of machine claims an operating head range of 1 to 200 metres with water flows of between 0.25 to $13m^3$/second (reference 10). As the flow through the turbine is essentially two dimensional, part load control is readily achieved by splitting the inlet pipe to the runner into several parts - each controlled by a separate vane. The usual method is a two section inlet, one twice the width of the other. Thus at two thirds full load, the smaller section is closed off completely and the water enters only through the larger section. At one third full load the larger section is closed and the smaller section passes all the water used by the turbine. By controlling the flow in this way, and by adjusting the inlet guide vanes at intermediate flows, the efficiency of the turbine can be held at around 80% down to 20% load.

2.2 Reaction Turbines

Reaction turbines operate fully submerged, thus, provided an efficient draught tube is used, the full available head at the turbine is utilised. This together with the higher specific speeds of the reaction turbines makes them generally more suitable for low-head applications than are impulse machines.

Two types of reaction turbine are commonly used, the Francis and the propeller. There are however several subdivisions within each type making it possible to cover a range of heads from 150 metres down to 1.5 metres.

2.2.1 The Francis Turbine

The origins of this machine have been discussed earlier in section 1.6.2.

Modern Francis machines are available as high head (up to 150m) medium head and low head (down to 2m).

High head machines have narrow runners in which the flow is mainly radial. These are enclosed in fabricated spiral casings. Specific speeds are typically around 90.

The runners of medium head machines are deeper than those of the high head turbines and allow radial and axial flow to occur within the runner. These runners are also enclosed in spiral casings. Specific speeds of about 220 are achieved. Low head turbines use very deep runners with a large area available to the incoming water. Flow in the runner is mainly axial. Often these machines are installed in open pits to minimise costs and reduce head losses. High specific speeds in the range 300 to 450 are obtained.

Figure 2.3 shows a low-head Francis installation.

Variable inlet guide vanes are universally used on Francis machines to give reasonable part load performance. The complexity of these variable vane mechanisms together with the manufacturing costs of spiral casings and multi vane runners make the Francis turbine a very expensive machine. This is offset to some extent by the general ruggedness of the runner which experience shows to have a long life - 50 years or more - with minimal maintenance.

If part load control is not required it is possible to use a centrifugal pump in place of the Francis turbine. Hydraulically these machines can be very similar, but as pumps are mass produced the costs are considerably less than for a tailor made machine.

2.2.2 Propeller Turbines

This turbine is most commonly used for low-head sites as very high specific speeds can be obtained, up to 900 in metric units.

The earliest machines - often known as Lawaczech turbines - had fixed blades - usually four or six in number. In 1913 however, Dr V Kaplan, patented a variable pitch propeller turbine with adjustable inlet guide vanes. This made possible a low-head installation with an almost flat efficiency curve at between 40% and 100% load.

Whilst propeller turbines have been used at heads over 50 metres, problems with cavitation at the high specific speeds of these machines make them more commonly used at heads of 20 metres or less.

Propeller turbines with both adjustable runner blades and inlet guide vanes are usually referred to as Kaplan turbines. The variable angle guide vane mechanism is expensive to manufacture - adding of the order of 50% to cost. Thus the semi-Kaplan - with fixed guide vanes and variable pitch runner blades is often used. The cost of producing this type of machine being approximately 25% more than the fixed blade machine but with extremely good turndown characteristics. The turn down performances of a Kaplan, a semi-Kaplan, an adjustable inlet guide vane fixed runner and a fixed guide vane-fixed runner machine are compared in figure 2.4.

For small-scale hydro propeller turbines are usually offered as package units when the electric generator, the transmission system and the propeller are delivered to site as an integrated unit. These units are designed to minimise losses across the installation as in low-head situations small losses are a substantial percentage of the total available head.

Four available types of installation are shown in figure 2.5. These are the tube (or S type) turbine, the right angle drive unit, the bulb turbine and the rim drive turbine. Each has certain advantages over the others. The rim drive machine provides the cleanest water passage and so has minimum losses for very low heads. The S type usually has greater losses in the water passage but allows the mechanical parts to be sited out of the water leading to a cheaper capital cost. The right angle drive machine is very flexible in that the generator can be situated in several positions around the turbine casing and a suitable speed increase can be incorporated in the drive gears.

The bulb turbine is usually only available for 50kW or less. It is however extremely compact and relatively simple to install.

All the above units are usually of the semi-Kaplan type and incorporate a shut-off valve or gate on the upstream side to isolate the machine for maintenance purposes.

2.3 Power for Small Communities

The power generation and distribution systems for cities and large communities can be easily funded by the number of consumers concentrated in a relatively small area.

Hence large capital sums can be spent on generation plant and transmission systems. Large hydro-electric plants are usually remote from centres of population but an expensive transmission system is made possible by the value of the electricity which passes through it.

Small communities, which are remote from major centres of population, neither can afford nor require large amounts of power. Thus the cost of building a transmission line to serve them from a large power source cannot be justified on purely economic grounds. In developed countries the building of such lines is often subsidised by the rest of society - either as a general increase in the cost of power or by means of a government subsidy.

In less fortunate societies, however, no funds are available to support small communities in this way and the provision of power becomes a local responsibility.

The worldwide nature of the problem is illustrated by delegates from over thirty countries attending the fourth Energy for Rural and Island Communities Conference in Inverness in 1985 (reference 11).

Numerous routes are being followed to attempt to solve the problem. The above conference included papers on photo-voltaics, biofuel gas generators, hydro and tidal power, wave power, wind power, geothermal power and direct solar based systems.

Many of these are aimed at under-developed countries but some, as for example - photovotaics, rely heavily on high technology for the hardware.

Small hydro power installations, however, can be designed for manufacture by societies where only relatively low technology processes are available. This is appreciated by International and National aid agencies who are increasingly supporting conferences devoted entirely to small hydro power (ref. 12, 13). These agencies now recognise that small scale hydro power schemes that is below 5MW - have to be considered against different criteria to larger schemes. For example the design costs are a higher portion of the total costs and the unit cost of the power produced is greater. Most low-head hydro schemes fall well below 5MW many being below 500kW. These form a very important section in the provision of power to remote communities as most people live in the lowland areas where only low-head schemes are possible.

Whilst the problem of supplying power to small communities is most acute in under-developed countries there exists a real need in the island communities of Scotland and Ireland where the tides may be the most reliable source of power. Thus low-head generators should be suitable for both river and tidal sites if they are to be universally useful.

2.3.1 Traditional Machines and Low-Head Hydropower

The major problem with low-head hydropower is that the energy available in the water source is small per unit volume. Hence to produce worthwhile energy outputs large volumes of water must be used.

Conventional rotodynamic hydraulic turbines generally have small water passages with complex shapes which makes them unsuitable for heads of 3 metres and below. In developing countries the problem is compounded by the available technologies often being inadequate to manufacture turbo machines of any size or complexity.

2.3.2 Water Wheels

The undershot water wheel is the best known ultra-low head hydropower machine. As discussed in Chapter 1 the efficiency of these machines is very low - typically only 20%. Also the rotational speed is low making electricity generation difficult in that high ratio step-up gear boxes are required. However because of the simple construction it is possible to build large machines with rudimentary materials. For example the wooden wheel built by John Smeaton in 1768 for London Bridge Waterworks was 9.75m (32ft) in diameter and 4.7m (15'6") wide with 24 flat wooden paddles each 1.37m (4'6") deep. This revolved at 5rpm and was in use for 49 years from 1768 to 1817 (reference 14).

Water wheels are commercially available today made of glass re-inforced plastic materials which the manufacturers claim give a long, trouble free life (reference 15).

These are intended for overshot operation when efficiencies of 70% are claimed. Standard diameters are 2.4m (8 feet), 4.9m (16 feet) and 6.1m (20 feet). Only the 2.4m diameter wheel, when overshot may be considered to be for low-head use and even then a further 0.9m clearance below the wheel is recommended to give clearance at time of floods. This gives a nominal working head of 3.35m. Thus, overshot water wheels, even relatively small ones require a head that is above that normally considered to be ultra low.

Hence if water wheels are to be considered for low head sites they must be undershot or breastshot.

The efficiency of a well designed low breast shot wheel is probably marginally less than an undershot wheel of the Poncelet type. The efficiency of both types is in the range 35 to 65%. The higher efficiencies are obtained at the bigger heads - say over 1m. The actual wheel for both types of machine would be very similar - each being furnished with between 30 and 50 curved vanes. The traditional (that is Victorian) method of construction is to cast each wheel rim in two or more sections often complete with rim gear drive. These sections are then riveted or bolted together and bolted to the spokes supporting the axle.

The buckets are formed from sheet steel rivetted to sockets cast into the rims. The basic soundness of this design is underlined by the hundreds of Victorian wheels of this type of which the rims and axles survive to this day awaiting only the replacement of the sheet metal buckets to restore them to working order see figure 1.12.

The technology to construct wheels of this type is certainly available in countries such as India and Egypt where iron foundries exist. These countries also have many low-head sites on their irrigation canal systems (reference 16) which would be suitable for water wheels.

The prefabricated construction would enable such wheels to be transported to remote sites by Land Rover or local animal transport. Local craftsmen could easily cope with the erection and maintenance of water wheels of this type. The relatively low speed of rotation - say between 10 and 15 rpm for a 1 metre head would only be a disadvantage if it were required to drive high speed pumps or synchronise electricity production with a diesel generator. For most rural power requirements up to 10kW, wheels of this type would be perfectly suitable and as efficient as other available turbo machines. The robustness, ability to cope with detritus and flotsam and ease of local repair should make water wheels a popular source of power in rural locations in developing countries. It is unfortunate that the wish to provide twentieth century technology to these communities has overlooked the possibility that suitably modified nineteenth century equipment would be more appropriate for small low-head schemes.

2.3.3 Crossflow Turbines

The crossflow turbine is perhaps the easiest rotodynamic machine to make using the technologies available in the developing countries. Plans suitable for local manufacture are available from several sources (reference 17). These designs, however are usually for heads above 6 metres. The peripheral speed of a crossflow runner being approximately $.5\sqrt{2gh}$ where h is the head. Thus for a 6 metre head a peripheral speed of 5.4m/second is typical. Hence a 0.3m diameter rotor will have a rotational speed of 346 rpm. At a 1m head the rotational speed drops to 140 rpm. However to pass 1 cubic metre of water per second under a one metre head a .3 metre diameter machine would be 7 metres long. Clearly this is not practical.

A solution to this problem using large diameter rotors is discussed in section 3.2.

Small crossflow turbines typically have efficiencies in the range 60-80% (references 18, 19). If loss of head due to setting the runner above tailwater is considered for low head applications then the gross efficiency can fall below 50%. The provision of draught tubes for crossflow turbines only partly solves this problem.

This, coupled with the undoubted short lives of locally manufactured crossflow turbines in developing countries should make this machine a questionable proposition compared to a good water wheel. A major factor, however, in the promotion of this turbine by development agencies is the intention to improve local craft skills in the areas of welding and metal fabrication.

It is, however, unlikely that the conventional crossflow will be used widely for low-head hydropower.

2.3.4 The Propeller Turbine

Although propeller turbines are used widely for low-head hydropower in developed countries they have not been found suitable for local manufacture in developing countries.

The major advantage of a propeller turbine is its high specific speed which enables a relatively small unit to produce reasonable power at a low head and also to reduce the necessity for a gear-box when coupled to a generator.

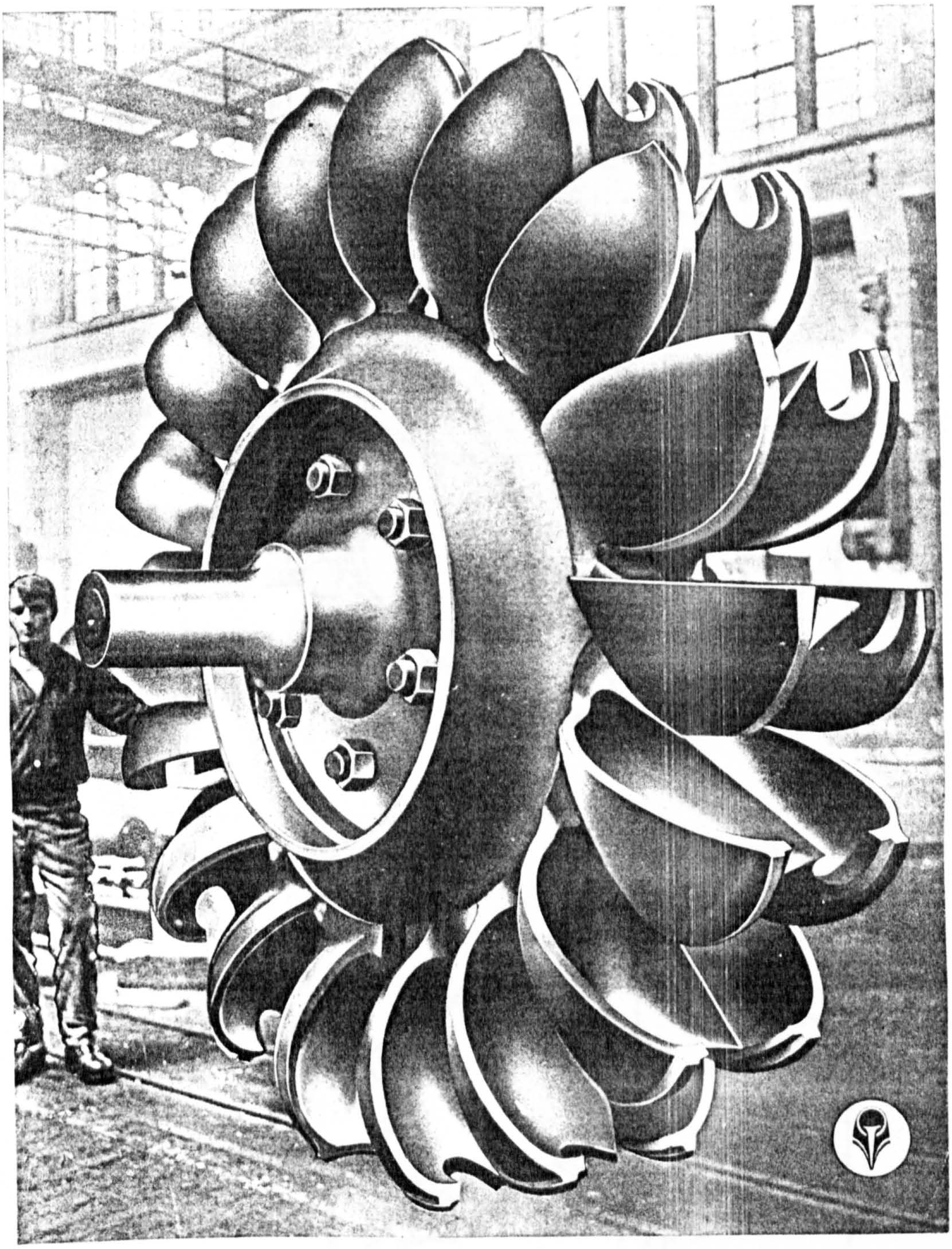
Propeller turbines, however, do require careful design and manufacture of hydraulic profiles to obtain good efficiencies. Even small machines manufactured in industrialised countries for use in developing countries, can have efficiencies in the low sixties (reference 20).

The setting of propeller turbines also requires more care than for other types. Small changes in head and flow produce large changes in efficiency and power output. The provision of Kaplan style adjustable vanes is usually beyond the capabilities of local manufacture.

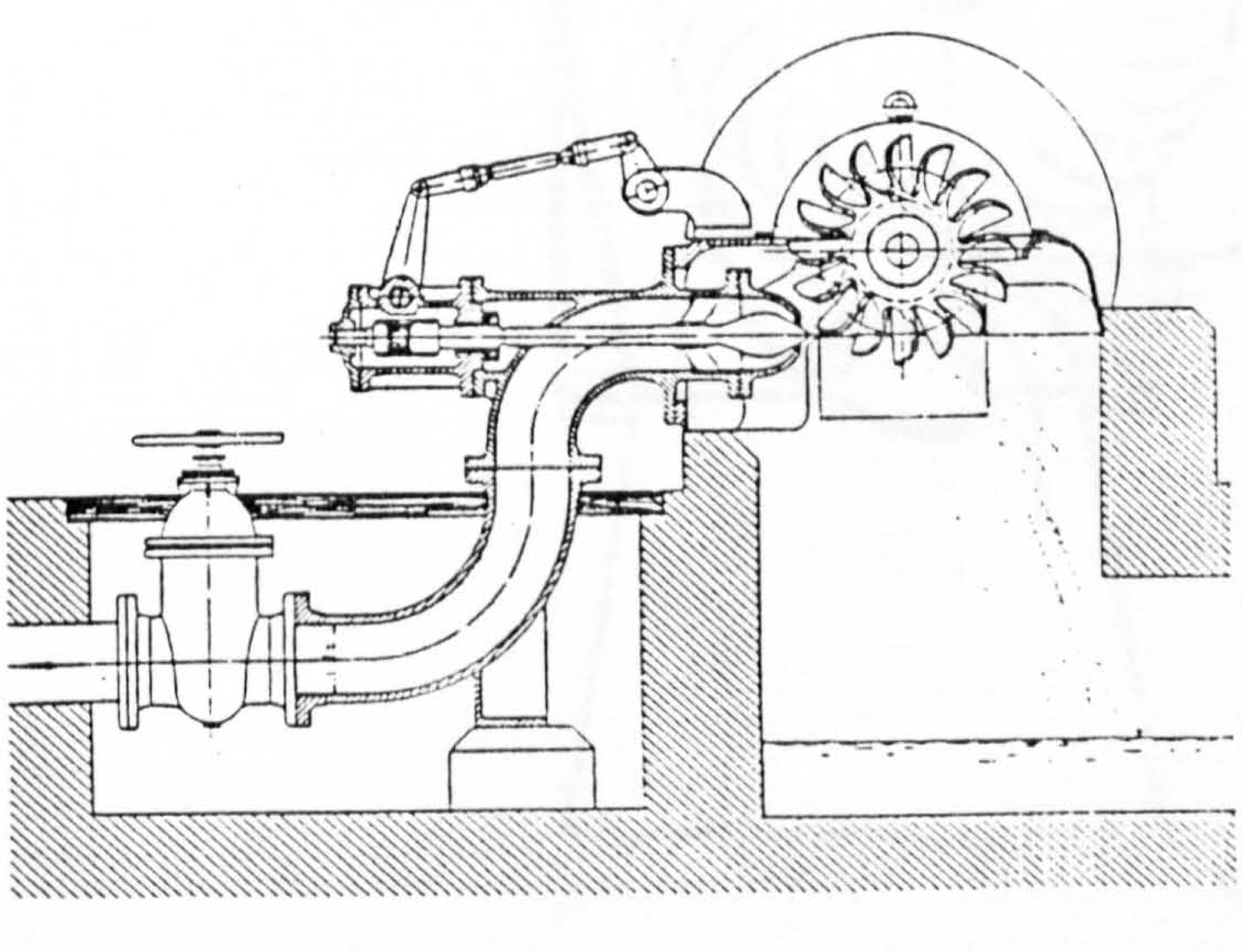
A further drawback to the propeller turbine is its sensitivity to water-borne debris. Solid debris, because of the high rotational speed, will often cause serious mechanical damage to the runner.

Hence better provision must be made for trash racks and silt traps than with more robust machines.

A small machine, designed for local manufacture is described in reference 21. This machine is designed for a 2m head with a rotational speed of 1000 rpm. Individual adjustment of the inlet guide vanes is possible. By using the machine to generate D.C. electricity and charge batteries the designer has avoided the need to maintain an accurate control of the rotational speed.

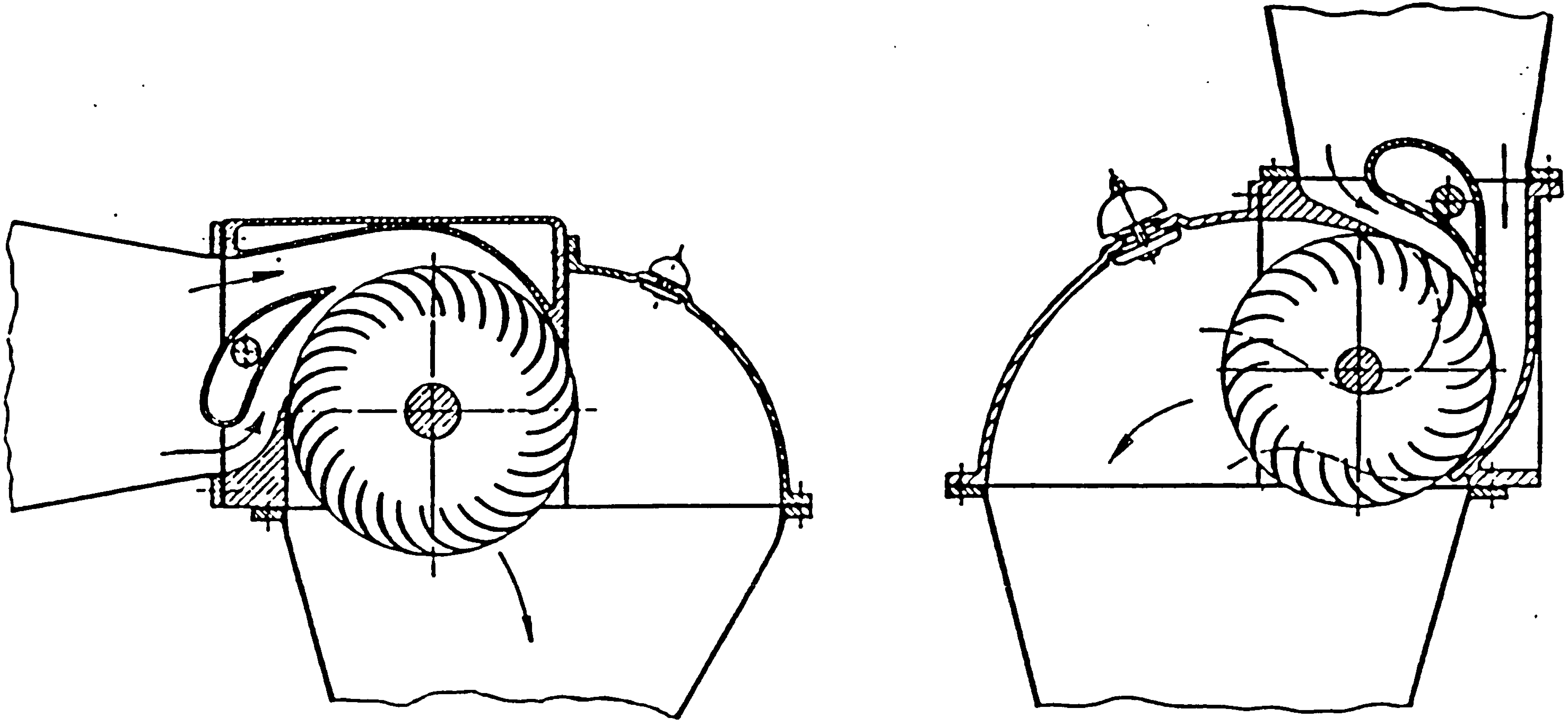


Cast steel Pelton wheel.



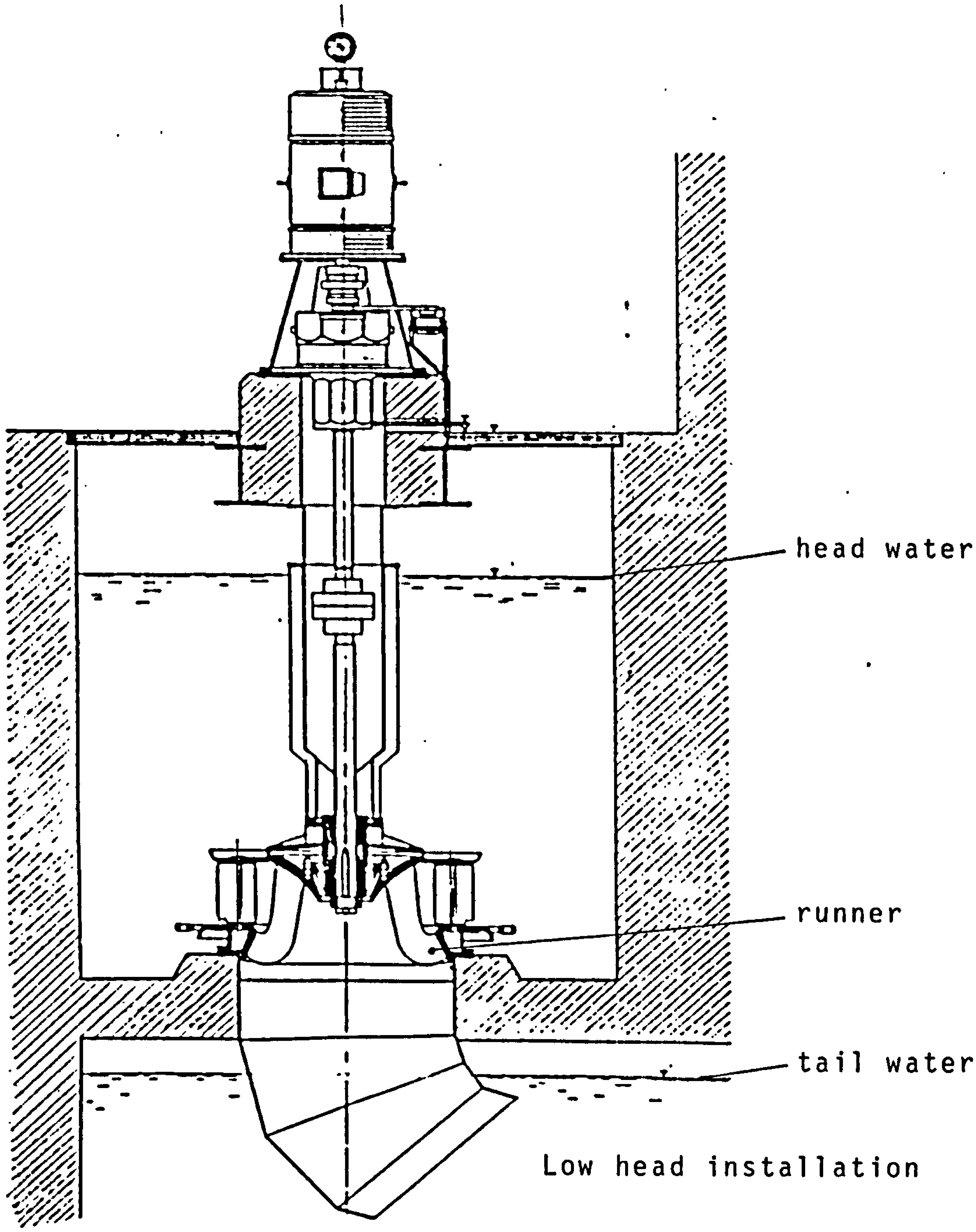
Pelton turbine.

Figure 2.1

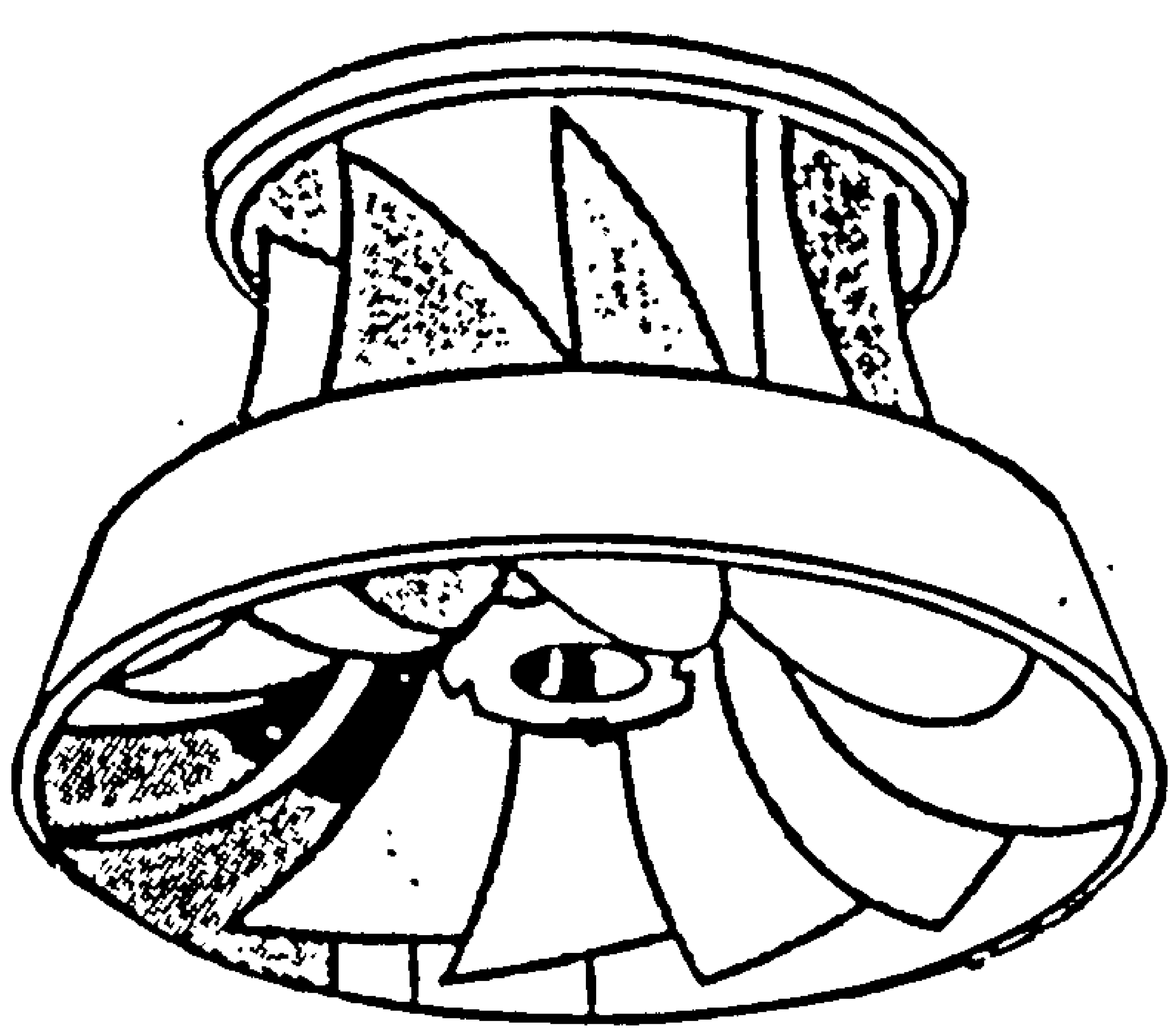


CROSSFLOW TURBINE

Figure 2.2



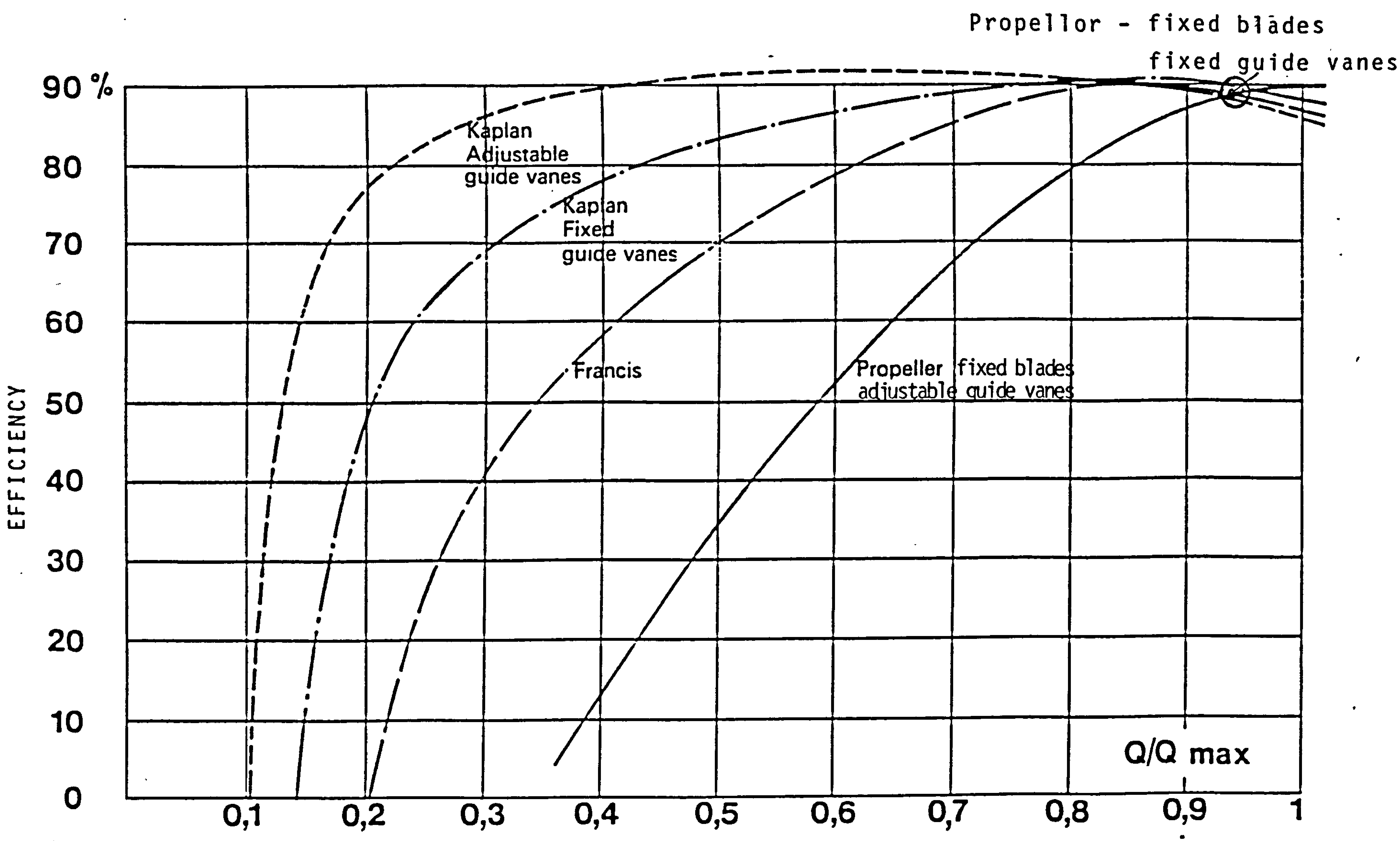
Low head installation



Francis runner for
low head application

Francis turbine

Figure 2.3



Part flow efficiency for low head turbines

Figure 2.4

Propellor turbine installations

Figure 2.5

CHAPTER 3

Modern Unconventional Hydro-power Generators

3.0 Introduction

Over the centuries many attempts have been made to improve on existing hydro power technology by looking outside what is considered conventional. Barkers' Mill, although it enjoyed only a small success was an early attempt to escape from the conventional. Recent efforts have often looked to aerodynamic technology to produce devices such as the vertical axis turbines and the hydro pneumatic machines. Cash encouragement has been given to the protagonists of many such devices by governments who wish to see alternative energy sources developed to appease the more environmentally conscious sections of the developed world's population. This chapter outlines the more successful and better known unconventional devices.

3.1 The Vertical Axis Water Turbine (VAWT)

These machines are identical in operation to the Darrieus or vertical axis wind turbines. A series of aerofoil sections are suspended vertically on arms protruding radially from the central axle. (Figure 3.1).

The hydrodynamic forces acting on the aerofoil when suspended in a flowing stream gives rise to a component of lift at right angles to the arm thus producing rotation.

The magnitude of this component varies as the aerofoil rotates on the arm and the relative direction and velocity of the stream flow to the aerofoil changes. The net hydrodynamic torque on the turbine is obtained by integrating the forces on all the blades during a complete revolution. In order to maximise the torque produced the speed of the blade is usually two to five times faster than the velocity of the water through the turbine.

The machine is hydrodynamically symmetrical about its vertical axis, thus the direction of rotation is the same regardless of the flow direction of the water.

Two types of this machine have been proposed, one a light structure - usually suspended below a barge for use in rivers, the other a substantial machine contained in a barrage across a river or tidal creek.

3.1.1 The Barrage VAWT

The suggested configuration of this is shown in figure 3.2.

Laboratory models have developed net efficiencies of 50% (that is using the head across the turbine). Gross efficiencies, however, using the head across the barrage were reduced to a maximum of 37% due to entry and exit losses.

Problems with this machine found from tests on a 2.4m diameter prototype are uneven torque output, fatigue of the blades and cavitation problems at heads over 2m. To obtain maximum energy at tidal sites the speed of rotation must be adjusted to suit the available head as efficiency drops sharply if the optimum ratio of blade velocity to water velocity is not maintained.

In spite of these drawbacks proposals have been made to construct tidal barrages containing these machines with a rotor diameter of 12m and a height of 14m. The relatively simple construction making such a large machine possible.

It is difficult to imagine that this machine, despite its simple construction, will be a practical proposition at small scale low-head sites.

3.1.2 Free Stream (Barge mounted) VAWT

Several prototype machines of this type have been built. A small one, producing approximately 1kW is described in reference 17.

In all designs the rotor is suspended from above, that is cantilevered from its support bearings. In small machines the blades themselves are cantilevered from the radial arms making the machine extremely fragile and susceptible to damage from water borne debris. In spite of this it is possible that a suitably designed machine may have a use on water courses where it is not possible to build a barrage or where a degree of transportability is required.

These machines are approximately 35% efficient - water to electricity. A stream flowing at a velocity of 1 metre per second will contain a kinetic energy of ½ x 1000 x 1^2 = 500 watts per cubic metre of water. Thus a device with an area of 1 square metre (say one metre diameter with one metre deep blades) would produce 0.35 x 500 = 175 watts.

The volume flow per second is the area of the device multiplied by the stream velocity, hence the power produced by a device in a stream is $\eta \times 0.5 \times A.V.V^2$

where η is the efficiency (0.35)
A is the Area of the device (Diameter x blade depth)
V is the stream velocity

Thus Power kW = $0.175\ AV^3$

As an example of a practical unit consider a machine 4m diameter, 1.5m deep in a stream flowing at 1.5m/second.

$$\text{Power} = 0.175 \times 4 \times 1.5 \times 1.5^3 \text{ kW}$$
$$= 3.5 \text{ kW}$$

The speed of rotation - assuming a ratio of 3, blade to water velocity - is 21 rpm. Thus a considerable step up is needed to connect the rotor to a standard 1500 rpm generator.

The power output of the machine improves rapidly however with increase in water velocity - an increase in velocity of 25% doubles the power produced.

It is possible that these machines could find an application as temporary sources of power at times of emergency or for specific tasks when their light weight and portability gives them an advantage over other devices. It is difficult to imagine that they could be made robust enough for long term or permanent use.

3.2 The Momentum Water Wheel or Undershot Crossflow Turbine

This machine combines the features of a crossflow turbine with those of an undershot water wheel of the Poncelet type.

The wheel is similar in construction to a Poncelet wheel but the buckets are open at the inner face thus allowing some water to pass through to the interior and out through the opposite side in the manner of a crossflow turbine. There is no casing to the wheel and control is limited to the size of sluice opening in the same way as the Poncelet wheel.

Two arrangements for the device have been proposed (reference 24). These are shown in figure 3.3.

In the first arrangement the wheel is set down stream of a sluice which is raised to allow water to strike the underside of the wheel. A shaped masonry or concrete sill below the wheel is required to maximise the performance. In the second arrangement water flowing over the crest of a weir runs down the shaped front face to strike the underside of the wheel.

Laboratory tests on 1/5th scale models have shown little difference in efficiency between the two types - both having a maximum of around 65%.

The arrangement with the sluice gate gives a degree of control, although unlike the crossflow turbine part load control cannot be obtained by allowing water onto only a part of the length of the wheel as it rotates in the tail race. Churning losses in the unused section would reduce any advantage gained from the reduced flow.

Although allowing the wheel to be partly submerged does reduce part load control it has the advantage of ensuring that full benefit is obtained from the available head.

An analysis of this type of wheel is given in reference 25. This enables wheel diameter relative to head and flow to be established. For heads in the range 1 to 3 metres a ratio of wheel diameter to head of 2 gives high theoretical efficiencies.

This is supported by the laboratory tests which on a proposed installation in Egypt gave a wheel of 3.5m diameter by 4.5m long for a head of 1.9m passing $17m^3$/sec.

The above indicates that this wheel is somewhat smaller than a Poncelet wheel for the same duty. Typically a Poncelet wheel would have a diameter of at least twice the available head. The shroud depth (blade depth) being about one eighth the radius with the water jet about one half of this. Thus a Poncelet wheel to compare with the above would be approximately 4m in diameter with a width of 11m.

The ability of this wheel to pass more water is due to the use of less blades (about half the number of a Poncelet) without a loss in efficiency over the Poncelet.

This machine is suitable for sites with a fairly constant flow when only limited control is required. It should be possible to manufacture and erect in developing countries and is robust enough to survive with little maintenance.

3.3 Hydro Pneumatic Devices

These devices use a head of water to compress air which is then passed through a suitable machine to do useful work. In this way it is intended to reduce the mechanical complexity by omitting gear boxes or other speed increasing devices.

The idea of using water to compress air is not new. The hydraulic air compressor or trompe was used in Italy in the sixteenth century for blowing forges. It was later widely used throughout Europe for iron smelting.

The trompe is shown in figure 3.4. Water from an overhead channel enters a vertical downpipe at the top of which are openings; through these atmospheric air can be entrained. Bubbles of air are thus carried by the water to the bottom of the pipe where they are released into a separating chamber. The air passes from this chamber via a pipe to the tuyere of the forge.

The pressure of the air is only dependent upon the height of the tail race above the separating chamber - not upon the upstream head, although the greater the flow down the pipe the more air is delivered to the forge.

More modern versions of this device, constructed in the early twentieth century claim efficiencies of over 80% at pressures of 8 bar (120 psi). (The efficiency being the ratio between the energy gain of the air by isothermal compression and the energy lost by the falling water). These have been used to power pneumatic tools in mines and quarries where electric powered equipment would be undesirable.

Perhaps the largest of these was built to supply power to a silver mine at Cobalt Ontario, Canada in 1910. Water was dropped 105 metres down a vertical mine shaft, the air separated out in an horizontal gallery 300m long the water being expelled upwards through another shaft. This device worked on a head of 21m producing 19m^3 per second of air at 8.28 bar. Chapter 7 expands on the possible use of a trompe for low-head hydropower generation.

Modern hydro pneumatic devices received a revival of interest with the upsurge of research into wave energy in the 1970's. Various devices were tried which relied on the waves forcing air from a closed chamber through an air turbine. Initially these air turbines were of a conventional type which would accept air flow from one direction only. Thus the suction effect of a receding wave was either wasted or required a series of one way air valves to correct the direction of the air flow.

A self rectifying air turbine (ie. one that rotates in the same direction regardless of the way the air is flowing) invented by Professor Wells of Queen's University, Belfast (reference. 26) removed the need for these valves and thus simplified the conversion of wave energy to rotational motion. These Wells turbines form the basis of recent low head hydro pneumatic devices in which high velocity air flow enable rotational speeds of several thousand rpm to be obtained.

A great deal of work has been carried out on hydro-pneumatic devices for low head hydro applications by the Coventry Lanchester Polytechnic Energy Systems Group - both in the laboratory and on prototypes (ref. 27).

Two basic machines have emerged from this work - the box and the bag.

3.3.1 The Box

This device is shown diagramatically in figure 3.5.

It consists of a chamber with two water control valves at its base. One connects the chamber to the upstream water, the other to the downstream water.

At the upper end of the chamber is a duct in which is set a Wells turbine connected to a generator. The outer end of the duct is connected either to atmosphere or to another similar chamber.

The mode of operation is as follows:-

a) The downstream water control valve is closed and the upstream one opened. Water flows into the chamber forcing air out through the turbine.

b) Just before the water in the chamber reaches the upstream level the upstream water control valve is closed and the downstream one opened. The water flows out drawing air into the chamber through the turbine.

The cycle is then repeated.

A modification to this simple box is to connect two boxes together so that they work in anti-phase. One filling when the other is emptying. The duct containing the turbine is connected between them. In this way the turbine experiences twice the pressures across it compared to a single box arrangement. Thus the size of the turbine (an expensive item) is reduced for a given power output.

It is claimed by the Lanchester group that a box of area 10 square metres operating on a 2 metre head would pass 1.33 cubic metres per second of water and develop an electric power of 11kW using a Wells turbine rotating at 6000 rpm.

This gives a net efficiency of 42%. However this estimate is based on a power to the gates of only 50W. The writer's experience with the AUR Water Engine, hydraulically a similar device, suggests that this is too small by at least a factor of 10. This would reduce the efficiency to 40%. Further doubts are raised as to the mechanical complexity, life and cost of the turbine and generator rotating at 6000 rpm.

Whilst this device may have had a limited success in the laboratory it requires considerable development before it could be offered as a solution to the energy needs of the developing world; if indeed it can ever be made robust and reliable enough.

3.3.2 The Bag

This is described in reference 27. It is intended for similar heads to the box but as the water flow is continuous it can be used for bigger installations. The heart of the device is a fabric reinforced rubber bag or sheet which serves to separate the water from the air inside a sloping tunnel connecting upstream water to downstream water. The bag has two sections, air being forced by the water from one to the other through a duct containing a Wells turbine. The design of the bag is such that discrete slugs of water pass down the tunnel so causing the air in the bag to oscillate from one section to the other.

A prototype machine on the river Derwent near Derby has been designed for a head of 2.75m at a flow of 9 cubic metres per second. Electric power output is predicted at 150kW to give a net efficiency of 62%. The air turbine is designed to rotate at 3000 rpm.

The water tunnel containing the bag is 30m long by 5.5m wide by 2.75m deep. The nylon reinforced rubber bag which lines the tunnel has a predicted life of only 5 years.

It is difficult to imagine how a device of such size, doubtful life and undoubted cost can compete with the more conventional machines available.

3.4 Positive Displacement Devices

A positive displacement water pressure engine was described by Belidor (reference 6) in 1739 but it was not until 1750 that an actual machine was built by Hoell to pump water from a mine at Schemnity in Hungary (Ref. 28). Similar engines are described by the mining engineer Francis Thompson of Ashover, Derbyshire who referred to them as pillar engines in his journals dated 1788. Basically these machines used the high head of water across the piston in a cylinder in a similar manner to the steam in a steam engine. This water cylinder drove a rod which worked a pump cylinder.

The entry and exit of water to the power cylinder being controlled by automatic valves.

An engine described by Thompson worked on a head of "40 yards" (37m) and had a power cylinder of 8 inches (0.203m) diameter. Positive displacement machines for low head use need very large diaphragms to work against to produce a useful amount of power. Hence simple piston devices are not practical and other means have been used to obtain the necessary through flows of water.

The machines developed at Salford University are all of the positive displacement type. These are discussed fully in the following chapters - however they are briefly described here to set them in context.

3.4.1 The AUR Water Engine

This machine named after its inventor Alister Ure Reid, consists of a float in a closely fitting chamber into which water can be introduced and exhausted through large valves in the base (Figure 4.1). Power is extracted from the machine both as the float rises and as it falls by an ingenious system of hydraulic rams which match the bouyancy and gravitational forces acting on the float. The pressurised fluid generated in the rams is used to drive a hydraulic motor which can be used to turn a generator, a flour mill, a saw, a pump or whatever the local need.

A twin chamber prototype generating approximately 5kW has been built on the river Arun near to Billingshurst in Sussex.

The inventor considers that this machine is suitable for tidal use when devices of several hundred kW could be built. Such machines, however, would require floats of tens of square metres in area and be extremely cumbersome to build and maintain. It is possible that small machines of up to 30kW could be competitive with more conventional machines.

3.4.2 The Salford Transverse Oscillator (STO)

This machine was invented at Salford University by Professor E M Wilson and Dr G N Bullock. It overcomes the economic problem of moving large quantities of water through a machine by utilising simple passages made from relatively cheap materials such as concrete.

The device consists of a double wall barrage built across the direction of water flow. At regular intervals in the barrage are gates which connect the space between the two walls with the upstream and downstream sides of the barrage. A vertical paddle hangs into the space between each pair of gates so preventing water flowing along the barrage. The resulting head across the paddle causes it to move towards the gate open to the downstream water. By opening alternative gates to upstream and downstream water the paddles are made to oscillate backwards and forwards along the barrage. A carriage, running on the barrage wall connects all the paddles together and enables power to be extracted by hydraulic, mechanical or electrical means.

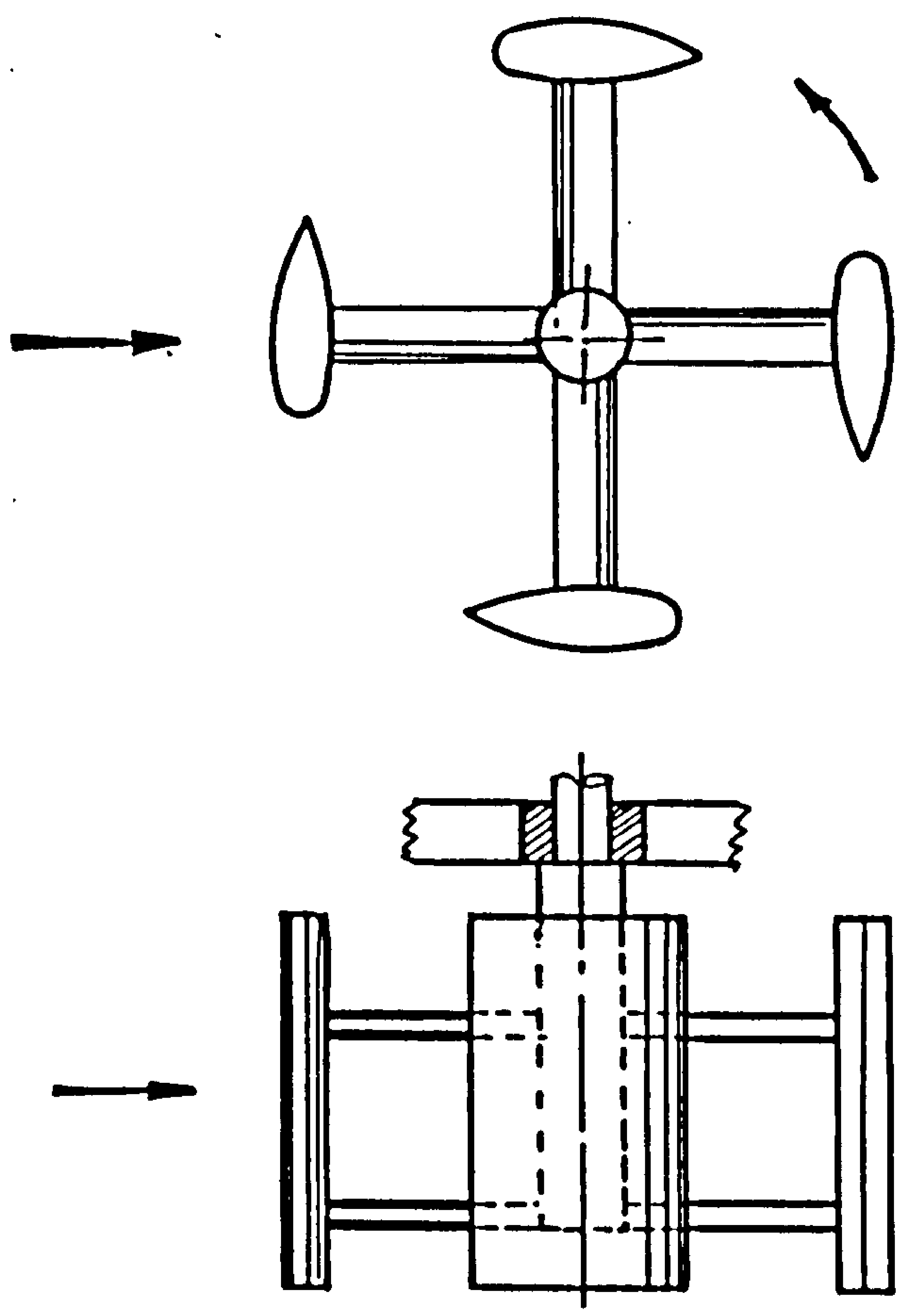
This device is suitable for tidal applications as flow direction can be reversed. It is intended for building with locally available materials in sizes from 20kW to several hundred kW.

3.4.3 The Underwater Motor

The AUR Water Engine and the STO are free surface reciprocating machines. The underwater motor is a fully submerged rotating device. It is similar in construction to a sliding vane pump with the water passages and ports modified to allow the passage of large volumes of water.

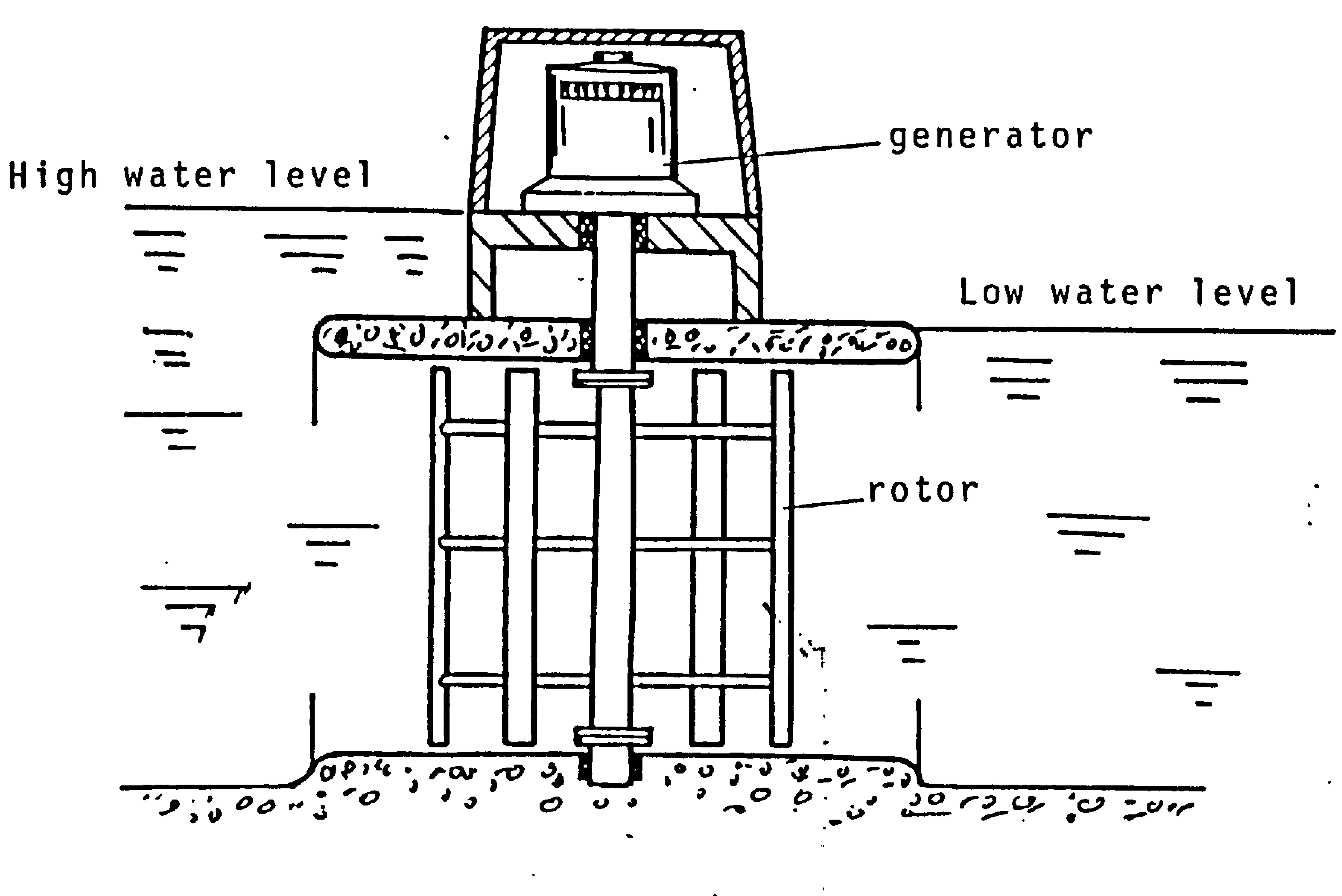
The power output of the machine depends upon the pressure across the vanes which slide in and out of a rotor. Hence changes in water level do not affect the machine provided the difference in head remains the same.

This machine is intended to be built in small factories in developing countries and transported to site as a unit to be fitted onto a prepared base. Power outputs of a standard, easily transported machine would be up to 20kW depending upon the available head.



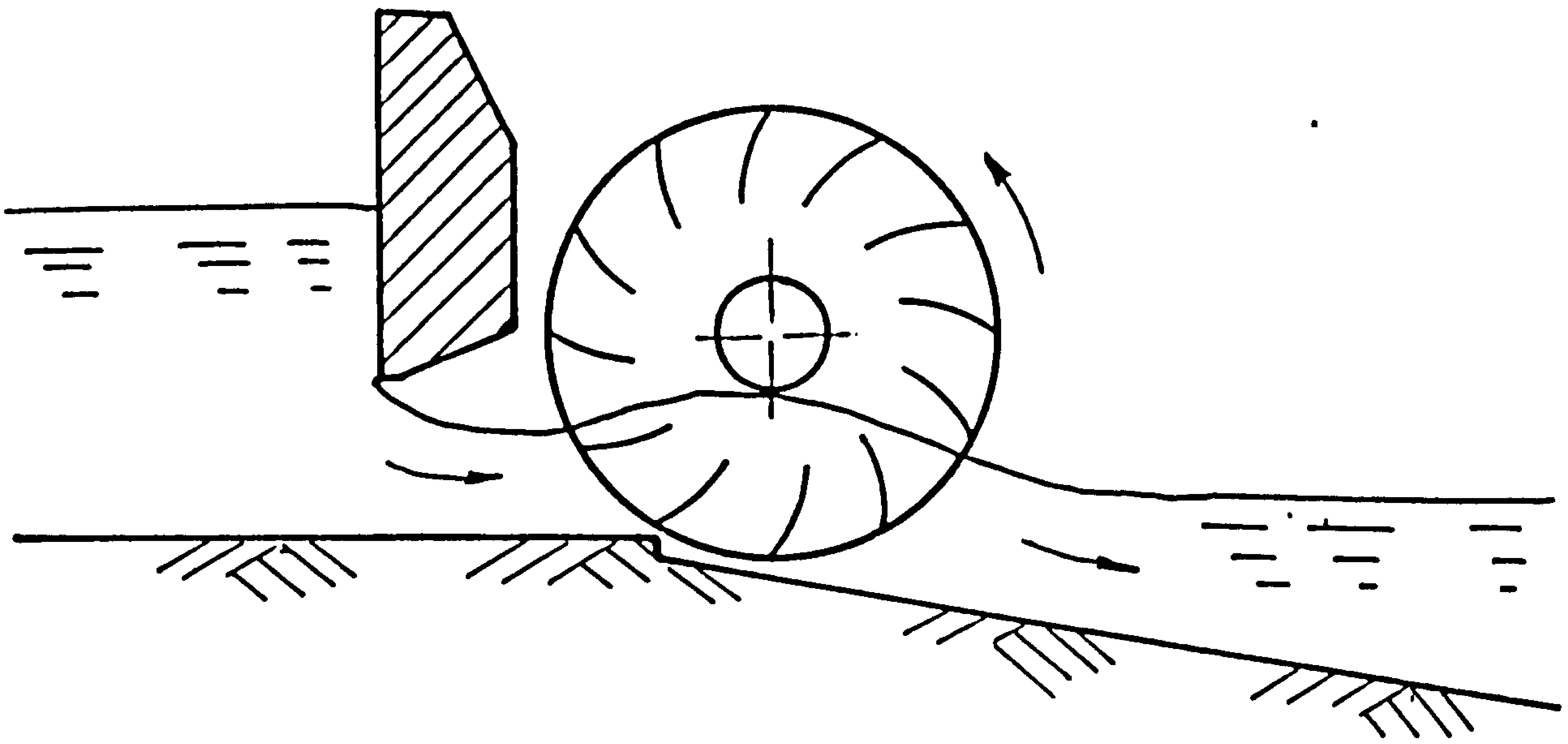
The vertical axis water turbine

Figure 3.1

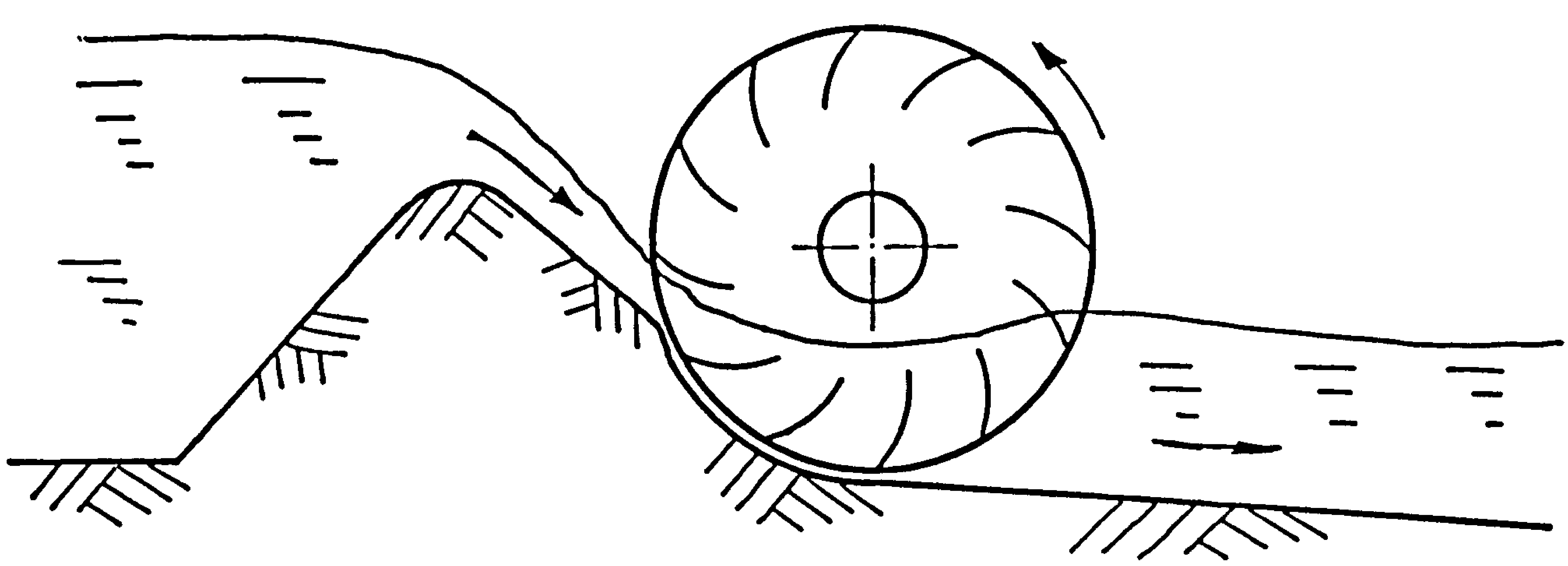


The barrage V.A.T.

Figure 3.2



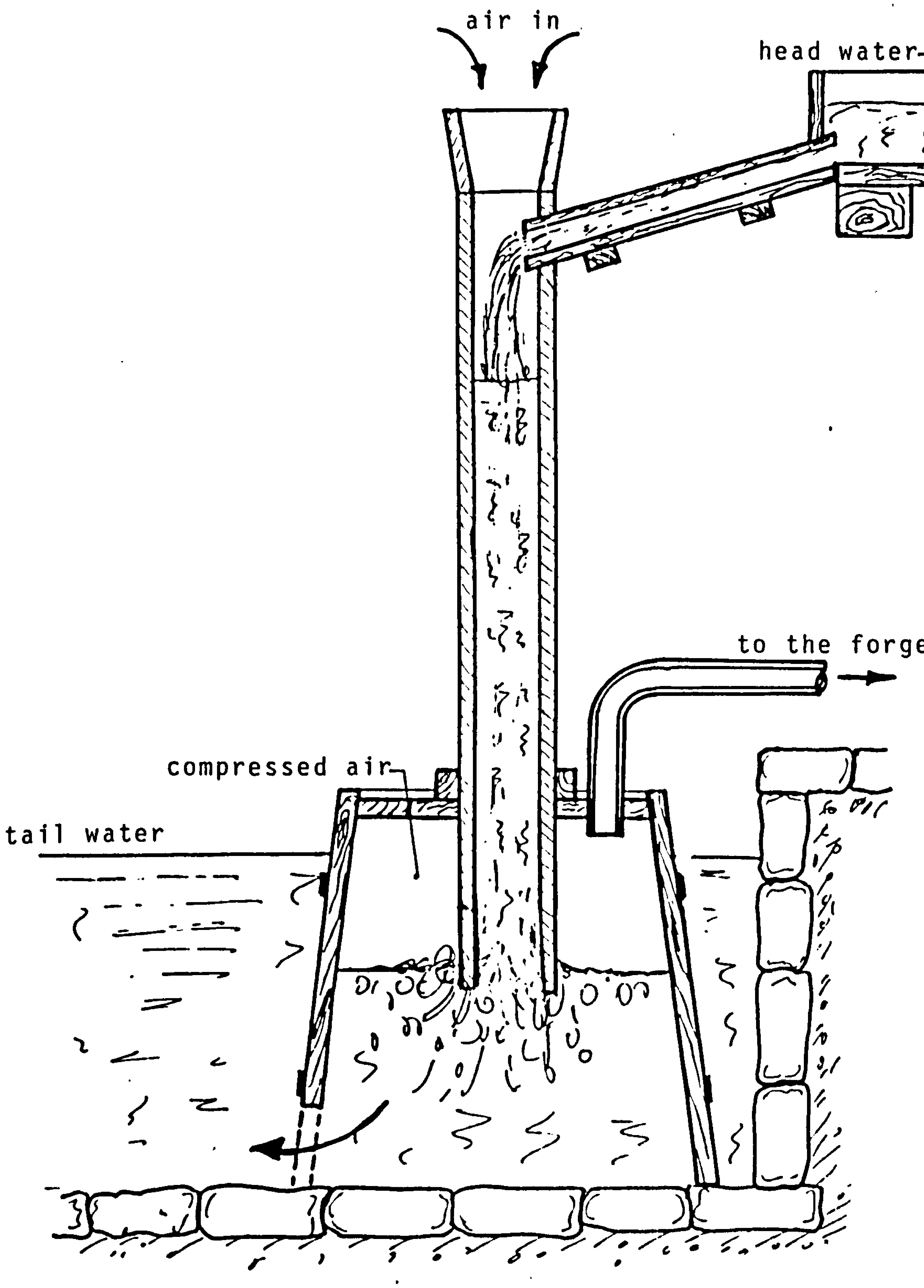
Bottom jet type



Fall jet type

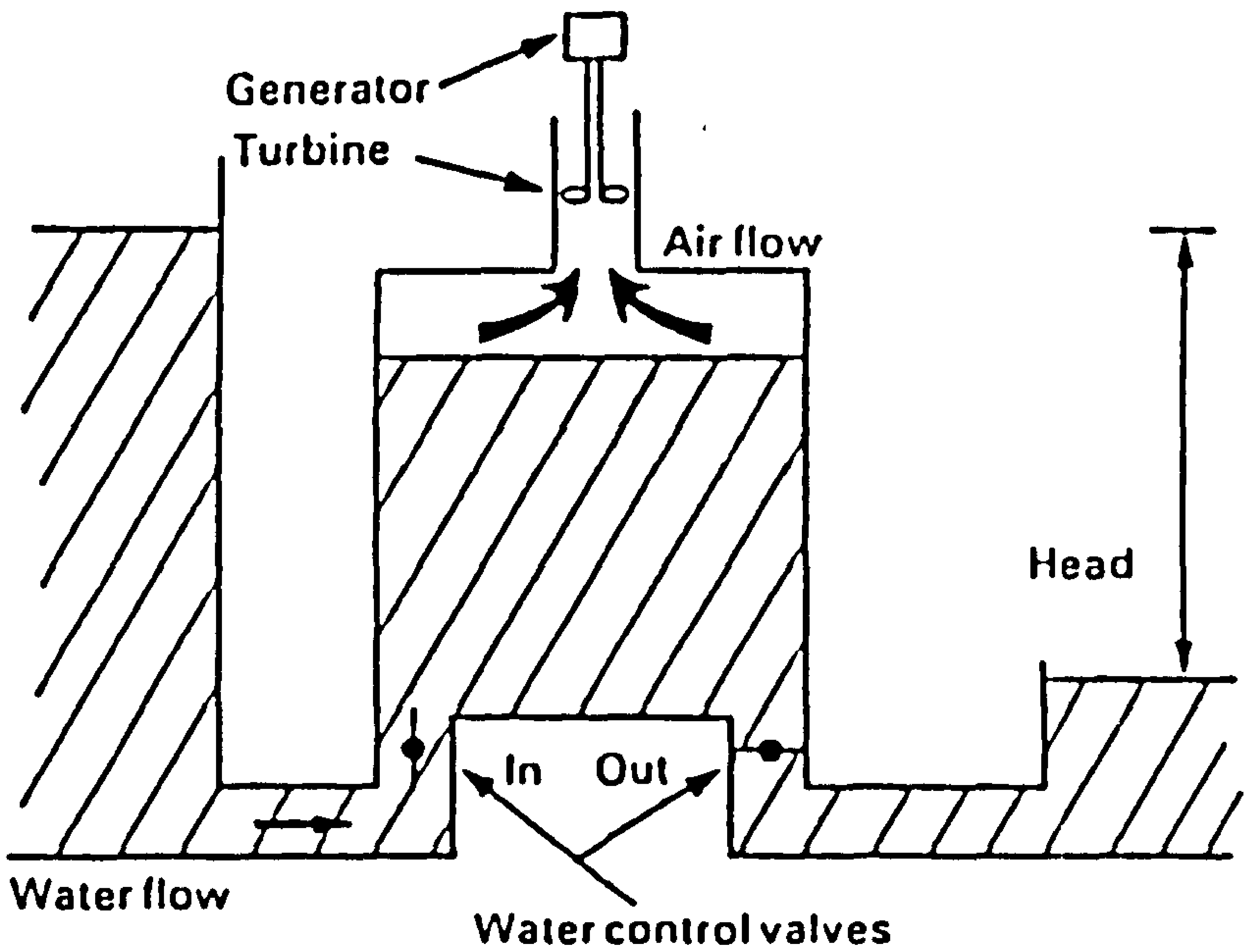
The momentum water wheel

Figure 3.3

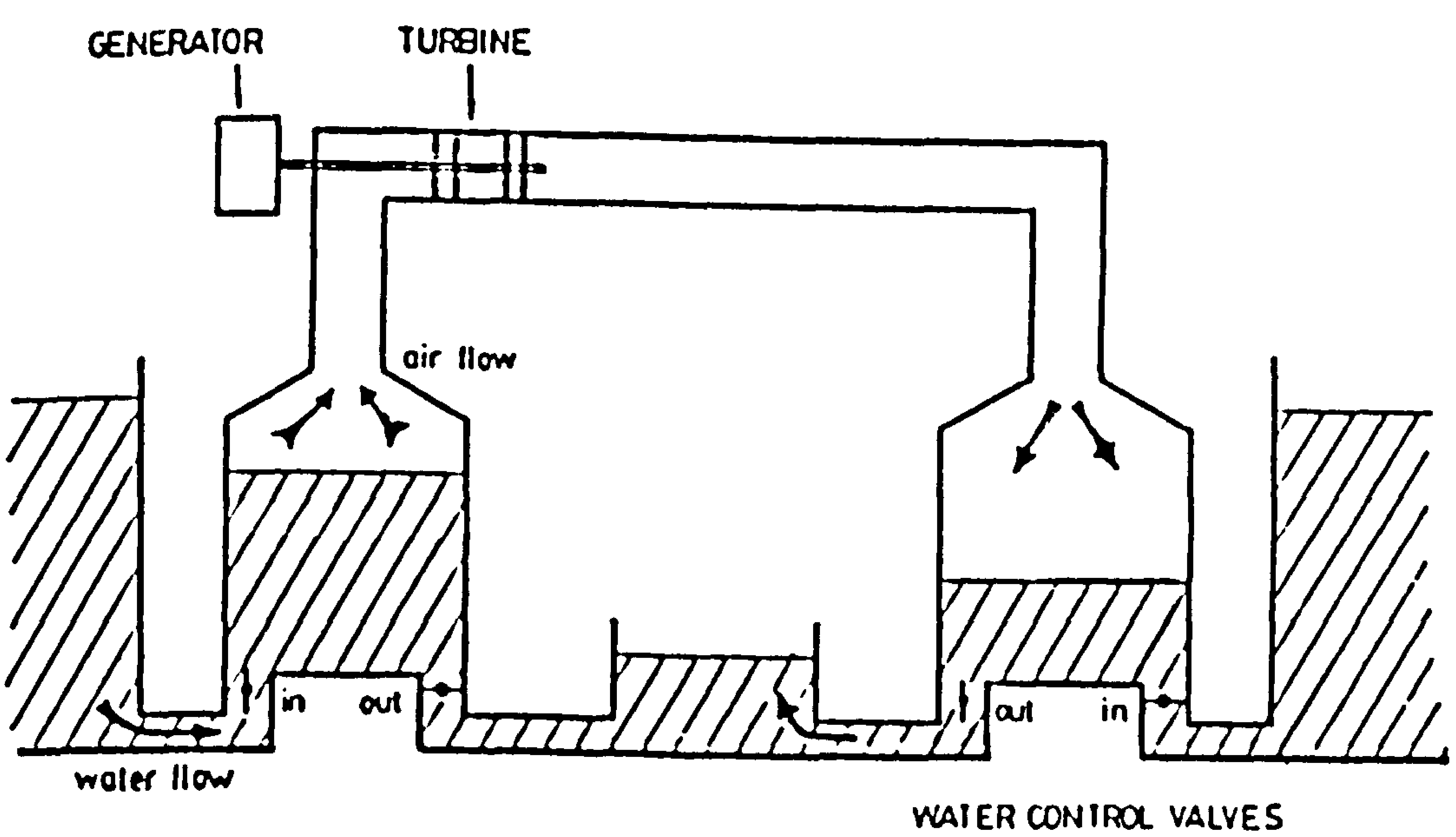


Trompe for blowing a forge

Figure 3.4



Single chamber



Twin chamber

The box hydro-pneumatic generator

Figure 3.5

CHAPTER 4

The AUR Water Engine (AURWE)

4.0 Introduction

The AUR Water Engine (AURWE) is named from the initials of its inventor Mr Alister Ure Reid and was patented by him in 1975. Although a full size machine was built by Mr Reid and his partner Mr T L G Deuce at Ripley Mill in Surrey in 1978 it was not until 1980 that a programme at the University of Salford was started to develop the machine. This programme was funded jointly by AUR Hydropower Limited and the Department of Energy.

Table 4.1 is a summary of the work carried out at Salford between 1980 and 1984 on the machine and its progress from a simple laboratory model to a full size prototype.

This chapter outlines the work undertaken to create a practical machine with minimum complexity whilst at the same time improving the power output and efficiency. A more complete record of the tests carried out and the computer analysis and simulation work can be found in reference 29.

The work at Salford was carried out in the Department of Civil Engineering under the direction of Professor E M Wilson, Dr G N Bullock was Project Manager from October 1980 to July 1983. The writer was involved in the project from February 1982 as scientific officer with responsibility for the design and construction of the machines, including the Gibbons Mill prototype.

4.1 The Operating Cycle

Energy is extracted from the water by harnessing the work done against an applied resistance, by the vertical displacement of a float within a chamber. Water flow into the chamber from the higher level and out to the lower level is controlled by a gate system at the base of the chamber. The float is linked to a power take off mechanism which incorporates a number of hydraulic rams. As the float responds to changes of water level in the chamber, so the rams deliver a supply of pressurized fluid to the load.

4.1.1 The Power Take Off System

The effectiveness of this machine lies in the simple way in which the power take off system resistance matches the float displacement forces. This is illustrated in the working cycle Figure 4.1.

At (1) the float is in its bottom position, and the upstream gate is opening. The lower pair of hydraulic rams are fully compressed, the upper pair fully extended. The annulus round the float quickly fills to the upstream level, which subjects the float to a buoyancy force.

(2) shows the float rising under the buoyancy force pressurizing the oil in the upper pair of rams: at the same time the lower pair of rams are being extended and drawing in oil.

At (3) the float has reached the top of its stroke, thus actuating the mechanism which will close the upstream gate and open the downstream gate. The upper rams are now fully compressed, having expelled all their oil under high pressure, while the lower pair are fully extended and filled with oil.

(4) The upstream gate is completely closed and the downstream gate is opening, thus lowering the water level in the annulus to the downstream level. The float is now subjected to a downward gravitational force equal and opposite to the buoyancy force described in (1).

(5) shows the float on its way down. The downstream gate is wide open: the lower pair of rams are being compressed and the upper ones extended.

(6) The float has reached its bottom position, again actuating the gate mechanism which will close the downstream gate and open the upstream gate to repeat the cycle.

The rams are single acting and only resist the float motion on the compression stroke. One way valves on the inlet and outlet ports ensure that oil can only flow in the correct directions.

The maximum upthrust (buoyancy) of the float occurs at the instant the rising water in the annulus reaches its highest level. This occurs when the rams and float are as shown in Figure (1). That is when the downward component of the ram resistance is a maximum. As the float rises in the water, Figure (2), the buoyancy force is reducing but the downward component of the upper ram's resistance is also reducing as they approach the horizontal.

In Figure (3) the float has risen to its equilibrium position and, therefore, is no longer pushing upwards on the rams. The rams are now horizontal and thus do not exert any downwards force on the float.

Figure (4), (5), (6) show the float on the second half cycle. The water falls away down the annulus and the weight of the float works against the lower pair of rams.

The above describes the ram configuration used on the early machines and as described in the original patent; later machines obtained the same effect with fewer rams.

4.1.2 Basic Principles of the Machine

A single vertical ram could be used to harness the energy of the float but this, if it operated at a constant pressure, would not produce a resistive force so nearly matched to the forces on the float.

A simple static analysis of the forces on the float demonstrates how the basic dimensions of the machine are arrived at.

The equation of motion of the float within the chamber can be described by Newton's Second Law.

$$M \ddot{z} = F - M g - \text{Frictional Forces} - \text{Resistance of Rams} \quad [1]$$

where

F = force of water on the float
g = acceleration due to gravity
M = mass of float
z = position of float from its lower equilibrium position

The term F is a complex function which may be derived by the use of Bernoulli's Equation extended to cover unsteady flow, the mass of the dynamic system, and empirically determined discharge coefficients. These are outlined in Ref. 29. The essence of it can be understood by a static examination of the forces acting on the float.

If the motion of the float is to be symmetrical, the forces on the float during upstroke and downstroke should be equal.

Let

H = available head
p = density of water
A = area of float
Y = draught of float at equilibrium

$M = YpA$

The maximum net forces on the float occur a positions 1 and 4 of figure 4.1.

Net force on float = Buoyancy Force - Weight of Float

At 1 (Figure 4.1)
Maximum Force (upwards) = $(Y + H)\ pgA - Mg = HPgA$ [2]

At 4 (Figure 4.1)
Maximum Force = $(Y - H)\ pgA - Mg = -HpgA$ [3]

To obtain these forces the float freeboard at equilibrium must be at least equal to H and the submergence Y must also be at least equal to H to prevent water falling below the float. Hence the minimum height of the float is 2H and it should be designed to float at mid height. This is achieved on the laboratory models by water ballast inside the float.

At positions 3 and 6 of Figure 4.1 the net force on the float is zero. Assuming the force on the float decreases linearly, a force diagram (Figure 4.2A) is obtained.

Figure 4.2B shows the arrangement of the rams. For the top and bottom pairs to be horizontal at each end of the stroke the vertical separation of the pairs must be H.

Hence the resistance of the rams to the float is

$$2\ Pa\ \sin\theta$$

where

P = fluid pressure
a = area of a ram

and

$$\theta = \tan^{-1}\frac{2(H - z)}{X}$$

where X is the horizontal distance between ram pivot points.

If θ_{max} is less than 45° then $Pa \sin\theta$ varies approximately linearly with z, a similar result is obtained when the float is moving downwards. A graph of the variation of force along the stroke for varying ratios of H/X is given in Figure 4.26. (H/X = 0 implies infinite horizontal separation when $2\ Pa \sin\theta$ is a straight line).

Thus by adjusting the pressure in the rams to the correct value for the head, the resistance felt by the float can be matched to the buoyancy and gravitational forces on it. This results in a constant float velocity for most of its travel with rapid acceleration at the start of a stroke when frictional forces are at a minimum.

4.2 The First Laboratory Model

The first model built in the hydraulics laboratory was intended to give basic data necessary to check the accuracy of the mathematical model of the AURWE which had been produced by Dr Bullock. It was hoped that the mathematical model be a useful tool to design installations for specific sites.

Other objectives of this model were to investigate the optimum gate size, the effect of varying the pressure in the rams and the effect of a piped water supply to and from the chamber.

The machine was small, float size 500mm x 500mm, to enable it to be made quickly and to make alterations and adjustments easy.

Construction Details

Construction

Power take off rams	4 off 40mm diameter bore, 22mm diameter piston rod, single acting trunnion mounted. Each with two internal check valves. The power produced was dissipated by a throttle valve mounted in the circuit see figure 4.4.
Float construction	Wooden frame with 12mm pvc sides - 500mm x 500mm x 1200mm.
Chamber construction	19mm thick clear perspex and grey pvc - 510mm x 510mm.

Gates	Vertical sluice type, constructed of PVC plate with stiffeners and operated by externally powered pneumatic cylinders - 500mm wide - variable vertical opening.
Power take off frame	Scaffold structure, which allowed the ram geometry to be adjusted.
Supply tanks	2 off fibre glass, maintained at different levels by side weirs.

<u>Instrumentation</u>

Water levels	Measured both with scales and capacitance gauges.
Pressure Transducers	Three Druck PC 10a, 350mb transducers attached to the float base.
Position Transducers	The float and gate were fitted with rotating potentiometer position transducers.
Oil Pressure Gauge	Oil pressure in the hydraulic circuit was measured with a 0 to 100 psi gauge.
Strain Gauges	Mounted on the float mast to measure force transmitted to the power take off rams.

Computer Controller	All the instrumentation was connected to the laboratory's Texas 960 A process control computer. This operated the machine activating the gates with respect to float position.

See Figure 4.3 for a general view of the model

See Figure 4.4 for schematic of the hydraulic system.

4.2.3 Results from the Model

The model and instrumentation were found to have a number of shortcomings. The more important of these were:

a) The power take-off rams were of too large a diameter for the float area. This meant a low working pressure - which was predicted - but also a disproportionate amount of seal friction occurred which was difficult to quantify.

b) The strain gauges on the mast suffered from signal noise which could not be separated from the small strain signal.

c) The three pressure transducers in the float base each gave a different reading due to the pressure distribution on the base. Hence only a qualitative assessment of mean pressure could be made.

d) The float was not guided at its base, hence during tests with gate overlap (upstream and downstream gates open at the same time) jamming occurred due to the lateral forces.

Despite the above, however, a number of important conclusions were obtained from the results

1) Output power increased with gate opening up to a maximum at about 300mm.

3) The sudden closure of the gates caused water hammer in the inlet pipe.

The mathematical model was in close agreement with the physical model in predicting periods and water flows. It was unstable however for certain input conditions and required a large quantity of empirical derived data to calibrate it. The data from the physical model was not accurate enough to enable this to be done.

Maximum power obtained from the model was 105 watts based on the pressure signals from the base of the model. Maximum efficiency was 80%; efficiency at maximum power was 68%. These efficiencies are based on the displaced water volume of the float, they do not include any leakage or gate change over losses.

4.3 The Second Laboratory Model

The second laboratory model was designed using the basic information obtained from the first model. This enabled gate size, ram size and float guidance methods to be identified as important to the success of the device.

A machine with a power output of about one kilowatt was the largest that could be accommodated within the physical space and water flow available in the hydraulics laboratory. This was a reasonable power output to enable accurate results for the efficiency of the device and its component parts to be obtained.

A considerable effort was made to make this machine a realistic example of a possible production model. Concurrent with the laboratory work an investigation was carried out to find a suitable site for a prototype machine and to consider the most economic means of installation.

The first model had shown that a machine whose gates opened directly to upstream and downstream water gave a better power output than one with a piped inlet and exit flow. It was considered, however, that the overall installation costs and maintenance would be lessened if the machine could be built in the bank at the side of a weir with inlet and exit pipes above and below the weir. These pipes should be as short as possible to minimise the inertia of the water contained in them.

The configuration of the model and an artists impression of a bankside prototype are shown in figures 4.5 and 4.6.

4.3.1 Design Details

The design philosophy was to produce a machine with low maintenance and long life using standard components where possible.

Chamber and Float

These were made from pieces of standard diameter glass fibre reinforced plastic pipe supplied by Redland Limited. The pipe was of glass and resin construction only. The alternative type of glass fibre pipe with a sand and resin core was rejected as being heavier and therefore adding to installation costs.

Concrete pipe was rejected as being too heavy and difficult to modify. Armco type steel pipe was available in the required sizes but was not considered to be as durable as glass fibre.

The base of the chamber was made as a separate piece to aid assembly. Two stub pipes for water entry and exit were bonded to the base at 90° to one another.

Gates

The first laboratory model had gates worked by an external air supply. On the second model however the gates were to be powered by the engine itself - hence a design that used little power but gave a good flow area was required.

The design chosen was the circular sleeve gate shown in figure 4.7. This consists of a length of glass fibre pipe of the same diameter as the float, stiffened internally with steel ribs and suspended from a central post by radial spokes attached to the ribs. The sleeve has two cut out sections which allow water into and out of the chamber when the gate is rotated to align one of the cut outs with the supply or exit pipe.

Bearings and Support Structure

A central stub bolted to the base of the machine carries the lateral forces on the sleeve valve. The valve rotates on plain, water lubricated bearings of Nylatron (molybdenum filled nylon) on a stainless steel sleeve fitted over the stub.

A flexible coupling of the rubber tyre type connects the valve to the central actuator post which serves both to turn the valve and act as a guide for the float.

The float bearings are of Nylatron. These are split axially to allow removal for examination without dismantling the machine.

The inner ends of the power take off rams are mounted on a post pivoted at the top of the float to compensate for any misalignment.

Hydraulic System and Control

The hydraulic power take off system is shown in figure 4.8.

The four rams are single acting with self aligning eyes at both ends. These are connected by standard 3/4" flexible hoses to the static elements of the system.

A spool valve, actuated by float movement transfers high pressure oil to the gate cylinders to turn the gate. Typically 5% of the total high pressure oil is used to work the gate.

Initially this spool valve was mechanically operated with adjusters to alter the point at which it operated relative to the float rise and fall. Later, to speed adjustment, it was replaced with a solenoid valve actuated by reed switches triggered by the float.

The accumulator smooths out the flow of oil to the swash plate motor driving the AC generator the output of which is dissipated in a bank of light bulbs. Control of pressure in the circuit is by a combination of the load on the generator and adjustment of the angle of the swash plate of the hydraulic motor. A low swash plate angle demands less oil but at a higher pressure - thus the resistance to the movement of the float can be controlled.

To minimise flow losses in the hydraulic circuit feed pipes of 1" nominal bore are used with delivery pipes and fittings of 3/4" nominal bore.

Despite the large delivery pipes problems were initially experienced with dissolved air boiling out of the oil at the sub-atmospheric pressures in the rams at the start of the suction stroke. Typically hydraulic oil can contain up to 10% by volume of dissolved air at atmospheric pressure. If any of this comes out of solution, the volumetric efficiency of the rams and hence the efficiency of the whole machine is reduced.

This problem was overcome by sealing the low pressure reservoir and inserting an air bag into the space above the oil. This bag is inflated to 0.25 bar thus effectively pressurising the feed circuit and preventing sub-atmospheric pressures occurring.

The alternative to this is to have larger feed pipes and a reservoir at a high level - say 3 metres above the machine. The pressurised bladder is a more practical solution for a prototype machine.

The hand pump in the circuit is used to power the gate actuator when starting up.

4.3.2 Alternative Hydraulic Fluids

The fluid in the hydraulic system serves three purposes; it transmits power, lubricates and cools working parts.

Mineral based oil is the fluid generally used in industrial systems. It is widely available and performs well.

A consideration with the water engine is that some oil may leak into the water course and pollute it. Hence alternatives fluids were examined.

The most common alternatives are water based fluids. These are used where there is a risk of fire - such as in mines or warships. Additives are mixed with the water to enhance its properties usually in small quantities - some as low as 5% by volume. The disadvantages however are that the additives are expensive, not always availabe in remote parts and usually toxic. Furthermore special fittings, generally free of copper alloys, have to be used, and moving parts - such as ram seals and motor bearings often have a shorter life when using water based fluids than when using mineral oil.

Recently extravagant claims have been made for components that will operate using pure water as a fluid. These items are presently extremely expensive and have no track record.

For the above reasons it was decided to use standard mineral oil in the AURWE.

4.4 Major Modifications to the Initial Design

Gate

The sleeve gate was designed initially so that at no time during its operation were the inlet and outlet ports open to the chamber at the same time.

Increasing the size of the cut-outs in the sleeve introduced a sector in the arc of rotation where inlet and outlet were open together, called the overlap.* This was found to reduce the delay caused by the water's inertia within the dynamic system. The effect on model performance was a decrease in period, a substantial increase in power, reduced gate rotation and a slight reduction in efficiency because of the loss of water during overlap.

Testing on the sleeve gate ended in December 1982 and a new gate was installed known as the shield gate. This consisted of a rolled steel plate, 900mm square and fitted into the annular space between the chamber and the float. Its manner of operation was similar to the sleeve gates except that it relied on an oscillating blockage rather than an oscillating hole to control water flow.

The construction of this gate allowed the float base to travel 0.26m below the depth of the inlet and outlet pipes, thus reducing the depth of construction of a prototype with consequent cost savings. This had four main effects on the machine.

a) The float was subject to direct lateral forces from the inflow and outflow. In the first model this caused jamming of the float. The second model, with a round instead of a square float and with better bearings did not suffer from this.

* In oil hydraulics the term "underlap" is used to describe the condition when two ports in a valve are open together in this way.

b) The overlap of the float reduced the effective area of each port at the beginning of its upward stroke and at the end of its downward stroke. However, no reduction in power output was noted.

c) The construction of the new valve was inherently less stiff than the sleeve valve, hence clearances were greater to allow for flexing. Leakage was between 50 and 70 l/s compared with 35 to 40 l/s for the sleeve. It would be possible to reduce the leakage substantially in a prototype version by careful design.

d) The amount of overlap was greater, ie. there was a larger sector in the valve's arc of rotation where both inlet and outlet were open simultaneously. This caused losses of between 30 and 50 l/s according to gate speed.

Figure 4.9A shows the sleeve and shield gate.
Figure 4.9B illustrates the effect of increased overlap.

Throttle around the Float

Previous tests had allowed an unrestricted flow of water within the annulus, consequently the water moved rapidly up the annulus at the beginning of each half cycle. At high operating pressures in the hydraulic system the float was slow to respond to buoyancy forces and the kinetic energy of the water in the annulus caused it to overshoot its equilibrium position and overtop the float. A throttle consisting of a 150mm deep strip of foam filled hairlock (glued horsehair) was installed around the circumference. This substantially reduced the water level fluctuation in the annulus. This was tested on the model using a heavily ballasted float, so that when the float was in its lower equilibrium position the top of the float was 0.2m below the upstream level, at no time during the water engine's normal cycle did the water within the annulus come within 0.3m of the top of the float with the throttle in place.

Power output was slightly increased by the throttle, this was thought to be due to the marginally increased area presented to the water. With the throttle the float became subjected to a combination of buoyancy and pressure forces instead of just a buoyancy force.

Thus the throttle allows the float to be shorter than twice the head without loss of power or effiency. This means that the machine can be smaller and hence cheaper.

Summary of Test Results

A more complete set of results is given in reference 29. The main parameters affecting performance are noted below.

Operating Pressure

Two important operating conditions were identified, these being a point of maximum power and a point of maximum efficiency. These points were not sensitive to small changes in pressure. In general, a 15% change in pressure about the optimum caused a 2.5% reduction in power output. Overall efficiency (net oil power/total water consumption) was approximately 45% at maximum power and 50% at maximum efficiency (net oil power is total oil power less that taken to turn the water control valve). This efficiency includes leakage and other water losses.

Ram Geometry

Figure 4.10 shows the effect of vertical ram separation (Y) on gross oil power, net oil power and conversion efficiency. the tests indicate an optimum vertical ram separation of 1.05 metres when operating at a head of 0.85m. A further series of tests investigated the effect of altering the horizontal ram, separation (X). It was expected that there would be some improvement in output power, as X increased due to an improved profile of ram resistance. This effect was not observed in the model.

Float Stroke

With the solenoid operated spool valve activating .the gate it was possible to control float travel to less than the head. A series of tests were carried out with float travel of between 50% and 100% of the head.

From these tests a number of points were noted.

1. The shorter the stroke the higher the stall pressure.

2. The shorter the stroke the shorter the period and hence the greater the number of gate operations in a given time. This reduces both the net oil power available to the hydraulic motor and the overall efficiency.

Gate Speed

The speed of the gate changeover is dependent on its own moment of inertia, inertia of any water it disturbs, the distance to be moved and the force acting on it. If speed is increased then a larger percentage of the gross oil production will be taken, which may reduce the net efficiency if there is no accompanying increase in gross output. The times to move the gate varied from 0.5 seconds to 1.0 seconds.

No consistent increase in power was noted due to faster gate speed.

Conclusions

The model had a maximum efficiency of 50%. This could probably be improved on a prototype by careful design of the gate system to reduce leakage.

The model demonstrated that it was possible to adjust the operating pressure, gate speed and float stroke to maintain this efficiency at times of reduced flow.

4.5 The Third Laboratory Model - The First Twin Chamber AURWE

Introduction

Investigation into prototype costs showed that the civil works were a large proportion of the total installation costs. Costing carried out on the machine itself revealed that a concrete or steel chamber with steel floats would be cheaper than glass fibre reinforced pipes. It was also necessary to reduce the number of parts on the machine itself to keep down manufacturing costs.

The twin chamber machine, shown in outline in figure 4.11, offers advantages over the single chamber machine. These advantages are:-

1) Reduction in the number of power take off rams.

2) Reduction in the number of water control gates.

3) The water feed to each chamber would reduce inertia changes in the feed channel.

4) Improved operational characteristics if the stage of the river changes. With a single chamber machine changes in operating water levels require changes to the ram positions to obtain best efficiency. With a linked twin chamber machine, because the floats are balanced, one against the other, changes in water level do not affect the working stroke.

5) For a given power output the component parts are generally smaller thus easing site assembly.

4.5.1 Design Details

The design chosen for the third model is shown in figure 4.12.

The chambers and floats are fabricated from mild steel sheet painted with chlorinated rubber paint to resist corrosion. The chambers are each 1.25m square internally with a centreline separation of 2.3m. The floats are 1.2m square by 1.8 metres high. Each float has a central tube to which is fitted Nylatron bearings. These bearings guide the float on a central post. Small wheels are fitted at two corners of each float to prevent rotation.

To synchronise the floats a large, double acting ram is fitted inside each central post. The piston rod of each ram is attached by a mast, equal in length to the stroke of the float, to each float. These two rams are piped together in such a way that as one float rises the other float is forced by the pressure transmitted to the other ram to fall.

These rams also transmit the forces from the falling float to the power take off rams on the rising float. The configuration chosen allows the machine to be built very low in the water being very little higher than the upstream water level, thus giving minimum blockage at times of flood.

A single large butterfly gate sited between the chambers at the base directs water into either one or the other of the chambers. This gate is mechanically simple and gives a reasonable flow path into and out of the chamber. The floats can also be allowed to pass below the top of the gate so reducing the construction height.

Set against these advantages is a large operating torque. A trial gate was tested between two tanks. This gate required a torque of 540Nm to operate at a one metre head.

A view of the finished model is shown in figure 4.13.

4.5.2 Test Results and Problems

The central balancing rams

There were a number of problems associated with these. It was decided for the reasons stated below that linking the floats hydraulically was unsatisfactory.

1) Air in the system - Initial filling of the hydraulic circuit was difficult and purging the air from the system was slow and laborious.

2) Air entrainment - This occurred during running. Air entered through the nose seal at the front end of the cylinders on the suction strokes. This was prevented by providing a reservoir of oil around the seal.

3) To reduce the effect of residual air in the system and to prevent air being entrained the balance ram circuit was pressurised. This was done by connecting a small line from the power take-off circuit to the front side of the balancing rams. This resulted in a decrease in output power due to increased friction on the ram seals.

4) <u>Creep</u> - It was noted that the floats tended to creep together. No leaking joints were noted, and it was deduced that this was due to leakage across the piston seals and the end seals. Creep rates were:

   0.5mm per hour - when the AURWE was stopped
   15mm per hour - when the AURWE was operating.

<u>The Butterfly Gate</u>

A typical set of test results using this gate are given in figure 4.14. The power required to operate the gate was considerable as can be seen from the difference between gross oil output and nett oil power, approximately 32% of total oil power produced by the engine.

At higher oil pressures the gate moved noticeably faster giving the engine a shorter cycle time and a better power output and efficiency due to lower losses of water during gate operation.

Leakage across the gate when closed was measured at 50 l/sec. Most of this was through the clearance spaces at the top and bottom of the gate.

4.5.3 Semi Circular Gates

To reduce the power absorbed by the gates and to reduce leakage, the twin vertical axis circular segment gates shown in figure 4.15 were fitted. Power consumption of these gates was about 6% of the power produced by the engine.

The table below compares the measured performance of the twin machine with the expected performance obtained by scaling the results of the single circular machine using the Froude scaling laws.

Head	Expected Power	Butterfly Gate		Circular Gate	
m	(Gross) Watts	Gross Pwr	Net Pwr	Gross Pwr	Net Pwr
0.6	1293	963	618	702	650
0.7	1630	1411	979	935	870
0.75	1807	--	-	1280	1200
1.16	3477	--	-	2248	2163

Gross powers for both gate types were less than expected. This was due to:

a) Poor internal hydraulics, the water has a tortuous passage to enter the float chambers. The butterfly gates gave a better flow than the circular gates and hence a better gross power.

b) Extra losses in the balancing ram system. Measured at about 10% of gross power.

c) Poorer chamber hydraulics, square chambers rather than circular.

Vertical Ram Tests

The mathematical model predicted that simple vertical mounted rams would produce lower power than the more complex pivoting ram arrangements. This model allowed the prediction to be tested by altering the hydraulic circuit, disconnecting the power take-off rams and piping the balancing rams into the main hydraulic system. This converted the machine into a twin chamber AURWE with unlinked floats and vertical power take-off rams. A few short tests were carried out and the results confirmed the predictions of the mathematical model of reduced power output, reduced stroke and reduced overall efficiency.

Figure 4.16 shows the comparison between the power from the vertical rams and that from the inclined rams at a head of 1.05m with semi-circular gates.

4.6 Model Studies of Gate Design

The results from the twin AURWE emphasised the importance of the gate design on the performance of the device. Three small model water engines were built from perspex sheet and tube to enable different gate configurations to be tested in the hydraulics laboratory flume.

One model was a twin circular chamber machine with gates between chambers, the second a square chamber machine with gates between chambers, the third a square chamber machine with gates below the chambers. Each could be fitted with a number of gates of different design. Gate area to float chamber area ratio of 1:4 was used for most tests. For a few tests this was increased to 1:3.

A scale of 16.8:1 was used, this being representative of a 3m diameter prototype at Adelphi Weir, Salford using 178mm bore perspex pipe for the model.

The square prototype was assumed to have 3.5mm square chambers and hence was modelled at 208mm square.

The head was 2m prototype scaled at 119mm.

The following points were considered when designing the gates tested:

i) Minimising flow losses
ii) Minimising the power to operate the gates
iii) Simple linkage mechanism
iv) Not prone to jamming
v) Ease of manufacture
vi) Ease of maintenance
vii) Minimising leakage

4.6.1 Testing

Tests were carried out both with and without a float. Gate movement was controlled manually and timed so that the water level in the chambers rose to upstream level and fell to downstream.

The most promising gate types are illustrated in figure 4.17 and listed below.

1. Horizontal axis circular segment gate in conjunction with a 15° ramp.

2. Horizontal axis circular segment gate with a vertical rubber sill.

3. Horizontal axis flat plate gate with a flat sill.

4. Twin horizontal axis butterfly gates, positioned close together.

5 Semi circular horizontal axis gate with a fillet across the chord.

6 Twin horizontal axis butterfly gates, with a wide separation.

7 Semi circular horizontal axis gate without a fillet.

8 Twin vertical axis segment gates centrally positioned.

9 Centrally positioned vertical axis butterfly gate.

4.6.2 Discussion of Results

The best gates with regard to the hydraulics and resistance to jamming were the horizontal axis butterfly gates with wide separation. Although these types of gates have large power requirements, it was decided to test the gates on the large laboratory model. The gate dimensions were deemed sufficiently different (being long and thin) from the vertical axis gates already tested to be worth a full investigation.

The horizontal axis circular segment gates tested had good hydraulic characteristics although they tended to jam. Jamming was caused by small particles which tended to be swept into the clearance between the chamber floor and the underside of the gate. Installing a sill between the floor and the gate reduced the frequency of jamming. This was an ideal gate from all other aspects, being good hydraulically, simple to construct and likely to have a low power requirement. It was felt that a moving sill, provided by a second smaller horizontal axis gate located below the main gate would prevent jamming.

Thus it was decided to build the second twin chamber laboratory model with twin butterfly gates on one chamber and twin horizontal axis circular segment gates on the other.

4.7 The Fourth Laboratory Model

This machine is shown in figure 4.18.

The changes compared to the previous machine are:-

1. Gates below the chambers - one twin butterfly gate, one twin circular segment gate.

2. Rocking beam pivoted between chambers on an adjustable frame. This allows the pivot point to be raised or lowered. The floats and power-take off rams are connected to each end of this beam.

3. Taper floats - base 1.195mm x 1.195mm, top 1.195mm x 1.135mm, height 1.6m. The taper is necessary to accommodate the tilting which occurs as the beam rocks. Nylon pads, fixed to the bottom corners of the floats guide them up the chamber walls.

4. A double acting ram to operate the gates on each chamber.

5. This machine has only two power take off rams compared to four on the previous model.

4.7.1 Power Output

The model was tested at a head of 0.6m. A typical power and efficiency curve is shown in figure 4.19.

The improved period of operation of this machine meant that the laboratory water supply could not sustain the 0.7 metre head used for the third model.

Froude scaling of these results to 0.7m head indicates that performance is 25% better than the previous model and similar to the single chamber machine scaled to the same conditions.

4.7.2 Comparison of Gates

By disconnecting one of the floats and closing off the gate circuit for that chamber it was possible to operate the machine as a single chamber AURWE. Hence it was possible to test each gate independently of the other. The redundant gate aperture on the upstream side was sealed off with a blanking plate to minimise leakage.

The results were as follows:

1) Static leakage through each gate system at 0.6m head

   - butterfly 24 l/sec
   - circular 45 l/sec

   The butterfly gates were fitted with rubber sheet sealing strips on their horizontal edges which account for the smaller leakage. This strip however was fragile and not suitable for a prototype machine.

2) Total loss, static plus changeover loss, through each gate system was approximately equal. This was because the circular gates moved faster than the butterfly gates. As the rams working each set of gates were of the same size, this indicated that the closing force required for the circular gates was less.

3) The butterfly gates moved more slowly at high heads than at low heads. The circular gate speed appeared unchanged with head.

4) Power output was the same for each gate system at 0.6m head.

Conclusion

It was concluded that the circular segment gates absorbed less power than the butterfly gates and that the power requirement was largely independent of head. Thus, provided the design can be improved to reduce static leakage, the circular segment gates are superior to butterfly gates.

4.7.3 Short Float Tests

Tests on the single chamber machine with a throttle around the float showed that provided water did not fall below the float base on the down stroke, allowing air beneath it, then the height could be less than twice the head.

Investigations into prototype costs showed that by shortening the machine in this way savings could be made on machinery costs, site preparation costs and installation costs.

Flexible throttle strips, shown in figure 4.20 were fitted around the base of each float. The hinge was made of rubberised canvas with nylon scraper strips. This throttle allowed water freely up the float but restricted the down-flow so giving a seal and preventing the ingress of air.

Tests were carried out on the effectiveness of this throttle by lowering the water levels in the upstream and downstream tanks by an amount equal to the head so that at the lowest position the float base was only just submerged. No air was entrained and power and efficiency were the same as previous tests.

Conclusion

A short float with suitable seals at the base gives the same results as a long float.

4.8 Prototype Work at Gibbons Mill

Appendix 1 contains details of possible prototype sites investigated in detail concurrent with the laboratory programme.

The machines proposed for each of these sites represents the state of the art at the time the investigation was carried out.

The Gibbons Mill site on the river Arun near to Billingshurst, West Sussex was offered as a prototype site in 1984. The owner of the site, Mr Peter Adorian, is a keen advocate of low head hydropower and he was willing to accept the inconvenience which a prototype machine would entail. The site owner also had "Miller's Rights" which meant that Water Authority consent was not required to abstract water.

The area was surveyed, hydrological data was obtained from the Southern Water Authority and a flow duration curve was produced for the river at Gibbons Mill. This information was then examined with a view to installing the fourth laboratory model or a modified version at the mill. The salient points considered were as follows:-

1. The available head at Gibbons Mill is approximately 2.0m. The laboratory model could stroke a maximum of 1.0m. It was not possible to adapt it to stroke the full head. However tests on previous models showed that running the AURWE at a head approximately equal to twice the stroke did not have a serious effect on power output, although efficiency was likely to be less than the maximum possible.

2 Calculated flow rate at optimum power of the laboratory model was 0.8m^3/second at 1.9m head.

3 From the flow duration curve the AUR would operate at full power:-

   i approximately 50% of the time - if all water were available.

   ii approximately 40% of the time allowing both for compensation flow and an existing turbine at the mill having preference for the first 0.6 cumecs.

4 The lower flow limit for the machine would be the 58 percentile exceedence flow.

5 Electrical output at 1.9m head would be approximately 5.0kW.

6 An AURWE machine using a short float would be more suitable in this location. This would reduce the excavation depth and formwork to shore the excavation sides. This was particularly important as the site owner intended to install the machine himself, using agricultural equipment.

   Maximum component weight was to be less than 1 tonne.

4.8.1 Modifications to the Laboratory Model

Modification work commenced in July 1984. The machine was delivered to site in late September when a lull in the normal agricultural work allowed time for the owner to carry out the installation.

Modifications were:-

1) New base chamber with circular sector gates - centre line separation distance increased.

2) Valve linkage modified to raise the operating rams above the tail water.

3) Larger power take off rams to increase the oil flow to the motor. These rams are sited between the chambers to reduce the overall machine size and to concentrate the loads at the centre of the structure.

4) A new beam fabricated.

5) The float masts extended and a large rod end self aligning bearing fitted at each float to beam junction.

Construction

The complete machine was delivered to site on a ten tonne truck in September. It was installed as shown in figure 4.21 in a specially excavated channel at the side of the mill in October 1984.

All work was carried out using an agricultural excavator with a backhoe.

The generator, hydraulic motor, reservoir and control systems were installed in a convenient room within Gibbons Mill itself, and connected to the AURWE by 10m of high pressure and low pressure tubing.

4.8.2 Modifications on Site to the Machine

During the first year of running the following improvements were made to the machine:

1) The beam was strengthened by welding a steel plate to the top section.

2) The lower gate was removed and replaced by a large rubber cill. This reduced the static leakage from 80 litres/second to 40 litres/second. It also reduced the complexity of the gates. Counterweights were fitted on the outside of the chambers to balance the moving gate.

3) Steel draught tubes fitted to the outlet ports. These were necessary to reduce the erosion occurring at the right hand bank and were successful in achieving this.

The configuration of the machine after these modifications is shown in figure 4.22.

4.8.3 The Hydraulic Circuit

The basic hydraulic circuit of the Gibbons Mill prototype is shown in figure 4.23. This is similar to the laboratory model circuit but with flow restrictors in the gate cylinder line to prevent too rapid movement of the gate and a solenoid operated shut off valve in the high pressure line to isolate the accumulators on shut down.

This circuit allows the machine to be controlled in the same manner as the laboratory model by adjusting the hydraulic motor swash plate angle thus altering the working pressure and hence the stroke and period of the floats.

An automatic control circuit was superimposed onto this system. This circuit received signals from pressure transducers registering the upstream and downstream water levels.

An electric motor controlled by these signals is used to alter the swash plate angle to maintain pre-set water levels. If insufficient water is available the system shuts the water engine down until such time as water levels return to normal.

This control system contains a dump load of air cooled resistors which absorb the generator output when the applied load is unsufficient. Thus the load is matched to generator output.

4.8.4 Gibbons Mill Prototype Results

A typical set of results taken in April 1985 is given below and figure 4.24.

The water flow was measured at the downstream weir.
The oil pressure by calibrated gauges.
The head by ranging poles.
The stroke of float and rams by direct measurement.
The electricity output by moving coil voltmeter and ammeter.

Test No.		1	2	3	4	5	6
Head	m	1.65	1.67	1.66	1.59	1.53	1.66
Weir flow	m^3/sec	0.66	0.6	0.55	0.71	0.83	0.79
Cycle times	sec	5.6	6.0	6.3	6.4	4.1	5.0
Oil pressure	bar	69.0	75.9	80.3	51.7	52.6	56.2
Float stroke	cm	71.0	66.0	57.5	95.0	58.0	83.0
Ram stroke	cm	32.6	30.3	26.6	44.8	26.6	38.6
Float displacement	m^3/sec	0.365	0.317	0.263	0.428	0.407	0.478
Gross oil power	kW	4.52	4.12	3.41	3.70	3.43	4.36
Nett oil power	kW	4.17	3.78	3.09	3.49	3.11	4.08
Float disp. power	kW	5.91	5.19	4.28	6.68	6.11	7.78
Total water power	kW	10.7	9.83	8.96	11.1	12.5	12.9
Electric power	kW	2.46	2.24	1.79	1.87	1.97	2.68
Efficiency water/wire		0.23	0.23	0.20	0.17	0.16	0.21
Efficiency - conversion		0.77	0.79	0.80	0.55	0.56	0.56

The gross oil power is the total volume displaced by the power take off rams at the system pressure.

$$\text{Gross oil power} = \frac{\text{Volume displaced per cycle x pressure}}{\text{Cycle time}}$$

The nett oil power is the gross power less that used in the rams operating the gates.

The float displacement power is the power in the water actually displaced in the chamber. The total flow at the weir is this flow plus leakage at gate change over plus leakage when the gates are closed.

The conversion efficiency is the ratio of gross oil power to float displacement power. This represents the maximum possible efficiency that could be achieved by the machine if water leakage were reduced to zero and the oil were used directly as a source of power rather than be converted to electricity.

The water to wire efficiency uses the total water flow and includes the losses in the motor and generator.

The conversion efficiency rises with oil pressure to a maximum of 0.80 at the highest test pressure of 80 bar. The lowest pressure of 52.6 bar also has the lowest efficiency of 0.56.

These two tests were at the same float stroke and operating head but, because of the pressures the cycle time is different. The slower operation giving the higher efficiency but lower power output.

This is significant in so much as that at times of low water flow the oil pressure can be raised to slow the machine but give better efficiency. When enough water is available the maximum power can be obtained by lowering the pressure to speed up the machine.

It is clear from these results that considerable gains in efficiency can be obtained by improving the design of the gating system. If only one gate is used to control inlet and outlet of water then there will be losses at change over. The alternative of two independent gates per chamber will improve efficiency at the expense of greater complication.

Significant power is wasted in the gate rams at the higher operating pressures when the fixed geometry of the gate system demands more oil than is required to turn the gates, the gate ram size being fixed at the lowest anticipated working pressure. Hence a system that adjusted the ram stroke to the working pressure would be beneficial.

Overall water to wire efficiency is reduced by both the generator and hydraulic motor working away from their design points. The efficiency of these two items together for the results shown is about 58%.

If both were near design point then an efficiency of about 72% would be obtained. This would raise the water to wire efficiency to a maximum of 28%.

Thus with a redesigned gate system, water to wire efficiency could be:- 0.8 conversion efficiency less 0.05 gate power times 0.72 oil to electricity efficiency = 0.54.

This figure is not unreasonable for any low head hydropower generator and certainly better than any of the esoteric machines described in Chapter 3.

The Gibbons Mill machine has demonstrated that an AURWE can be made sufficiently robust to survive floods and flotsam. A number of mechanical weaknesses have been revealed in the original design. These have been mainly bearings, subjected to alternating loadings, which have had a shorter life than anticipated. Larger bearings of more suitable materials will undoubtedly cure these problems.

The company, AUR Hydropower Limited, at the present time - January 1989 - are planning the installation of a 70kW machine. This has been fully costed by the company at a repetition cost of £133,000 or £1900 per kW installed.

4.9 Conclusions

The AUR Water Engine has a certain emotional appeal that other hydropower generators - apart from the water wheel do not have. The slow steady oscillation of the beam and the rise and fall of the floats is reminiscent of a nineteenth century steam engine. Perhaps, like the steam engine and the water wheel, the water engine will find application at sites where a visual presence is required in addition to the purely practical.

It is undoubtedly true that slow reciprocating machinery attracts a degree of care and attention from its user that high speed revolving machinery does not.

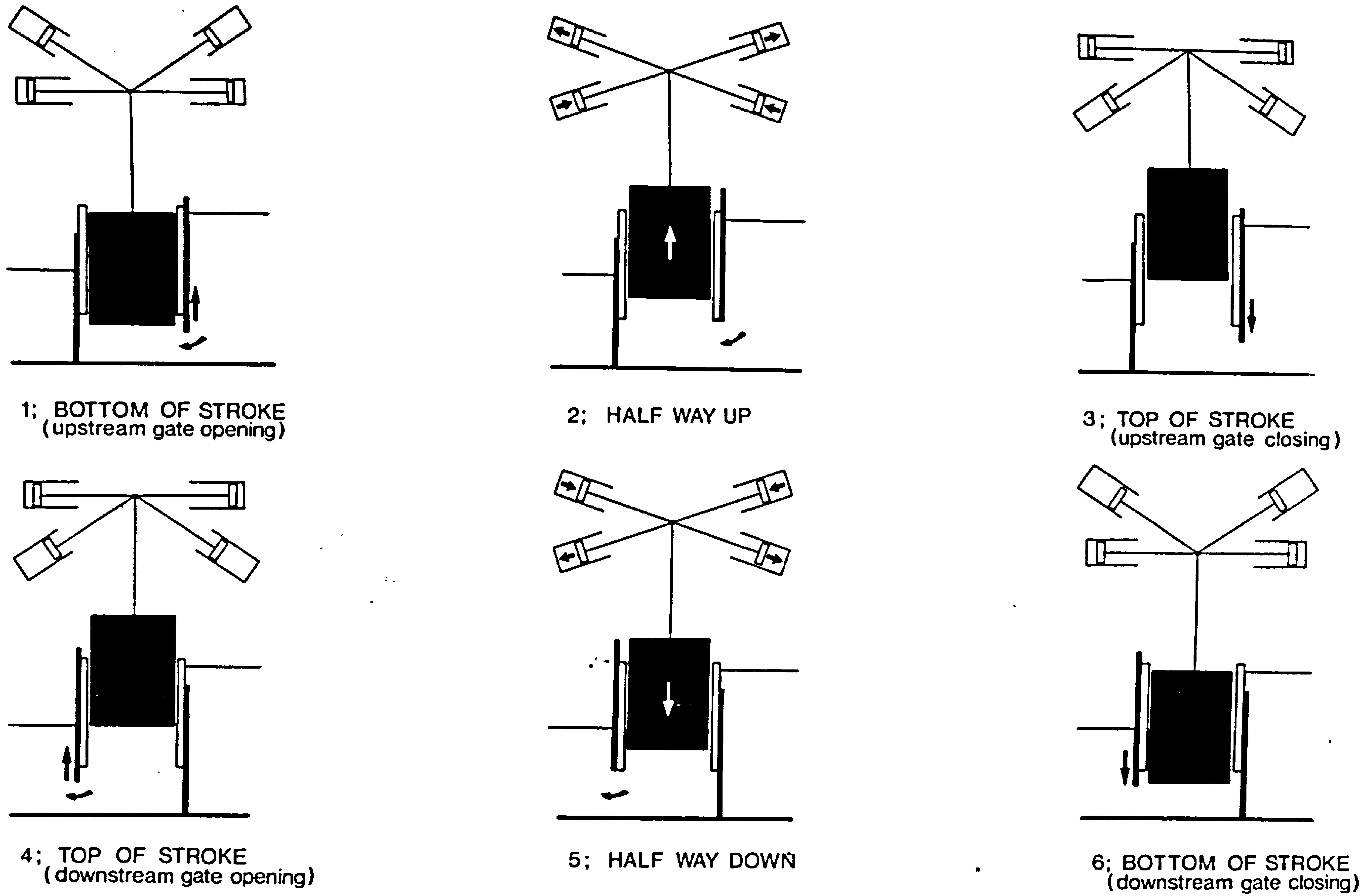
Recent examples of this observed by the writer were the lovingly polished diesel engines, often manufactured in the nineteen thirties preserved on the Sri Lankan tea estates. These machines, though rarely used, were in excellent condition. The water turbines, whether used or not, were invariably neglected and more often than not totally ignored.

Thus an argument may be offered that in developing countries, where failure of western technology due to local indifference is not uncommon, a machine with such a basic appeal may succeed where more bland equipment has failed.

Laboratory Model and Gibbons Mill Prototype, Summary of Construction Features

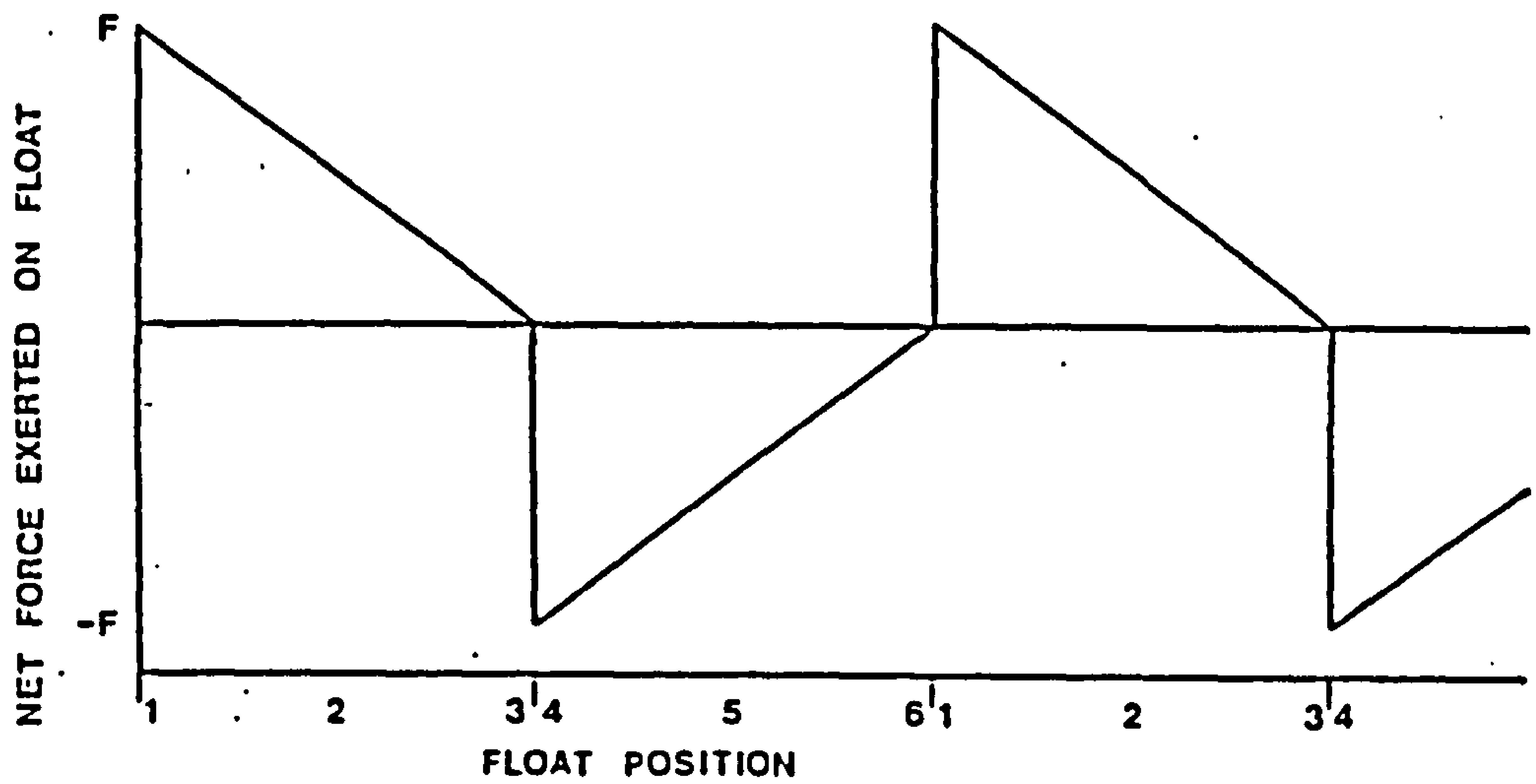
	First Laboratory Model	Second Laboratory Model	First Twin Chamber Model	Second Twin Chamber Model	Gibbons Mill Machine
Float chambers (m)	0.51 x 0.51	1.5 diameter	1.25 x 1.25 x 2	1.25 x 1.25 x 2	1.25 x 1.25 x 2
Float Area (m)	0.5 x 0.5	1.42 diameter	1.2 x 1.2 x 2	1.95 x 1.95 x 2	1.195 x 1.195 x 2
Float Height (m)	1.2	1.8	1.8	1.6	1.6
Construction	P.V.C.	G.R.P.	Steel	Steel	Steel
Gate type (1) (2)	2 off sluice	Sleeve then shield	Butterfly then 2 off segment type	2 off butterfly & 2 off segment	4 of segment then 2 off segment
Power take off rams (No)	4	4	4	2	2
Dia (mm)	40	50	50	50	80
Piston (mm)	22	28	28	28	56
Stroke (mm)	300	500	500	500	700
Gate Rams					
Type	Pneumatic	Hydraulic	Hydraulic	Hydraulic	Hydraulic
No.	2	2	2	2	2
Dia (mm)	63	40	28	28	40
Piston dia (mm)	12	22	12	12	28
Max. stroke (mm)	500	300	200	200	200
Linking Arrangement	-	-	2 off 50 mm dia. rams	Beam	Beam
Maximum working head (m)	0.5	1.0	1.0	0.7	1.9
Maximum discharge (m^3/s)	0.05	0.28	0.45	0.45	0.8
Gross oil power (kW)	-	1.21kW at 0.85m	1.4 at 0.7m	1.3 at 0.6m	7.2kW at 1.8m

Table 4.1



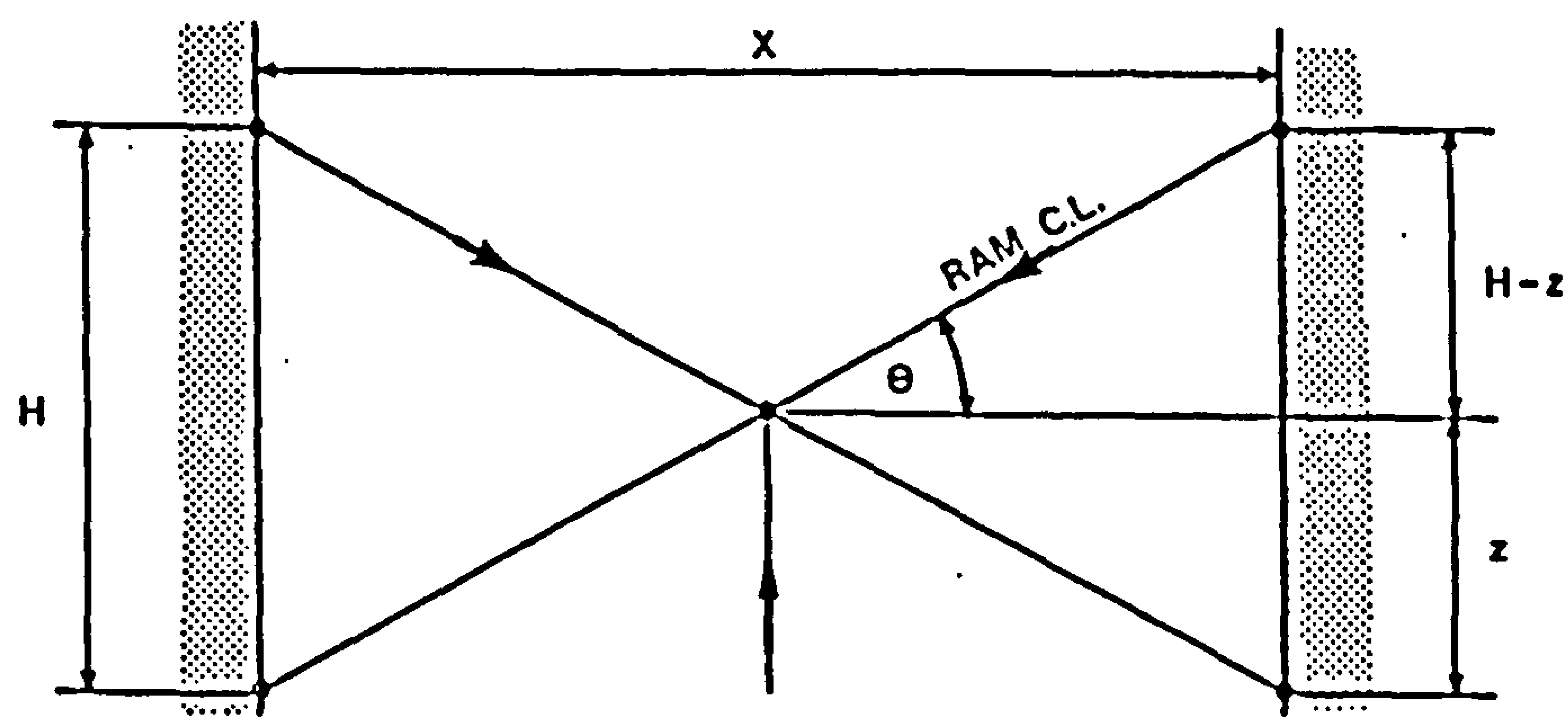
OPERATING CYCLE

Figure 4.1



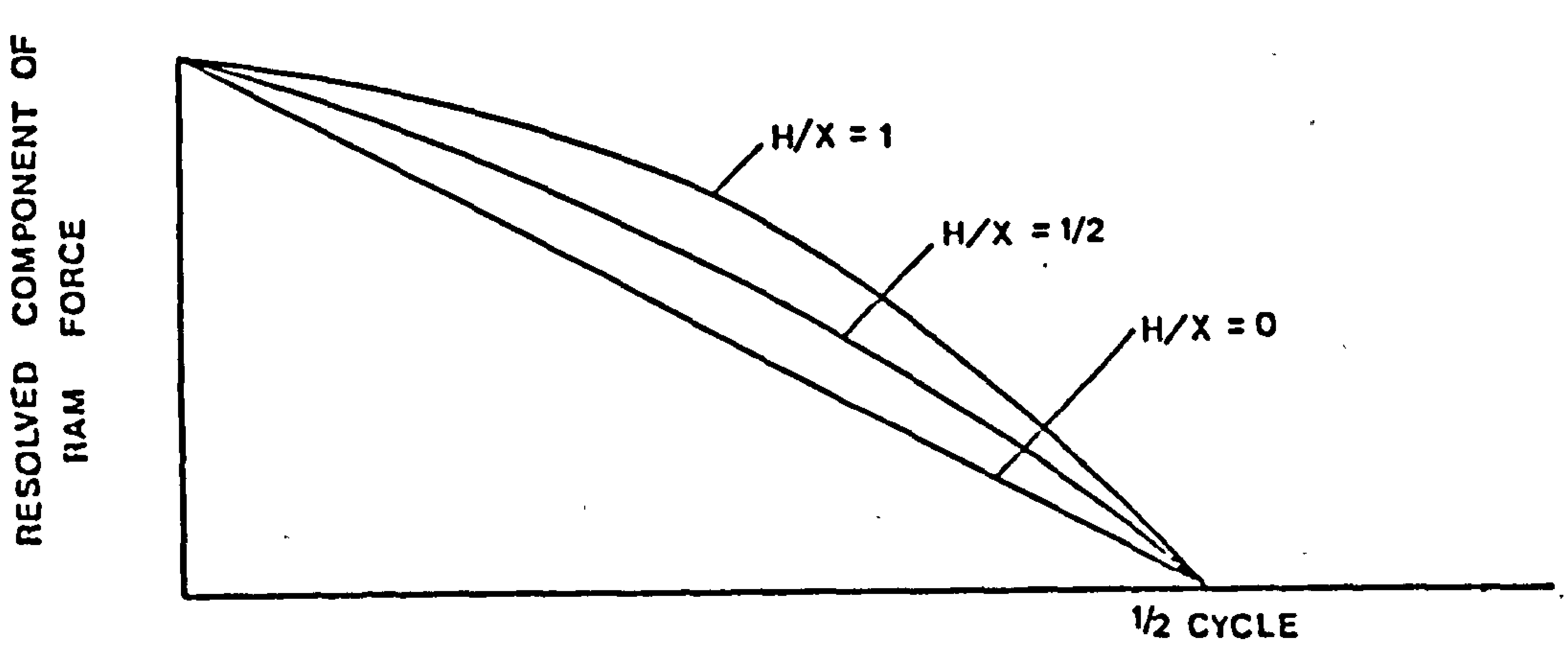
VARIATION OF NET FORCE EXERTED ON THE FLOAT WITH POSITION

Figure 4.2a



RAM GEOMETRY

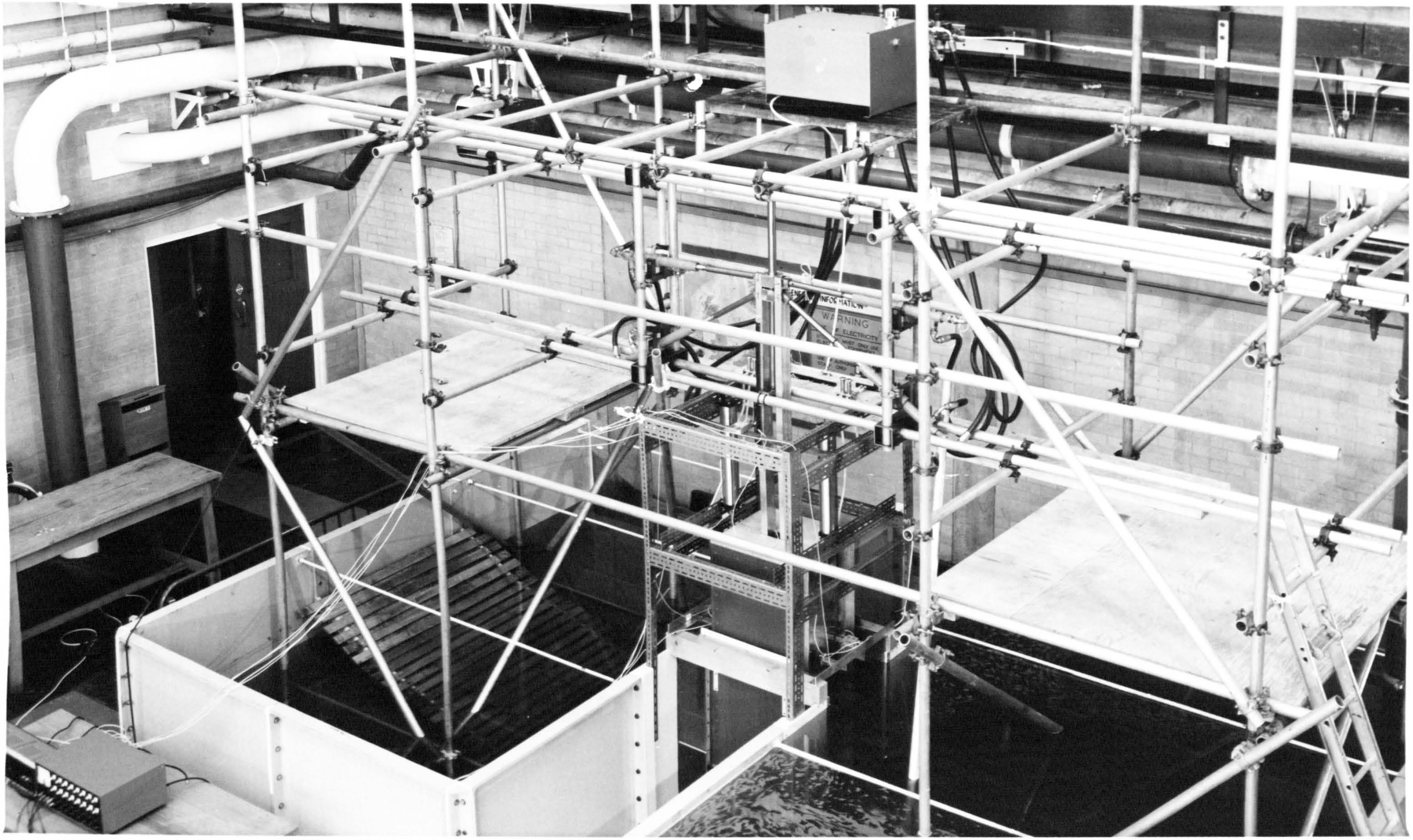
Figure 4.2b



VARIATION OF RAM FORCE WITH H/X OVER ONE HALF CYCLE

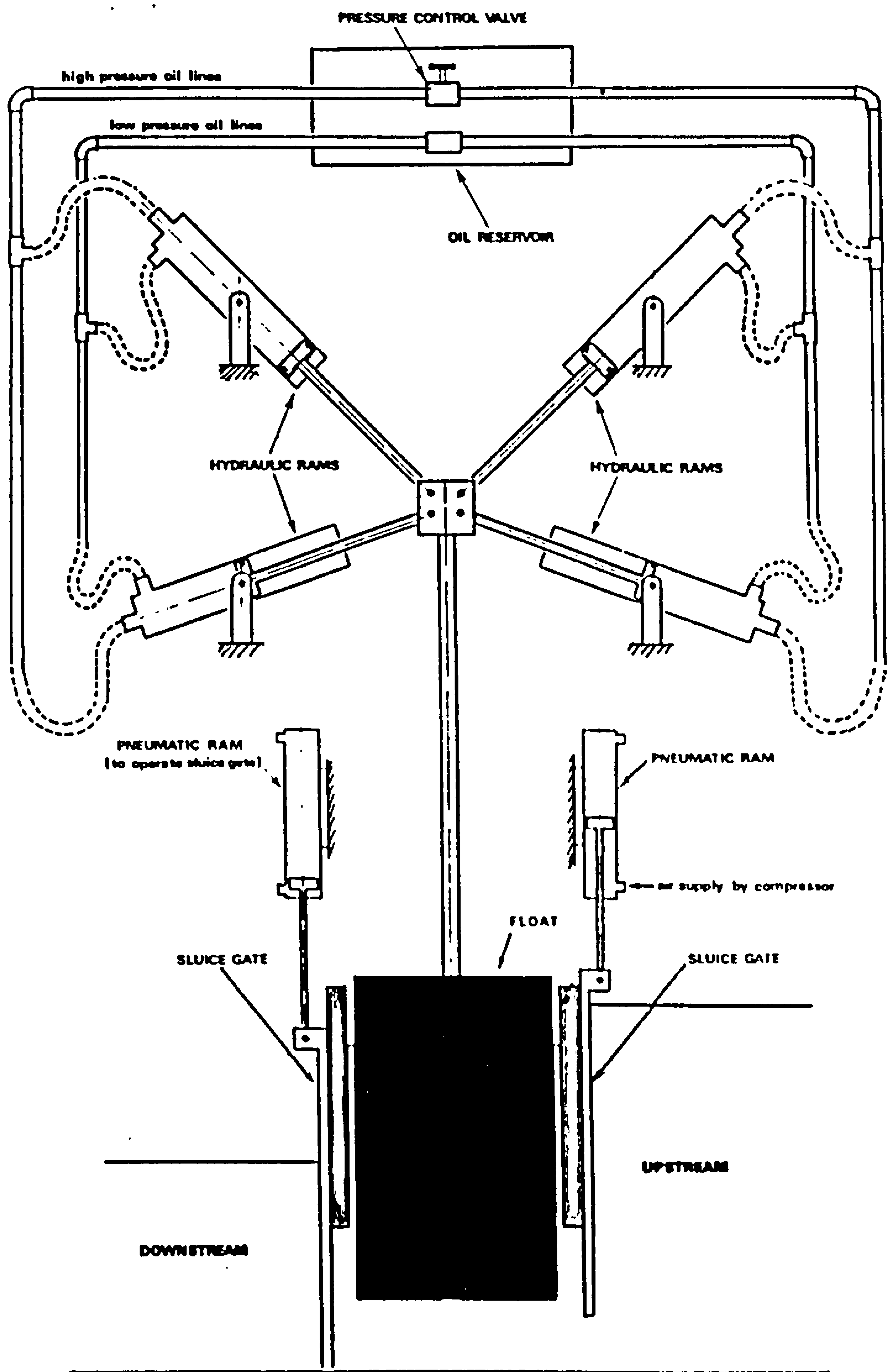
Figure 4.2c

Forces on the power take off rams



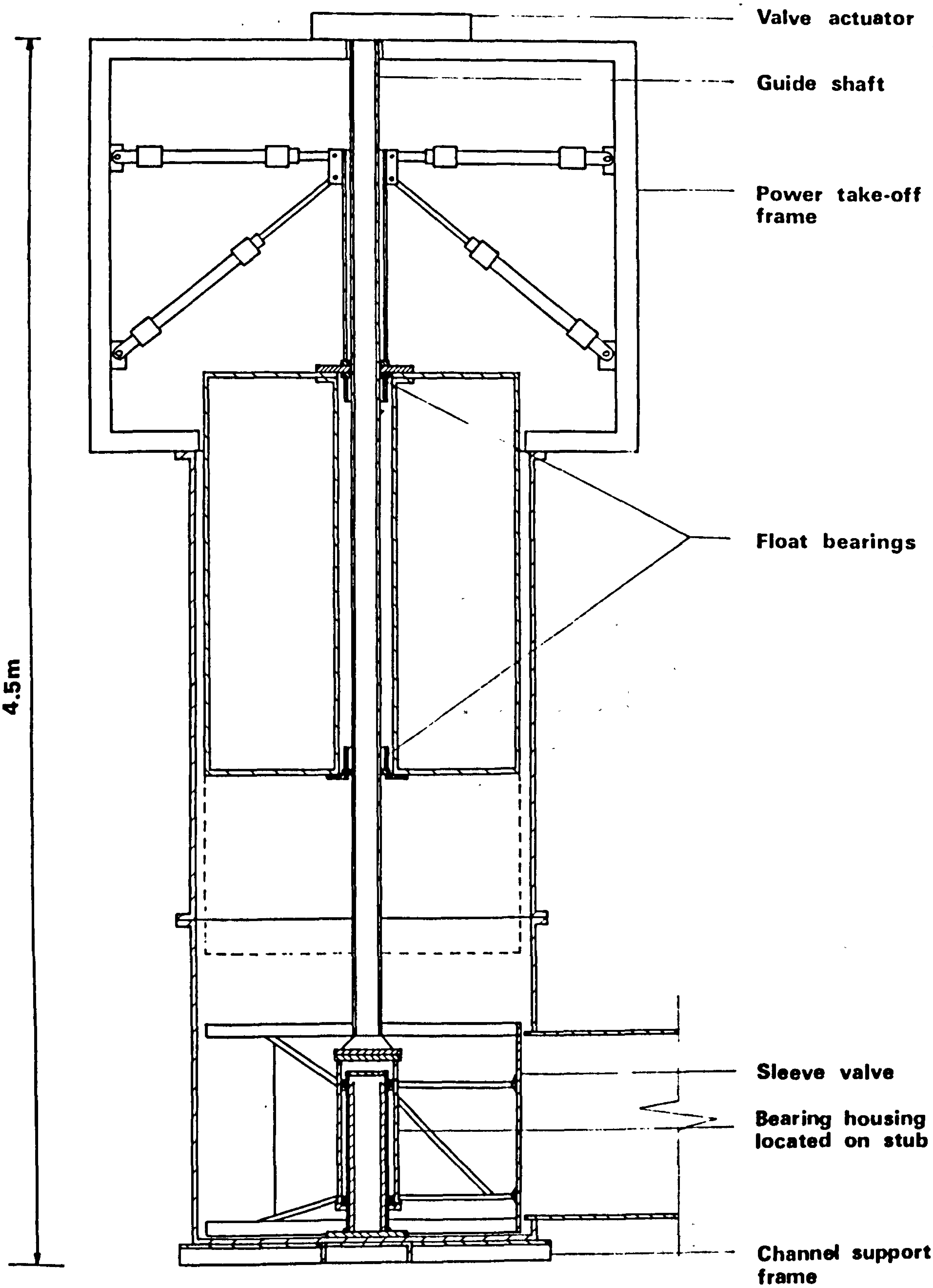
View of the first laboratory model AUR water engine

Figure 4.3



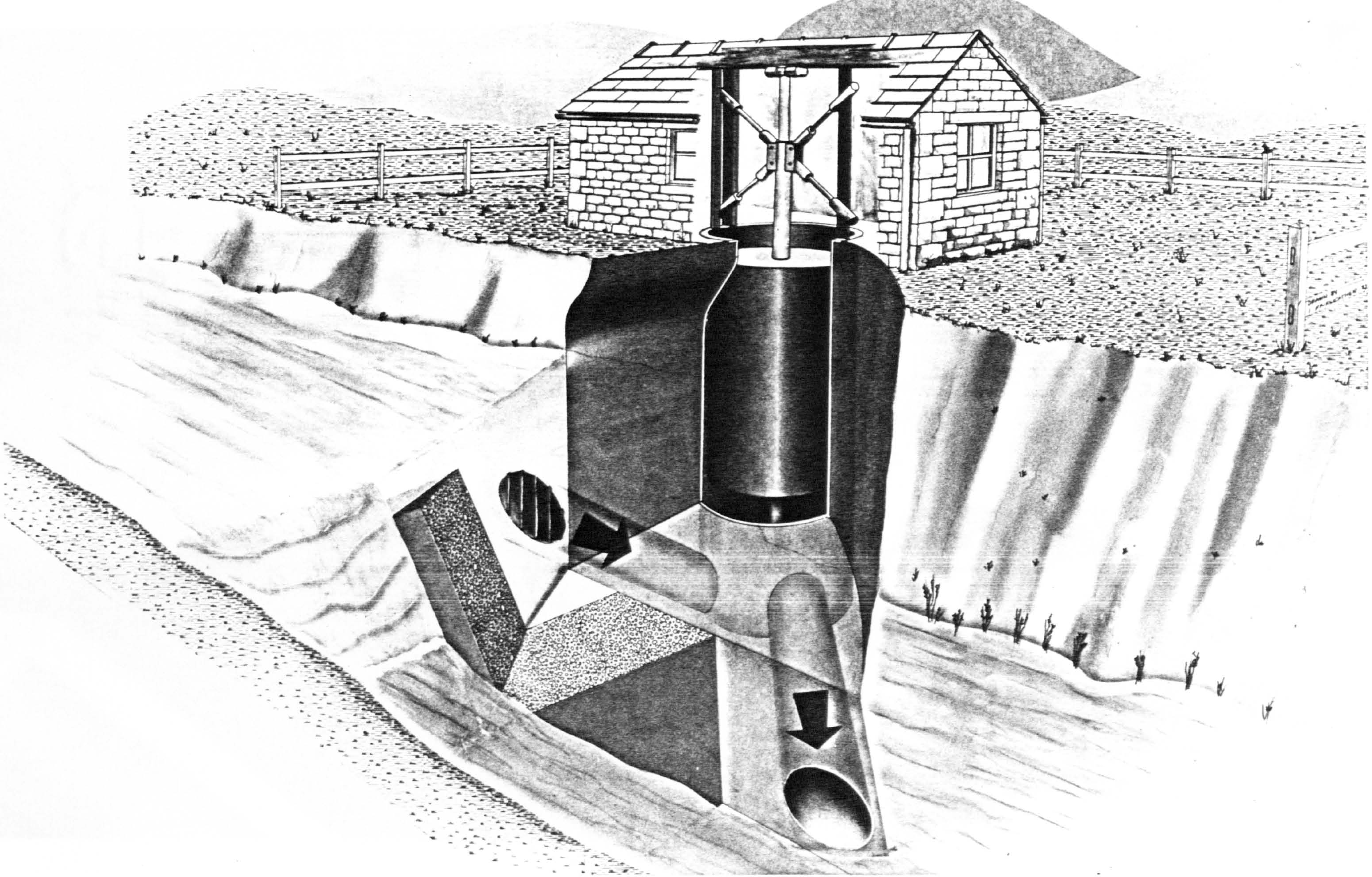
Hydraulic power take off system of
the first laboratory model water engine

Figure 4.4



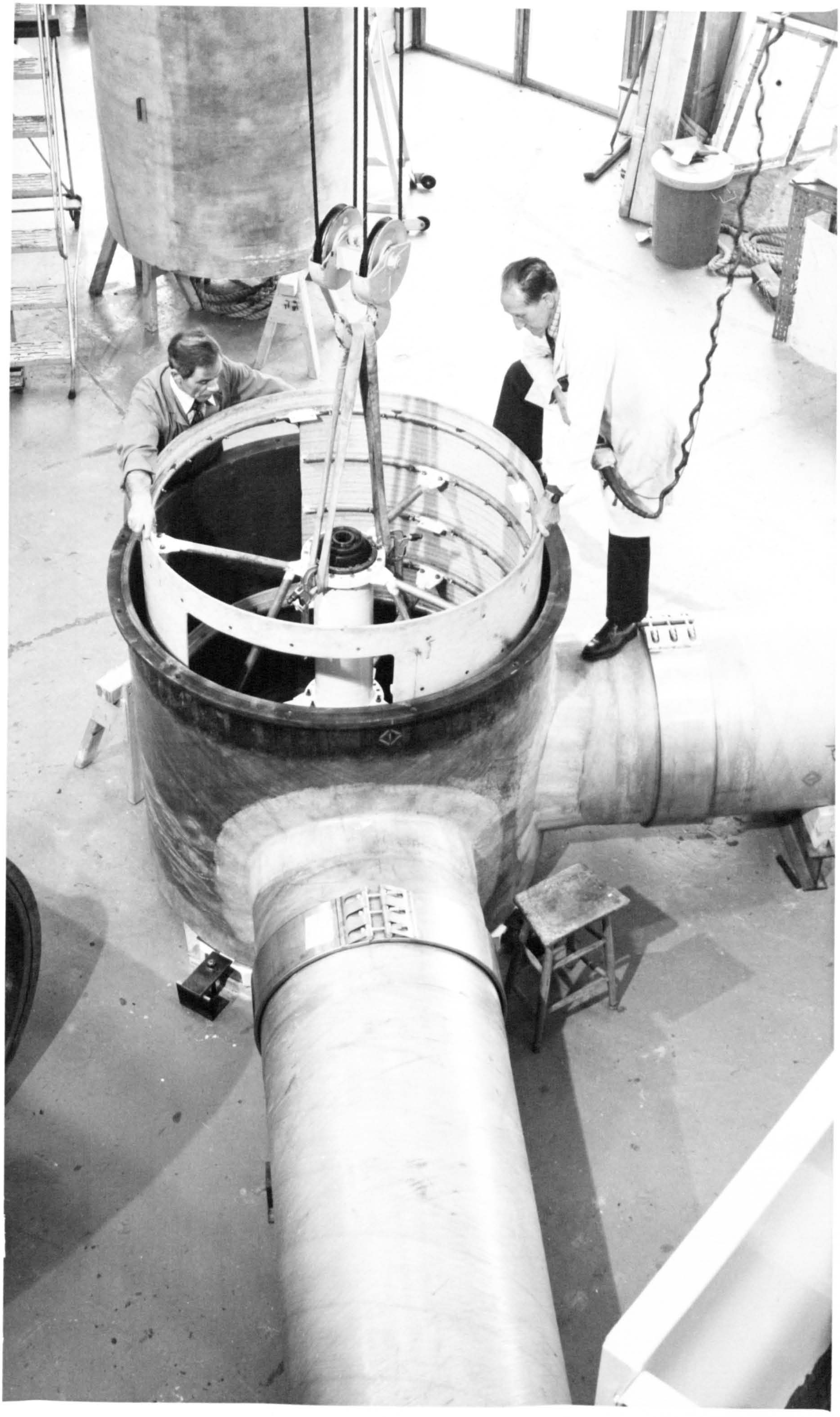
Arrangement of the second laboratory model

Figure 4.5



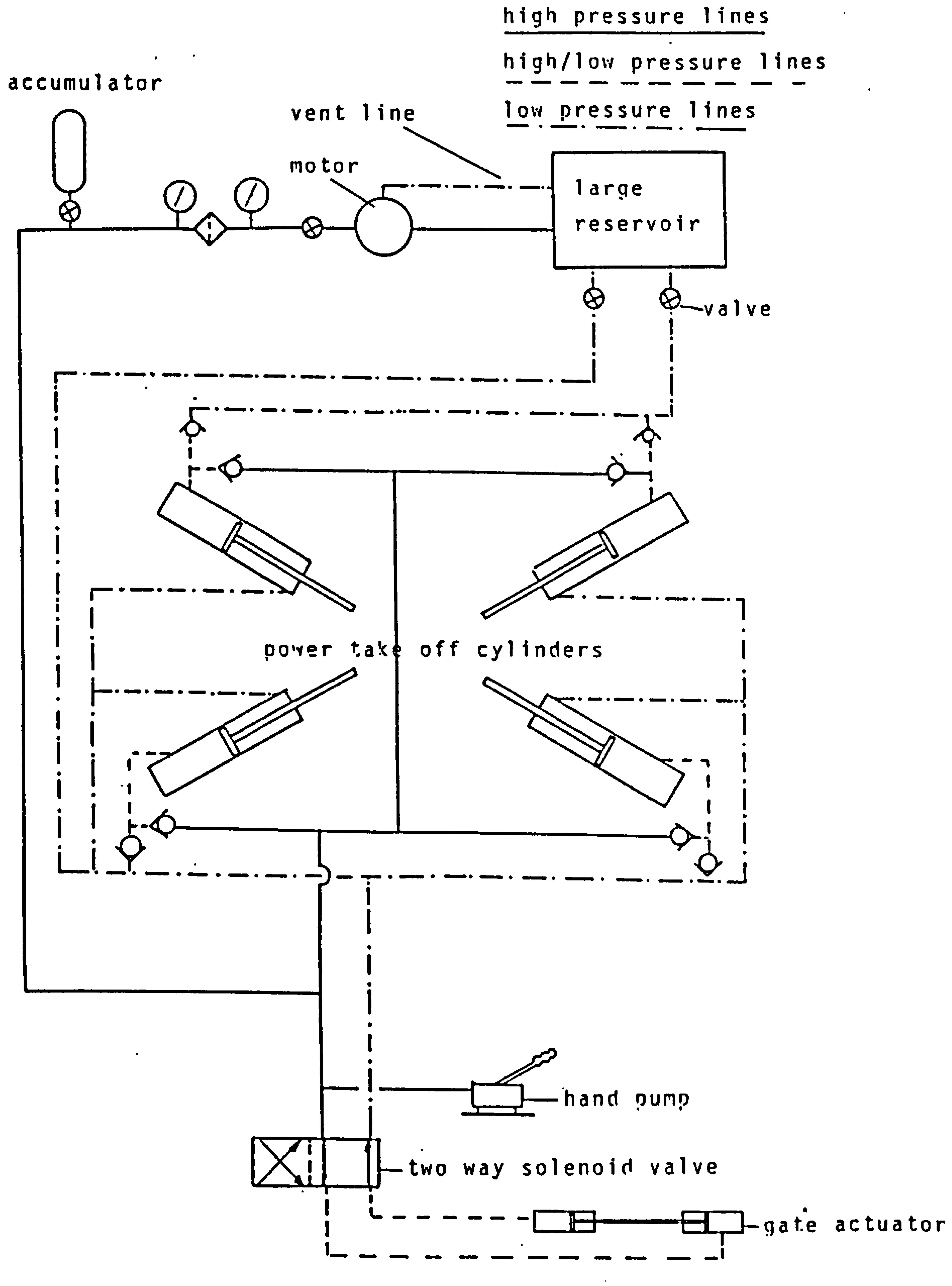
Proposed prototype water engine at Adelphi weir

Figure 4.6



The circular sleeve gate

Figure 4.7



Hydraulic power take-off system of the second laboratory model

Figure 4.8

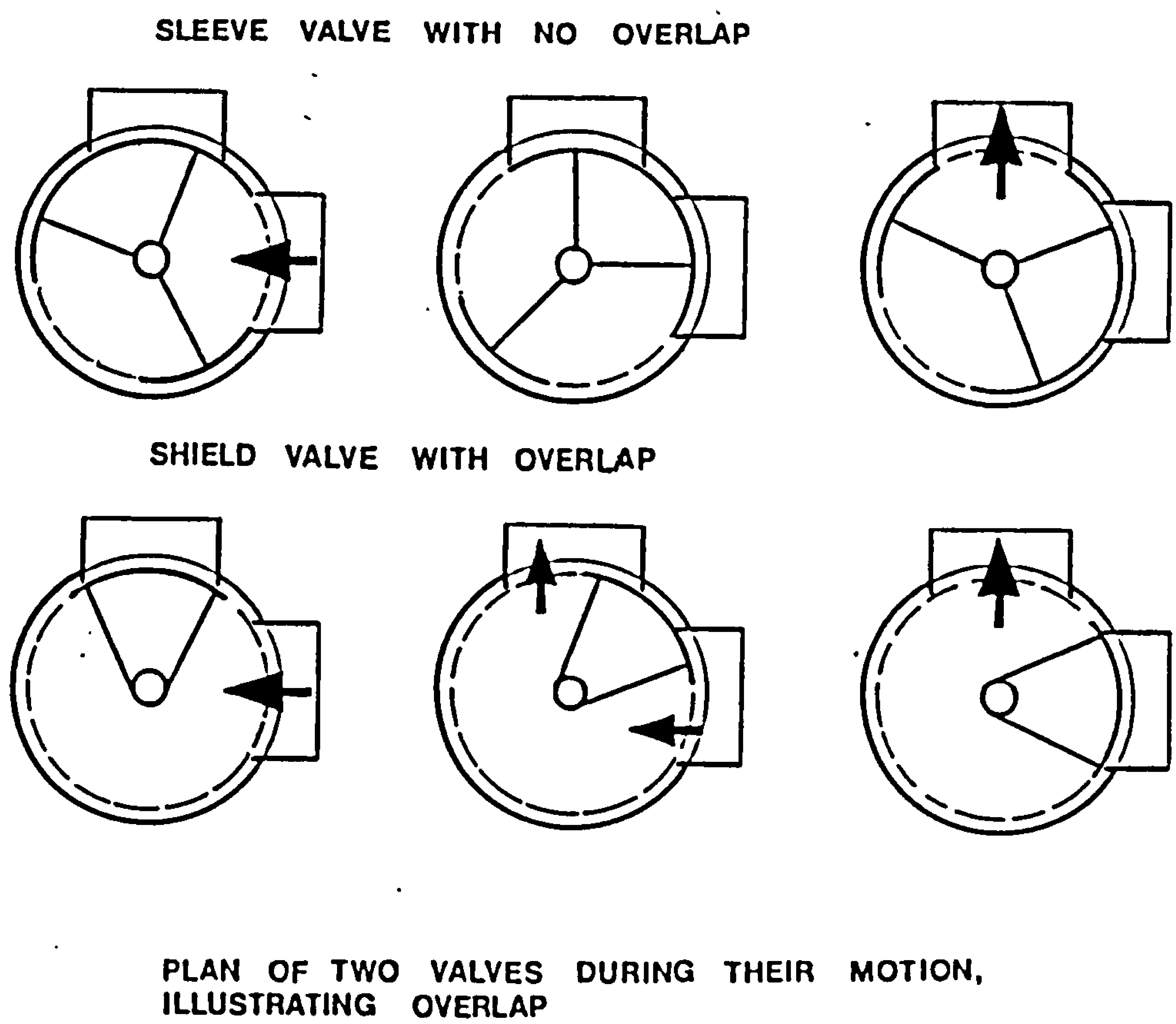
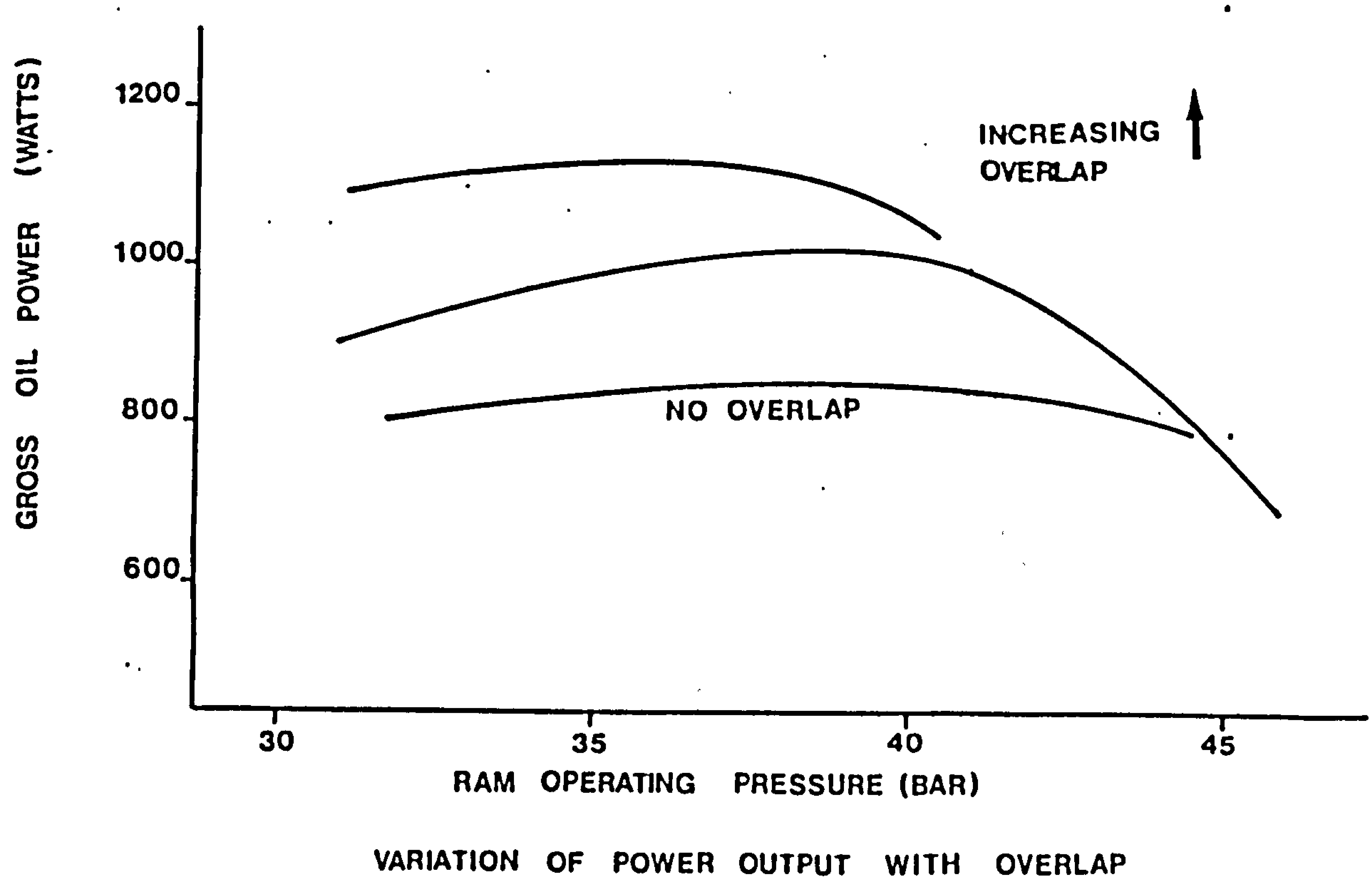
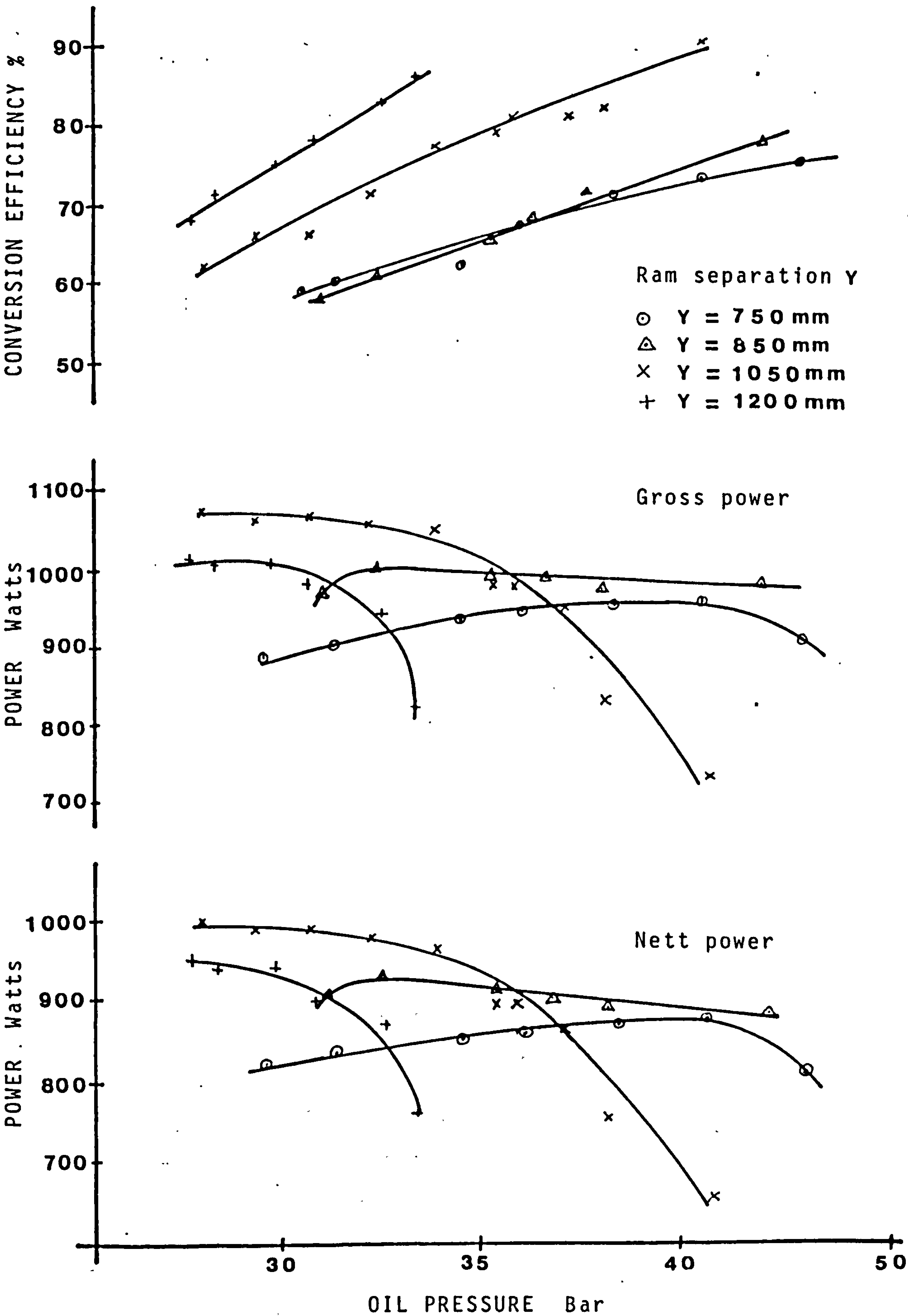


Figure 4.9A



Sleeve valve, shield valve and the effect of overlap on power output

Figure 4.9B



Effect of vertical ram separation on power

Figure 4.10

Diagram 1

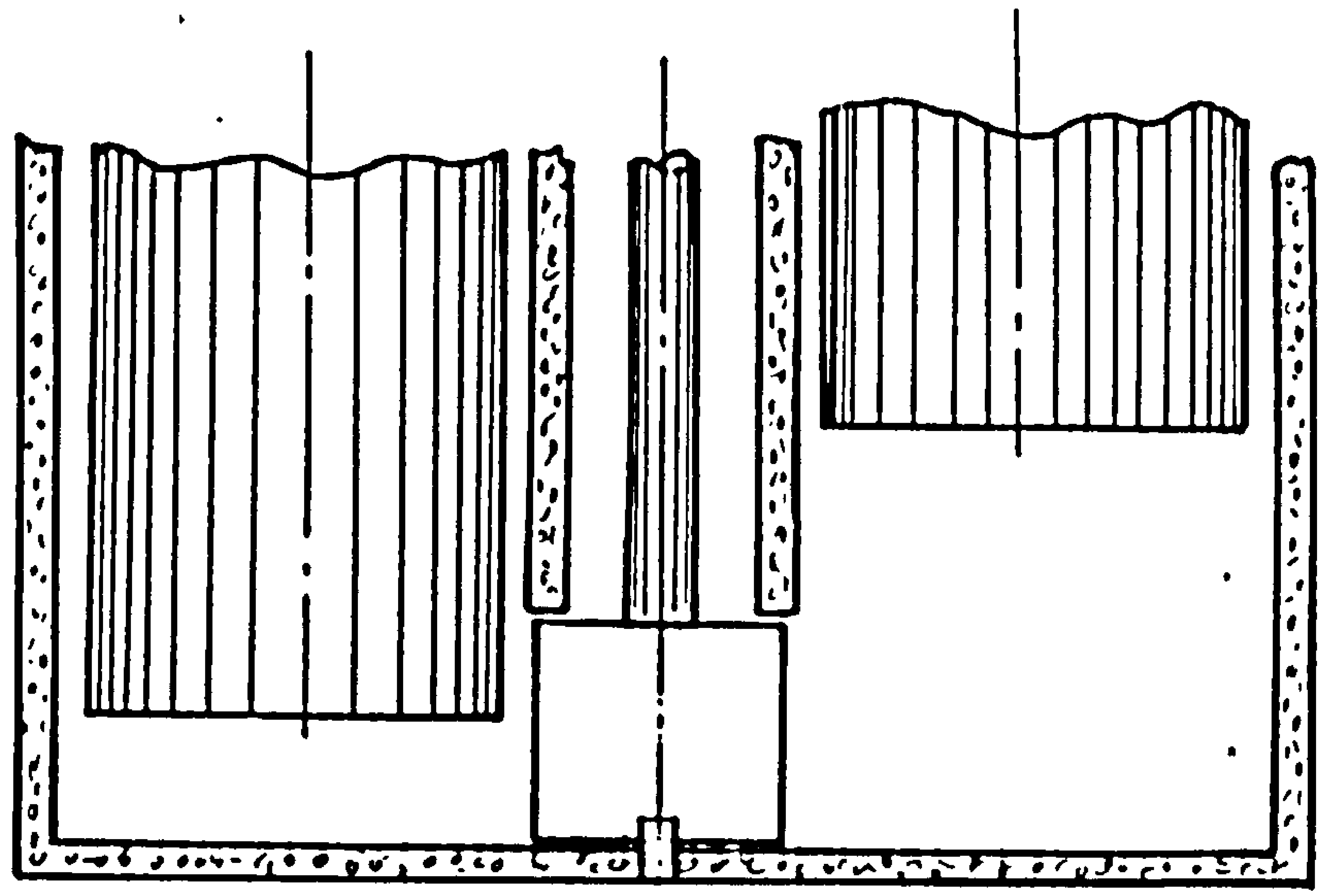


Diagram 2

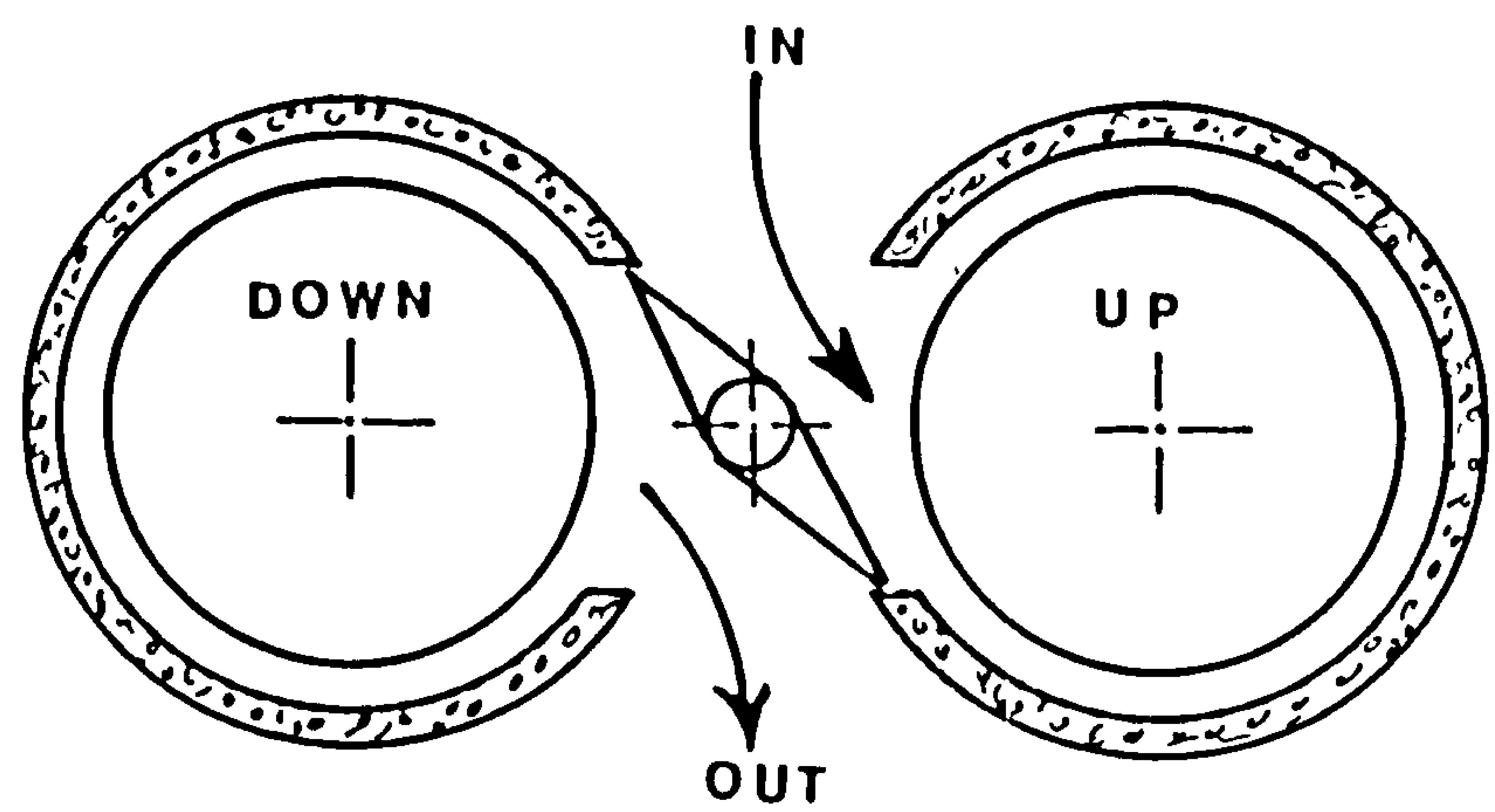


Diagram 3

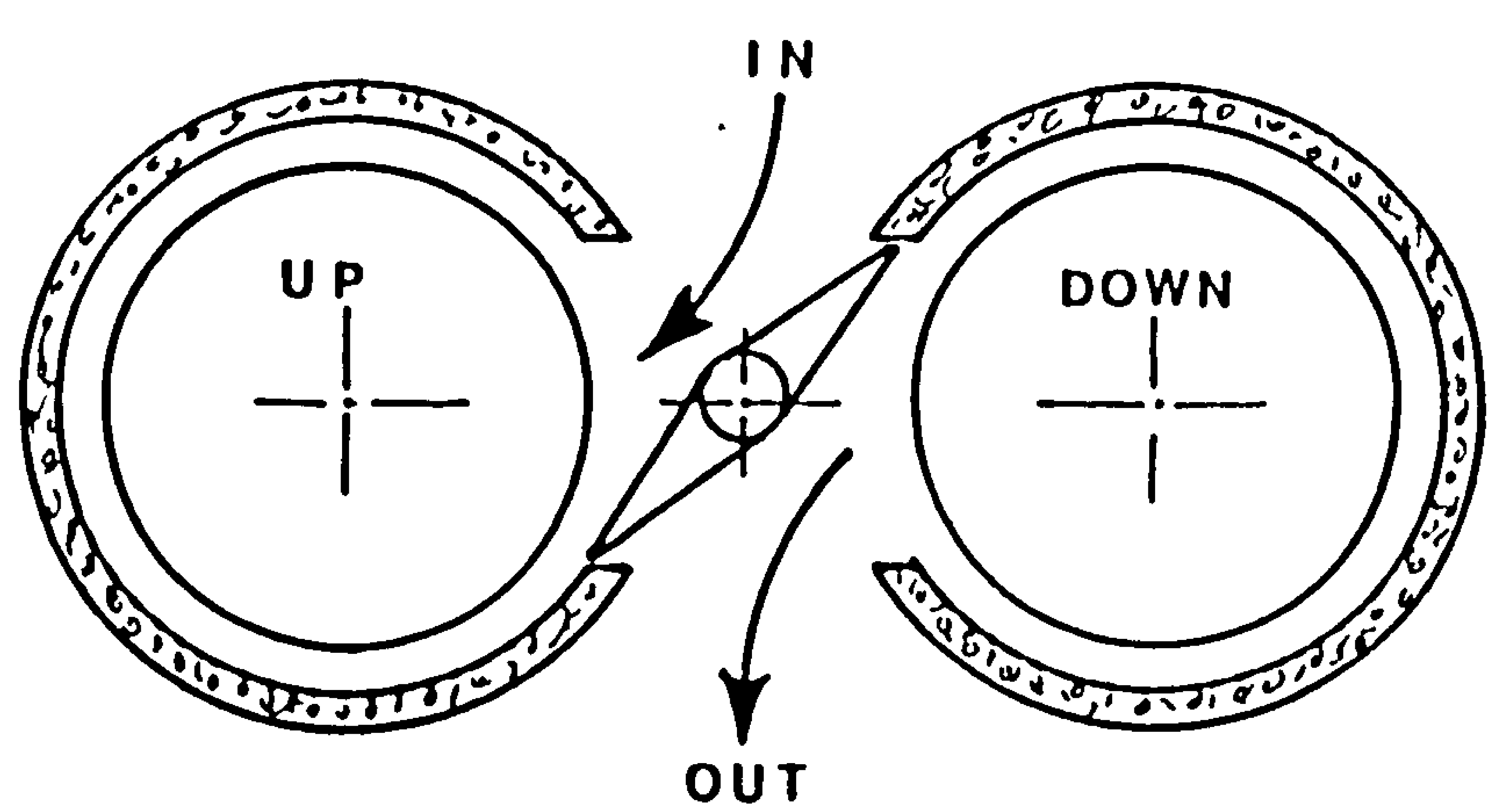


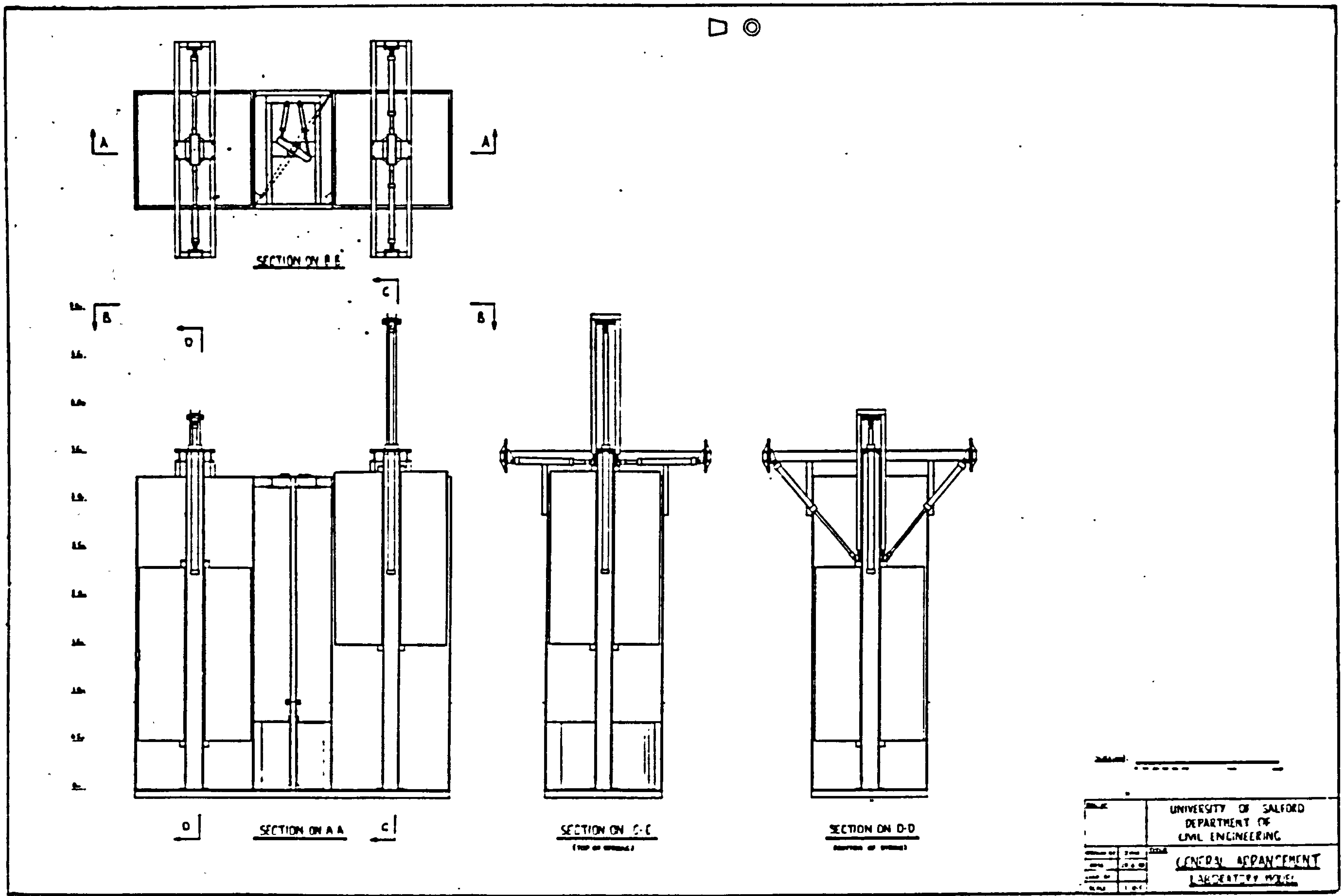
Diagram 1 - section through chambers

Diagram 2 - gate in position for floats as diagram 1

Diagram 3 - gate in alternative position floats change position

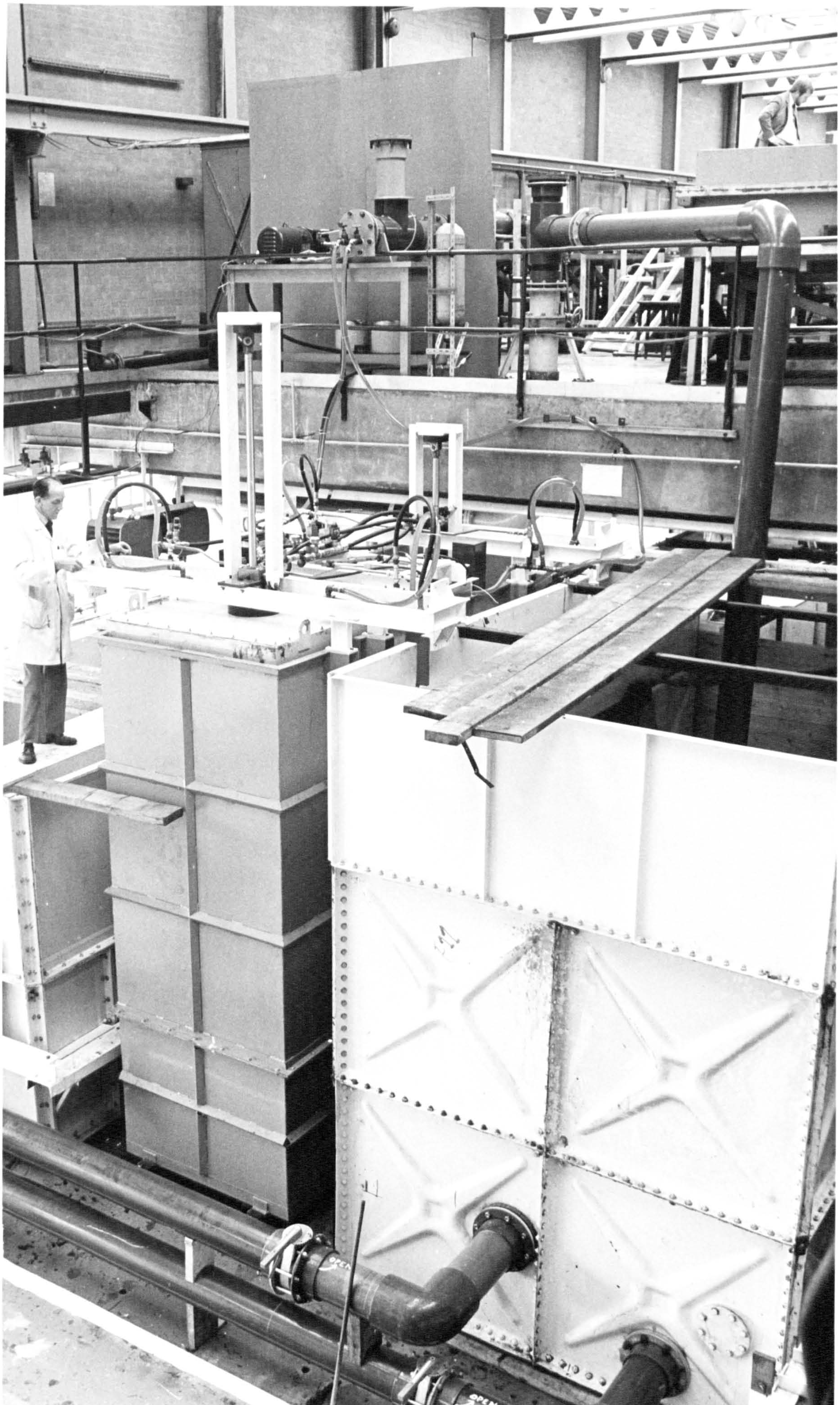
Basic twin chamber Water Engine

Figure 4.11



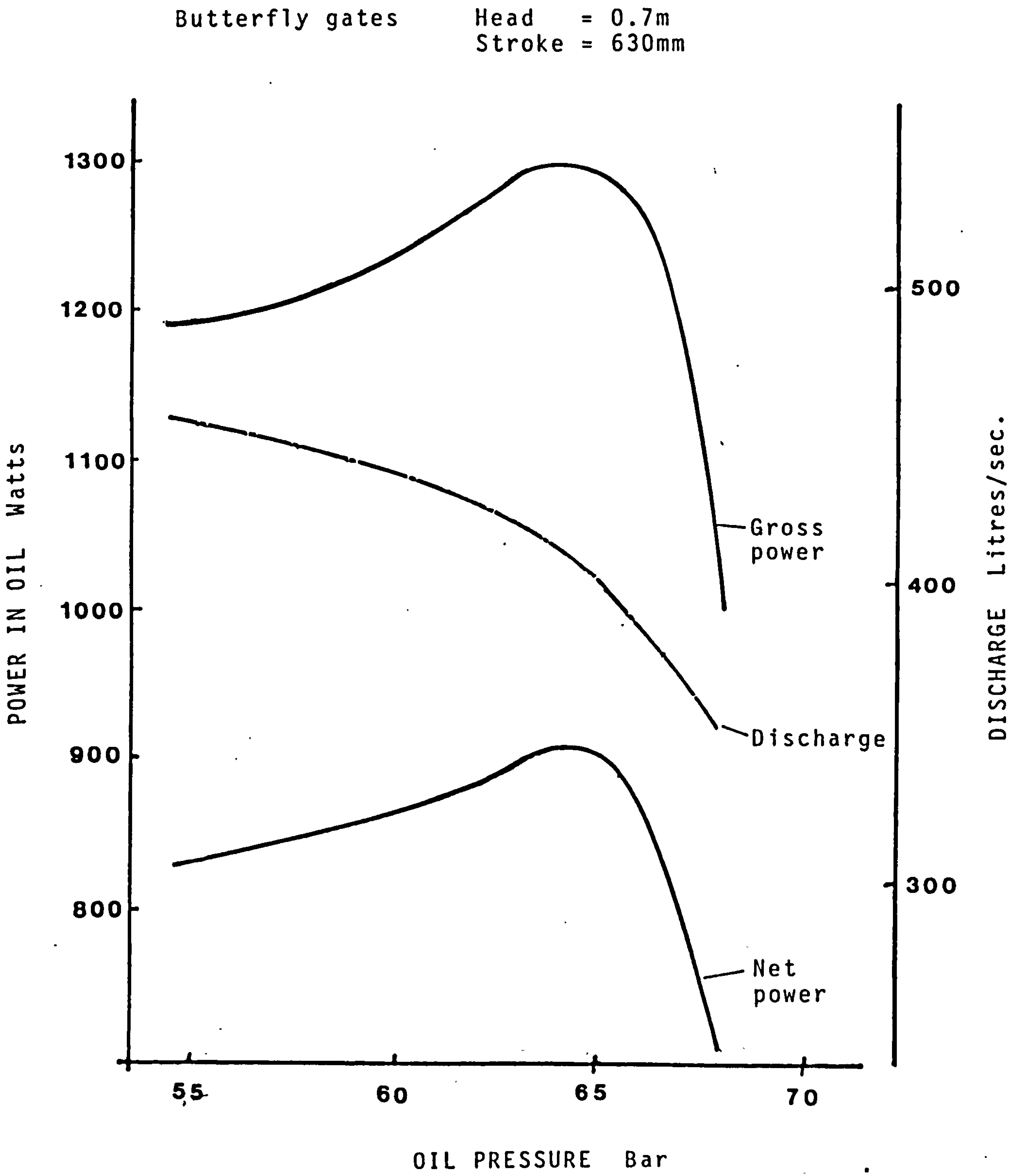
Arrangement of the third laboratory model AUR water engine

Figure 4.12



View of the third laboratory model

Figure 4.13



Typical power curve for twin chamber machine with butterfly gate

Figure 4.14

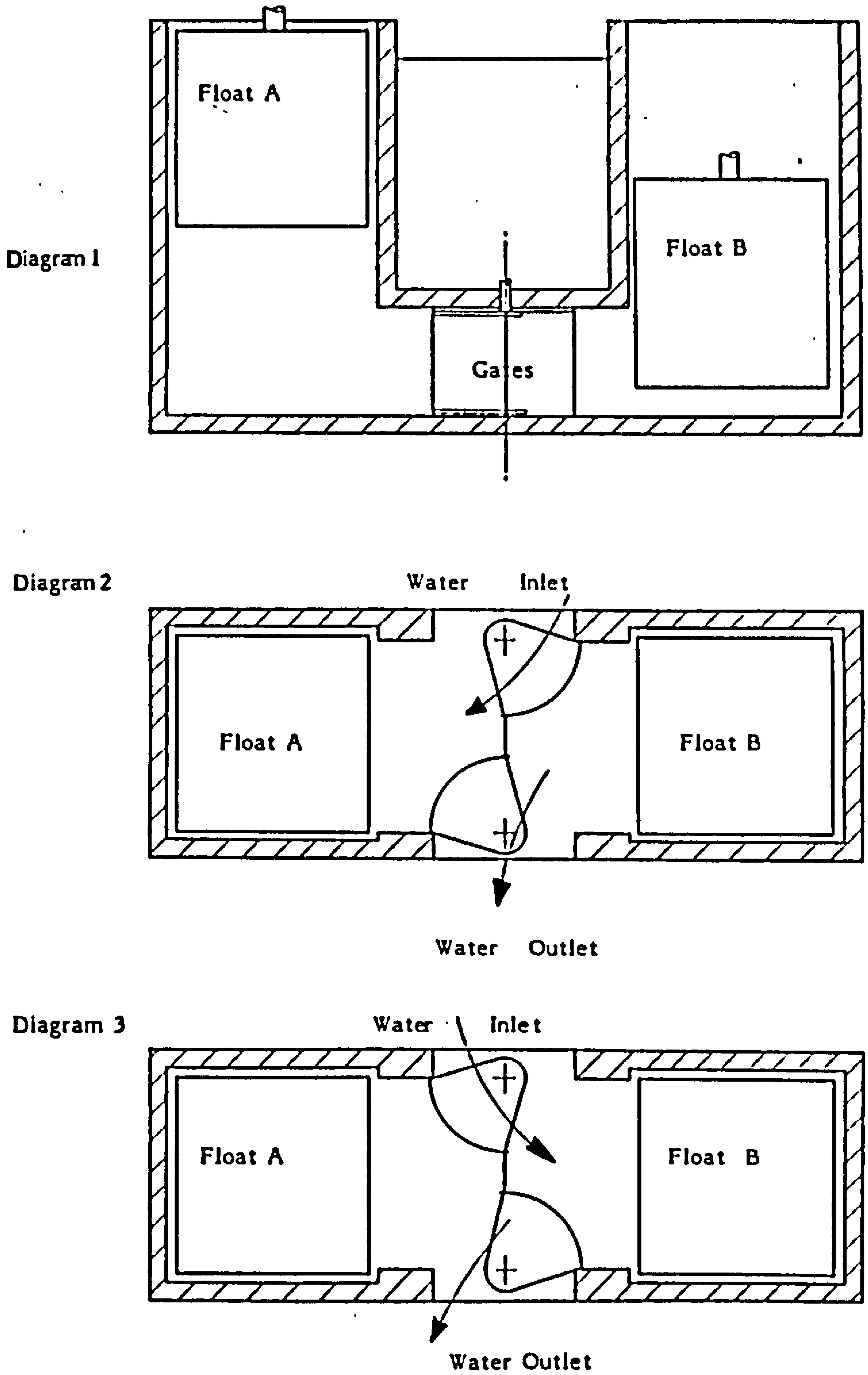


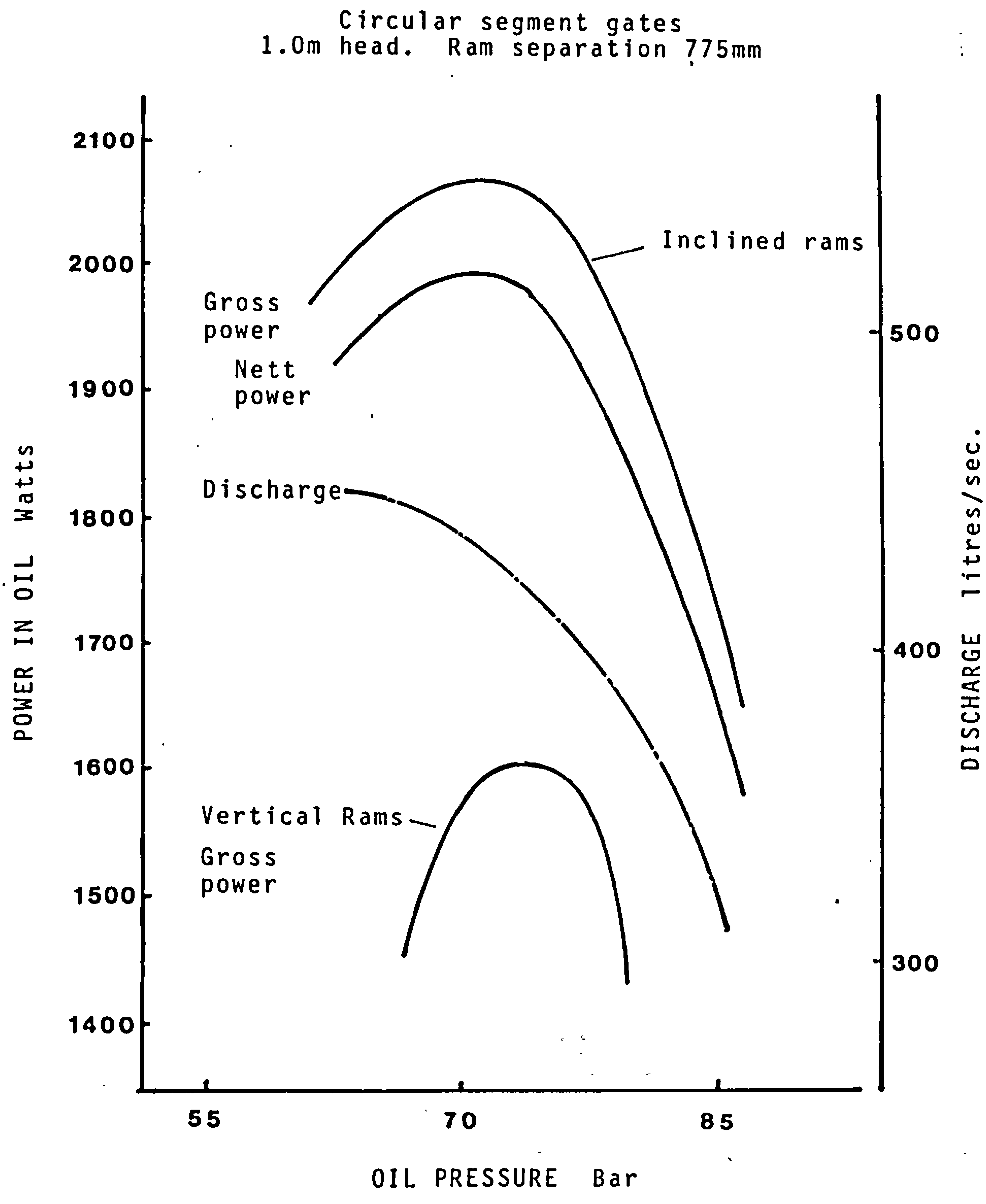
Diagram 1 - section through centre of chambers

Diagram 2 - plan section - floats in position shown in diagram 1

Diagram 3 - gates switched over - float A moves down, float B moves up

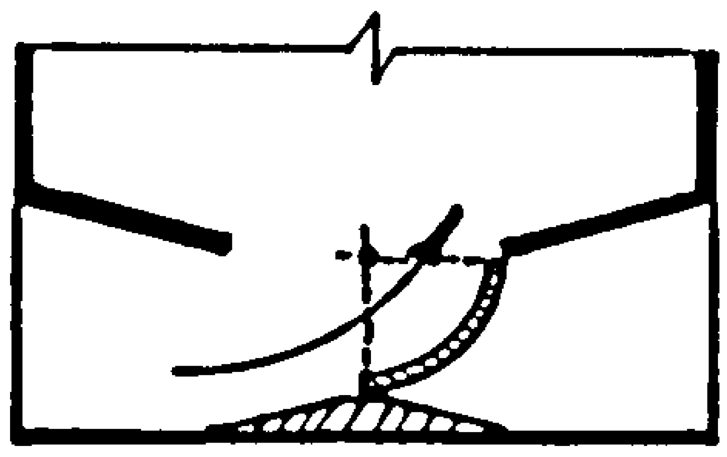
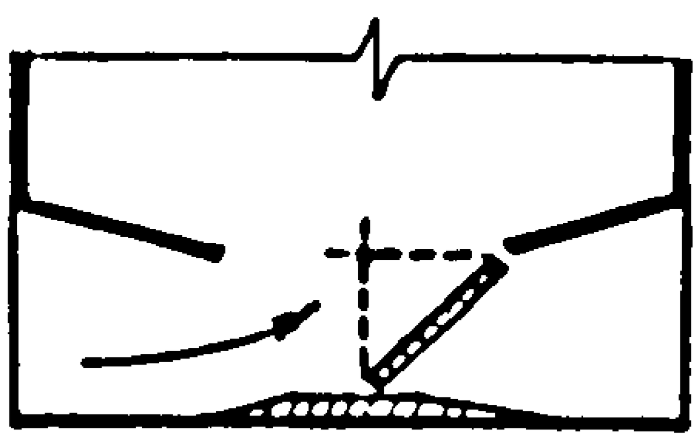
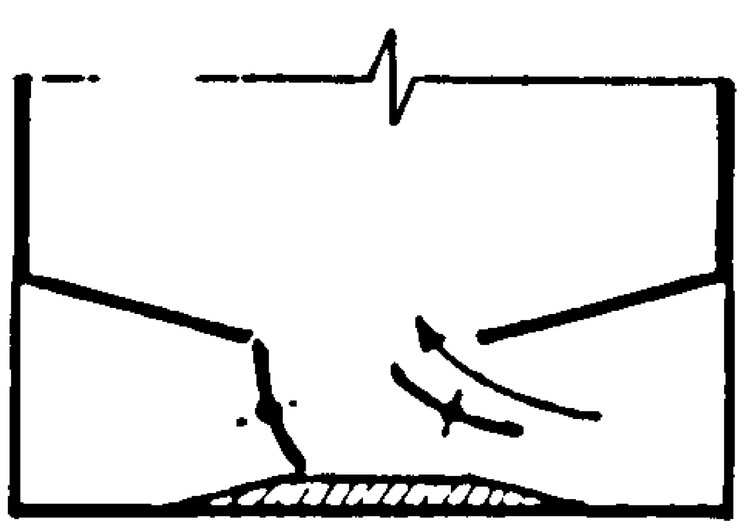
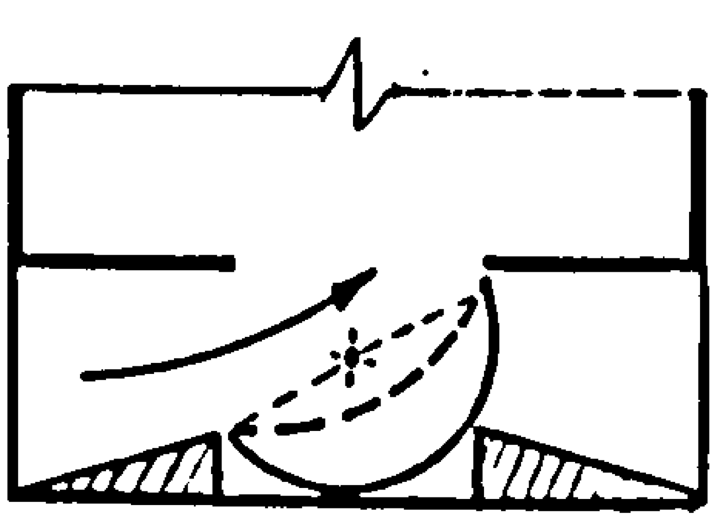
Vertical axis circular segment gates

Figure 4.15



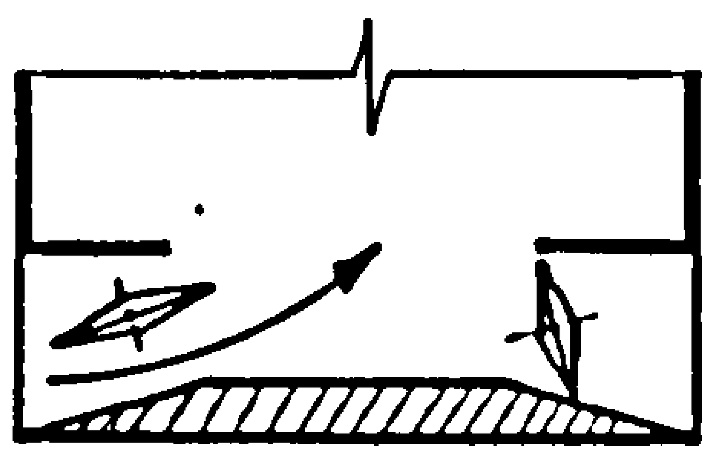
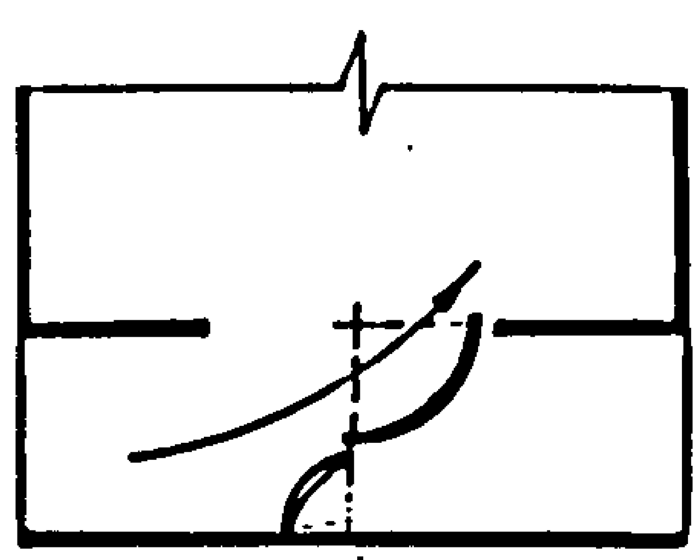
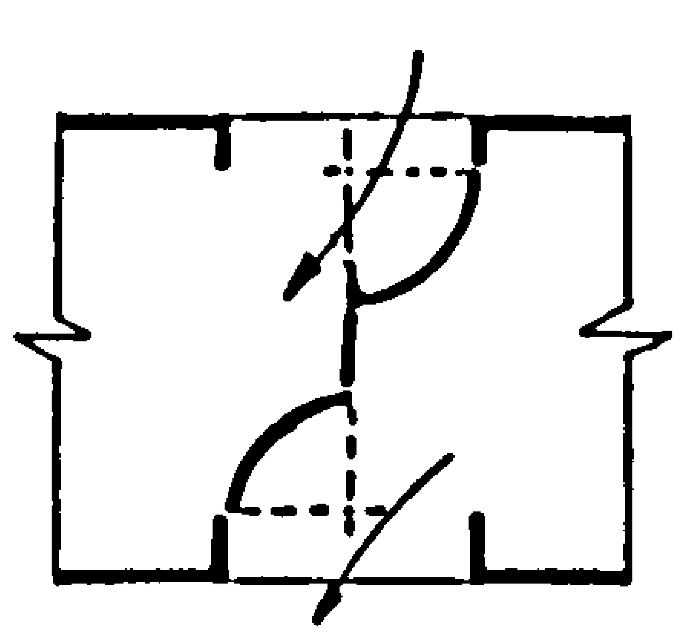
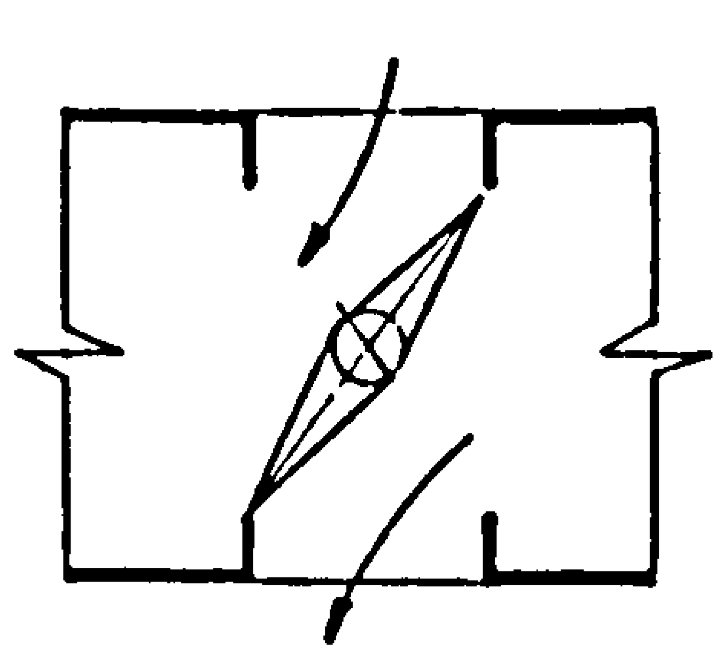
Power output of vertical rams compared to the power output of inclined rams

Figure 4.16

Gate type	Period at gate to chamber area ratio		Comments Prototype - 2 chambers 3.5m square
	0.25	0.33	
 Horizontal axis circular segment gate a) with 15° ramp b) with vertical sill	 1.5 to 1.7 1.5 to 1.7	 - -	Small particles jam between gate and base. Larger particles present no problems. Jamming less frequent with sill than with ramp. Prototype size 3.5m long x 0.75m radius
 Horizontal axis plate gate - one per chamber	1.65 to 1.75		Jams very easily with all sizes of particles between plate and chamber Prototype size 3.5m long x 0.75 radius
 Twin curved butterfly gates a) moving sequentially b) moving simultaneously	 1.9 to 2.2 1.75 to 1.9	 1.65 to 1.8	No jamming until particles are half the gate width. Prototype size:- 3.5m long x 0.875m wide
 Semi-circular horizontal axis cylinder with ramps a) with a fillet b) without a fillet	 2.3 2.0	 1.7 to 1.8	Jams with small particles between the gate and the ramp. If the fillet is excluded, debris collects in the concave part of the gate. Prototype size:- 3.5m long x 1.0m radius

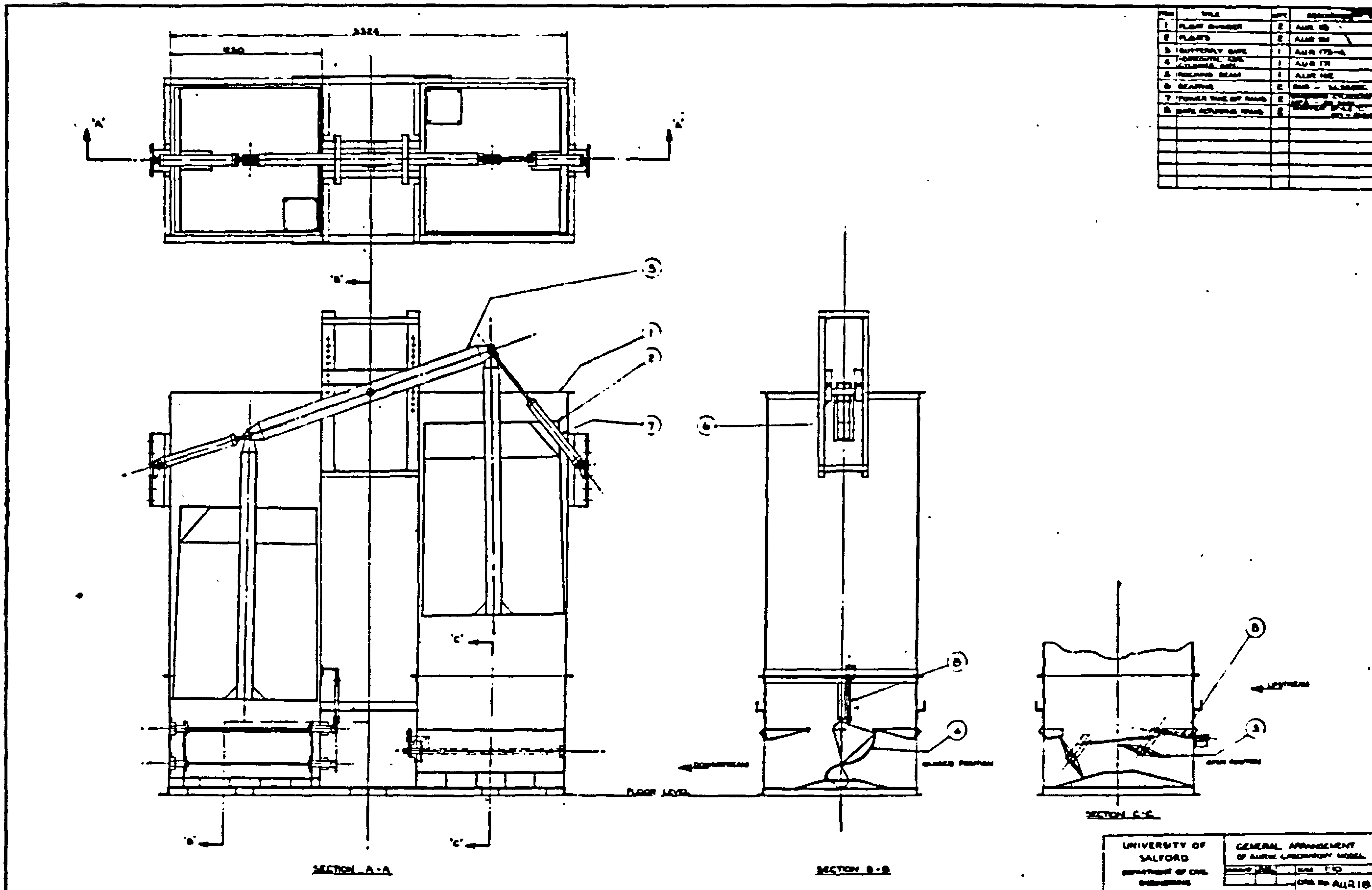
Summary of tests on small scale model gates

Figure 4.17a

Gate type	Period at gate to chamber area ratio		Comments Prototype - 2 chambers 3.5m square
	0.25	0.33	
 Twin butterfly wide separation Two per chamber	1.4 to 1.6	-	No jamming occurs until the particle sizes are at least half the gate width. Prototype size 3.5m long x 1m wide
 Twin horizontal axis segment gates	-	-	Tested on large laboratory model. No tendency to jam noted. Prototype size 3.5m long x 0.6m radius and 0.3m radius
 Twin vertical axis circular segment gates between Chambers	2.0	-	No jamming Prototype size 1.2m high x 0.9m radius
 Vertical axis butterfly gate between chambers	2.0		No jamming Prototype size 1.2m high x 3m long

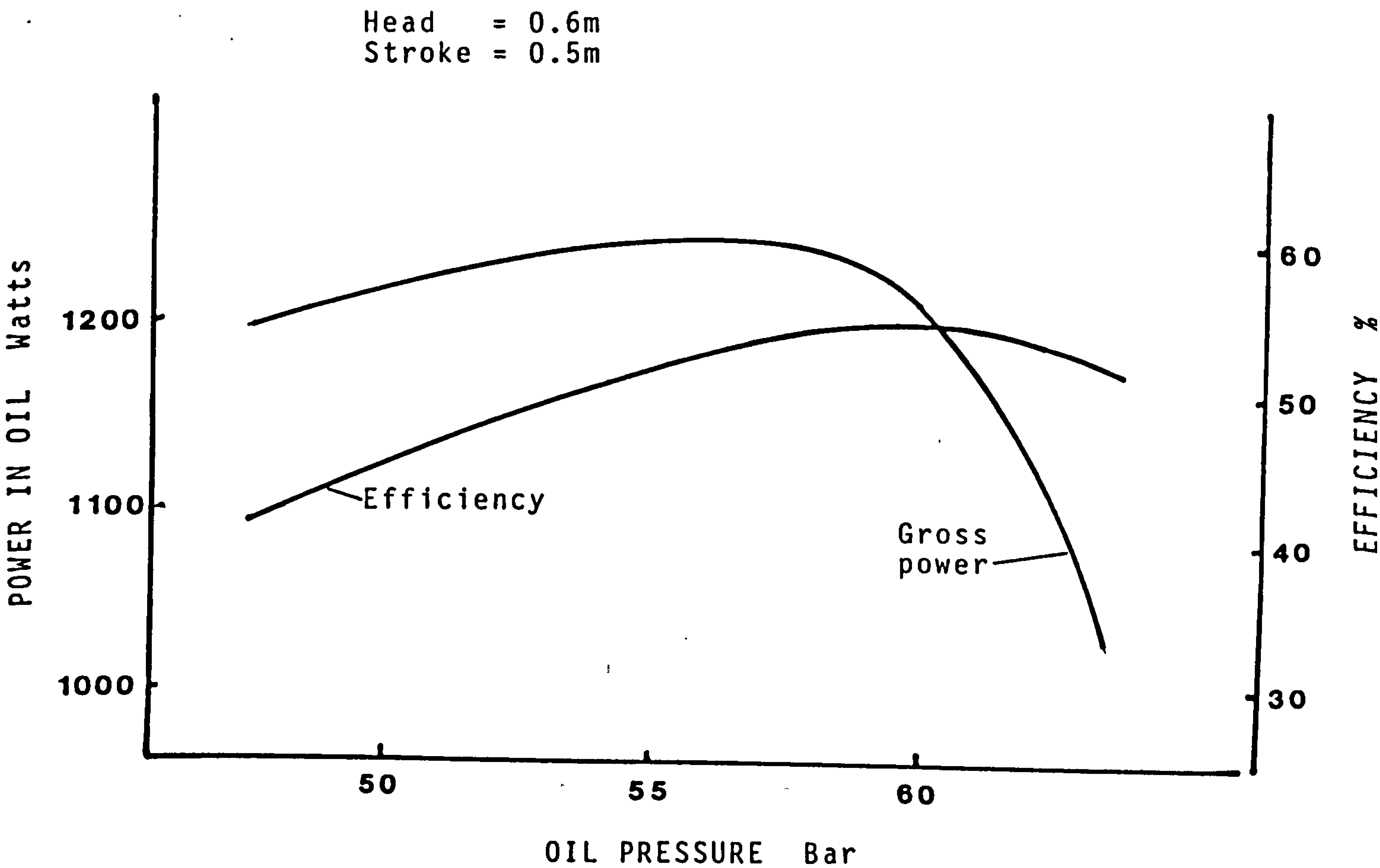
Summary of tests on small scale model gates

Figure 4.17b



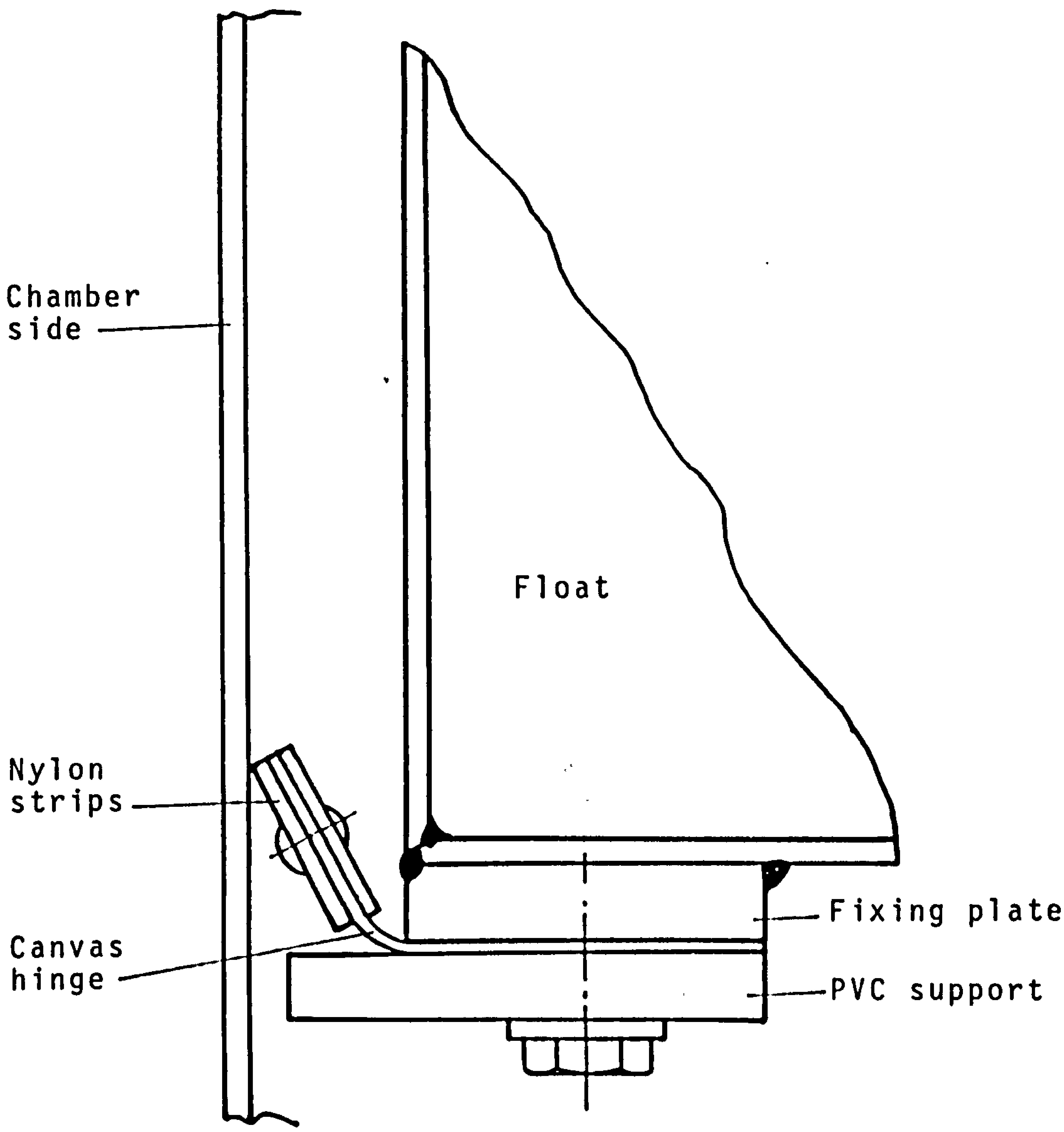
Arrangement of the fourth laboratory model AUR water engine

Figure 4.18



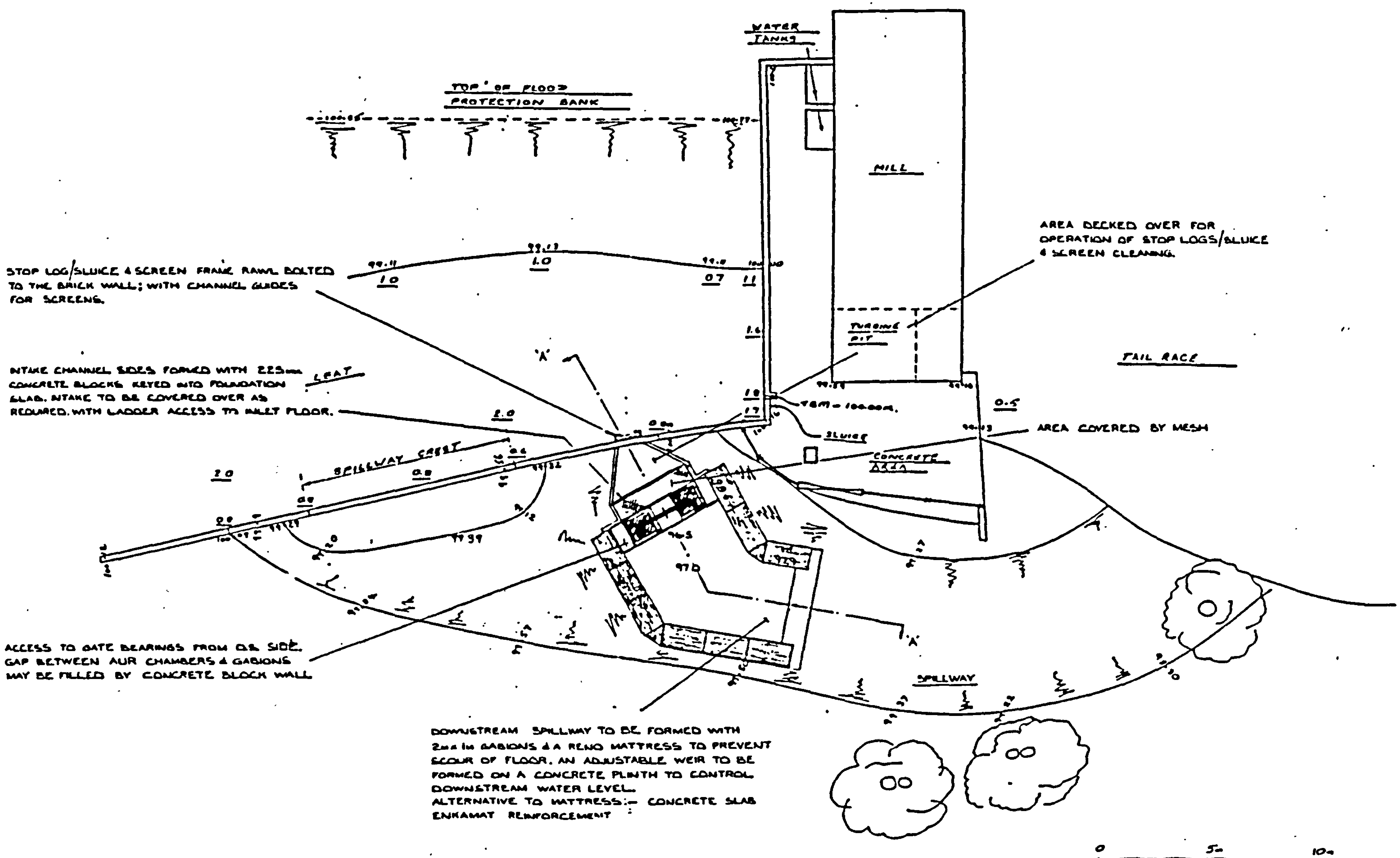
Typical performance curve of fourth laboratory model

Figure 4.19



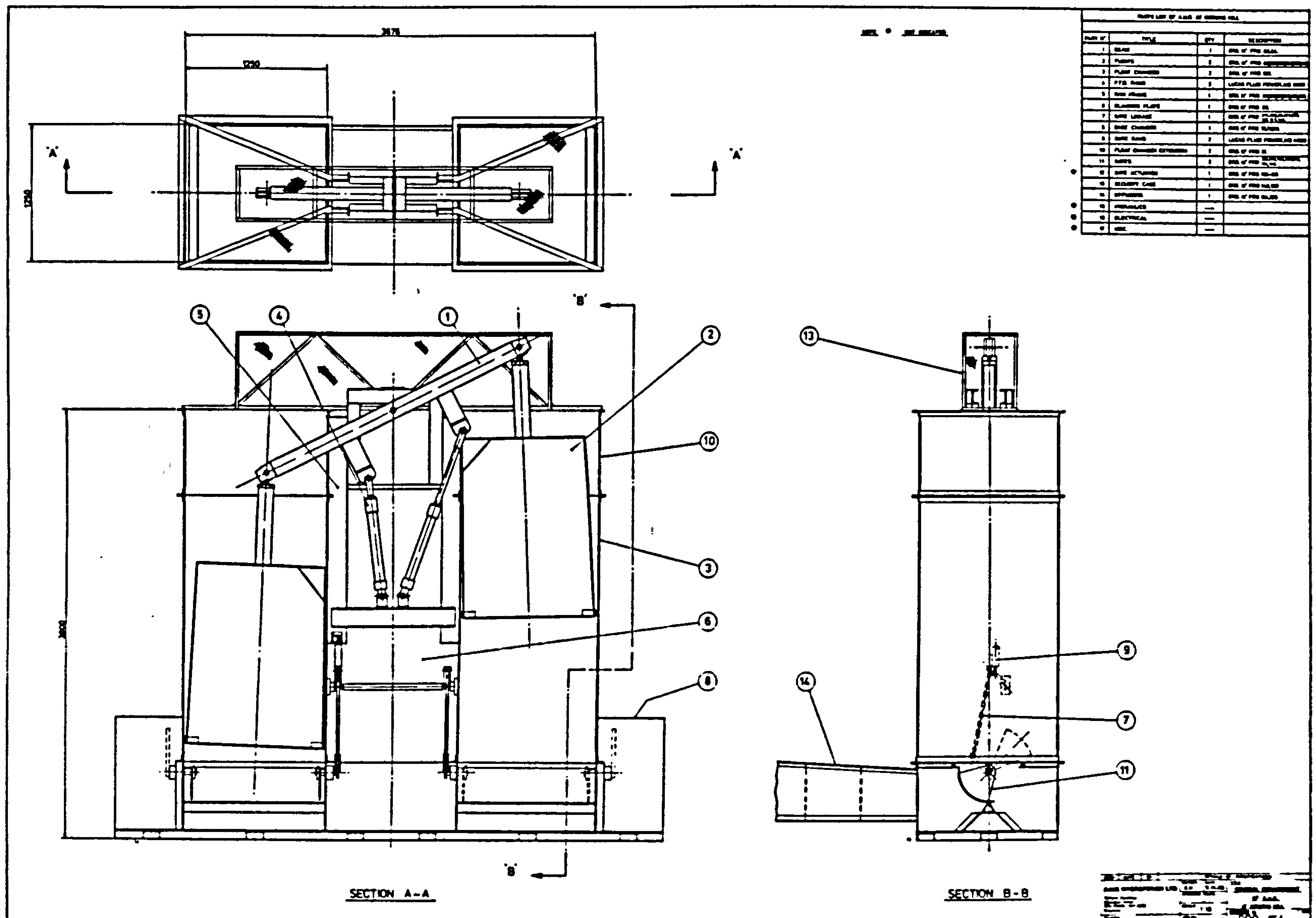
Seal at float base

Figure 4.20



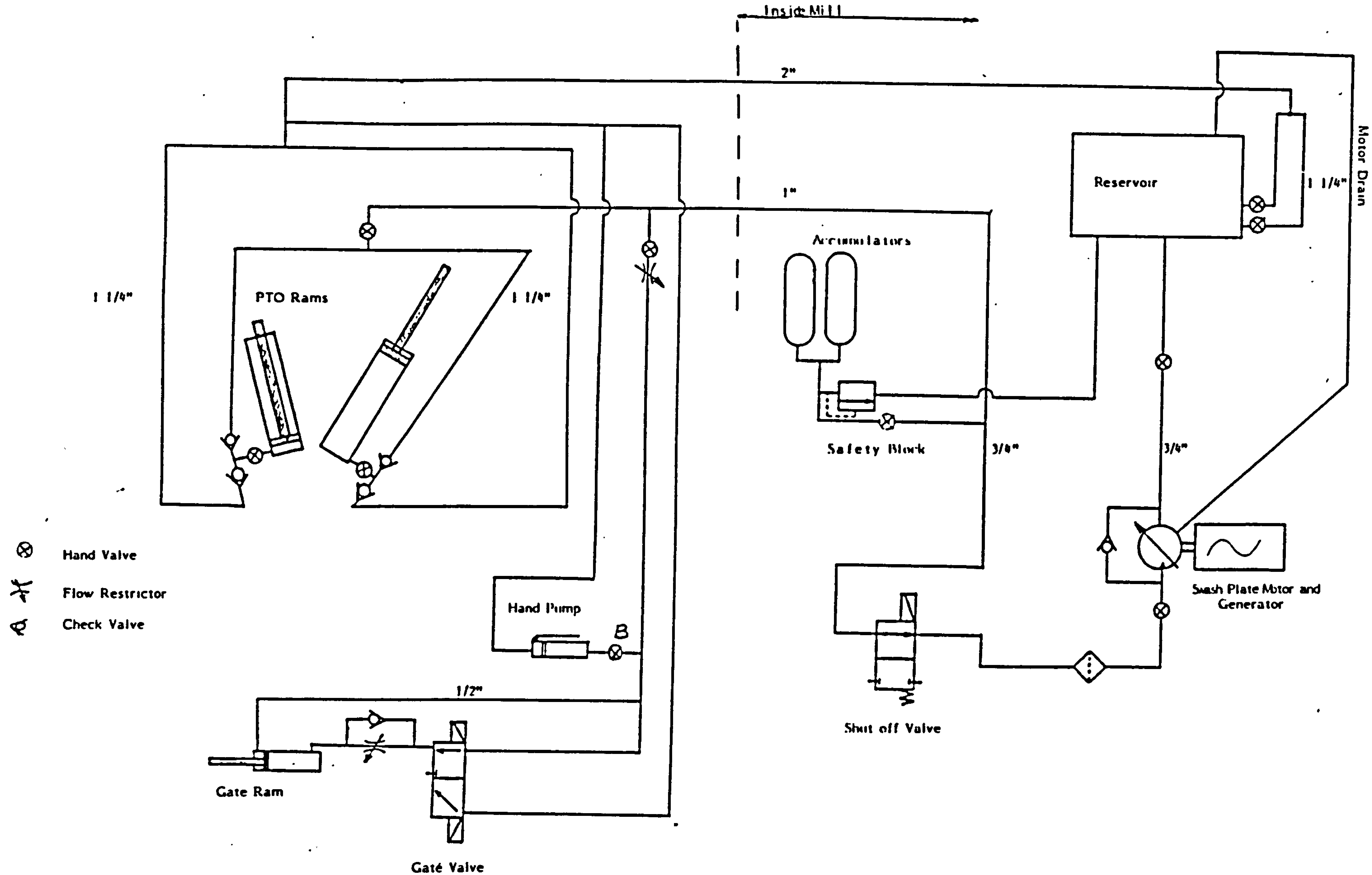
Site plan of Gibbons Mill

Figure 4.21



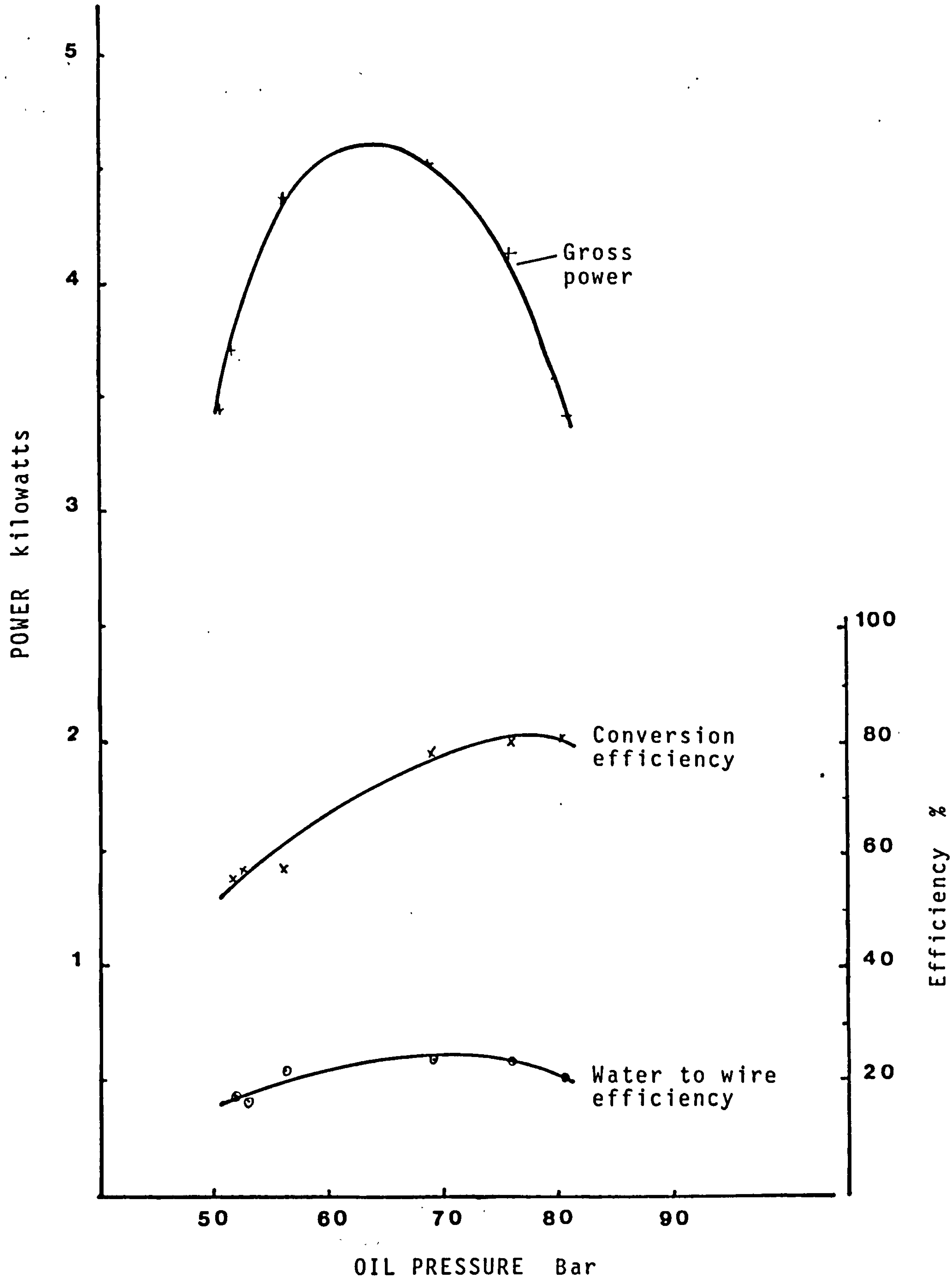
Arrangement of Gibbons Mill prototype AUR water engine

Figure 4.22



Hydraulic circuit at Gibbons Mill

Figure 4.23



Typical results from Gibbons Mill prototype

Figure 4.24

CHAPTER 5

The Salford Transverse Oscillator

5.0 Introduction

The Salford Transverse Oscillator (STO) was invented by Professor E M Wilson and Dr G N Bullock of the University of Salford. It solves the economic problems associated with moving large quantities of water through a low-head hydro power generator by replacing the expensive convoluted passages and runner of conventional machines by simple rectangular passages of concrete or other relatively cheap material.

Rectangular paddles, suspended in these passages are moved by the water and this movement is converted into usable power.

The water flow in the passages is controlled by a series of gates which can be open alternatively to upstream or downstream water. The oscillating movement of the paddles across the direction of water flow gives the device its name.

The writer started work on this project in February 1982 under the direction of Professor E M Wilson with responsibility for the design and construction of the second laboratory model and close involvement in the design of prototype machines.

5.1 Description of the STO

Figure 5.1 shows a schematic plan view of a STO in a water course.

The machine consists of a double wall barrage built across the direction of water flow. At regular intervals along the barrage are gates connecting the space between the walls with the upstream and downstream sides of the barrage. The sections between the gates are called cells. The size and number of cells depends on the conditions at a particular site. Into each cell is dropped a paddle. This fills the gap between the walls and can move along the cell between the gates. All the paddles in the barrage are connected together by a carriage running along the cell walls.

Diagram 1

The gates are positioned to allow upstream water into the left-hand side of each cell and out of the right hand side to downstream. This creates a hydrostatic differential across each paddle and forces it to the right.

Diagram 2

The paddles have reached the extreme right of the cells and the gates are changing their position.

Diagram 3

The gates are now in position to allow upstream water into the right-hand side of each cell and out at the left to downstream level. Hence the paddles move to the left.

The paddles are linked together by a carriage running along the top of the barrage wall. Power is extracted from the movement of the carriage by hydraulic or possibly electrical means.

The gate movement is linked to the paddle position and its speed along the barrage.

As the machine is symmetrical it will work equally well with the flow from either direction. Hence it is suitable for tidal applications.

A pseudo static analysis of the theoretical efficiency of the STO is given in Appendix 2.

This shows the theoretical power output to be proportional to $d_1 \times (d_1 - d_2)$ where d_1 is the upstream depth and d_2 the downstream depth.

The efficiency is $0.5 + 0.5 \frac{d_2}{d_1}$

Thus for a given head the depth of water either side of the paddle should be maintained at the maximum possible to obtain the highest possible efficiency from the machine. The minimum theoretical efficiency is 0.5 when the downstream depth is zero.

For practical reasons of cost and construction difficulties it is unreasonable to expect the downstream depth at any site to be more than the head under normal circumstances

When d_2 = head, maximum efficiency = 0.75.

5.2 The First Laboratory Model

Although a small wooden model had been tested in the one metre wide flume in the hydraulics laboratory of the Department of Civil Engineering in February 1981, the first model of significant size was constructed between April and September 1981.

This was constructed to demonstrate that the machine would work and collect basic information for a more sophisticated model.

5.2.1 Tank & Model Details

Number of gates	-	4
Number of cells	-	3
Cell depth	-	600mm
Cell internal width	-	300mm
Gate centre distance	-	700mm
Maximum carriage stroke	-	320mm

The model was built across the centre of a tank 2.4m wide by 7.3m long.

The upstream side of the tank was fed from the laboratory low head water system through pipes fitted with pitot tubes to measure flow.

The downstream side of the tank was connected to the sump.

Upstream water level was controlled by valves on the inlet pipes, downstream level by weir boards on the tank outlet.

Brief details of the model are given below. Full details are given in reference 30.

5.2.2 Construction

Figure 5.2 shows the basic model with the carriage raised. The cell walls and top are made of wood, the carriage and paddles of aluminium. The carriage runs along the wheels set on the cell walls.

The gates, made of aluminium plate, can be seen in Figure 5.3. They are linked together with a bell crank mechanism and driven by an air cylinder. This cylinder is supplied with air from the laboratory compressor via solenoid valves triggered by two-micro-switches fixed at each end of the first cell.

When the carriage moves over the switches the solenoid valves are triggered and the cylinder changes the gates over so directing water into the opposite ends of the cells and reversing the carriage.

(Note: The power to change over the gates comes from the laboratory air supply - not from the STO itself).

5.2.3 Power Take Off System

Four rams are fastened to the top of the cell walls (2 per side), the rod ends of which are bolted to the carriage and move with it.

As the carriage reciprocates, water is alternatively drawn into the rams from the upstream end of the tank through a system of pipes with non-return valves and pumped out to a container at a high level; thus work is extracted from the water passing through the STO. This system allows the load on the carriage to be adjusted by simply changing the height of the container.

5.2.4 Testing

Parameters measured during tests were upstream water levels, downstream water levels, discharge volumes, periods of oscillation and stroke lengths.

The stroke length was adjusted by moving the micro-switch triggers. For results to be comparable, it was necessary to have a constant stroke. This was chosen as the maximum possible and the triggers repositioned to give this stroke as the head across the machine was varied.

5.2.5 Results

Due to the small size of the model and the relatively crude means of measuring the output, the results obtain could only be considered as giving general information on machine characteristics and pointing the direction for further research.

Details of the test results can be found in reference 31. The main conclusions reached from the tests were:-

1. The dead space at each end of the stroke should be minimised as the water in this dead space does not contribute to the power output of the STO.

2. Maximum efficiency does not occur at maximum operating speed. This is due to the flow losses being greatest at the highest speeds.

3. Gate aperture has a significant effect on output and efficiency. Too small an aperture means a significant head loss across it, too large an aperture allows a greater loss of water at changeover.

4. The pause at each end of the carriage stroke should be as short as possible. This can be controlled by careful triggering of the gates.

5. Leakage losses, across the paddles and on gate changeover, are significant and careful detail design is required to minimise these.

5.3 Gates

The gates on the first laboratory model were not powered by the STO itself. Thus the energy absorbed was not subtracted from the output of the machine. In a full size machine, however, the power absorbed and hydraulic characteristics of the gates are very important. Before the design of the second machine was finalised a number of tests were carried out to determine an optimum design. Points considered were:-

a) Power required to move the gate
b) Head loss across the gate
c) Leakage during operation
d) Tolerance to debris in the water

5.3.1 Horizontal Axis Gate

This was only tested in model form. The construction can be seen in Figure 5.4. It is basically a diagonal slice of a cylinder equal in diameter to cell width and as long as the gap between the gate piers. The base of the machine under the gate is suitably shaped to seal against the gate. The horizontal axle is held in bearings supported on beams set in the cell walls.

The hydraulic characteristics were good with few eddies in the cell.

No power was needed to turn the gate as the flow of water through the cell rotated it.

This gate would be expensive to make full size because of its complex shape.

Small stones under the gate caused it to jam.

It was concluded that a gate of this type was not suitable due mainly to the ease with which it jammed.

5.3.2 Butterfly Gates

A number of these were tested on the first laboratoy model.

Reduced Height Butterfly Gate

This gate was made with its top level with the downstream water and a roof fitted over the aperture so submerging the inlet and outlet to the cell. By eliminating surface effects it was hoped to improve entry and exit coefficients.

A further benefit would be the reduced power needed to turn the shorter gate.

Tests were carried out on simple plate gates of heights 27cm and 18cm with central pivots (full height of cell 60cm).

For all tests there was a choking of the water flow into the cell. The marginal reduction in operating power could not justify the greater structural complexity.

Offset Pivot Gate

A simple flat plate gate was tested with the pivot point offset to the upstream side of the cell, thus giving a weather vane effect to the gate.

Offsets of 12mm, 25mm, 28mm and 50mm were tested.

The upstream opening was held at 152mm and the rear opening adjusted to obtain a seal at the piers.

The torque required to turn the gate against a series of heads was then measured.

It was found that one torque was needed to reach an equilibrium position and a second torque to fully close the gate from this position. The position of equilibrium and relative values of the two torques varied with offset. From the results - given in reference 31 - it was concluded that the optimum offset was 3.8% of the full gate length. At this value the initial torque and closing torque were approximately equal and about half the value needed for a gate with no offset.

5.3.3 Rotor Gates

These are butterfly type gates but designed to be self rotating in the stream of water passing through the cell - thus minimising the energy required to operate them.

The shapes tested are shown on Figure 5.5 a, b, c, d.

These gates did self rotate to a certain extent. Each however, had a stall point part way through the cycle and required some energy to close. The profile d required the smallest closing torque.

A wider gate opening was needed for these gates, due to leading edge thickness, to pass the same water as a plate gate. This would mean a longer barrage for a given carriage stroke.

Full size construction of these could be costly.

Tests were carried out using profile d with an offset pivot. Results were similar to the flat gate offset pivot tests.

There are no obvious advantages with these gate shapes over flat gates.

5.3.4 Rotor Gate with Flaps

Profile d was fitted with small flaps as shown in Figure 5.5f. These were intended to open when the gate stalled (ie. did not self rotate), and cause the water to force the gate shut.

In practice, this did not work - the flaps created eddies which reduced the entry coefficient of the gate opening.

Various positions and sizes of flaps were tried without success.

5.3.5 Aerodynamic Profile Gate

This was tested using an offset pivot in an attempt to improve the water flow around the gate. A fat, symmetrical profile with low drag coefficient was chosen - NACA No. 664-021.

No change in required closing torque was found relative to the flat plate gate. However, an improvement in flow through the gate opening with less swirling was noted.

Conclusion on Gate Tests

1. Vertical axis butterfly type gates are superior to the horizontal axis type tested.

2. Offset pivot of 16mm (3.8% of total gate length) gives lower closing energy requirements for unidirectional water flow than a gate with a central pivot..

3. Aerofoil section gives improvement in water flow through the gate opening.

5.4 The Second Laboratory Model

5.4.1 Design Brief for the Second Model

The second laboratory model was built with two basic intentions:

a) to collect performance data

b) to test and develop the machine elements prior to building a prototype.

The first of these objectives required that the machine be large enough to allow accurate measurements of power output, velocities, water leakage, etc., to be made.

The second objective required that the design should allow for changes to be made to the elements of the machine. This would allow new ideas to be tried out as they came along.

Important points to be considered during the design were:-

1.	Gates	- Design for minimum energy loss, both in water flow and actuation.
2.	Gate control mechanism	- The system for linking gate acutation to paddle movement should be simple and reliable.
3.	Kinetic energy recovery system	- A system for recovering or absorbing the kinetic energy of the water/paddle/carriage at the end of each stroke should be incorporated into the design.

4. Improved paddle/carriage/power take off system.

5. Allow for investigation of the effect of cell size and gate design on power output.

6. Prototype costs - The laboratory model should be designed with due consideration for the materials and methods that could be used in a prototype eg. standardised units, existing technology, off the shelf components.

5.4.2 Size

The maximum flow from the laboratory pumps is $0.35 m^3/sec$. The machine was designed to use this as a maximum throughput.

Number of Cells & Gates

There is one more gate than there are cells in a STO barrage. The STO could work with only one cell and two gates. This configuration, however, would give a relatively large paddle and gates for a given flow of water.

A two or three cell machine was considered to be more practical.

A three cell machine was finally fixed upon as probably being typical of a full-size installation. This layout also ensured that the two central gates were free of "bank effects" which could affect the flow into the end cells.

Dimensions

The machine dimensions were fixed from the above considerations, using data obtained from the first model:

cell width - 445mm
cell depth - 880mm
cell length - 830mm
gate centres - 1090mm

The tank to hold the machine was built of 4ft square (1200mm) glass fibre panels. Internal dimensions 12ft x 28ft (3.66m x 8.54m).

Control of Water-Levels

This was achieved on the upstream side by regulating the flow into the tank via geared butterfly valves, and on the downstream side by a drawbridge style weir frame with removable boards.

5.4.3 Designs Considered

Prototype Machine

It was proposed to build a prototype machine in the River Irwell at Adelphi Weir, Salford. Basic data for this was:-

3 cells
3m stroke
$10m^3$/sec maximum flow
2m maximum head

The laboratory model, it was decided, would be designed assuming it was a scaled down prototype. The figures given below refer to this prototype.

Paddles and Power Take Off System

The force on the paddle is constant over the major part of the stroke and is approximately 60kN per paddle. Total force is thus 180kN - approximate to 20 tonnes.

Investigation of various means of harnessing this power - large, slow moving forces - gave the conclusion that a hydraulic power take off system was the most practical.

Direct conversion to electricity could not be done using existing technology and available equipment.

Furthermore, these water forces give rise to large bending moments at the carriage to paddle connection if the thrust is reacted solely at the carriage. To reduce this moment, it is necessary to react the water forces nearer to the centre of pressure of the paddle.

Many options were considered; the three most likely to work were:-

1. Push-pull rods at the centre of pressure, carriage above water.

2. Carriage to run in the water near to the centre of pressure.

3. Wire ropes connecting the lower ends of the paddles via pulleys to the hydraulic system.

The wire rope system was rejected on the grounds that the fatigue life of the rope was not long enough. (The machine would perform about 3 million cycles per year).

The submerged carriage was considered to offer too many problems in terms of wheel bearing life and debris entangling with the mechanism.

The push-pull rod at the centre of pressure was hence adopted as offering the most satisfactory solution to the problem.

Several possible methods were available to connect this push-pull rod to the hydraulic power take-off system. The three considered most practical were:

a. A ram connected directly to the rod.

b. A swinging beam pivoted above the carriage with rams attached to it.

c. A rack and pinion to convert the oscillation to rotary motion to turn a hydraulic pump.

Option b was adopted as it gave great flexibility in that the ram positions on the beam could easily be adjusted to alter the resistance to motion. Also it allowed the rams to be above water level.

The tower supporting the beam could be sited on the bank or on the barrage above the cells. Siting on the bank gives a simpler structure, does not load the cell walls and does not obstruct the water course in time of flood. This configuration was chosen. Figure 5.6 shows the laboratory model and figure 5.7 a simplified arrangement drawing.

5.4.4 Construction Details of the Laboratory Model

Gates

These are of simple butterfly type with the spindle offset towards the upstream. The profile is a narrow lozenge formed from a central tube with two aluminium plates wrapped around it. This is considered to give the desirable flow properties of the aerofoil without the expense of making the true shape.

The push-pull rod passes through a horizontal slot in the central gate tube. This slot is sealed by a plastic bush inside the tube floating on the push-pull rod. Hence the push pull rod can pass through a gate and the gate can turn from side to side without excessive leakage.

Each gate is fitted with removable wooden leading and trailing edge pieces. These can be changed to assess the effect of their shape on water flow.

The internal space between the sheets forming the gate faces, is filled with polyurethane foam. This serves to stiffen the structure and keep out water.

The gate bearings are of self lubricating plastic on stainless steel spindles.

Paddles

These are constructed of 12.5mm thick Aerolam - a composite material produced by Ciba-Geigy Limited. The outer skins of glass fibre reinforced epoxy resin are separated by an aluminium honeycomb core.

This gives a very rigid but light structure - comparable in price to aluminium.

This was chosen as being a possible material to use on a prototype where lightness and corrosion resistance would be important.

Adjustable PVC rubbing strips were later fixed to the paddle sides and bottom edges. These were necessary to minimise leakage across the paddles.

Carriage

This is constructed of aluminium angles and fitted with three wheels at each of the four corners. (The first prototype had wheels fixed to the barrage walls). These wheels run on aluminium rails fixed to the cell walls and are positioned to prevent the carriage moving up, down or sideways.

Push-Pull Rod

This is of galvanised steel tubing in discrete sections between each paddle and between the last paddle and the swinging beam. Each joint between the paddle and rod allows the rotation of the rod through 360° plus a translation of up to 10mm. This is to allow for minor misalignment as the paddles move along the cells.

Swinging Beam

A standard hot rolled joist is used for this. The push-pull rod is connected to the beam via a steel pin with bronze rollers running in a slot at the bottom end of the beam. The beam is pivoted at its top end in bronze bushes fixed to the supporting A-frame.

Cell Walls

These are pre-cast concrete sections; four sections form each cell bolted to a marine ply base plate which is fastened to the floor of the tank.

The end of each cell wall has inserts cast in to take the bolts holding the wooden gate stops. These stops can easily be changed to test the effect of their shape on the water flow.

Power Take Off System and Hydraulic Circuit

The power is taken from the swinging beam by two single acting hydraulic cylinders fixed to it as shown in Figure 5.7.

Non return valves connect each cylinder to the reservoir on the low pressure side of the system and to an accumulator on the high pressure side. When a cylinder is extended by the movement of the beam oil is drawn from the reservoir. On reversal of the beam movement, the oil is pushed out of the cylinder at high pressure and stored in the accumulator.

From the accumulator a steady flow of high pressure oil is drawn off to drive a motor and generator unit.

Refer to Figure 5.8 for details of the hydraulic circuit.

Mineral oil based hydraulic fluid was used in this system. The reasons for this are discussed in section 4.3.2.

Gate Actuation

The gates are actuated by a link bar attached to the top of each gate. This bar is driven by two small cylinders attached to the gate nearest the power take off frame. Oil from the high pressure accumulator is fed to these cylinders via a spool valve triggered by the movement of the swinging beam. Thus as the beam reaches one end of its movement it triggers the valve which allows oil into one or other of the cylinders thus moving the first gate and, through the link bar, all the other gates. The timing of this triggering can be adjusted if necessary as the machine is running.

The pivot points of these gate cylinders can be adjusted to supply larger or smaller proportions of power to the gates. The adjustment allows a variation of between 5% and 20% of the available power to be directed into the gates.

Kinetic Energy Recovery System

An adjustable cushion is provided at each side of the swinging beam made from belleville washers and coil springs. This was designed to collect the kinetic energy of the carriage/water at the end of each half cycle and give a boost to the movement at the beginning of the next half cycle.

During tests no discernable benefits were noted using this device. The acceleration and deceleration of the carriage depended upon the pressure in the hydraulic system and the gate timing - not on this buffer.

5.4.5 Summary of the Second Laboratory Model Mechanical Details

Number of cells	- 3
Number of gates	- 4
Cell length	- 830mm
Cell width (inside)	- 445mm
Cell depth	- 880mm
Distance between gate centres	- 1090mm
Stroke	- 700mm
Gate aperture between cells	- 260mm

Hydraulic System

Power take off cylinders, single acting, 2 number, 40mm bore, 300mm available stroke, 170mm used.

Gate cylinders single acting 2 number, 12.5mm bore, 150mm available stroke.

Fluid	- Shell Tellus 37
Accumulator	- bladder type 4 litre capacity (Christie ref. AC04.00.00.20)
Fittings	- 3/4" on main system 1/4" on gate system
Operating pressure	- 35 bar to 105 bar

Materials

Cell walls	- Precast reinforced concrete
Paddles	- Ciba Geigy Aerolam F Board 12.5mm thick
Gates	- Steel central tube, aluminium skins on wooden ribs filled with PU foam.

Carriage	- Aluminium angle - bolted to paddles. Plastic wheels with synthetic rubber tyres 60mm diameter. 12 off.
Carriage runners	- Aluminium angle fixed to top of cell walls.
Swinging beam	- Rolled steel joist
Power take off frame	- Rolled steel sections
Push pull rod	- 1" N.B. galvanised tubing.

5.4.6 Testing

Measured Parameters

The first laboratory model had been tested by fixing the load on the paddles and measuring the period for a variety of heads. The results obtained were difficult to analyse because too many parameters were changing at each test point.

For the new model it was decided to fix several parameters at each point and record the change in the others.

The Carriage Stroke

This was fixed for all the first series of tests at 700mm by adjusting the position of the mechanical triggers which operated the gate valve using a simple hand wheel and screw mechanism. To ensure that any changes in stroke were noted two micro switches were positioned on the cell walls at each end of the carriage travel. These switched on a small lamp giving clear indication that the end of the stroke had been reached.

Load on the Paddle

Fixing the operating pressure of the hydraulic system effectively fixes the load on the paddles. During tests a needle valve was used to maintain pressure (rather than the motor generator system) as this was easier to adjust. At the start of each series of tests the pressure of the nitrogen in the accumulator was set at 90% of the operating pressure to ensure maximum smoothing effect from that size of accumulator.

Downstream Depth

This was reasonably easy to control at a constant value by raising or lowering the drawbridge weir. Small adjustments during testing did not disturb the other fixed parameters.

Upstream depth could not be controlled accurately without long delays between points; waiting for the system to settle and then re-adjusting.

Similarly it was found that discharge was also very difficult to hold constant without long delays between points and substantial adjustment of the inlet valves.

Thus the fixed parameters chosen for each series of tests were stroke, oil pressure and downstream depth. The variables measured were upstream depth, discharge and period.

From these values of head, power and efficiency could be calculated. (The power was calculated from the oil pressure and oil flow - this being fixed by the stroke of the power take off rams - a fixed proportion of the carriage stroke).

Instrumentation

Oil Pressure

Measured by a Bourdon type dial gauge 0-140 bar, 100mm diameter, calibrated against a master gauge.

Water Depths

By rule in stilling wells made from perspex tube.

Discharge

By calibrated pitot tubes in the feed pipes.

Period

By stop watch over several cycles.

Procedure

The test procedure was as follows:-

1. Set the weir to give the chosen downstream depth, (this required readjustment before the readings were taken).

2. Charge the hydraulic accumulator to 90% of the chosen operating pressure.

3. Open the valves in the inlet pipes to let water into the upstream side of the tank.

4. When the head is sufficient to move the carriage adjust the needle valve to give the chosen operating pressure.

5. Adjust the downstream weir to give the correct downstream depth.

6. Adjust the inlet valves to stabilise the flow.

7. Adjust the needle valve to maintain the oil pressure at the chosen valve.

8. Check the carriage stroke - make minor adjustments as necessary.

9. Allow the system to settle - make minor adjustments as necessary.

10. Record period, pressure, upstream depth, downstream depth and pitot manometer readings.

11. Increase discharge for the next set of readings by opening the water inlet valves.

Steps 5 to 10 were repeated until either the STO was overtopped or the maximum water flow was reached.

The procedure was then repeated for another value of oil pressure.

When all suitable pressures at that downstream depth had been tested, another downstream depth was selected and the procedure repeated.

The information was then processed to produce figures for head and power at various flows and water depths.

The results, together with those for machines of a different stroke to width ratio, are discussed later.

5.5 Modifications to the Model

The model as described worked reliably from a very early stage in its development. There were however, a number of areas where details of the design needed to be improved or alternatives investigated.

5.5.1 Water Leakage

Water leakage was considered especially important, it occurred at three points.

a) Across the paddles

b) At the gate to pier joints when the gates were closed.

c) As the gates changed over at each end of the stroke.

Water which leaks across the paddle lowers the effective head and hence the power output. The greater volume thus required to fill a cell results in higher velocities and hence greater losses through the gates - the gate to pier seal losses also have this effect.

The change over leakage is a loss of potential energy and reduces the overall efficiency of the machine.

Tests were carried out to measure these leakages by comparing the measured discharge with theoretical discharge for a given upstream and downstream depth. This gave a total loss.

The across paddle loss was then estimated by measuring the leakage volume across the static machine for the same upstream and downstream depths as above with the paddles free and again with the paddle edges and bases sealed with mastic.

The gate pier seal loss was measured by repeating these tests with the pier to gate joints sealed with mastic.

Results gave the following mean values:

45% of the theoretical flow leaked away

This was broken down into:- 6% lost across paddles
27% lost at pier/gate joints
12% lost as the gates changed over.

Remedies

Loss across paddles - this was improved by making adjustable edge pieces on the sides and bases of the paddles. These could be individually set to minimise leakage.

Loss at gate piers - this was the largest loss. Several methods were tried including soft rubber sealing strips, loose rubber flaps and rebates for the gate nose/tail to rest in. The rubber seals were found to have only a limited life. The rebate for the gate ends gave the best all round results. (No rebate was made on the floor of the cell - hence losses continued to occur here).

Loss as gate switches over - this is a loss that cannot be completely eliminated. Attempts were made to speed up the gate movement but this absorbed more power without a noticeable reduction in the loss.

5.5.2 Effect of Pier End Shape

In addition to providing a seating area for the gate the pier ends direct the water into and out of the cell. Thus the shape of the end should combine the minimum possible constriction compatible with a good seal for the gate.

Four combinations of pier shape were tested at shown in figure 5.9 a, b, c and d. These pier ends were of wood bolted to the concrete walls and adjusted to ensure that the gate closed against all four points.

Type a was the shape used for initial tests. Type b showed no clear improvement over type a.

Type c showed a definite loss of power at the same heads. This was due to the poor profile presented to the exit water causing a constriction to the flow.

Type d gave better results than either a or b and all reported tests used these shapes of pier ends.

With the gate in place there was a certain amount of constriction. The distance between piers across the opening was 26cm whilst the minimum distance from gate to pier was 17cm.

5.5.3 Friction & Hydraulic Line Losses

These were measured by simply pulling the carriage along with a spring balance.

Results were:-

To pull the carriage, swinging beam and draw oil through the power take off system (but not compress the oil) - 16 kgf.

To move the carriage and swinging beam - rams disconnected 11 kgf.

The above results were obtained after every effort had been made to minimise hydraulic and friction losses. The non return valves in the suction lines were positioned so that the springs could be removed. The feed tank was placed above the rams and 3/4" nominal bore feed pipes used.

The beam and ram end bearings were oiled frequently and checked for any misalignment.

5.5.4 Gate and Carriage Velocities

Transducers linked to the carriage and gates were used to obtain information on velocities and acceleration of these components.

Plots of these are recorded in Figure 5.10.

These show that the carriage rapidly accelerates over the first fraction of the stroke, runs at constant speed and then rapidly decelerates. The major part of the stroke is at constant speed.

The actual time to accelerate and decelerate depends upon the head and working pressure.

The gate moves quickly over the first part of its stroke hesitates then closes more slowly over the last 15%. This appeared to be a function of the gate pier geometry and did not materially change with working pressure.

These plots enabled the optimum point of the stroke to be found for triggering the gates. This point gave the minimum time at which the carriage was at rest as it changed direction at the end of each stroke.

5.5.5 Symmetrical Gates

Early tests were carried out using offset pivot gates. These are suitable for one way flow only. In tidal situations it would be desirable to have a symmetrical machine suitable for ebb and flow generation.

The model gates and piers were modified in August 1983 to this configuration.

A series of tests were run to compare the power output and gate operating torque (as measured by the amount of high pressure oil needed to activate them) with previous results. No significance difference was noted.

The original torque tests had been carried out with flat gates in a straight through flow situation. The model test used a thick gate with flow into and out of the cell.

Conclusion

The STO can be made symmetrical for two way flow operation without loss of performance.

5.5.6 Wider Gate Opening

Tests on gate pier end shape showed the importance of gate opening on machine performance. In September 1983 the gate openings were increased to 33cm (from 26cm) and the machine retested.

Improvements in carriage speed were noted but efficiency fell dramatically.

These tests indicated that the performance of the machine depended upon the relative proportions of the gate opening, stroke and cell length.

Hence it was decided to investigate this aspect of the machine before fixing the proportions of a prototype.

5.5.8 Models of Different Geometry

To test the hypothesis that the geometry of the machine has a significant effect on its performance a series of tests were carried out by modifying the model.

These tests are described in detail in reference 30.

A short stroke model was made by shortening the carriage and reducing the distance between the gates. This gave less power and had a lower efficiency than the original model. This was due to the extra power used for the more frequent gate openings and water losses at each opening.

A narrow width model - that is one with a relatively longer stroke for its width - was made by moving the barrage walls in. The results for this were then compared with the original model using Froude law scaling on the ratio of widths. This machine had longer cycle times, operated more slowly and needed a higher operating pressure.

The power envelope was similar but fall off in power with increasing head was more noticeable.

For this reason the original geometry was considered to be superior

5.6 Characteristic Performance Curves

A graphical representation of the laboratory test results is given in Figure 5.11 where power is plotted against head - the control parameters being pressure and downstream depth. The lines consist of three sets -

each at a different downstream depth. Within each set, the different symbols represent the system pressures which vary between 28 bar and 105 bar in 7 bar intervals. These results are shown characteristically in Figures 5.12 and 5.13.

It can be seen that for any value of head and downstream water level, it is necessary to place a specific load on the paddles to ensure maximum power output. The applied load is a product of ram area and system pressure, thus, since for any given system, ram area is constant, pressure is the controlling factor. Further, since pressure determines speed of gate operation and applied load determines period of carriage oscillation, the discharge is also specified at any point within the diagram. A series of these diagrams can be produced for different values of downstream water level, and thus the operating parameters for the STO under any condition may be specified.

A STO operating characteristic to give maximum power under any condition can be computed by producing a family of maximum power envelopes for different downstream levels (Figure 5.12), noting variation in applied load and discharge along each curve and then superimposing each curve onto a single diagram (Figure 5.13).

This characteristic performance curve can be used to predict the performance of any geometrically similar machine. A machine of different geometry - that is cell width to length would require the generation of another characteristic curve.

The laboratory results gave a maximum efficiency - oil to water of 40%. The efficiency at maximum power was 35%.

The theoretical efficiency - oil to stroked water flow was as high as 70% - this can only be achieved if all water losses are eliminated.

5.7 Prototype Designs

A number of sites were investigated concurrently with the laboratory work. At two of these sites, Adelphi Weir and Sponish, site surveys were carried out, bore hole samples taken and detail design work started.

A full record is contained in Reference 30.

5.7.1 Adelphi Weir

This site is on the river Irwell adjacent to the University of Salford.

The proposal was for a machine capable of producing 30kW at a 2m head.

Unfortunately the Irwell is subject to very rapid rises in water level due to urban run off. This has resulted in flooding of domestic properties in the past.

Water Authority restrictions imposed on the construction of a STO at this site finally made it uneconomic to proceed.

5.7.2 Sponish

This site is at a tidal inlet on North Uist in the Outer Hebrides.

The proposal was for a 270kW machine supplying the local hospital. Substantial support was obtained for the machine at this site from the EEC Energy Demonstration Fund.

Design work was carried out by Crouch and Hogg - Consulting Engineers and James Howden and Company Limited of Glasgow.

Although a tidal site gave the opportunity to test the prototype at a variety of upstream and downstream water levels it meant that the machinery had to withstand a very hostile environment. Furthermore the control of the oil pressure to ensure that maximum efficiency was obtained at all stages of water was a major problem.

These factors made a large contribution to the cost of the installation. The price of energy from the STO at Sponish was calculated at 7.3p/kWh this compared with a buy in rate of 2.1p/kWh from NSHEB.

Thus for economic reasons it was decided not to go ahead with the project.

A detailed analysis of the computation of the energy output of the STO at Sponish is given in Reference 32.

5.8 Future Work

Recent developments in the control of generators by electronic means make it possible to produce a steady controlled supply from a relatively unsteady prime mover.

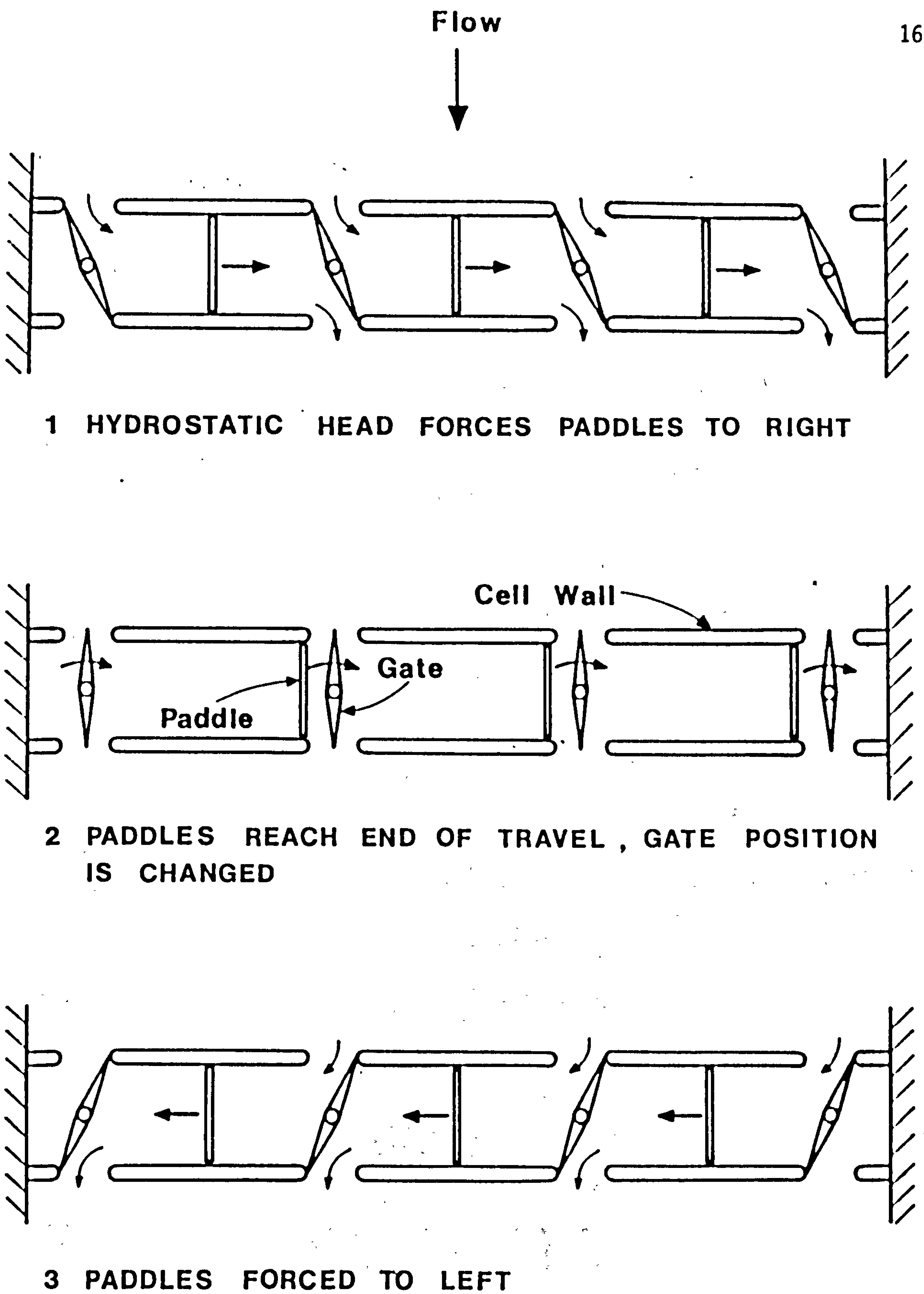
Work is presently being carried out in the Department of Electronic and Electrical Engineering at the University of Salford on such a system for use on a STO. This would enable electricity to be produced directly without the intermediate expense and complication of a hydraulic system.

Figure 5.14 shows a proposed configuration for such a machine with a generator mounted on the barrage driven by parafil cables. This design eliminates the complication of the push-pull rod on the paddles thus simplifying the gate system.

A further development is the use of gates which open across the flow instead of along the flow direction. This gives less loss of water at gate changeover and hence higher efficiency. Control flaps at the leading faces of the gates have been devised to harness the energy of the water flowing past the gates to assist in opening and closing the gates.

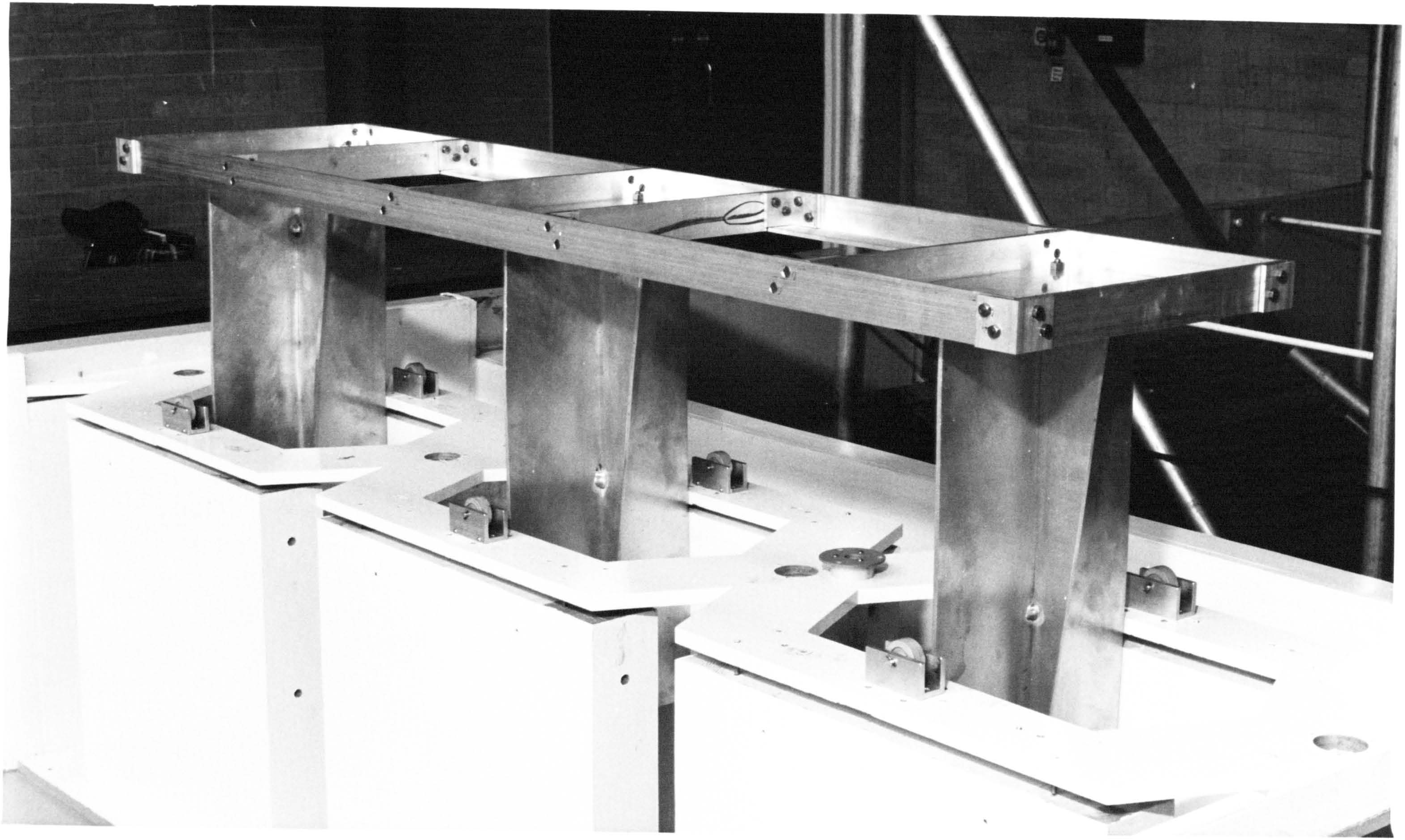
These factors all contribute to reducing the cost of an installation and will be laboratory tested during 1989.

It is hoped that EEC funding will be made available to construct a 20kW prototype to this design. Only after the first prototype has been built and operated for some time will it be possible to judge the STO against more conventional small head hydro machinery.



OPERATION OF THE STO

Figure 5.1

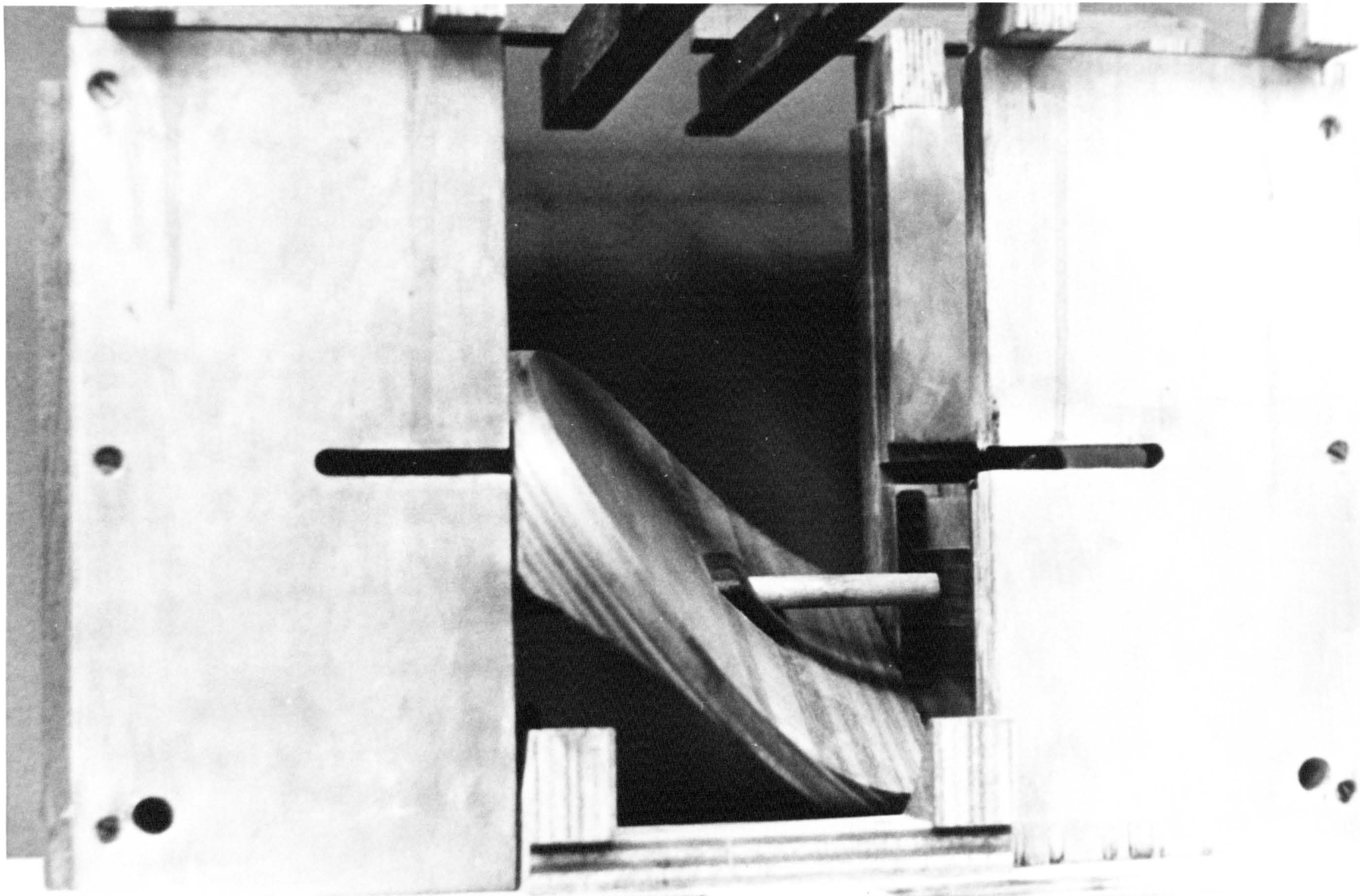


STO - First laboratory model cariage

Figure 5.2

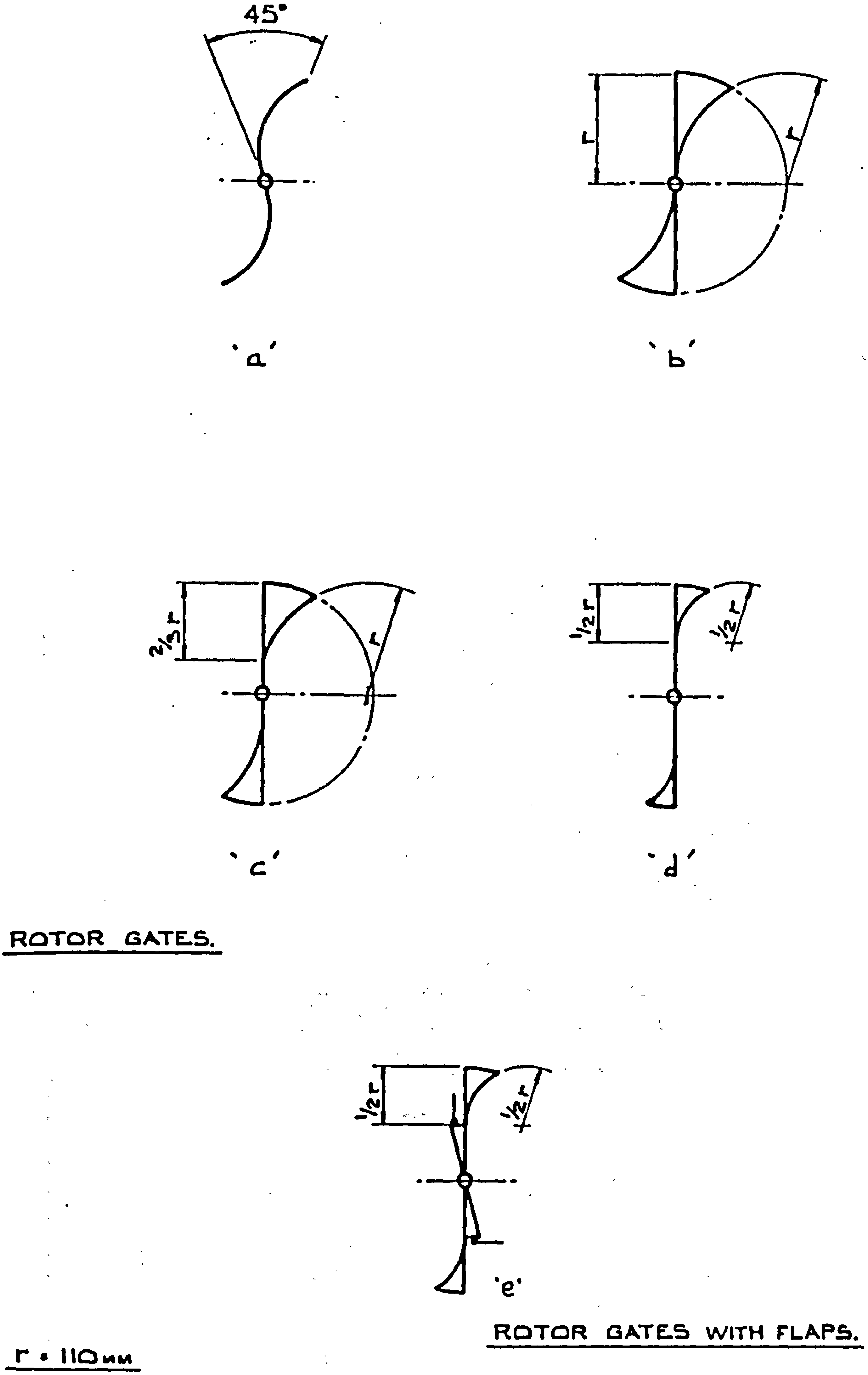
First laboratory model STO

Figure 5.3



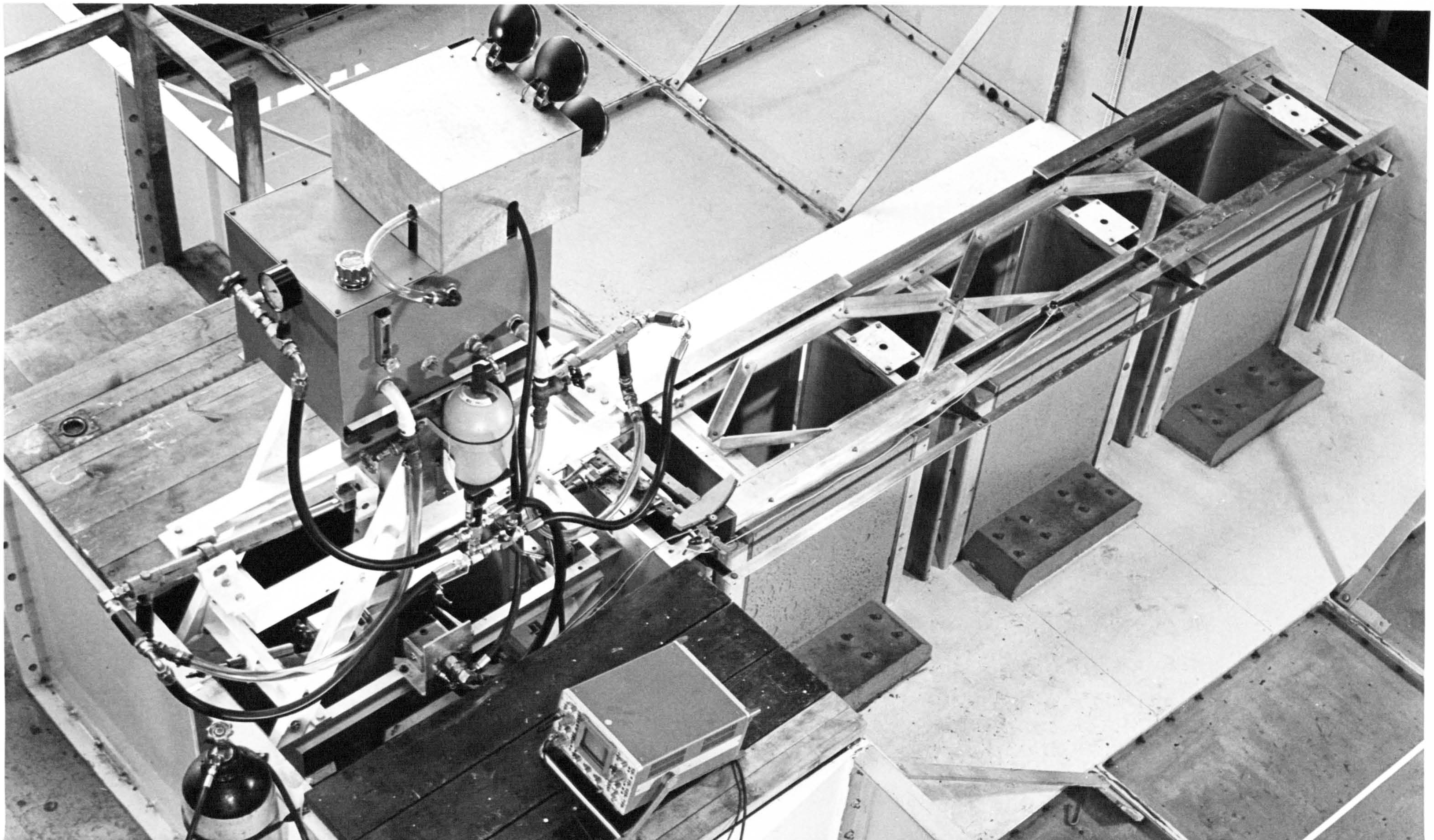
Model of a horizontal axis gate

Figure 5.4



Types of rotor gate tested

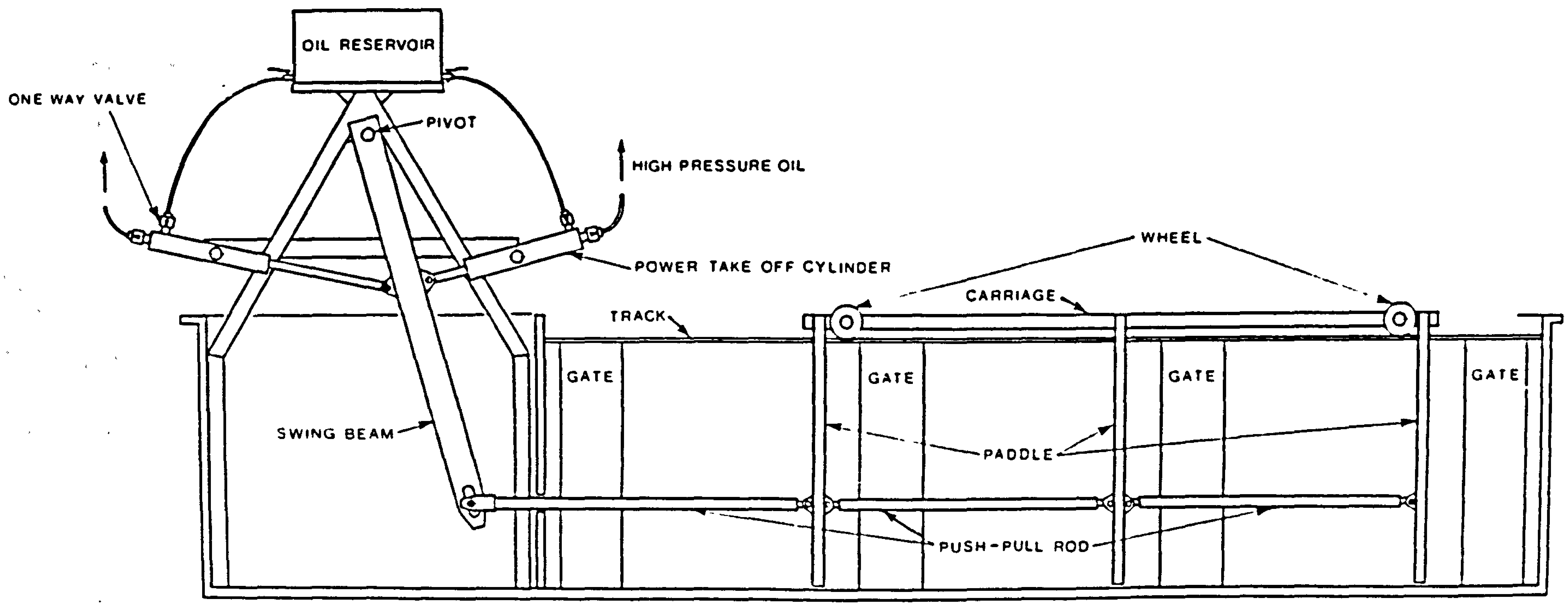
Figure 5.5



View of the second laboratory model

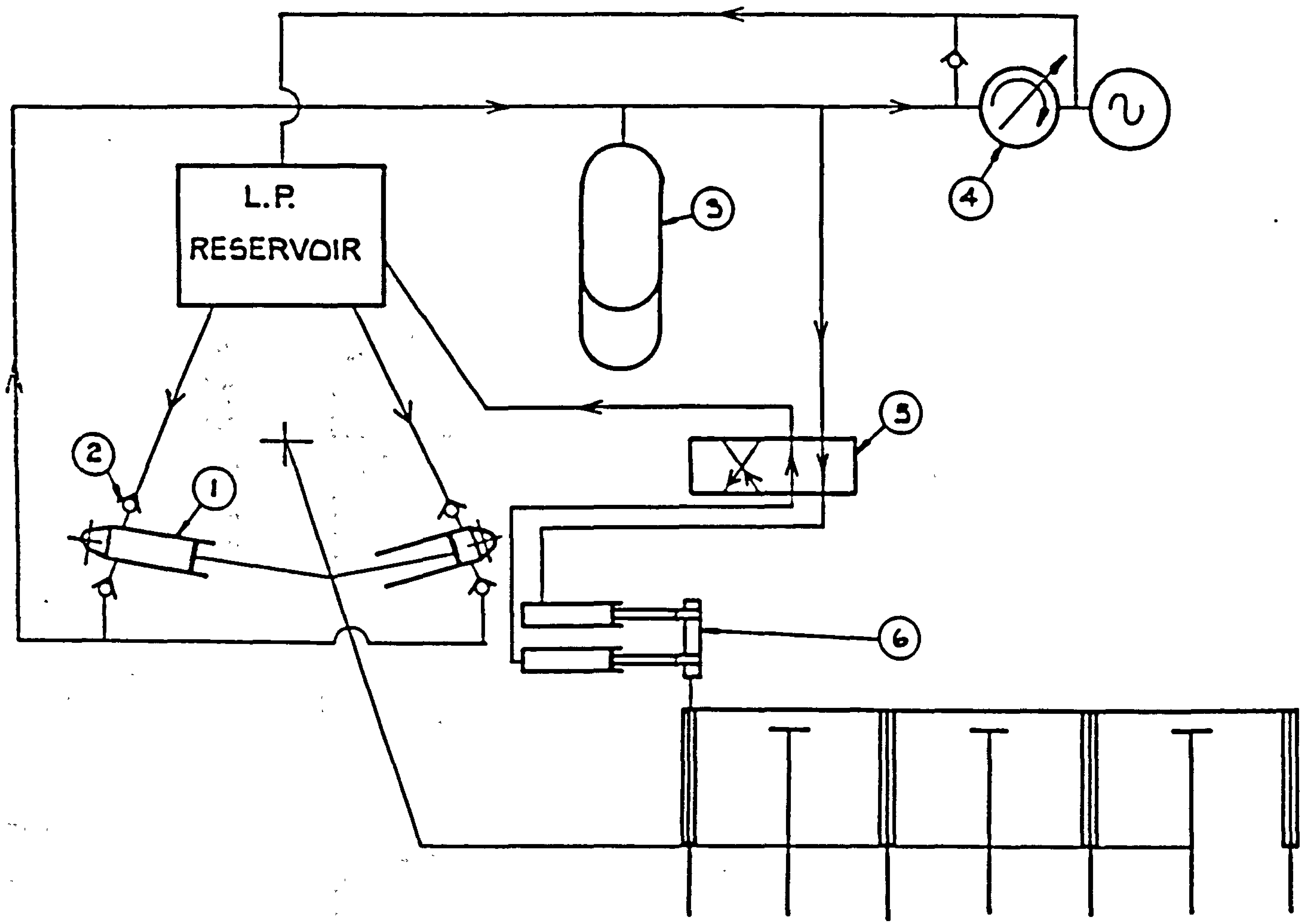
Figure 5.6

THE SALFORD
TRANSVERSE OSCILLATOR



Arrangement of the second laboratory model

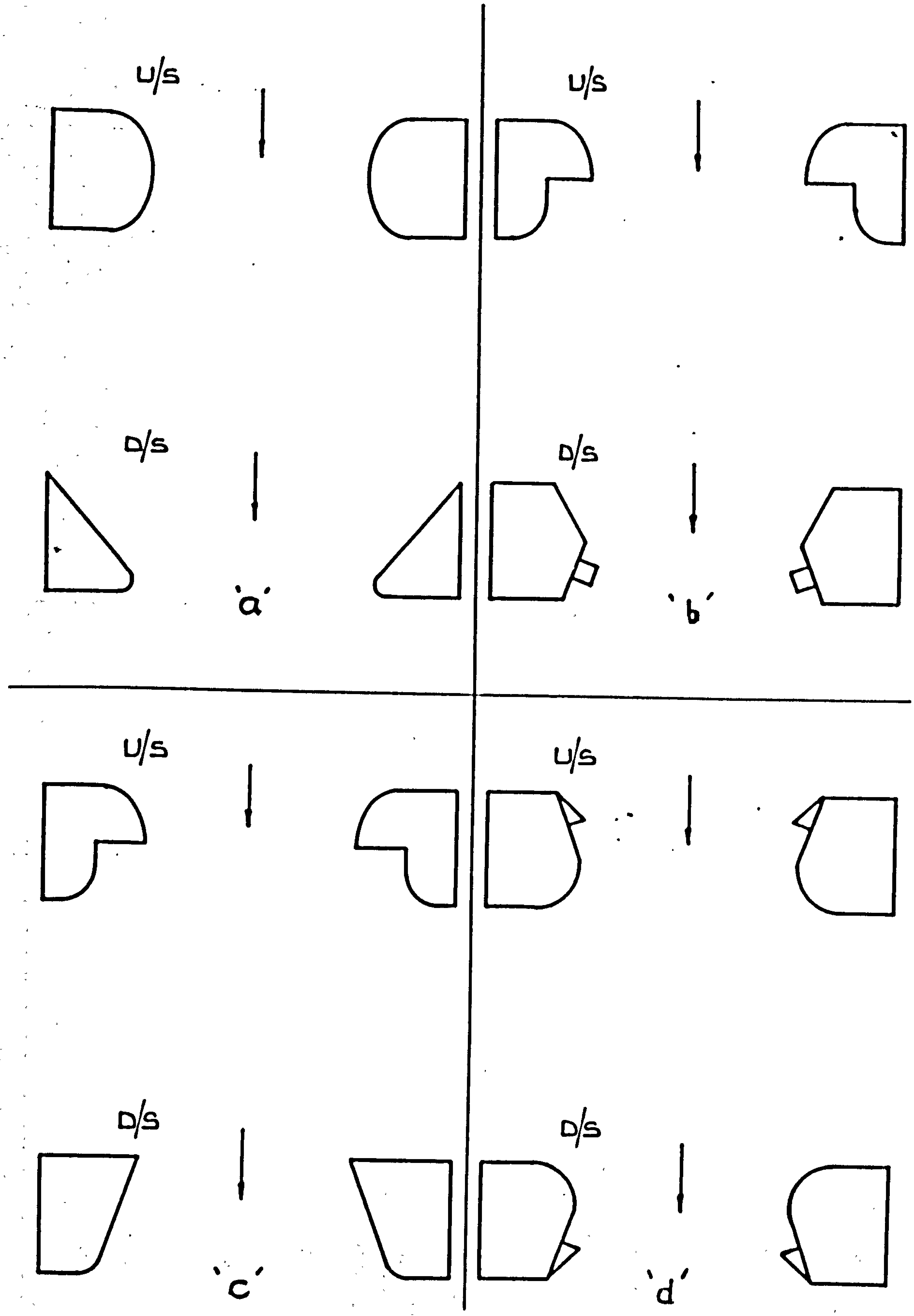
Figure 5.7



ITEM	TITLE	QTY.
1	POWER TAKE OFF CYLIDER.	2
2	NON RETURN VALVE	4
3	ACCUMULATOR	1
4	MOTOR	1
5	SPOOL VALVE	1
6	LINEAR ACTUATOR	2

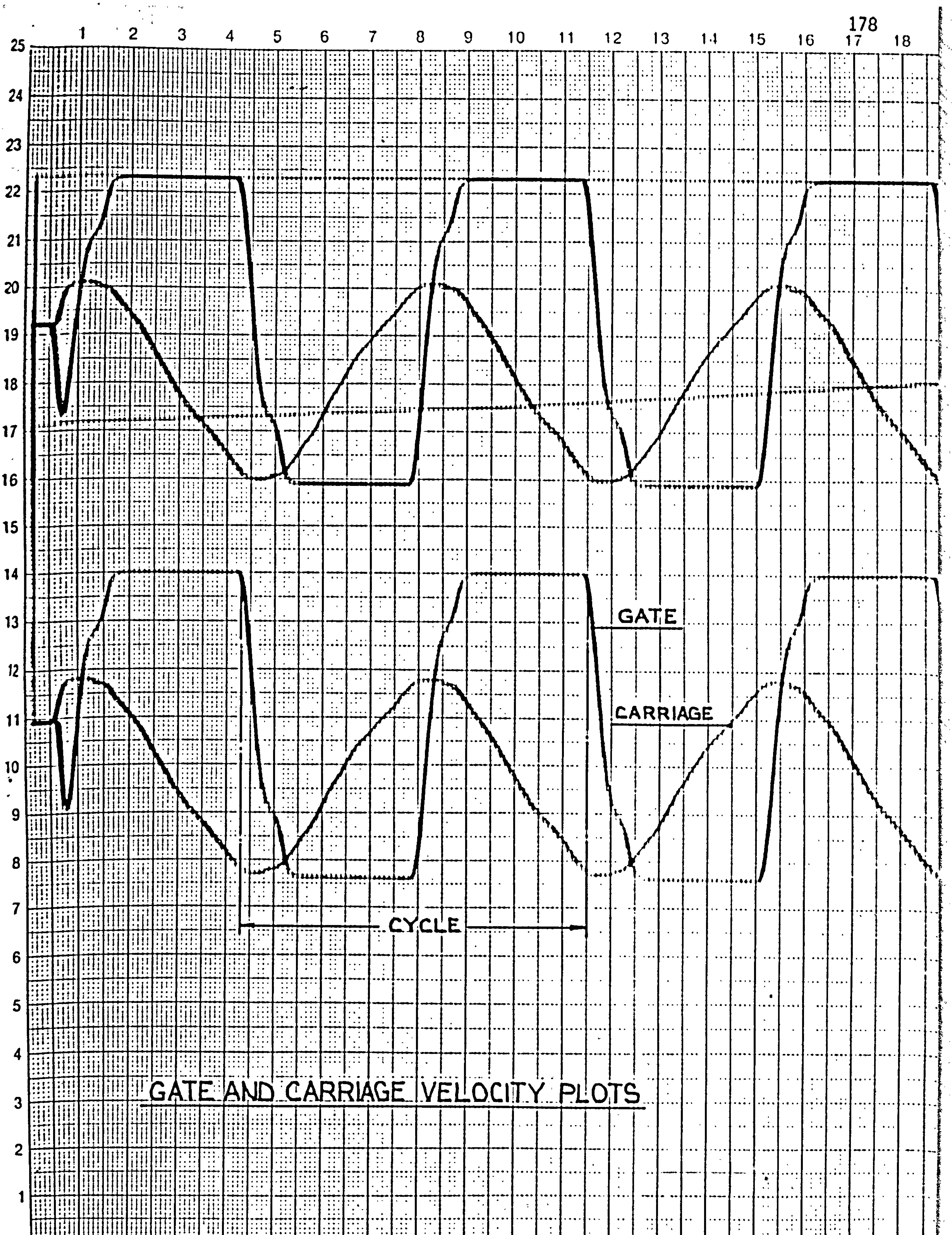
Hydraulic circuit, second laboratory model

Figure 5.8



GATE PIERS

Figure 5.9


GATE
CARRIAGE
CYCLE
GATE AND CARRIAGE VELOCITY PLOTS
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
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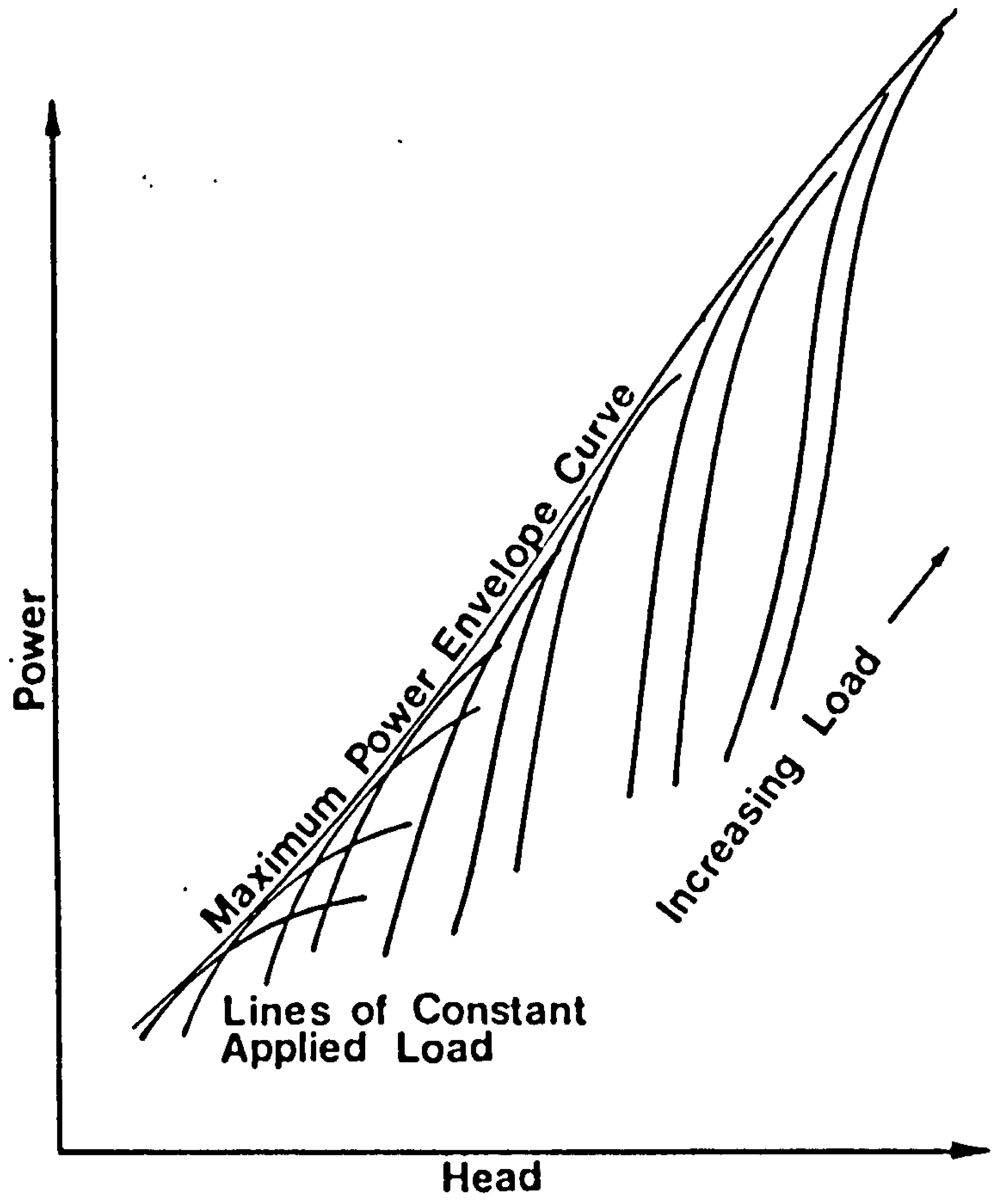
24
25

Figure 5.10

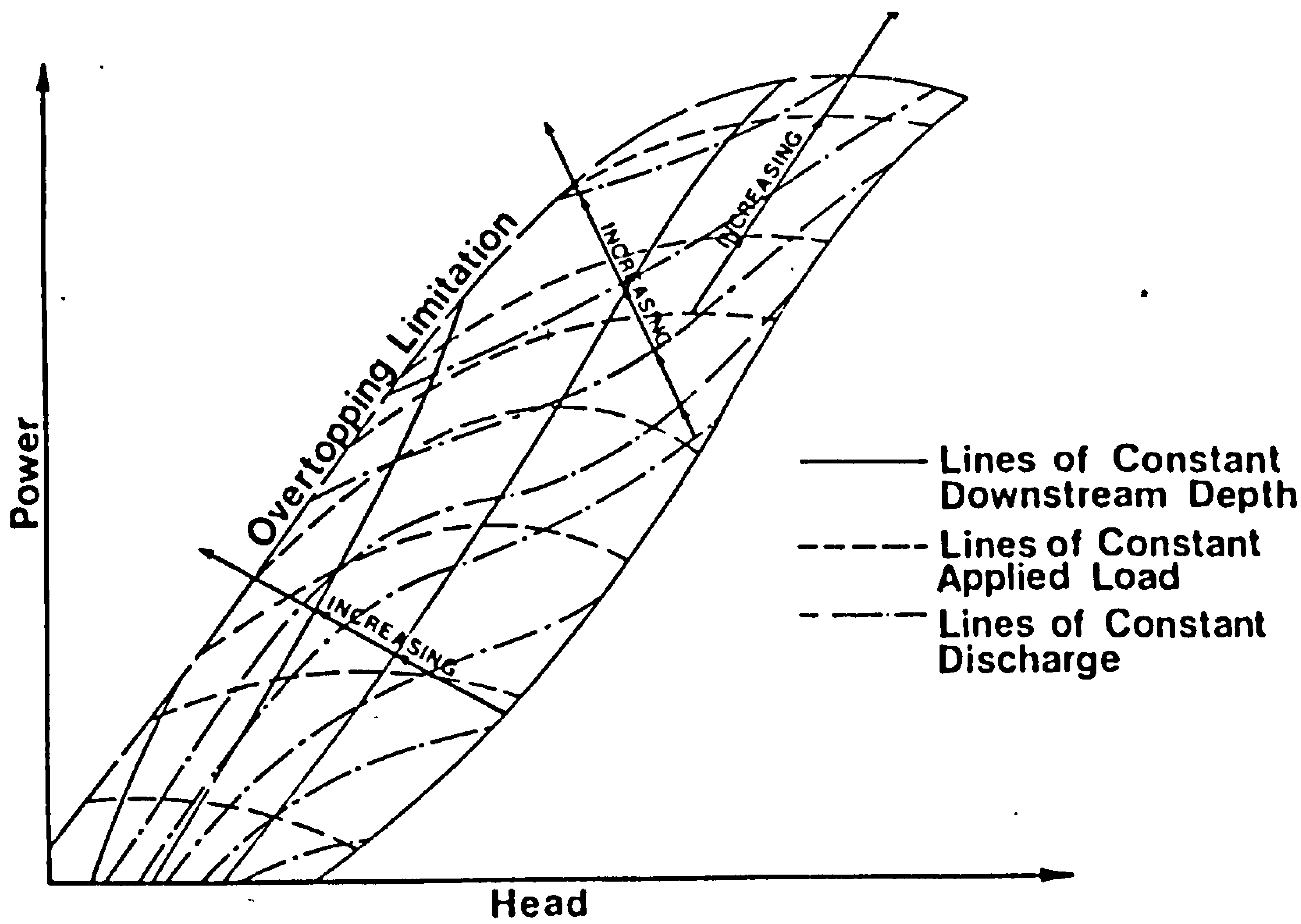
POWER V.S. HEAD -- D/S = 300,400 & 500

Figure 5.11



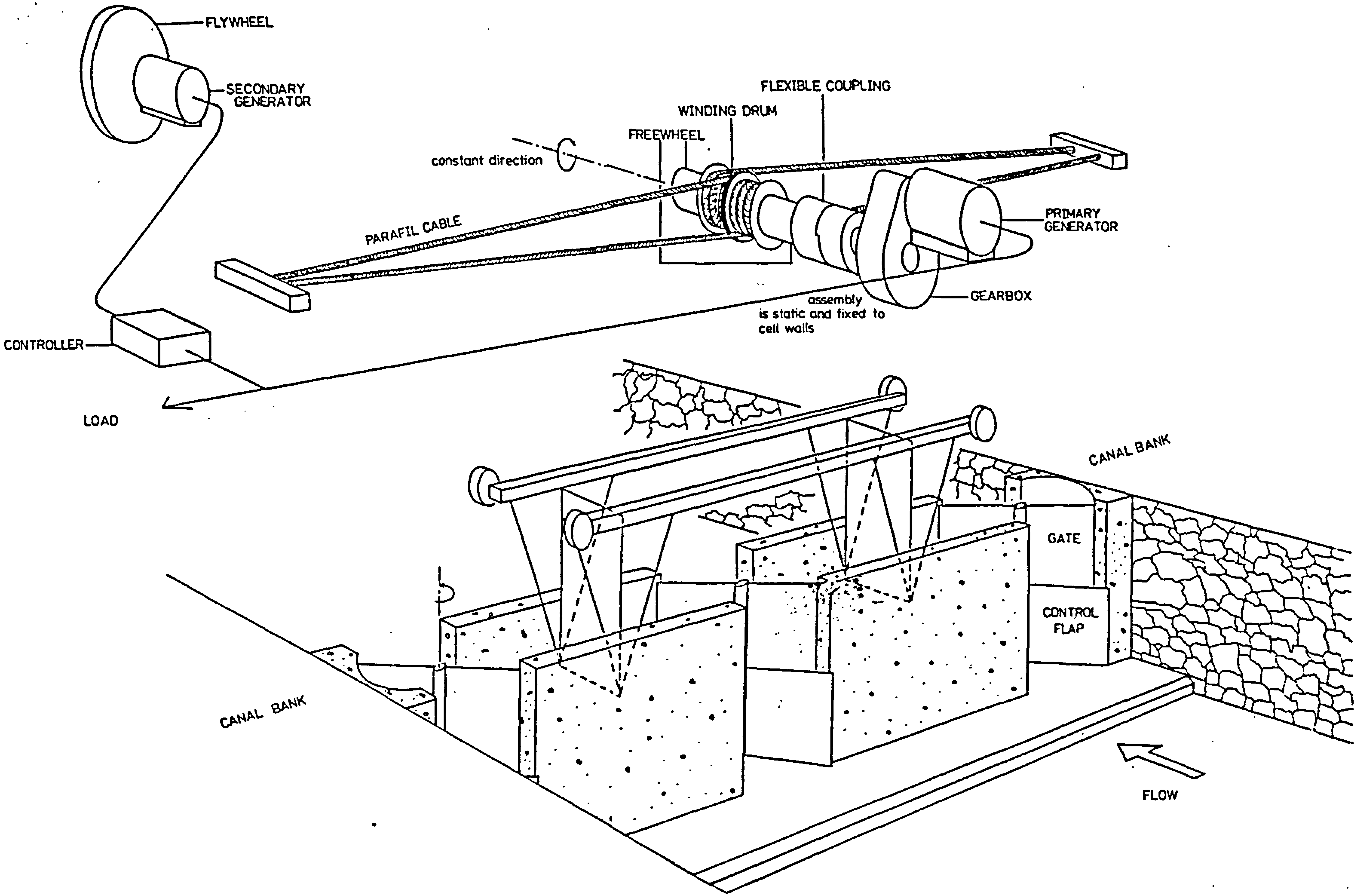
STO Power Output Against Head for Constant Downstream Depth and Varying Applied Load

Figure 5.12



STO Operating Characteristics

Figure 5.13



Proposed STO with cable drive to generator

Figure 5.14

CHAPTER 6

The Under-Water Motor

6.0 Introduction

The results of the work on the AUR Water Engine and the STO have convinced the writer that for small scale low head sites a simple machine that rotates rather than reciprocates is needed.

This machine is an attempt to combine the simple technology of the water wheel with the compact size and higher rotational speed of a cross-flow turbine.

The major drawbacks of the water wheel are:-

a) Bulk - The large components make transport to remote sites difficult and complicate erection.

b) Low rotational speed - whilst few low head hydro-power devices can be coupled directly to a standard electric generator, the slow speed of a water wheel means that step up ratios of the order of 150 to 1 are required.

c) Loss of efficiency at changes in water levels - overshot water wheels do not benefit from increases in the upstream water level; breast shot and overshot wheel performance is reduced by a rise in downstream level when the paddles trail in the tail water. If the wheel is set above the tail water to minimise this then potential power is lost.

The crossflow turbine has the following disadvantages at low heads:-

a) The runner must run in air for best effiency - thus there is a substantial loss of working head relative to the available head, even when a draught tube is fitted.

b) Long runners are necessary to pass the required flows at low heads - this results in mechanical problems and short life.

6.1 Design Criteria

To be a successful low-head generator suitable for use and manufacture in developing countries the following are essential points:-

a) Simplicity

The basic workings of the machine should be easily understood by local craftsmen.

b) Suitable materials

The materials of which the device is constructed should, ideally, be readily available in developing countries. At the very least they should be materials that can be repaired and maintained by local skills.

c) Robustness

Although the equipment may be essential to improve their way of life, it is unfortunate that local people do not always have the training and temperament to properly maintain hydro equipment and ancillaries such as trash screens and silt traps. Hence it is necessary that the machinery is robust enough to withstand poor maintenance and the passage of detritus without breakdown.

d) Maximum efficiency

Whilst it may be argued that some power is better than no power at all it cannot be argued that potential power should be wasted. Hence a low head hydro device should be designed to achieve maximum possible efficiency to ensure that the best use is made of the materials transported to remote sites. An improvement of 5% on a 10kW machine could mean that five more homes are able to have an electric light.

6.1.1 The Basic Design

The Underwater Motor is based on similar design principles to hydraulic and pneumatic motors of the moving vane type. Pumps and motors of this design have been known for many years but used only for relatively small volumes of fluid. Figures 6.1a and 6.1b show simplified sections through the machines.

One of the first records of a sliding vane pump is shown in Le Diverse et Artificiose Machine by Agostino Ramelli published in 1588. Many of the drawings in Ramelli's book were speculative and the machines shown did not exist at the time. Indeed his vane pump is shown connected directly to a water wheel. It is doubtful that this pump would work at such slow speed without some mechanism for moving the vanes out of the rotor.

This sliding vane pump of Ramelli's predates Dennis Papin's fixed vane centrifugal "Hessian" pump by a hundred years.

Sliding vane pumps are positive displacement machines. The rapid development of centrifugal pumps in the nineteenth century closed the gap between high pressure piston pumps and large volume lower pressure centrifugal pumps so that there was very little call for the rotary sliding vane pump in large sizes. Hence they were generally used for small volume medium pressure applications such as portable pumps for fire fighting.

In the fields of air compression and oil hydraulics vane pumps and motors are used whenever an inexpensive small machine is required. Perhaps the largest pumps of this type are the exhausters - once widely used for evacuating gas holders in the days of manufactured gas for industrial and domestic use. These machines, often called Beale exhausters after their inventor, usually have only two vanes and can deliver gas at up to 0.5 bar pressure.

A more complicated gas pump, the crescent chamber pump, is of similar design but can have up to twenty four vanes. Machines of this type will deliver gas at pressures of up to 4 bar.

Thus, although the principle is well known, there does not appear to be evidence of any attempt to harness hydropower using such a machine. A patent application has been filed in the writer's name for the device and the searches carried out in patent office records have yielded no other device that resembles it in application.

6.2 The Small Model

A small scale model, approximately one fifth full size was made to carry out preliminary tests in the hydraulics laboratory. This model has a rotor 100mm wide and 110 mm diameter. The working head is 200mm.

Two rotors were made for this model one with straight vanes one with curved vanes, see figures 6.2 and 6.3.

6.2.1 Straight Vane Rotor

The rotor for this machine is made from solid nylon bar with four slots, machined at 90° to each other. PVC vanes slide in these slots. The rotor sits in a basically circular casing but its axis is displaced such that the gap between rotor and casing is very small at the one side but large at the other. As the rotor revolves the vanes move out of their slots and protrude into the space between the rotor and casing. Water flowing from the upstream level to the downstream level pushes on these vanes thus turning the rotor. Energy is taken out via a pulley fixed to the axle of the rotor outside the casing.

Initial tests were carried out with the vanes free to extend from the rotor by centrifugal force alone as the rotor turned. It was observed, however, that a considerable gap was left between the casing and the ends of the vanes allowing a large volume of water to pass through the machine without doing useful work. (Hydraulic vane motors and air vane motors work in this fashion, but their very high rotational speeds - usually several thousand rpm - mean that the forces holding the vanes out against the casing are much higher).

An obvious solution was to set springs at the inside edge of the vanes to push them out of the rotor. This was rejected however as being unsuitable for a full size machine where considerable energy could be lost in the friction that would occur between vanes and casing.

The solution adopted was to fit a pin in the outside edge of the vane running in a groove cut into the casing side. This constrained the vane to move in and out of the rotor in the manner of a tappet following a cam. The end clearance between vane and casing could thus be as small as the accuracy of machining would allow without touching and hence frictional losses could be minimised.

6.2.2 The shape of the vane control groove

Initally the groove was made circular, concentric with the casing, so that the ends of the vanes followed the casing exactly.

Observations during testing showed that considerable churning of the upstream water was occurring as the vanes emerged from the rotor. Modifications to upstream entry geometry by ducting the water so that it flowed in the direction of the vanes did little to improve the situation.

The profile of the groove was then altered to give it an elliptical shape by flattening off the top and bottom of the circle. This retained the vane inside the rotor during the time it would have been churning the head water, allowing it out only when required to close off the space between rotor and casing.

Similarly at the downstream side of the rotor the vane is swiftly drawn back into the rotor to allow the water to escape with minimum impedance.

Several shapes of groove were tested to obtain the optimum shape. The final shape being that used for the half size machine described later.

6.2.3 Curved Vane Rotor

Superficially the curved vane rotor has several advantages over the straight vane rotor. These are:-

a) The rotor is a simple cylinder and hence easier to make than the slotted rotor.

b) If four vanes are used the length of each can be almost $\pi/2$ x radius. The straight vanes can only be a maximum length equal to the radius. Thus a curved vane machine could pass more water for a given size of rotor so having the potential to develop more power.

c) The friction losses at the pivot point are less than the sliding friction losses for the straight vanes.

Initial tests with this rotor allowed the vanes to swing freely, relying on centrifugal force to hold them against the casing. This did not work. The rotor rotated but the vanes did not touch the casing sides so allowing water to flow unimpeded through the machine.

Pins were then introduced into the ends of the vanes, these ran in slots in the side casing in the same way as the straight vane machine.

This worked insofar as the vanes now completely filled the gap between rotor and casing. It was noted however that the device ran very unevenly. Investigation using a video recording showed that the uneven running was caused by the vane unfolding from the rotor acting as a brake on the motion. Changes were made to the groove shape in an attempt to reduce this. No significant improvement was obtained and as a result it was decided to abandon work on this rotor and concentrate efforts on the straight vane one.

6.2.4 Testing of the Model

The model was tested in the 150mm flume in the hydraulics laboratory.

As modifications were made to the machine the relative effect was judged by the increase or decrease in rotational speed at a given head. No attempt was made to measure the flow of water through the machine.

The head of water across the model was maintained at 200mm by use of the overflow weir at the flume discharge and by controlling the water entering the flume with a gate valve in the inlet pipe. Generally the upstream water level was 280mm above the base and the downstream level 80mm above the base.

A summary of the tests carried out and speeds achieved is given below:

1. Circular side groove - 50 rpm
2. Eliptical top part to groove - 88 rpm
3. Eliptical top to groove with pressure relief holes in the downstream side of the casing - 116 rpm
4. Eliptical top and bottom to the groove - 139 rpm
5. As above but without a curved plate at entry to the rotor to guide the water into the machine - 118 rpm

Test number 3 was carried out because it was noted that with a circular bottom section to the groove a small amount of water was being trapped by the vanes and forced round the small gap between rotor and casing.

This caused a loss of power. It did, however, also indicate that by suitably shaping the groove and casing the motor could become a combined pump and motor. Water was pumped through a pipe fixed to one of the pressure relief holes to a height of 420mm above the base of the flume. This idea was not pursued beyond this simple test.

6.2.5 Power Measurements

Very rudimentary power measurements were made by fixing a simple pulley and weight dynamometer to the rotor axis as shown in figure 6.4.

This power was compared with the potential power in the water flow through the machine to give an efficiency. This efficiency is based on the geometric water displacement of the machine - not the actual flow which would include leakage.

Diffuser

A simple retangular diffuser 210mm long with an angle of expansion of 6° was added after the initial tests.

This resulted in a significant increase in power - over 10%. A further series of tests were conducted at heads of 130mm to 200mm. The results are shown graphically in figure 6.5.

6.2.6 Discussion of Results

The results show that maximum power occurs at an efficiency of about 50%. The maximum efficiency of 63% occurs at a lower speed.

At a head of 200mm the velocity of a free water jet would be $\sqrt{2gh} = \sqrt{2g \times 0.2} = 1.98$ m/second.

The radius of the vane tip as the rotor rotates is 93mm. Hence the velocity of the vane tip V, is wr.

At a velocity of 1.98m/sec., w = V/r
= 1.98/0.093
= 21.3 radians/sec.
= 3.39 revs/sec.
= 203 rpm

Thus the maximum velocity (runaway speed) of the rotor ignoring losses, would be about 200 rpm at a head of 200mm. Impulse machines such as Pelton wheels and crossflow turbines operate at or just below 50% of the maximum runaway speed.

Maximum power of the underwater motor at 200mm head occurred at 75 rpm or at 37% of runaway speed.

6.3 The half size model

The inlet and outlet water passages of the motor are a simple rectangular shape that can easily be matched to a water channel using simple sheeting or wooden planks. The motor itself can be made to fit into a rectangular box with only the power take-off pulley exposed. In this way the machine could be taken to site and dropped onto a prepared base with the minimum of difficulty.

It was thus decided to carry out further tests on a half size model of a motor that could easily be carried to remote locations but would give a useful amount of power at heads of between 1 and 2 metres.

Assuming an efficiency of 0.6, a vane tip speed of $0.4\sqrt{2gh}$ and a design similar to the one fifth scale machine the dimensions for a full size machine could be:-

rotor diameter 0.60m
rotor length 0.90m
vane tip radius 0.54m

This would give an estimated power of 5kW at 45 rpm on a 2m head, passing 0.43 cumecs.

At one metre head the power developed would be 2kW.

A half size model of the above was designed and figure 6.6 is an arrangement giving the main dimensions. Figure 6.7 is a photograph of the completed model.

6.3.1 Design details of the half size model

Whilst the model was built to obtain more performance data it gave an opportunity to test the effectiveness of design details and materials.

Rotor - This is fabricated from galvanised sheet fastended around circular end formers made of steel plate. A steel tube axle with brass journals at each end is welded through the centre of the end plates.

The vane slots in the rotor have thin brass angles at each corner for the vanes to slide in. These slots are not radial but each passes under the axle and the end butts against the side of the adjacent slot. This gives the maximum slot length. It was initially intended to fill the rotor with concrete to increase its inertia and so minimise any uneven running. This was found to be unnecessary.

Vanes - These are of 12mm thick Fibrelam - a glass reinforced epoxy resin sheet with an aluminium honeycomb core. This material gives an extremely stiff but light vane. A light vane minimises the forces on the side grooves and rollers. Tufnol edging strips are used on these to run against the brass angles in the rotor slots.

Guide rollers - These are of Nylatron, nylon 66 filled with molydendum disulphide, running on brass pins. The brackets holding the pins are bolted to the vanes. These brackets can be easily unbolted thus allowing a vane to be withdrawn from the rotor without further dismantling. The rollers have conical ends to minimise friction on the casing sides.

Casing - This is in two pieces, flanged and attached to the casing sides with bolts.

Casing sides - These are of perspex to allow the water flow inside the machine to be observed. The side groove is made by bolting a second layer of perspex to the inside of the casing the shape of groove having been cut out of this second layer using a band-saw. This system makes changing the shape of the groove possible by replacing the inner layer of perspex.

Diffuser - This is of galvanised steel sheet, the expansion angle is 7° with an area ratio of 1 : 2 - inlet to outlet.

Main bearings - The bearings carrying the axle are standard Dulite bearings suitable for running with water lubrication. They are in cast iron self aligning housings bolted to the outside of the casing.

6.3.2 Testing of the Model

The model is sited in a large tank constructed to carry out tests on the Salford Transverse Oscillator. The tank is 3.7m wide by 7.2m long by 1.2m high. The model STO is built across the middle of this tank.

To accommodate the Underwater Motor the middle gate of the STO has been removed and an inlet chamber constructed from galvanised sheet on the downstream side of the gate opening. The motor is mounted below the inlet chamber as can be seen in figure 6.8.

The upstream water level can be controlled by means of valves on the inlet pipes. The downstream water level is controlled by raising or lowering the height of the boards on the overflow weir at the discharge end of the tank. The outgoing water falls into the sump beneath the laboratory floor.

The power output of the model is measured by means of a simple Prony dynamometer pulley running at twice the speed of the motor.

Speed is measured by timing a number of revolutions with a stop watch. Upstream water level and downstream water level are measured by scales fixed adjacent to the inlet chamber and on the downstream wall of the tank.

No attempt was made to measure water flow through the machine by direct means. Instead it was decided to use the geometric flow plus leakage calculated after measuring the gaps between rotor, vanes and casing.

This was considered to be more accurate than either measuring the discharge from the diffuser or by trying to measure the inflow into the machine.

The results below use efficiencies calculated from the geometric flow plus calculated losses.

Initial tests were carried out at a fixed discharge from the supply pumps, this however did not give a consistent set of results because of the variation in leakage rate across the barrage.

The later tests were carried out at fixed upstream and downstream levels the flow being adjusted as necessary. Heads of 0.2m, 0.3m, 0.4m, 0.5m were tested with downstream depths of 0, 0.2, 0.3 and 0.4m.

The results are given in appendix 3 and graphically in figures 6.9 and 6.10.

6.3.3 Tolerances of debris in the water

Any object smaller than the width of the water passage can pass through the machine without causing damage. This was demonstrated on the half size model by dropping stones, pieces of wood, plastic and balls of paper into the machine. Most passed through without problem. The paper however stopped the machine on several occasions when it was struck by an emerging vane and trapped between the end of the vane and the casing. This did not occur with hard objects as they tended to be deflected off the vane end rather than be trapped by it.

It may be concluded that stones and any other detritus would not be a problem. Polythene bags and other thin materials would also pass through, even if trapped under a vane end. Other thicker materials, such as brushwood, would almost certainly stop the machine. Hence a full size machine would require a coarse grill to stop brushwood entering it. Leaves would not be a problem and could pass through.

6.3.4 Results

The results of the tests are presented graphically in figures 6.9 and 6.10.

Figure 6.9 shows the variation of power with speed at different heads.

The efficiency of the machine increases as the speed reduces. Lines of constant efficiency drawn across the head curves illustrate this trend.

Maximum power occurs at an efficiency of between 50% and 60% for all heads. Maximum power occurs at about 45% of the runaway speed as shown by the table below:

Head m	Runaway speed rpm	Speed at max power rpm	% of runaway
0.2	70	35	50
0.3	86	40	46
0.4	99	45	45
0.5	110	48	44

6.4 A Prototype Machine

Figure 6.10 can be used to obtain the basic dimensions of a full size machine.

If a working point with an efficiency of around 60% is chosen - say

$$\frac{Q}{D^2\sqrt{H}} = 1.0 \qquad (1)$$

when

$$\frac{ND}{\sqrt{H}} = 16.8 \qquad (2)$$

say H is 2m and maximum flow is 0.5 cumecs

from 1 above $D^2 = \frac{Q}{\sqrt{H}} \times \frac{1}{1.0}$

$$= \frac{0.5}{\sqrt{2}} = 0.354$$

thus D = 0.595m, - say 0.6

from equation 2 rotational speed $N = 16.8 \times \frac{\sqrt{H}}{D}$

$$= 16.8 \times \frac{\sqrt{2}}{0.6}$$

$$= \underline{39.9 \text{ rpm}}$$

Power output = 0.5 x 2 x 9.81 x 0.6 kW

$$= \underline{5.9 \text{ kW}}$$

The rotor length would be in the same ratio as the model length to rotor diameter. That is 1.5:1.

Hence the length is 0.9m

A major drawback to this machine is the low rotational speed. To drive a 1500 rpm generator a step up of 37:1 is needed.

This could be obtained using a gear box or two pulley steps of 6.1:1. The pulleys, preferably with flat belts to minimise losses, would be better for developing countries.

If a standard machine of one size were built - say with a 0.9m long rotor, then greater flows could be accommodated by joining several machines together with simple couplings. Turndown could then be achieved by uncoupling and blanking off sections at times of low flow.

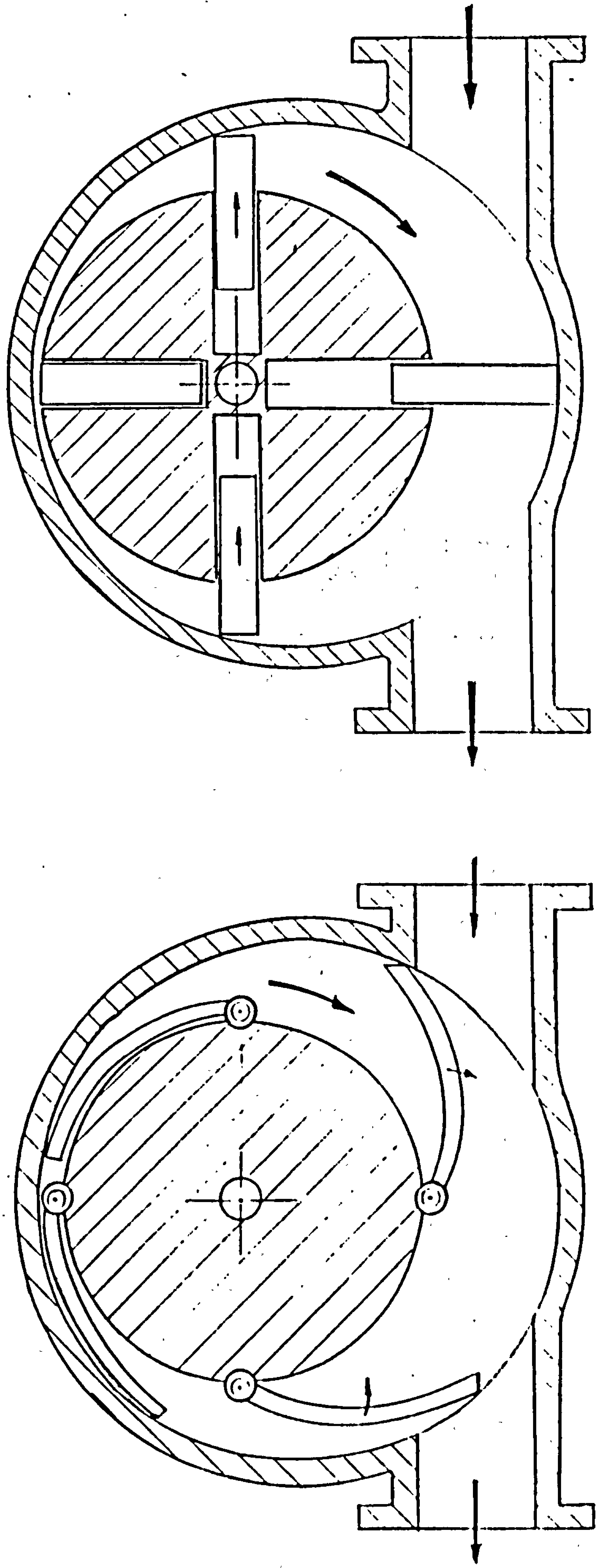
6.4.1 Prototype Materials

The point has been made that machines for developing countries should use locally available materials or materials than can be worked locally.

The Underwater Motor rotor and casing end plates could be made in cast iron. The rotor being made in four sections to simplify the machining of the slots.

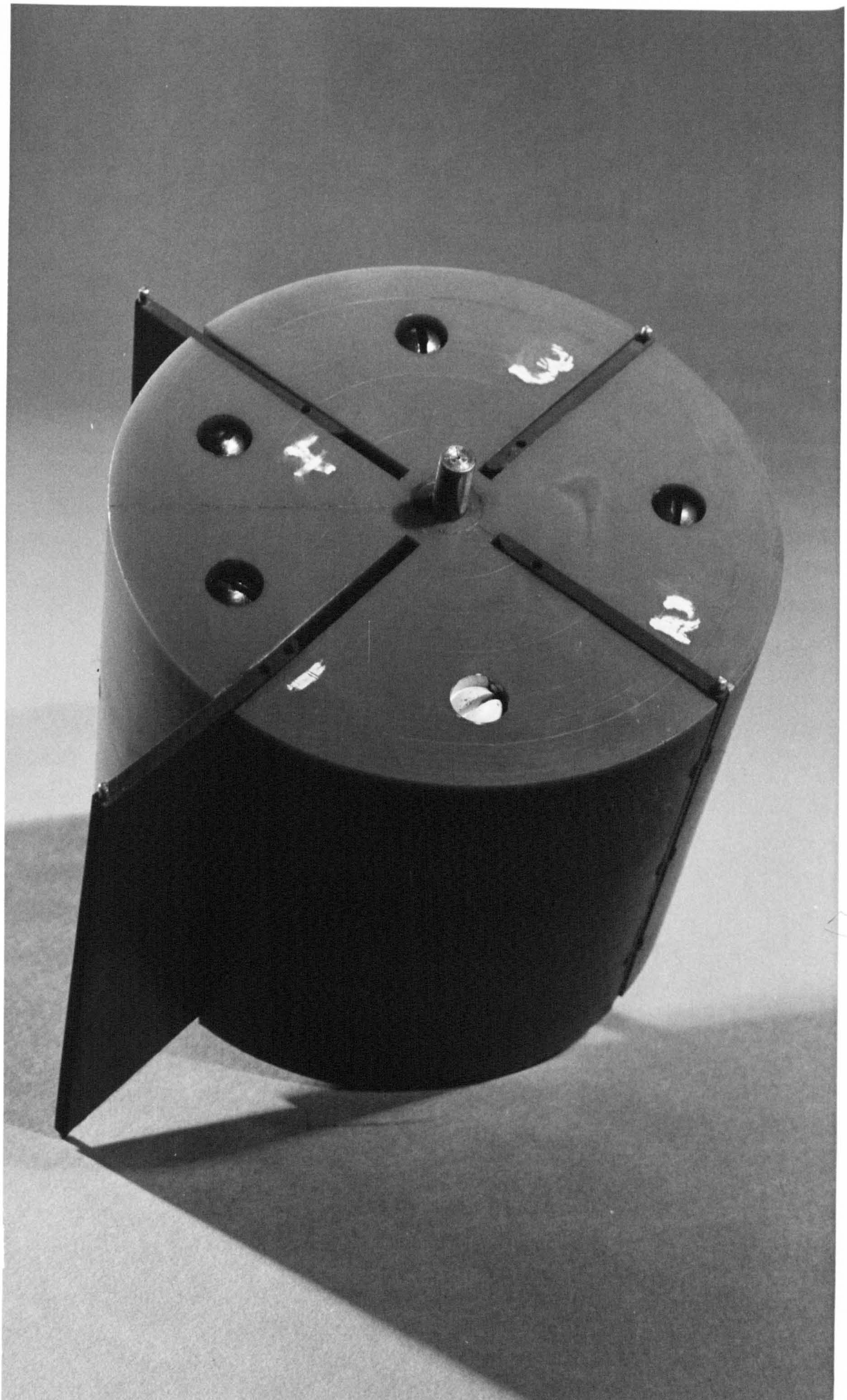
The guide groove in the casing ends could possibly be cast in without further machining provided the quality of casting was reasonable.

The vanes, rollers and other items could be of similar design and material as used for the half scale model. Whilst Nylatron and Fibrelam are not readily available in developing countries they are both very easily worked and could be acquired.



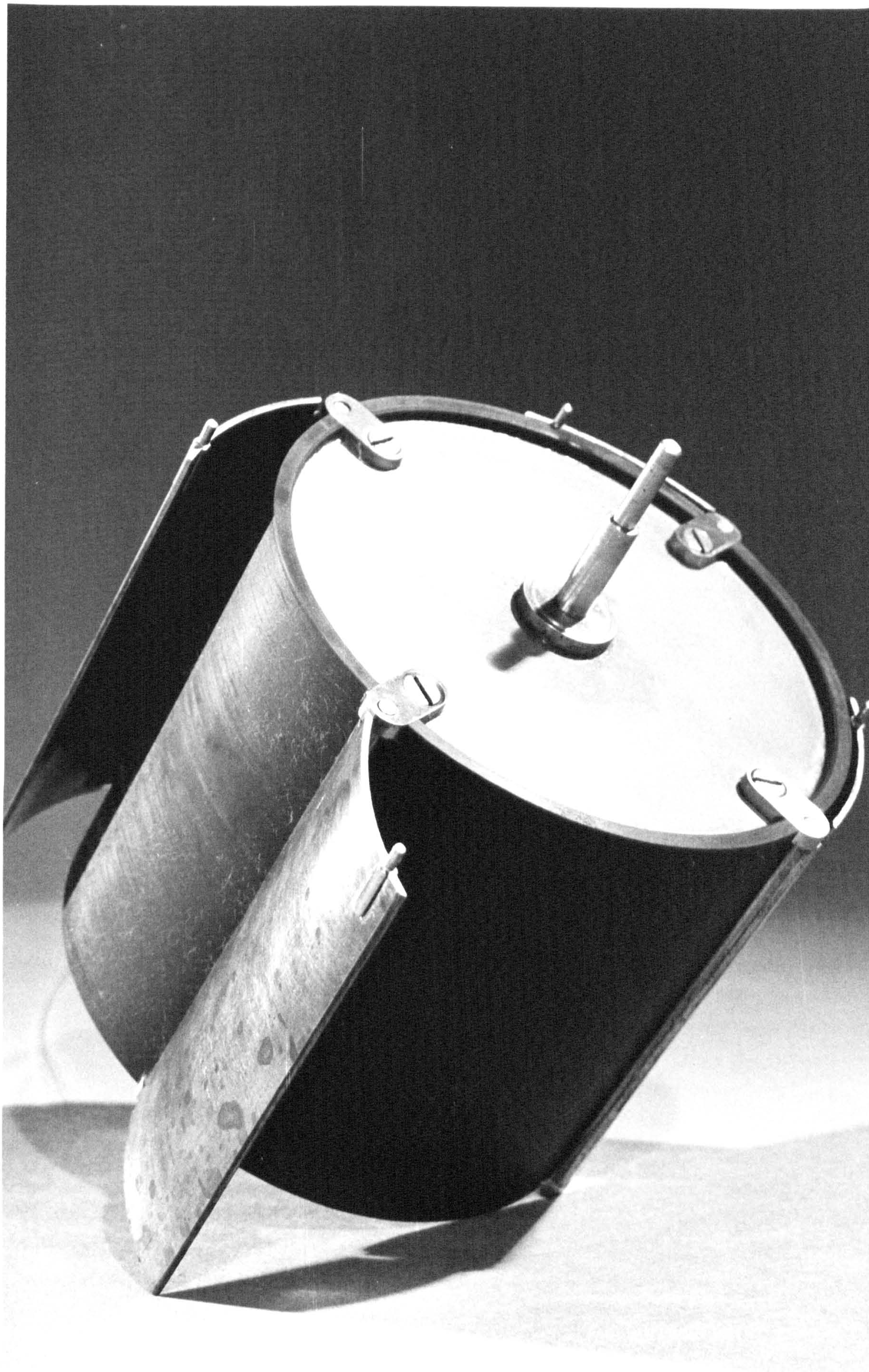
Machines with straight and curved vanes

Figure 6.1



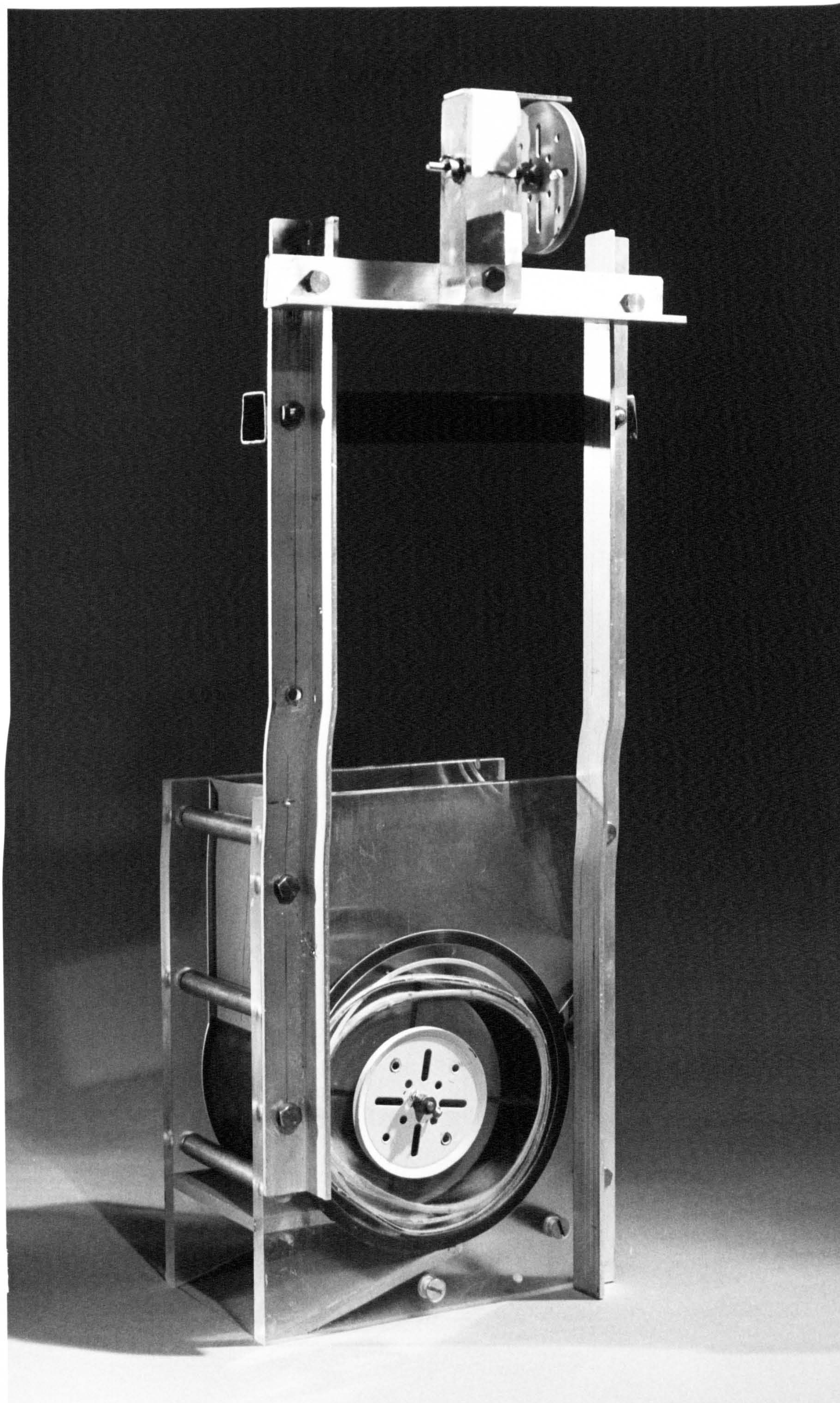
Rotor with straight vanes

Figure 6.2



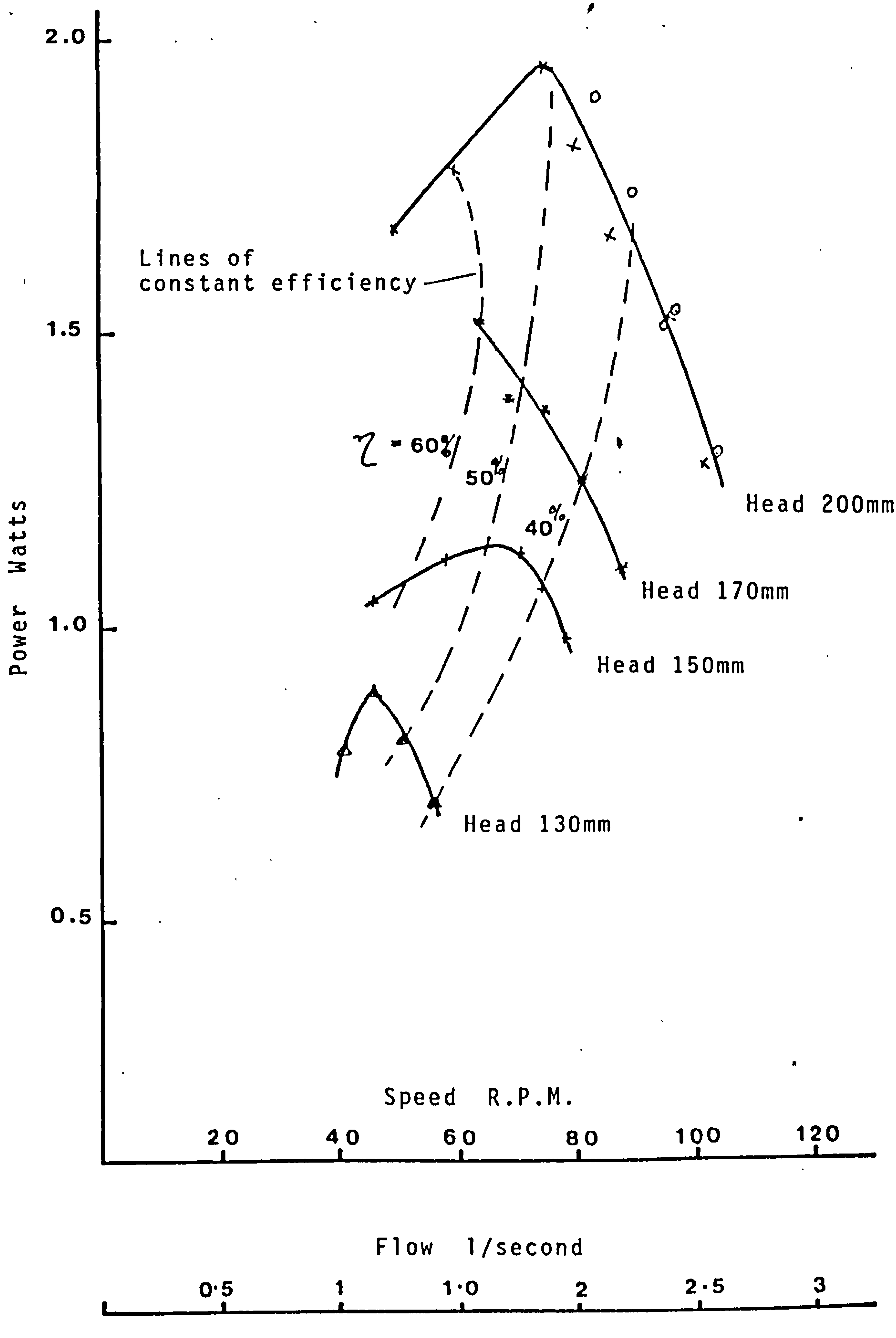
Rotor with curved vanes

Figure 6.3



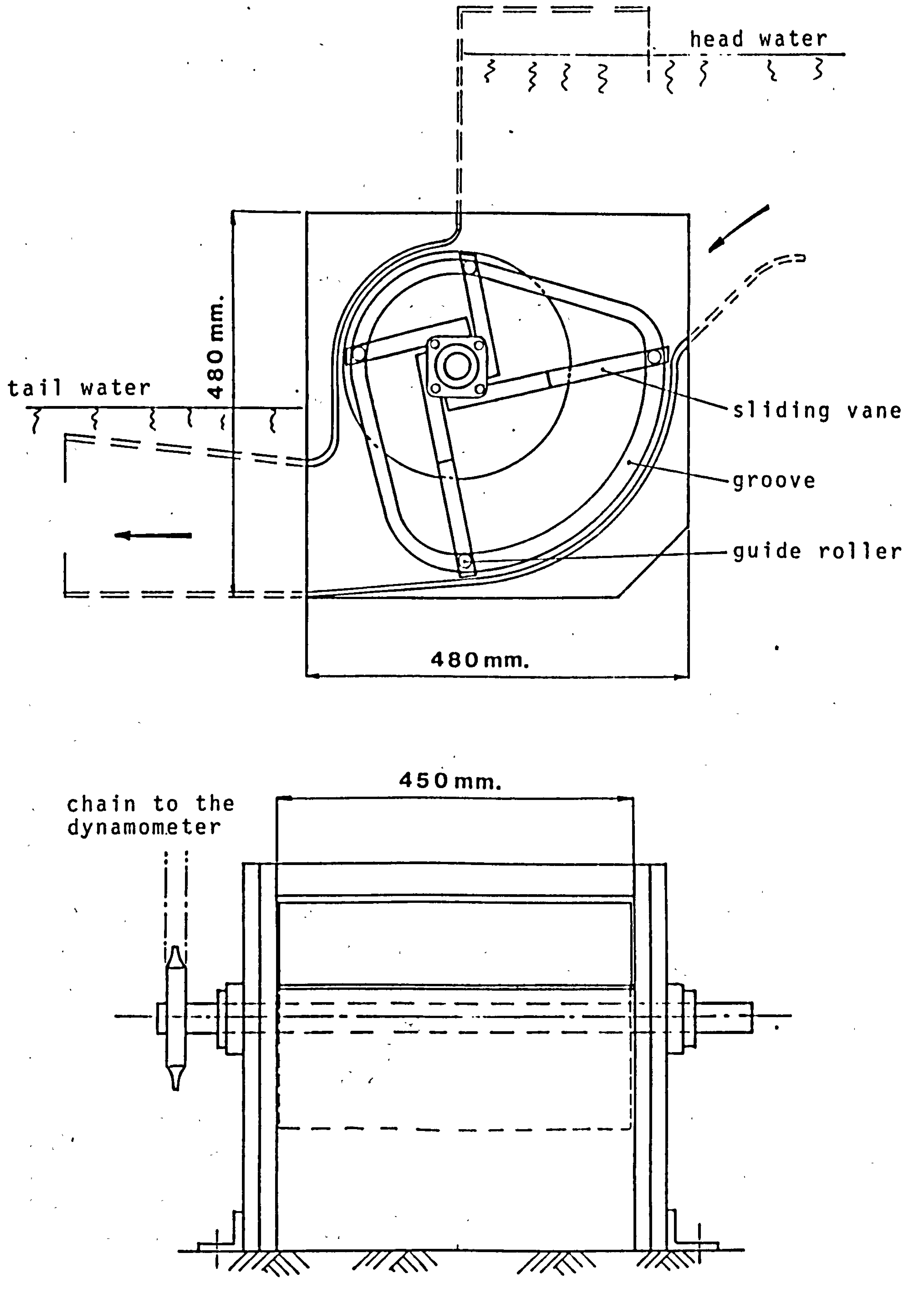
View of the one fifth scale model

Figure 6.4



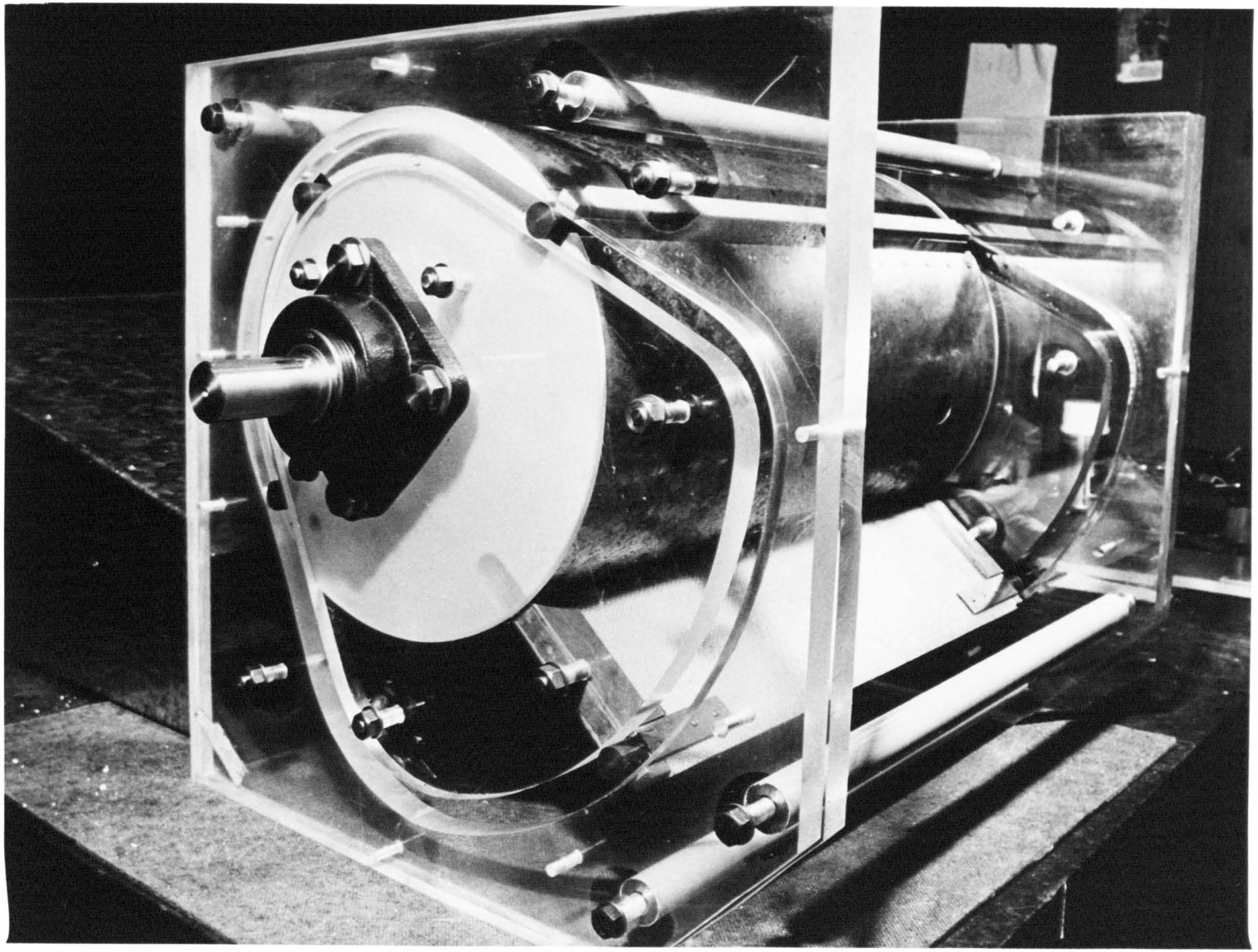
Power and efficiency - 1/5th scale model

Figure 6.5



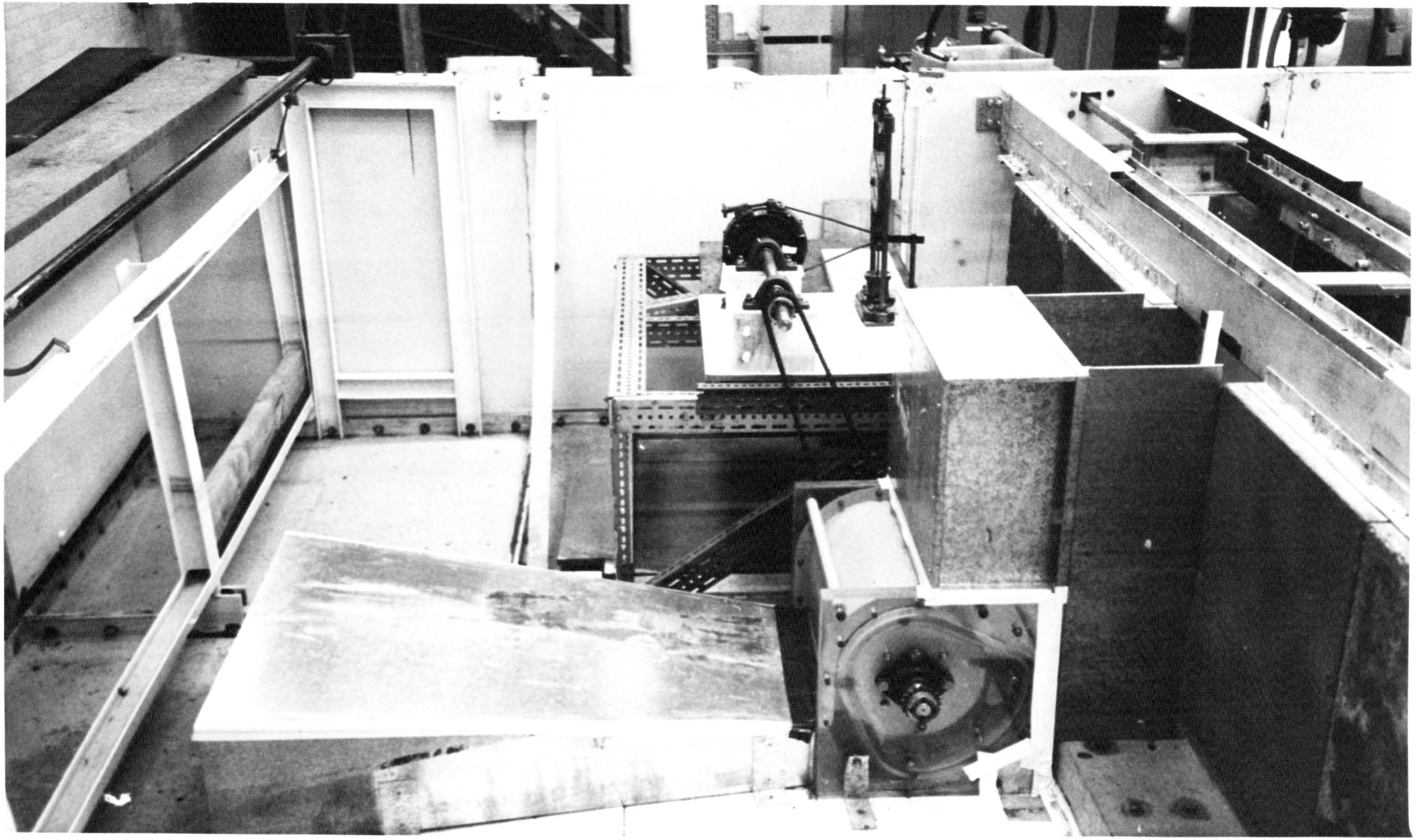
Arrangement of the half scale model

Figure 6.6



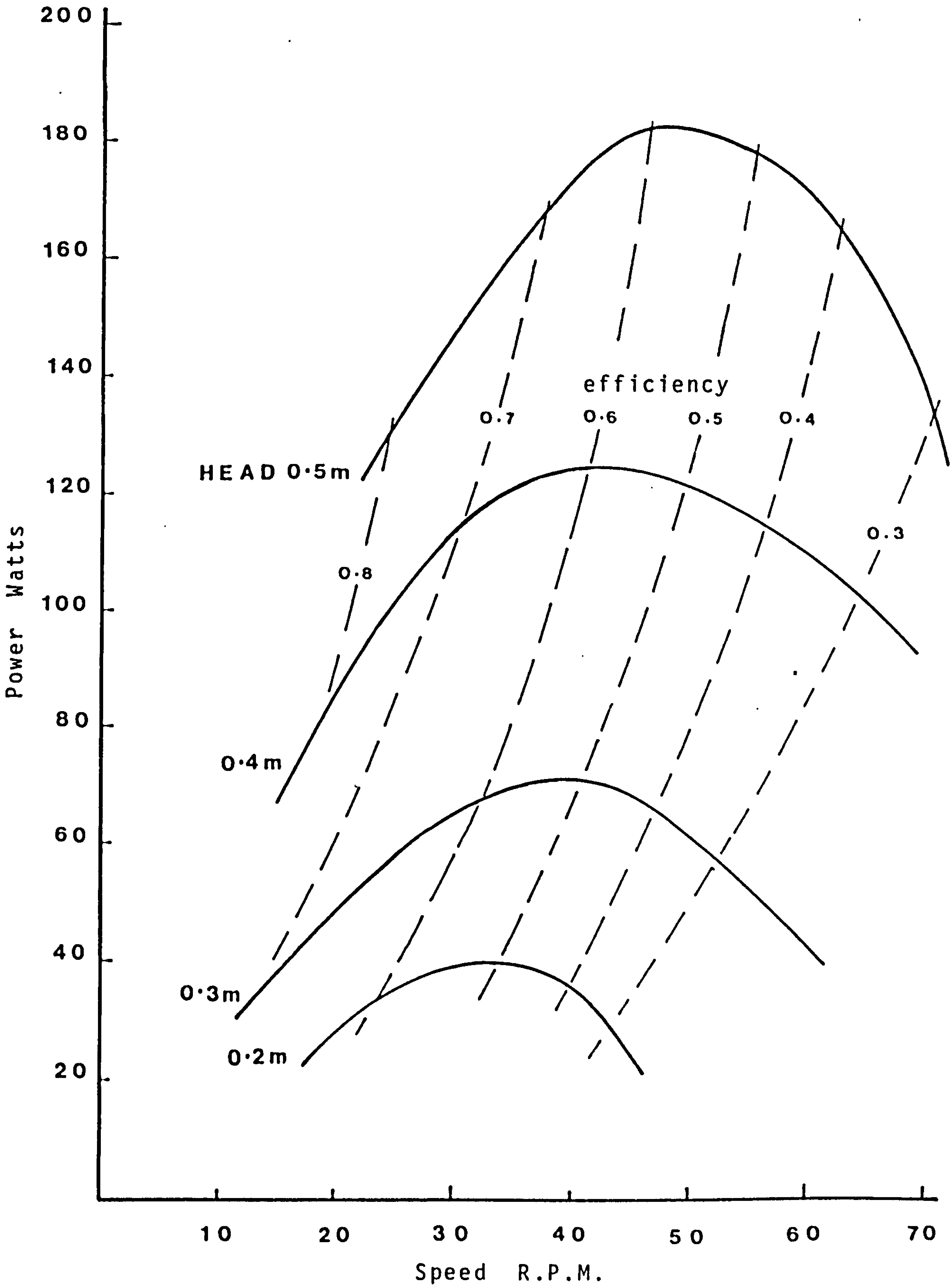
Completed model of the half scale underwater motor

Figure 6.7



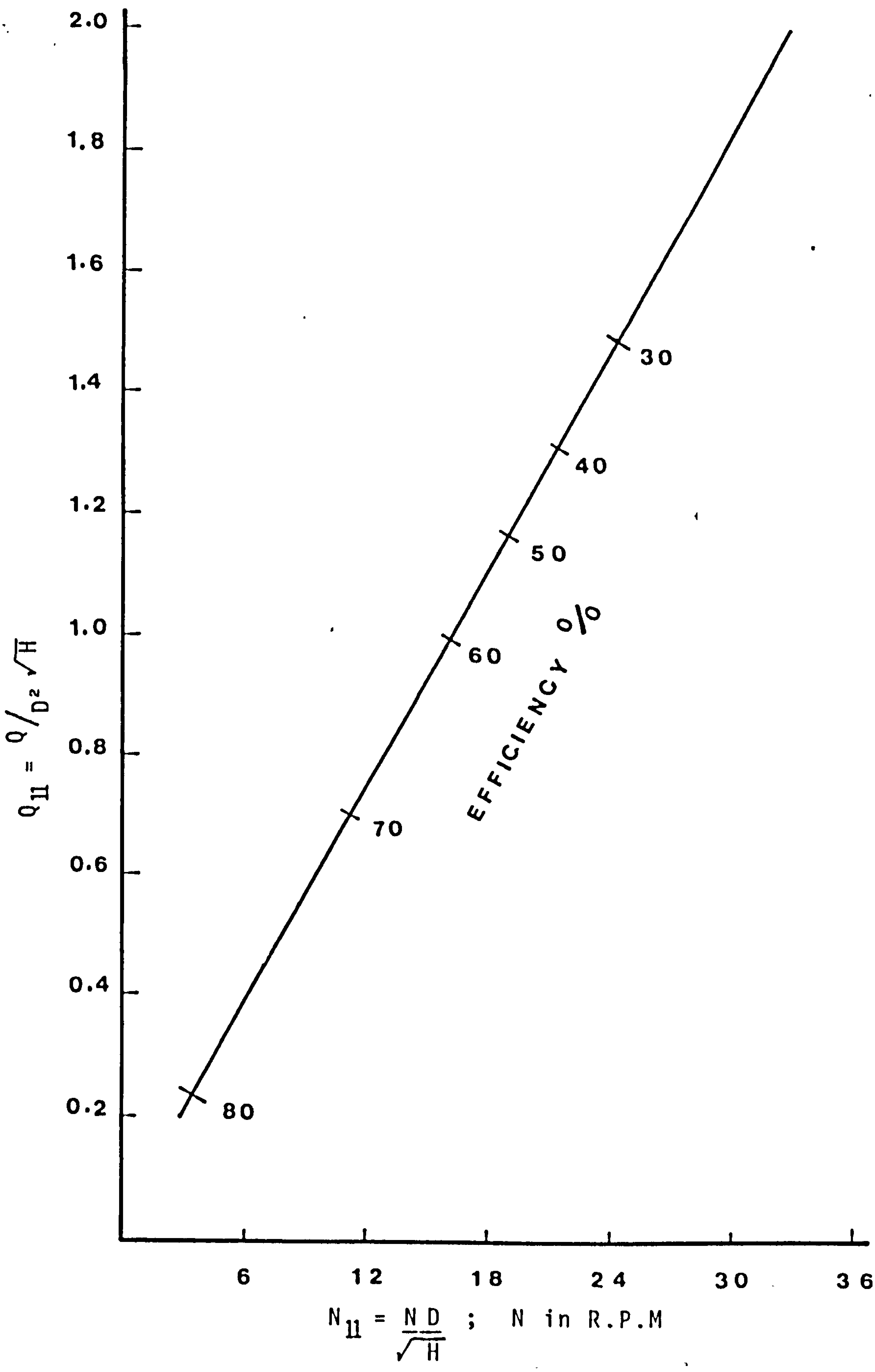
View of the half size model in the test tank

Figure 6.8



Power output and efficiency - half scale model

Figure 6.9



Dimensionless speed against flow

Figure 6.10

CHAPTER 7

A Hydro-Pneumatic Generator

7.0 Introduction

Section 3.3 introduces the box and the bag, hydro-pneumatic devices which are the subject of current research. Reference is also made to the hydro-pneumatic air compressor used in the mining industry to obtain high pressure air for pneumatic tools.

A British patent - number 183826 - for a trompe type compressor and turbine generator for tidal use was granted to Wilhelm Schwarzenauer in 1923. No details of the compressor are given, the invention relates to the idea of the barrage and control system to utilise the ebb and flow of the tide.

J. O. Boving patented an improved hydraulic compressor in 1925, Patent No. 260090. This gives clear information on the dimensions of the device. This was followed by a second patent for a two stage device for higher pressures.

Similar patents were taken out in both the USA and Germany for hydraulic air compressors at around this time.

In 1975 the idea was revived by Dr E M Wilson of Salford University who discussed the matter in some detail in private correspondence with Mr C S White of the British Columbia Hydro and Power Authority. Mr White's interest had been aroused by the construction of a model of the Ragged Chute Compressor at Cobalt Ontario (see 3.3) for the Ontaria Centennial Centre Museum. This compressor built in 1910 was still in use in 1975 (see reference 33 for a contemporary description of the plant).

Very high efficiencies, up to 85%, have been claimed for the Cobalt plant. If these are correct then the connection of such a device to an air turbine, for which efficiencies of 80% may be obtained, could result in a water to wire efficiency of 60%. This is probably better than any other non-conventional low-head hydro-generator and as good as a propeller turbine on a low head.

7.1 Description of the hydraulic air compressor

A brief description is given in reference 7.

Figure 7.1 shows the basic air compressor. Water from the upstream level passes down a vertical pipe to the base chamber. From the base chamber it rises up again to the downstream level, usually in a pipe concentric with the down flow pipe. The depth of the base chamber below the downstream level determines the pressure of the air produced by the device.

Air is entrained at the top of the down flow pipe, usually through a series of small holes or pipes. This ensures that the bubbles are small enough to be carried down the pipe by the flow of water. A 3mm diameter bubble will rise in water at about 0.25m/sec. Larger bubbles faster than this. The downward velocity of the water must be greater than the rising velocity of the bubble.

In the base chamber the velocity of the water is reduced, the air bubbles captured in an air chamber and the water exhaused to tailwater level.

A pipe from the air chamber carries the compressed air away to wherever it is needed.

The pressure in the air chamber is thus equal to H_2 metres of water, the volume flow of air down the pipe is a function of H_1, the head of water available

7.2 Basic Design Calculations

The energy available in the water at a low head site is ρQHg where Q is the volume, H the head and ρ the density.

The volume of air that can be compressed to a given pressure can thus be obtained by equating the energy in the falling water with the energy change of the compressed air.

Let the entrained volume of air be V_1 at a pressure (assume atmospheric) P_1. Let the compressed volume be V_2 at a pressure P_2. If the compression is isothermal - which it must be very close to as the water will remove any heat of compression, then

$$P_1V_1 = P_2V_2 = \text{constant } K$$

The work done going from state 1 to state 2 is:

$$\int_1^2 PdV$$

$$= \int_1^2 K/V \, dv$$

$$= K \, [\log_e V]_1^2$$

$$= P_1V_1 \, [\log_e V_2/V_1]$$

$$= P_1V_1 \, [\log_e P_1/P_2]$$

Thus $\rho QHg\eta = P_1 V_1$ [$\log_e P_1/P_2$]

where η is the efficiency of the device.

If Q and H are known and P_2 is fixed the entrained volume V_1 can be calculated for an assumed efficiency.

It can be seen from the above expression that the greater the compression ratio P_1/P_2 the smaller the entrained volume. Table 7.1 below gives the volume of air entrained (η = 1.0) for a flow of Q = 1 cumec at a head of 1 metre (H_1 = 1m) for various values of P_2 (H_2).

Outlet Pressure		Entrained Volume m^3/second
m. water H_2	bar gauge	
0.5	0.049	2.05
0.75	0.074	1.37
1	0.098	1.05
1.5	0.147	0.71
2	0.196	0.55
5	0.491	0.25
10	0.981	0.14
50	4.91	0.055
100	9.81	0.041

Thus if it is practical to excavate a pit 50 metres below tail water level only about one twentieth the volume of air is entrained compared to a device with an air chamber only one metre below tail water level.

It is reasonable to suppose that a low head hydropower device would be built with limited resources when a shallow chamber - say between one and two metres deep would be excavated.

This means that the volume of air entrained becomes a significant proportion of the water air mixture being carried down the pipe. Thus the density of the fluid mixture in the downpipe is considerably less than that of water - hence the pressure at the foot of the downpipe is reduced.

This will reduce the flow. The alternative is to inject the air at the base of the down pipe - this would seem to have the advantages of reducing the flow volume in the down pipe and reducing the possibility of small air bubbles coalescing into large ones and rising up the pipe.

7.3 Laboratory tests with a standard injector

Small jet pump type injectors are commercially available for use in laboratories. A section of one is shown in figure 7.2a. The nozzle is connected via a flexible pipe to a water main, the jet produced lowers the pressure locally in the chamber so drawing in air through the branch. It was convenient to use one of these of the dimensions shown for initial tests.

The test rig is shown in figure 7.2b. The supply tank can be raised or lowered to change the head above the receiving pipe outlet. The level of water above the injection pipe is altered by inclining the pipe to the vertical. Levels in the head tank and receiving tank are fixed by overflow pipes.

Air flow is measured by means of a positive displacement gas meter. Measurement of water flow is by collection in a calibrated vessel.

A series of tests were carried out at different water levels firstly with the injector at the foot of the receiving pipe and then repeated with the injector immediately below the headwater tank.

The results are shown in figure 7.3.

7.3.1 Discussion of Results

The results show clearly that a greater volume of air is entrained by a given volume of water when the ejector is at the lower level.

Furthermore the water flow rate, and hence the air entrainment rate is higher when the ejector is in the lower position.

At the upper position the water flow rate is constant regardless of the difference between upper and lower water levels. At the upper position the ejector is passing water at a rate fixed by the head above the nozzle which is constant. Thus no benefit is gained by increasing the difference between the upper and lower water levels.

The efficiency of the system is very low. At a head of 1m the efficiency is 10% with 0.5m tailwater depth and 12.4% with 0.75m tailwater.

Large losses cannot be attributed to friction in the water line where the velocity is only 0.33m/sec. The greatest losses are in turbulent mixing of the air and water streams and the relatively high velocity with which the water/air mixture enters at the base of the downstream waterpipe. A diffuser at this point would undoubtedly increase the efficiency.

The figure of 12.4% efficiency is not, however, exceptionally low for ejectors of this type. Gibson (reference 7) suggests that 25% is the usual efficiency for jet pumps, whilst later work by Cunningham (reference 34) indicates that higher efficiencies, up to 40% can only be obtained by designing for very specific conditions.

7.4 Venturi Based Entrainment

A number of the pneumatic air compressors described in the patent application taken out in the early years of this century used the low pressure at the throat of a venturi to effect the air entrainment. This would appear to be a simpler means of entraining air than the use of a nozzle type ejector.

The air lift pump, which is a compressor in reverse, uses an annular venturi to mix air with the water stream. Thus it is reasonable to assume that an annular venturi could be used in a compressor to create a large area of pipe surface for the entrainment of air.

The venturi would be followed by a diffuser to recover the kinetic energy of the mixed flow. As the volume of air and water is now considerably larger than the volume of water alone, the diffuser will in consequence be proportionally larger.

7.5 A Prototype Generator

Figure 7.4 shows the dimensions of a prototype device working at 2m head with 1.5m of tailwater above the air chamber passing $0.5m^3/sec$.

This at 80% compression efficiency would entrain $0.6m^3/sec$ of air. Assuming 75% efficiency of the turbine/generator unit, then 5.9kW of electricity would be produced.

A downpipe of 0.6m bore is shown which will give a mean velocity of 1.77m/second. An annular venturi is made inside this pipe to induce air into the flow. Maximum wall area for entrainment is obtained by admitting air to both the inner and outer regions of the venturi.

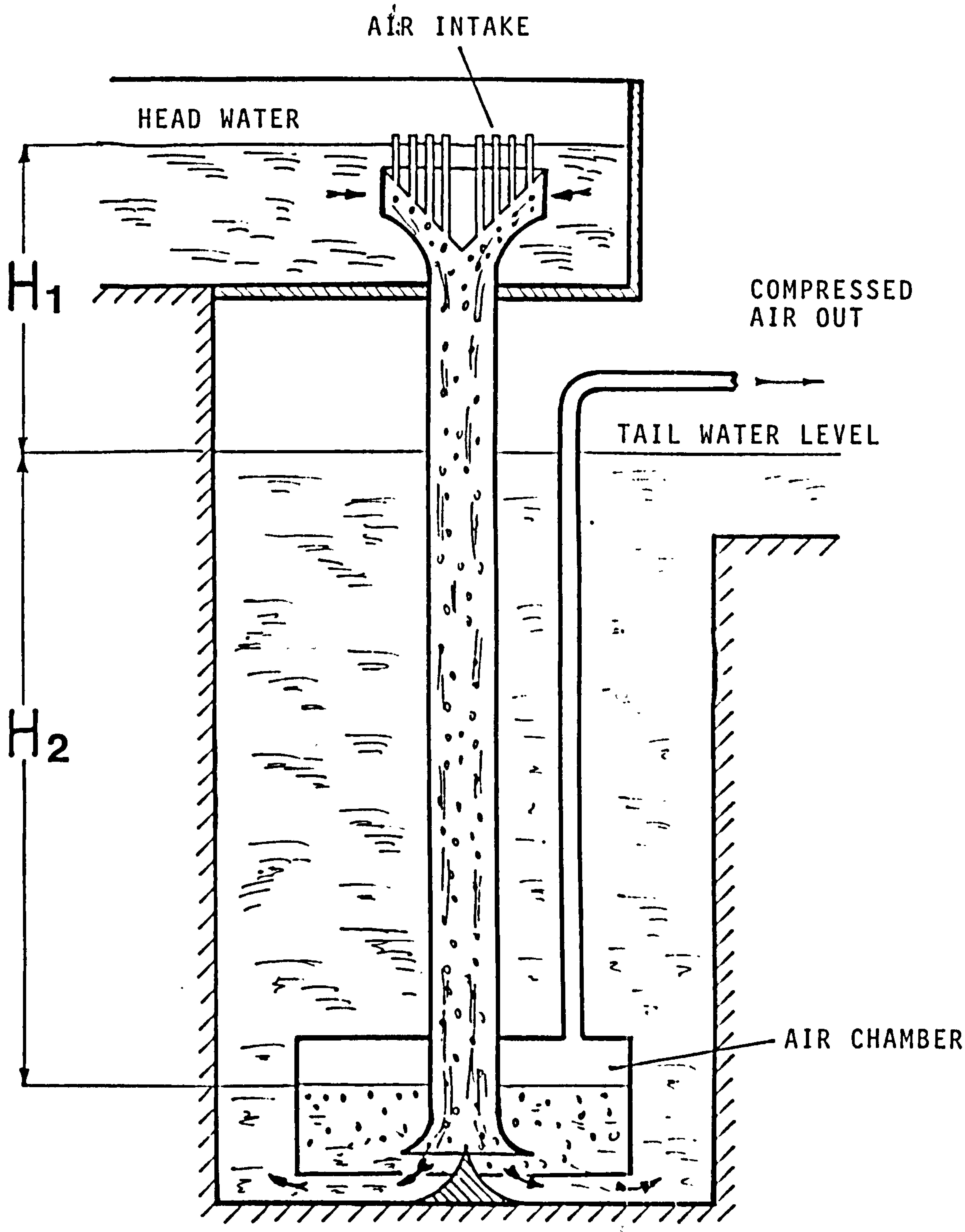
A 6° taper after the venturi enables most of the velocity to be recovered as pressure. However, as $0.6m^3$ per second of air have been entrained the increased volume means that the velocity at the end of this taper is 3.9m/second. Thus the taper must be continued to reduce the velocity to say 2m per second. The total length of the diffuser to achieve this is 3.6m.

To allow the air bubbles entrained into the base of the pipe to rise to the top and be collected a further length of pipe must be provided.

Assuming a rise rate of 0.25m per second (for a 3mm bubble) and a diffuser outlet diameter of 1.2m, then 5 seconds are needed to ensure all bubbles are collected. At 2m per second velocity a further 10m of pipe are required.

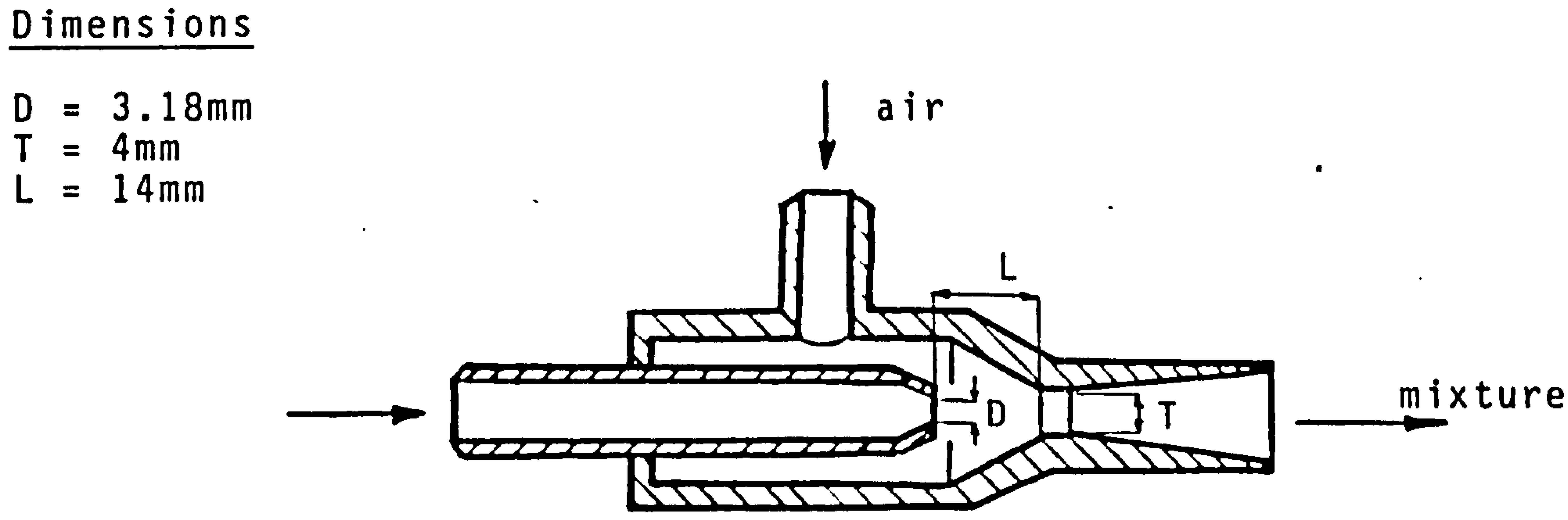
Thus the prototype compressor is of a substantial size to produce only 6kW. However the pipes are for the most part simple fabrications and at an estimate of the cost (at 1988) prices would be around £700 per tonne- say £5,500 total or about £1,000 per kW. To this must of course be added the cost of the air turbine and civil and electrical works.

It may be concluded however that it is worthwhile investigating the effectiveness of this hydropower generator. There is little doubt that, apart from the air turbine, the pipework and other items could be manufactured in developing countries.



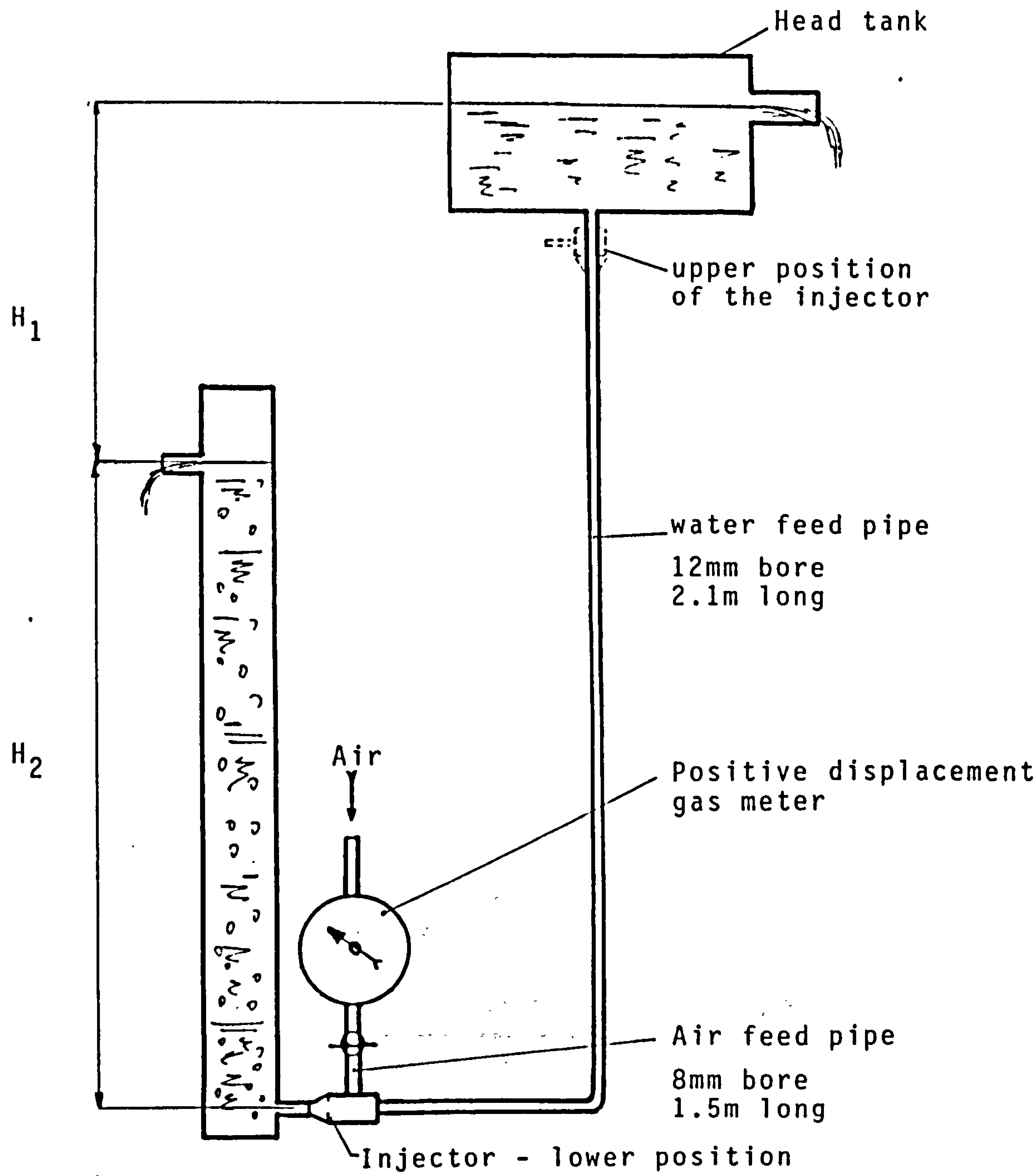
The hydraulic air compressor

Figure 7.1



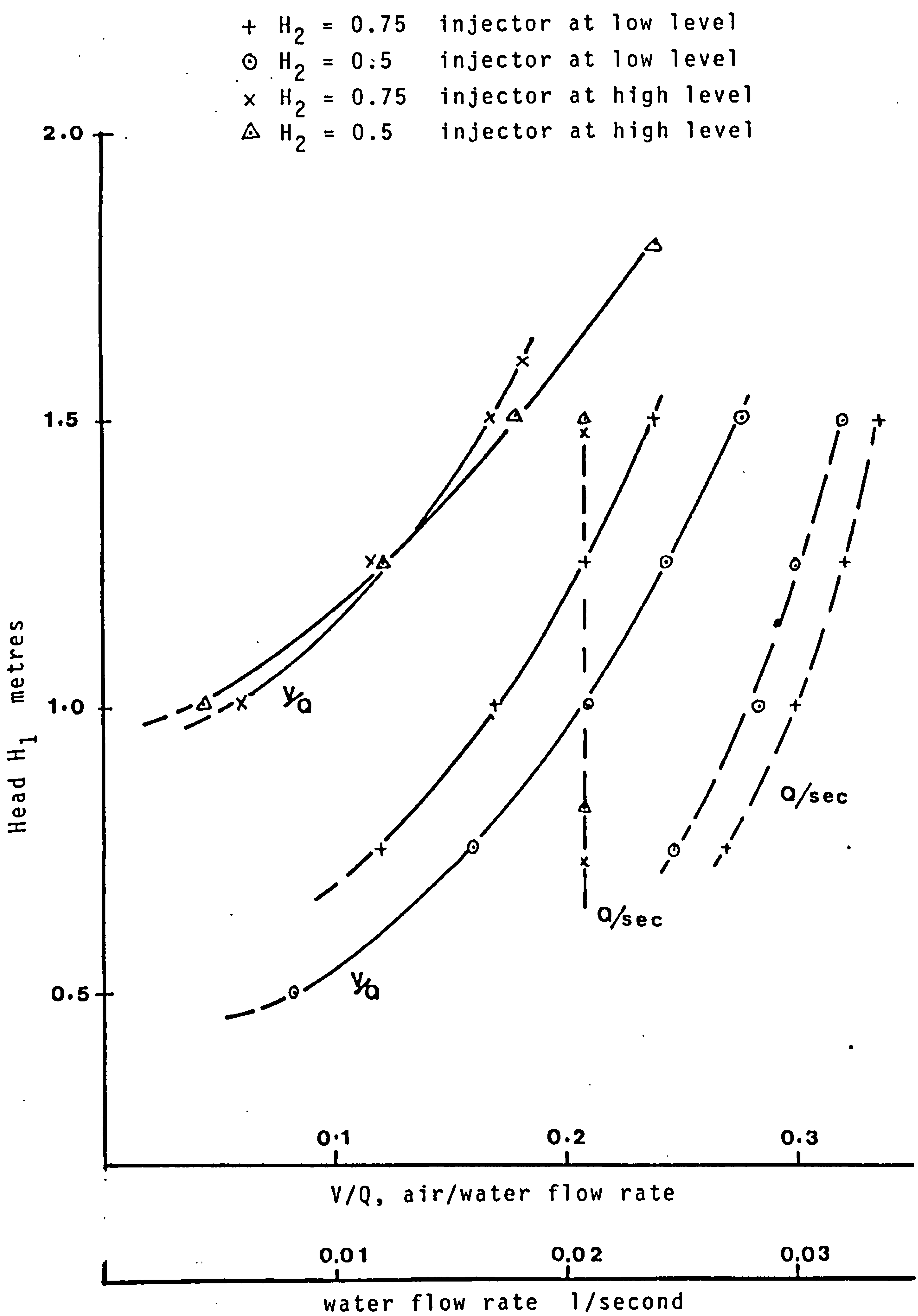
Section of laboratory injector

Figure 7.2a



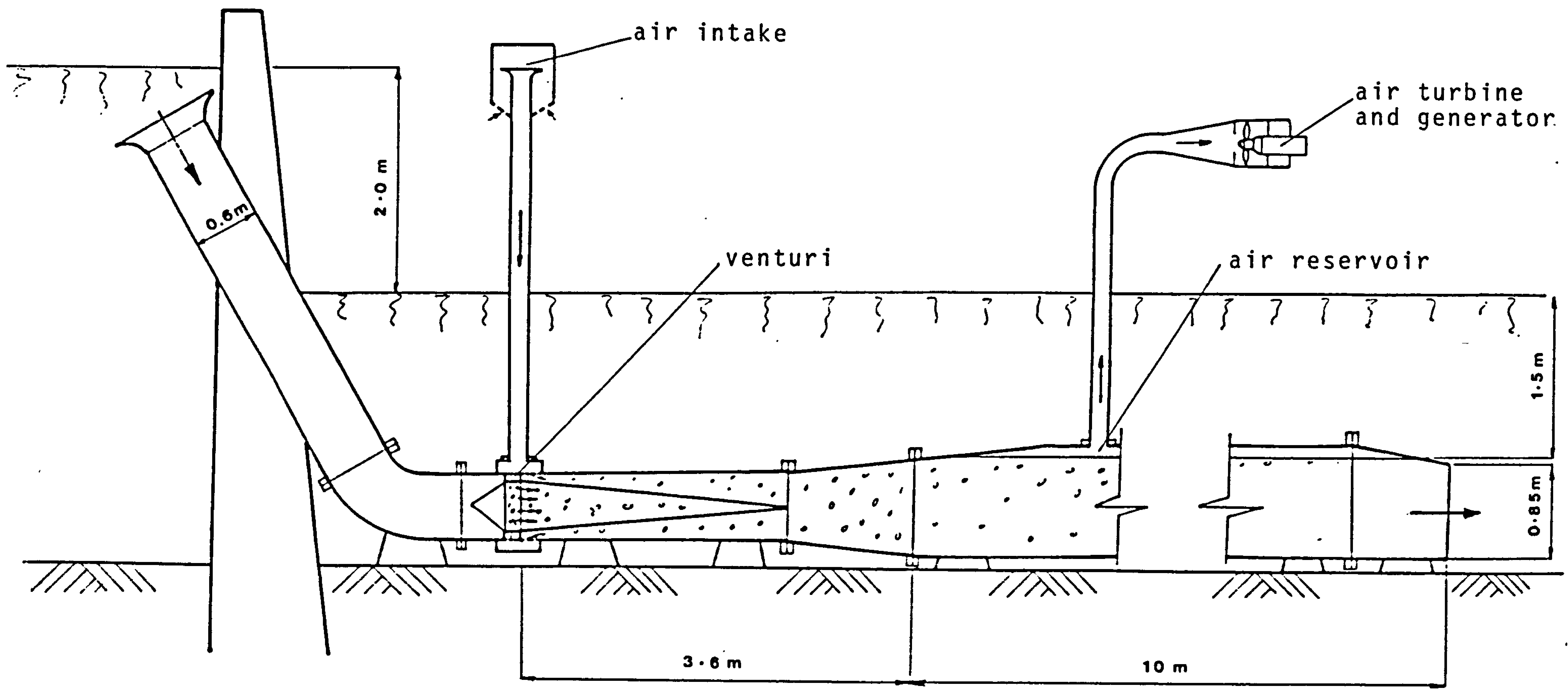
Air entrainment test rig

Figure 7.2



Test results

Figure 7.3



Proposed prototype hydro pneumatic generator

Figure 7.4

Summary of Low-Head Hydropower Generation

Low-head hydropower generation is attracting more attention as high head sites become developed and the world demands more energy. The current interest in the environment and the awareness of the unpleasant effects that fossil fuel powered generators are having on it has increased the need for an effective low head generator.

Non of the novel low head generators currently being developed stand out as being the perfect answer to the problem. The STO and AUR Water Engine are both practical machines in a technical sense but neither could be held to be better or even cheaper than a large propellor turbine.

The box and bag hydro pneumatic devices are on current evidence even less likely to provide a solution to the problem.

The writer believes that a short term solution would be the development of a large low cost propellor turbine which should be manufactured to have a limited life in line with modern financial accounting philosophy. Say ten years at the most.

The only realistic competitor to this is the water wheel - which would undoubtedly be suitable for most developing countries if designed for local fabrication using available skills and material.

The momentum water wheel, described in section 3.2 is essentially a traditional wheel improved to allow it to be set into the tailwater without loss of efficiency.

Long term solutions may be found in the results of current research but the present and short term needs can only be met by the two relatively conventional machines described above.

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APPENDIX 1

Prototype AUR Water Engine

APPENDIX 1

PROTOTYPE AUR WATER ENGINES

Introduction

A considerable amount of investigation and design work was carried out concurrently with the laboratory test programme. There was a two way exchange of ideas during this time. Laboratory results modified the prototype design work and practical considerations revealed by prototype work often altered the direction of the laboratory test programmes.

Although designs were produced for a number of sites, some of these were standard machines based on previously developed designs. The prototype machines described below were "state of the art" machines at the time and illustrate how the Water Engine developed during the testing work.

1. Single circular chamber - Adelphi Weir
2. Twin circular chamber - Adelphi Weir
3. Package Machine - twin square chamber
4. Package Machine - Byfleet Mill
5. Gibbons Mill - twin square chamber

Single Circular Chamber Machine at Adelphi Weir

Site - Adelphi Weir is on the river Irwell adjacent to the Civil Engineering Department of Salford University. It is a masonary structure, built around the middle of the 19th century to provide water to mills sited along the river banks. These mills no longer exist.

The head is approximately 2m at a flow of $17m^3/sec$. At higher flows the downstream water level backs up and the head reduces.

Situation of the Model

The machine was designed to be sited in the river bank adjacent to the weir, fed by a pipe on the upstream side and discharging via a pipe on the downstream. This configuration was judged to give the cheapest civil costs allowing excavation to take place in the bank in the dry. Figure 4.6 is an artist's impression of this machine.

Construction Details

Chamber - 3.1m internal diameter glass fibre reinforced pipe.
Float - 3.0m external diameter x 2.5m long glass fibre reinforced pipe.
Gate - Rotating cylinder type actuated through the centre of the float.
Power take off rams - 4 off - 100mm bore x 1000mm stroke.
Feed/discharge pipes - concrete
Output power 15kW @ 2m head.
Water consumption 1.7m³/sec.

Design work on this machine was overtaken by the twin chamber design.

Twin Circular Chamber Machine at Adelphi Weir

This machine was to be sited in the river together with a Salford Transverse Oscillator (STO). The situation was downstream of the weir in a head pond formed by the STO and a wall running along the middle of the river.

Materials

To investigate manufacturing costs, designs were produced for three machines.

1 Reinforced concrete chambers and steel floats,
2 All steel machine - chambers and floats.
3 Armco (corrugated steel pipe) machine - made from standard pipe sections.

Design

This is shown in figure A1.1. Vertical rams synchronise the movement of the two floats. A single central butterfly valve controls the water and each float has two power take off rams. The central synchronising rams also transfer the power developed by the falling float to the power take off rams of the rising float.

Entry of water is directly from the head pond and discharged directly to the river, there are no feed pipes.

Details

Chambers 3.00m inside diameter
Floats 2.5 m high
Central steel butterfly gate

Output power 35kW @ 2 m head
Water consumption 4.0m^3/sec.

Package Water Engine

Enquiries from overseas showed the need for a neat design that could be produced in quantitiy, easily shipped and quickly installed.

This coincided with the realisation that square chambers offered a cheaper construction cost per kW than round chambers.

Thus the package unit AURWE was conceived. The layout is shown in figure A1.2. In plan the two chambers are the size of a standard ISO freight container. The machine is formed from four separate units as indicated in the figure

These break down for transport into three sections each the size of an ISO container and are fitted with the necessary lugs and sockets to be transported, lifted and stacked with ordinary containers.

This design obviously has a size restriction, but the unit shown would produce upto 26kW on a head of 2m.

Byfleet Mill

Byfleet Mill near Weybridge Surrey was investigated as a potential site for a prototype package AURWE

Considerable design work and site survey work was carried out, estimates obtained and bore holes drilled.

The head at this site varied between 1.9 and 2.1m. A machine of installed capacity of 23kW was expected to give an annual energy output of 160,000 kWh.

Efficiency of Prototype Machines

The larger the output of the water engine the higher the overall efficiency. Effects of friction are proportionally less and larger energy convertors, generators and motors are more efficient than smaller ones.

Hydraulic Motor

These figures are based on the 'HD' series hydraulic motors manufactured by Commercial Hydraulics Limited, normally a motor will operate at 15oo rpm (50Hz) at between 3/4 and full swash plate angle.

Efficiency at full swash angle 92%
Efficiency at 3/4 swash angle 90%
Efficiency at 1/2 swash angle 85%

Say 91% overall, assuming oil at 50°C and 21 centistoke viscosity.

Generators

Precise generator efficiency depends on type, size, number of phases and quality of construction. Some typical figures are summarised below.

Make/Model	No. of Phases	Type	Power kW	Eff. %
Hawker Siddley Electric				
CRSIA	1	Synchronous	7.3	72.0
CRSIA	3	Synchronous	10.0	81.3
CRS4B	3	Synchronous	61.2	89.0
Leroy Somer				
LSP 160M	3	Induction	13.0	84.0
LSP 225M	3	Induction	60.0	91.5

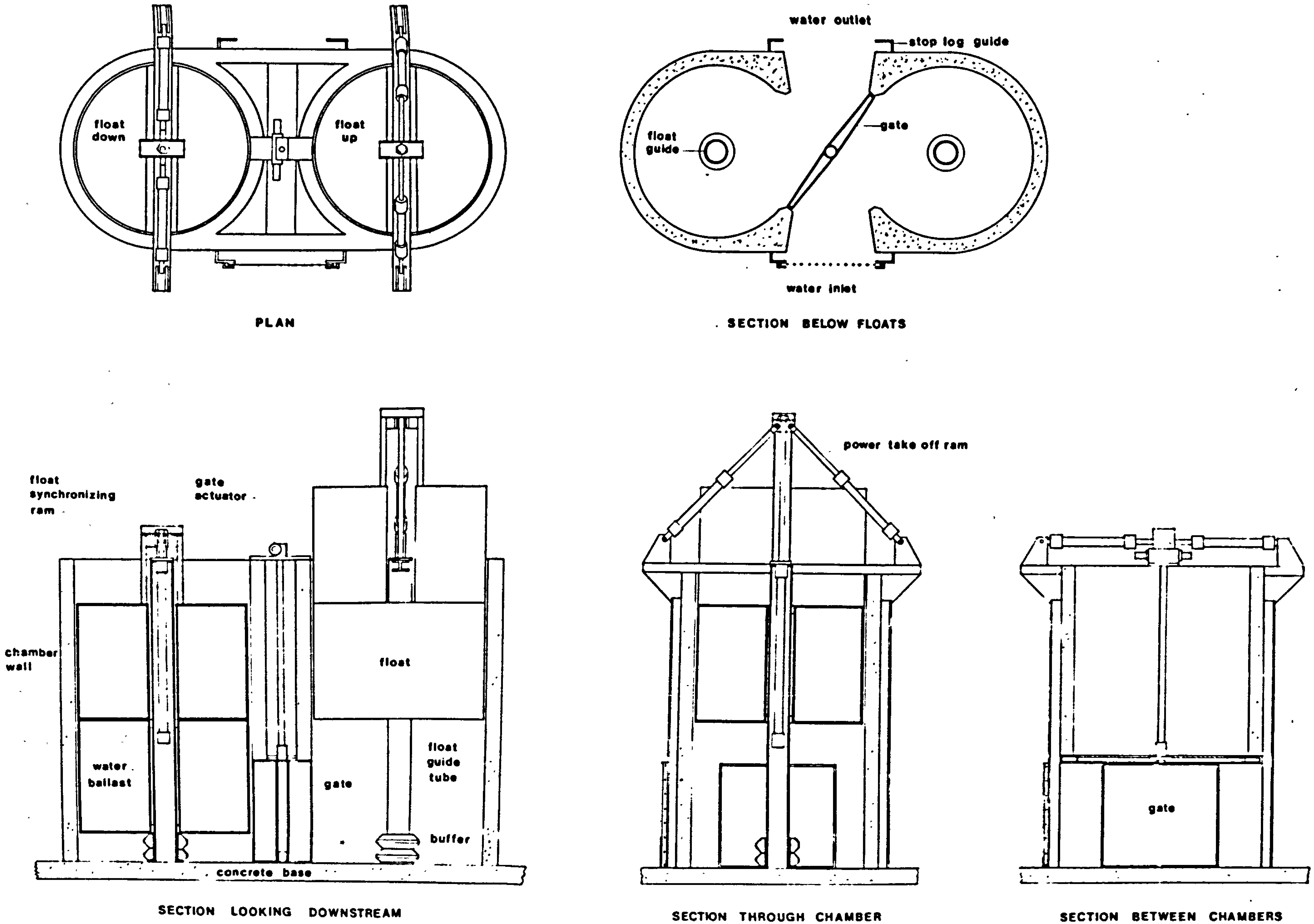
Water Engine Conversion Efficiency

Model tests and the Gibbons Mill prototype indicate that 80% water to oil conversion is the highest that is possible.

Overall Efficiency

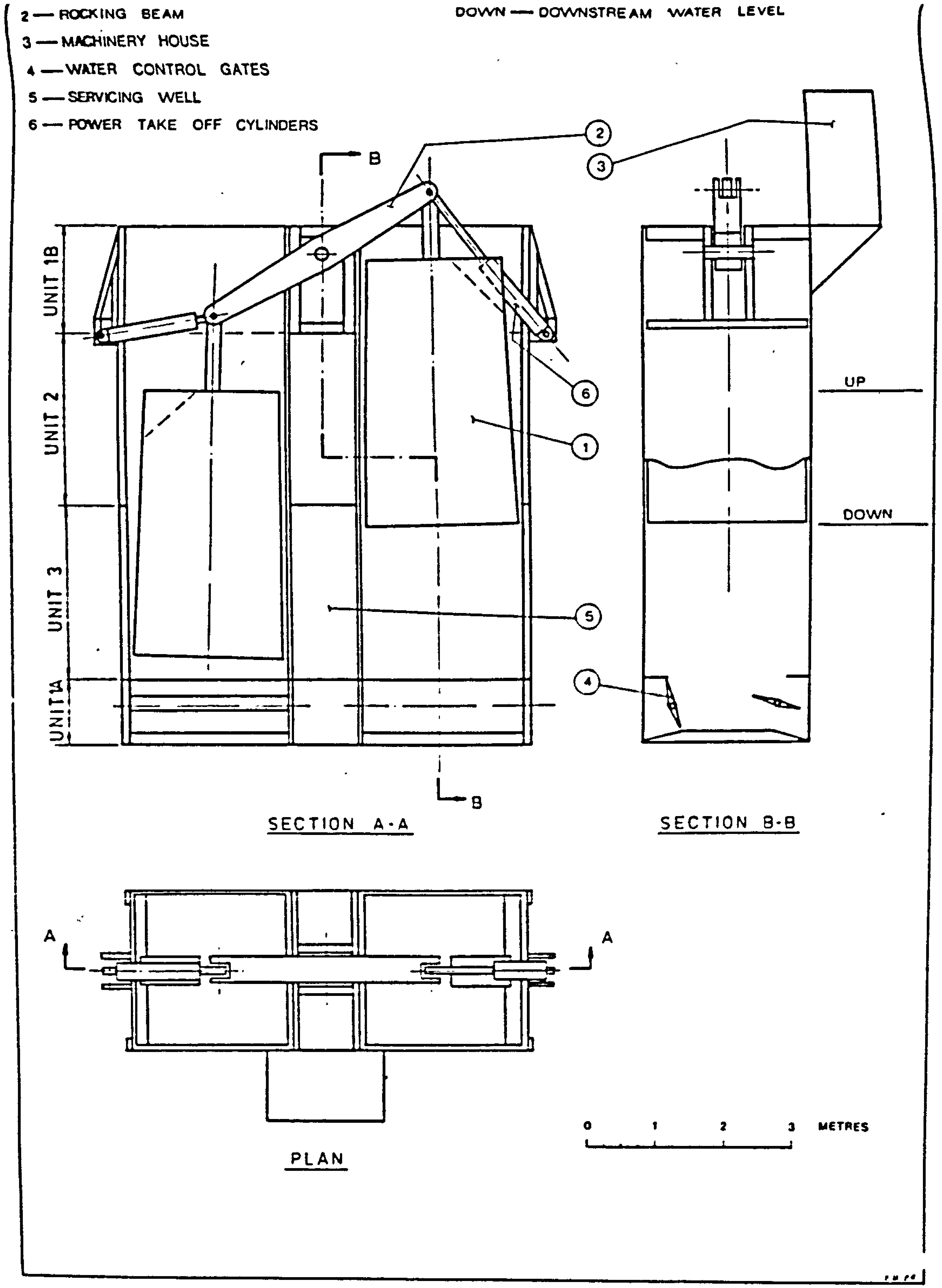
Thus the very best water to wire efficiency that could be expected for a 40kW machine is

.8 (water/oil) x .9 (hydraulic motor) x .9 (generator) = <u>0.648</u>

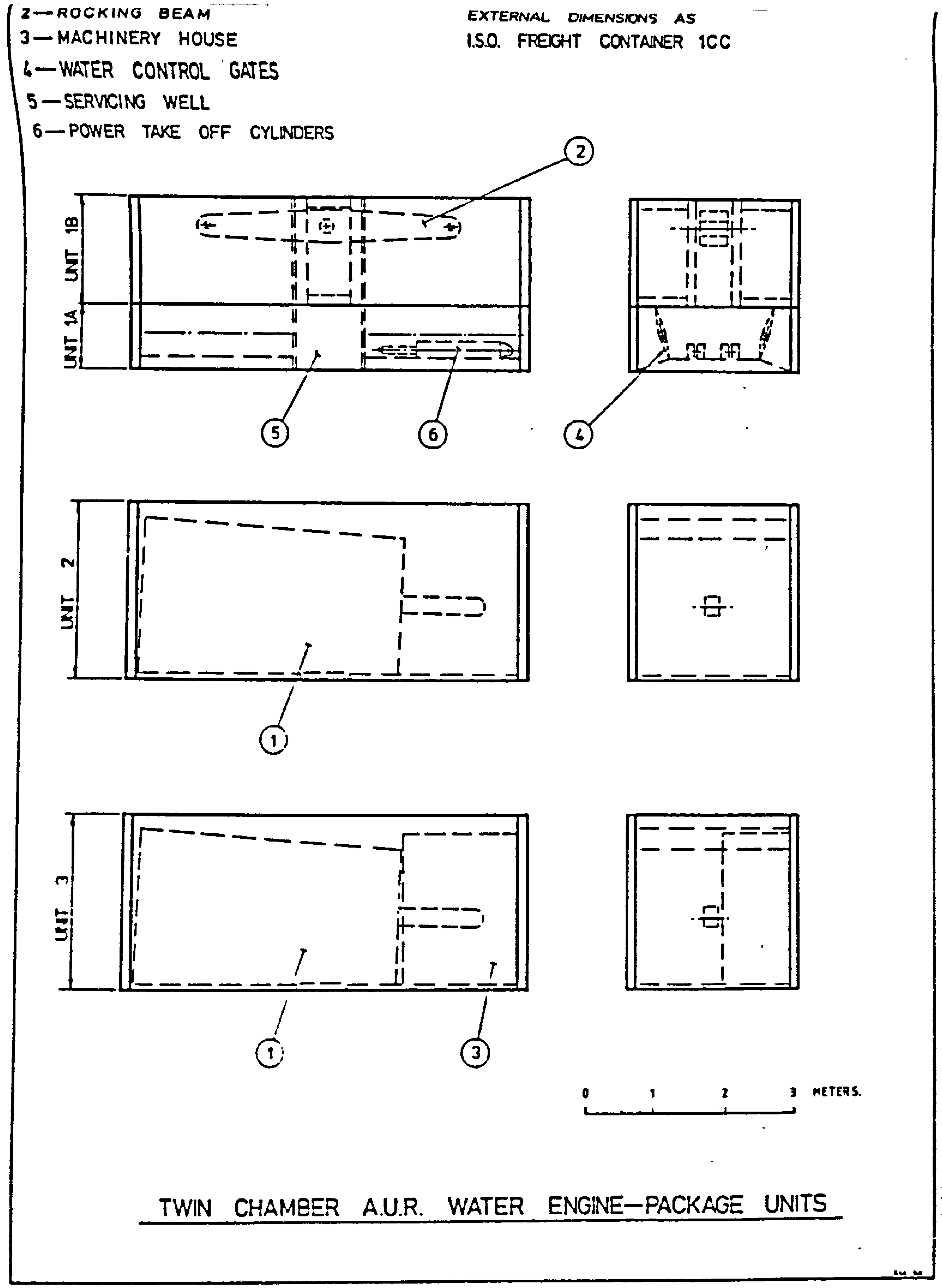


TWIN CHAMBER A.U.R. WATER ENGINE.

Figure A1.1



2 — ROCKING BEAM
3 — MACHINERY HOUSE
4 — WATER CONTROL GATES
5 — SERVICING WELL
6 — POWER TAKE OFF CYLINDERS
DOWN — DOWNSTREAM WATER LEVEL
UNIT 1B
UNIT 2
UNIT 3
UNIT 1A
B
A
UP
DOWN
SECTION A-A
SECTION B-B
PLAN
0 1 2 3 METRES



2 — ROCKING BEAM
3 — MACHINERY HOUSE
4 — WATER CONTROL GATES
5 — SERVICING WELL
6 — POWER TAKE OFF CYLINDERS
EXTERNAL DIMENSIONS AS
I.S.O. FREIGHT CONTAINER 1CC
UNIT 1B
UNIT 1A
UNIT 2
UNIT 3
0 1 2 3 METERS.
TWIN CHAMBER A.U.R. WATER ENGINE—PACKAGE UNITS

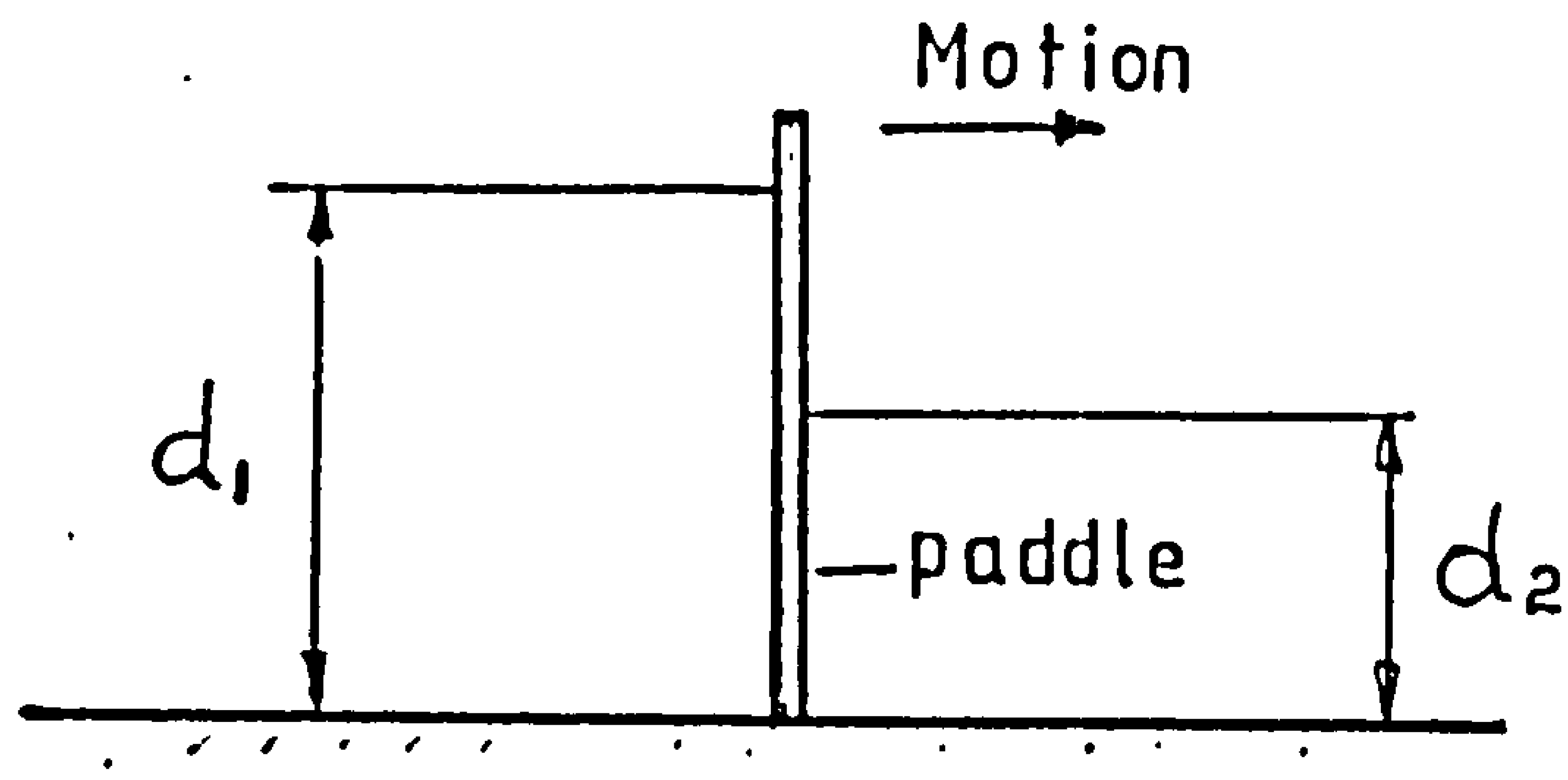
Figure A1.2

APPENDIX 2

STO - Pseudo Static Analysis to Obtain Theoretical Efficiency

Appendix 2

A PSEUDO STATIC ANALYSIS TO DETERMINE THE THEORETICAL EFFICIENCY OF A STO CELL



Notation

d_1 = depth of water on U/S side of paddle

d_2 = depth of water on D/S side of paddle

B = breadth of paddle

S = stroke

T = period of oscillatory motion

W = specific weight of water

Analysis

Resultant force on paddle in direction of motion

$$= \tfrac{1}{2} Wd_1^2 B - \tfrac{1}{2} W d_2^2 B$$

$$= \tfrac{1}{2} WB (d_1^2 - d_2^2)$$

Work done on paddle in one half cycle

$$= \tfrac{1}{2} WB ({d_1}^2 - {d_2}^2)S$$

Average rate at which work is done on paddle

$$= \frac{WBS}{T} (d_1^2 - {d_2}^2)$$

Head across barrage (neglecting losses)

$$= d_1 - d_2$$

Volume of water discharged for half cycle by each cell (neglecting gate volume)

$$= B\ d_1\ S$$

Average rate of discharge through cell

$$= B\ d_1 S \ / \ (\tfrac{1}{2}\ T)$$

Total Power available to cell

$$= W \left(\frac{2\ B\ d_1\ S}{T}\right) (d_1 - d_2)$$

Efficiency of cell

$$= \frac{\text{Average rate at which work is done on paddle}}{\text{Total power available to cell}}$$

$$= \frac{W\ B\ S\ (d_1^2 - d_2^2)}{T} \times \frac{T}{2W\ B\ S\ d_1\ (d_1 - d_2)}$$

$$= \frac{d_1^2 - d_2^2}{2d_1\ (d_1 - d_2)}$$

$$= \frac{d_1 + d_2}{2d_1}$$

$$= 0.5 + 0.5\ \frac{d_2}{d_1}$$

Notes

1. The theoretical efficiency is never less than 50% and will only equal 50% when $d_2 = 0$.

2. The theoretical efficiency is 100% when $d_1 = d_2$. When the upper surface of the water is at atmospheric pressure (as it is when the paddle moves in an open channel) $d_1 = d_2$ when the head across the barrage is zero and the total power available is zero. However, if a roof was added to each cell (so that the paddle in effect became a piston within a closed duct), d_1 and d_2 could be made equal at heads greater than zero.

3. Dynamic effects not included in this analysis must be expected to cause a reduction in efficiency. In particular, motion of the paddle will tend to decrease d_1 and increase d_2 relative to the depths of water U/S and D/S of the barrage.

APPENDIX 3

Underwater Motor Test Results

24th May 1988 with short diffuser

UPSTREAM DEPTH (m)	D'NSTREAM DEPTH (m)	HEAD (m)	BRAKE SPEED (rpm)	TORQUE (ft.lbs)	POWER (W)	GEOMETRIC FLOW (l/s)	TOTAL FLOW (l/s)	THEO. POWER (W)	EFFIC'Y	Q1	N1
0.700	0.260	0.440	88.2	10.000	125.2	52.3	53.89	232.6	0.54	1.113	0.598
0.680	0.265	0.415	90.2	9.000	115.3	53.5	55.03	224.1	0.51	1.170	0.630
0.665	0.265	0.400	92.3	8.000	104.9	54.7	56.25	220.7	0.48	1.219	0.657
0.635	0.260	0.375	92.3	7.000	91.7	54.7	56.20	206.8	0.44	1.257	0.678
0.620	0.260	0.360	96.8	6.000	82.5	57.4	58.84	207.8	0.40	1.344	0.726
0.595	0.260	0.335	97.6	5.000	69.3	57.9	59.27	194.8	0.36	1.403	0.759
0.575	0.260	0.315	100.8	4.000	57.3	59.8	61.12	188.9	0.30	1.492	0.808
0.560	0.260	0.300	103.4	3.000	44.0	61.3	62.63	184.3	0.24	1.567	0.850
0.535	0.260	0.275	106.2	2.000	30.2	63.0	64.24	173.3	0.17	1.678	0.911
0.520	0.260	0.260	107.1	1.000	15.2	63.5	64.73	165.1	0.09	1.739	0.945
0.830	0.280	0.550	91.6	14.000	182.1	54.3	56.10	302.7	0.60	1.036	0.556
0.825	0.280	0.545	99.2	13.000	183.1	58.8	60.60	324.0	0.57	1.125	0.605
0.820	0.280	0.540	103.4	12.000	176.2	61.3	63.08	334.2	0.53	1.176	0.633
0.820	0.280	0.540	113.2	11.000	176.8	67.1	68.89	364.9	0.48	1.284	0.693
0.825	0.285	0.540	121.2	10.000	172.1	71.9	73.64	390.1	0.44	1.373	0.742
0.820	0.285	0.535	125.0	9.500	168.6	74.1	75.88	398.2	0.42	1.421	0.769
0.815	0.290	0.525	134.8	8.000	153.1	79.9	81.68	420.6	0.36	1.544	0.837
0.820	0.290	0.530	141.2	7.000	140.4	83.7	85.48	444.4	0.32	1.609	0.873
0.810	0.295	0.515	144.6	6.000	123.2	85.7	87.47	441.9	0.28	1.670	0.907
0.730	0.265	0.465	94.5	10.000	134.2	56.0	57.68	263.1	0.51	1.159	0.624
0.715	0.265	0.450	100.0	9.000	127.8	59.3	60.91	268.9	0.48	1.244	0.671
0.690	0.270	0.420	100.8	8.000	114.5	59.8	61.33	252.7	0.45	1.296	0.700
0.670	0.270	0.400	105.3	7.000	104.7	62.4	63.96	251.0	0.42	1.385	0.749
0.645	0.270	0.375	105.3	6.000	89.7	62.4	63.91	235.1	0.38	1.430	0.774
0.620	0.265	0.355	105.3	5.000	74.8	62.4	63.87	222.4	0.34	1.469	0.795
0.600	0.265	0.335	109.1	4.000	62.0	64.7	66.09	217.2	0.29	1.564	0.848
0.585	0.265	0.320	112.1	3.000	47.8	66.5	67.83	212.9	0.22	1.643	0.892
0.565	0.265	0.300	114.3	2.000	32.5	67.8	69.09	203.3	0.16	1.728	0.939
0.815	0.270	0.545	84.4	14.000	167.8	50.0	51.82	277.1	0.61	0.962	0.514
0.780	0.270	0.510	91.5	12.500	162.4	54.3	55.97	280.0	0.58	1.074	0.577

7th Oct 1988 With 7 degree diffuser

UPSTREAM DEPTH (m)	D'NSTREAM DEPTH (m)	HEAD (m)	BRAKE SPEED (rpm)	TORQUE (ft.lbs)	POWER (W)	GEOMETRIC FLOW (l/s)	TOTAL FLOW (l/s)	THEO. POWER (W)	EFFIC'Y	Q1	N1
0.898	0.455	0.443	17.6	16.500	41.2	10.4	12.03	52.3	0.79	0.248	0.119
0.888	0.450	0.438	37.5	15.250	81.2	22.2	23.83	102.4	0.79	0.493	0.255
0.880	0.450	0.430	60.0	13.500	115.0	35.6	37.15	156.7	0.73	0.776	0.412
0.870	0.458	0.412	82.8	11.125	130.8	49.1	50.64	204.7	0.64	1.081	0.580
0.865	0.460	0.405	97.9	8.875	123.4	58.1	59.58	236.7	0.52	1.283	0.692
0.855	0.460	0.395	114.3	7.250	117.7	67.8	69.29	268.5	0.44	1.510	0.818
0.838	0.460	0.378	137.1	4.875	94.9	81.3	82.78	306.9	0.31	1.844	1.003
0.825	0.458	0.367	141.1	3.125	62.6	83.7	85.13	306.5	0.20	1.925	1.048
0.815	0.460	0.355	150.0	1.625	34.6	89.0	90.38	314.8	0.11	2.078	1.133
0.840	0.458	0.382	126.3	6.125	109.8	74.9	76.38	286.2	0.38	1.693	0.920
0.855	0.460	0.395	109.1	7.875	122.0	64.7	66.20	256.5	0.48	1.443	0.781
0.878	0.458	0.420	80.0	10.000	113.6	47.4	49.00	201.9	0.56	1.036	0.555
0.900	0.460	0.440	48.0	14.500	98.8	28.5	30.06	129.7	0.76	0.621	0.326
0.885	0.310	0.575	46.0	19.125	124.9	27.3	29.10	164.1	0.76	0.526	0.273
0.860	0.310	0.550	75.0	15.750	167.7	44.5	46.25	249.6	0.67	0.854	0.455
0.830	0.305	0.525	100.0	13.250	188.1	59.3	61.04	314.4	0.60	1.154	0.621
0.820	0.305	0.515	109.0	12.375	191.5	64.6	66.36	335.3	0.57	1.267	0.683
0.795	0.300	0.495	120.0	10.125	172.5	71.2	72.85	353.7	0.49	1.419	0.768
0.875	0.310	0.565	65.0	17.125	158.1	38.5	40.35	223.6	0.71	0.735	0.389
0.820	0.310	0.510	137.0	8.500	165.4	81.2	82.95	415.0	0.40	1.591	0.863
0.830	0.310	0.520	109.0	11.625	179.9	64.6	66.37	338.6	0.53	1.261	0.680
0.855	0.305	0.550	89.0	14.500	183.3	52.8	54.56	294.4	0.62	1.008	0.540
0.895	0.310	0.585	44.0	18.750	117.1	26.1	27.93	160.3	0.73	0.500	0.259
0.670	0.270	0.400	63.0	11.125	99.5	37.4	38.88	152.6	0.65	0.842	0.448
0.680	0.270	0.410	47.0	12.375	82.6	27.9	29.41	118.3	0.70	0.629	0.330
0.660	0.270	0.390	73.0	10.250	106.3	43.3	44.79	171.4	0.62	0.983	0.526
0.630	0.270	0.360	80.0	8.750	99.4	47.4	48.88	172.6	0.58	1.116	0.600
0.590	0.270	0.320	85.0	7.000	84.5	50.4	51.76	162.5	0.52	1.254	0.676
0.670	0.280	0.390	98.0	8.500	118.3	58.1	59.61	228.1	0.52	1.308	0.706
0.660	0.280	0.380	103.0	7.250	106.0	61.1	62.56	233.2	0.45	1.390	0.752

16th Oct 1988 With 7 degree diffuser

UPSTREAM DEPTH (m)	D'NSTREAM DEPTH (m)	HEAD (m)	BRAKE SPEED (rpm)	TORQUE (ft.lbs)	POWER (W)	GEOMETRIC FLOW (l/s)	TOTAL FLOW (l/s)	THEO. POWER (W)	EFFIC'Y	Q1	N1
0.568	0.268	0.300	70.6	6.875	68.9	41.9	43.18	127.1	0.54	1.080	0.580
0.570	0.264	0.306	55.8	7.875	62.4	33.1	34.42	103.3	0.60	0.852	0.454
0.576	0.276	0.300	86.9	5.625	69.4	51.5	52.85	155.5	0.45	1.322	0.714
0.584	0.282	0.302	104.0	4.000	59.1	61.7	62.99	186.6	0.32	1.570	0.852
0.590	0.290	0.300	120.0	2.625	44.7	71.2	72.47	213.3	0.21	1.813	0.986
0.578	0.272	0.306	80.5	6.125	70.0	47.7	49.06	147.3	0.48	1.215	0.655
0.570	0.268	0.302	66.7	7.000	66.3	39.6	40.87	121.1	0.55	1.019	0.546
0.578	0.278	0.300	93.8	4.875	64.9	55.6	56.94	167.6	0.39	1.424	0.771

24th Nov 1988 With 7 degree diffuser

UPSTREAM DEPTH (m)	D'NSTREAM DEPTH (m)	HEAD (m)	BRAKE SPEED (rpm)	TORQUE (ft.lbs)	POWER (W)	GEOMETRIC FLOW (l/s)	TOTAL FLOW (l/s)	THEO. POWER (W)	EFFIC'Y	Q1	N1
0.420	0.220	0.200	75.0	0.875	9.3	44.5	45.55	89.4	0.10	1.395	0.755
0.420	0.225	0.195	72.7	1.500	15.5	43.1	44.17	84.5	0.18	1.370	0.741
0.420	0.220	0.200	66.7	2.000	18.9	39.6	40.63	79.7	0.24	1.245	0.671
0.420	0.220	0.200	61.5	2.500	21.8	36.5	37.54	73.7	0.30	1.150	0.619
0.420	0.220	0.200	54.5	3.000	23.2	32.3	33.39	65.5	0.35	1.023	0.548
0.410	0.210	0.200	52.5	3.500	26.1	31.1	32.21	63.2	0.41	0.987	0.528
0.415	0.215	0.200	42.1	4.000	23.9	25.0	26.04	51.1	0.47	0.798	0.424
0.410	0.210	0.200	34.8	4.500	22.2	20.6	21.71	42.6	0.52	0.665	0.350
0.620	0.420	0.200	88.9	2.000	25.2	52.7	53.79	105.5	0.24	1.648	0.895
0.620	0.420	0.200	85.7	2.500	30.4	50.8	51.89	101.8	0.30	1.590	0.862
0.625	0.420	0.205	80.0	3.000	34.1	47.4	48.53	97.6	0.35	1.468	0.795
0.625	0.420	0.205	75.0	3.500	37.3	44.5	45.56	91.6	0.41	1.379	0.745
0.625	0.420	0.205	70.6	4.000	40.1	41.9	42.95	86.4	0.46	1.300	0.702
0.620	0.410	0.210	63.1	4.125	37.0	37.4	38.52	79.4	0.47	1.152	0.620
0.620	0.410	0.210	63.1	4.500	40.3	37.4	38.52	79.4	0.51	1.152	0.620
0.610	0.410	0.200	53.3	4.500	34.1	31.6	32.68	64.1	0.53	1.001	0.536
0.600	0.400	0.200	39.3	5.125	28.6	23.3	24.38	47.8	0.60	0.747	0.395
0.610	0.410	0.200	70.6	3.500	35.1	41.9	42.94	84.2	0.42	1.315	0.710
0.615	0.415	0.200	85.7	2.500	30.4	50.8	51.89	101.8	0.30	1.590	0.862
0.800	0.595	0.205	54.5	4.500	34.8	32.3	33.41	67.2	0.52	1.011	0.542
0.795	0.595	0.200	57.1	4.000	32.4	33.9	34.93	68.5	0.47	1.070	0.575
0.800	0.600	0.200	68.6	3.500	34.1	40.7	41.75	81.9	0.42	1.279	0.690
0.805	0.605	0.200	80.0	3.000	34.1	47.4	48.51	95.2	0.36	1.486	0.805
0.810	0.610	0.200	85.7	2.500	30.4	50.8	51.89	101.8	0.30	1.590	0.862
0.810	0.610	0.200	88.9	2.000	25.2	52.7	53.79	105.5	0.24	1.648	0.895

24th Nov 1988 With 7 degree diffuser

UPSTREAM DEPTH (m)	D'NSTREAM DEPTH (m)	HEAD (m)	BRAKE SPEED (rpm)	TORQUE (ft.lbs)	POWER (W)	GEOMETRIC FLOW (l/s)	TOTAL FLOW (l/s)	THEO. POWER (W)	EFFIC'Y	Q1	N1
0.700	0.390	0.310	27.9	9.000	35.7	16.5	17.88	54.4	0.66	0.440	0.225
0.695	0.395	0.300	52.2	8.000	59.3	31.0	32.27	95.0	0.62	0.807	0.429
0.690	0.400	0.290	70.6	7.000	70.2	41.9	43.16	122.8	0.57	1.098	0.590
0.690	0.400	0.290	82.8	6.000	70.5	49.1	50.39	143.4	0.49	1.282	0.692
0.710	0.410	0.300	92.3	5.000	65.5	54.7	56.05	165.0	0.40	1.402	0.758
0.710	0.410	0.300	75.0	4.250	45.3	44.5	45.79	134.8	0.34	1.145	0.616
0.720	0.420	0.300	114.3	3.000	48.7	67.8	69.09	203.3	0.24	1.728	0.939
0.710	0.420	0.290	120.0	2.250	38.3	71.2	72.45	206.1	0.19	1.843	1.003
0.720	0.000	0.720	120.0	9.875	168.3	71.2	73.20	517.0	0.33	1.182	0.636
0.730	0.000	0.730	109.1	12.125	187.8	64.7	66.75	478.0	0.39	1.070	0.575
0.750	0.000	0.750	109.1	12.250	189.8	64.7	66.77	491.3	0.39	1.056	0.567
0.770	0.000	0.770	109.1	13.750	213.0	64.7	66.80	504.6	0.42	1.043	0.559
0.750	0.000	0.750	92.3	15.500	203.2	54.7	56.81	418.0	0.49	0.899	0.480
0.760	0.000	0.760	49.0	17.000	118.3	29.1	31.15	232.2	0.51	0.490	0.253
0.745	0.000	0.745	53.3	18.500	140.0	31.6	33.68	246.1	0.57	0.535	0.278