Pumps as Turbines: A user's guide is a practical handbook for engineers and technicians involved in designing and installing small water-power schemes for isolated houses and communities. It concerns the use of standard pump units as a low-cost alternative to conventional turbines to provide stand-alone electricity generation in remote locations.

This book arises out of the practical experience of field work in village locations in a number of countries.

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Arthur Williams has worked with the Intermediate Technology Development Group on micro-hydro since 1987. He completed a Ph.D. at The Nottingham Trent University on the use of pumps as turbines, and continues to work there as a Senior Research Fellow. He has his own engineering consultancy, Hydro-Active Engineering, and has worked in Nepal, Pakistan, Sri Lanka and Guyana.

The Intermediate Technology Development Group (ITDG) was founded in 1965 by the late Dr E.F. Schumacher. Intermediate Technology enables poor people in the South to develop and use skills and technologies which give them more control over their lives and which contribute to the sustainable development of their communities.

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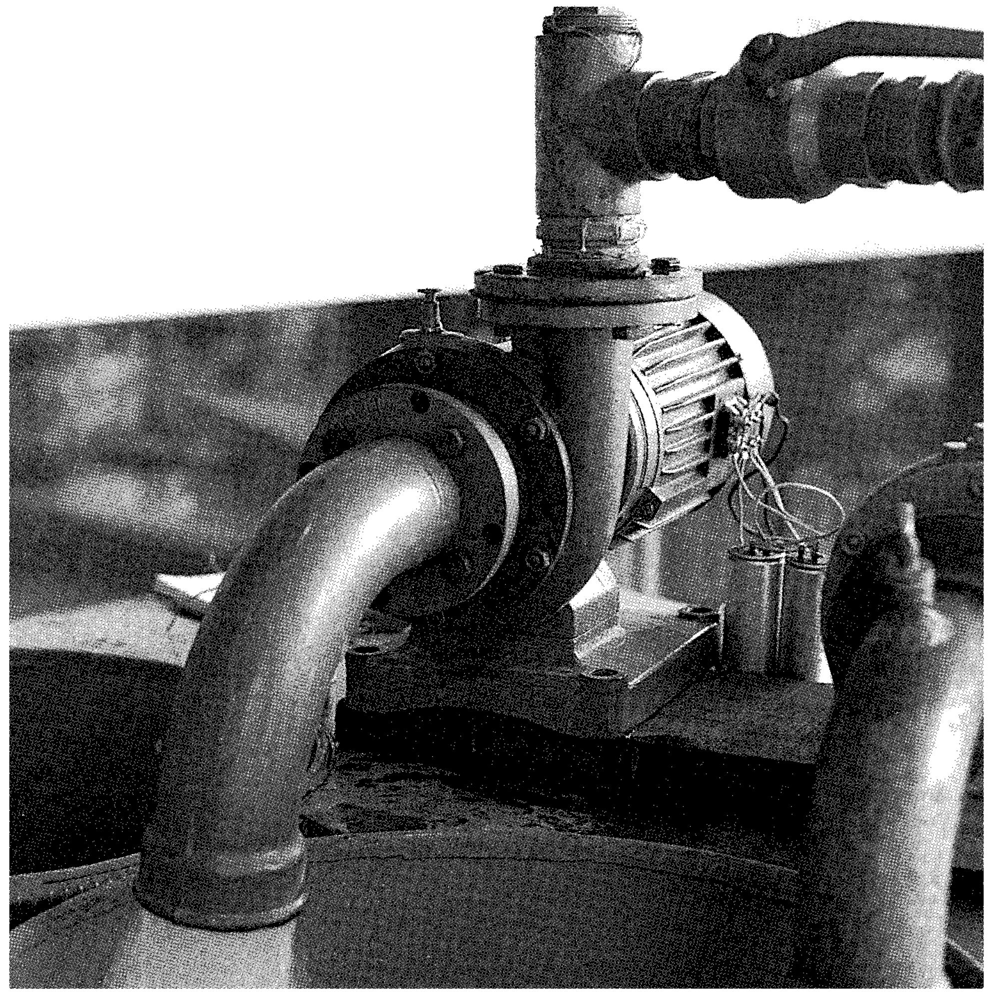

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Cover photo: A pump undergoing tests as a turbine at the MECO factory, Lahore, Pakistan. This pump was donated by the Pakistan Council for Appropriate Technology and later installed for field trials at a farm in North Yorkshire, UK.

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CONTENTS

PREFACE

Micro-hydro is a valuable source of energy for rural industries and village electrification schemes. It has been a traditional method of grain processing throughout the world and played a major role in modernization and industrial development in Europe and North America. Micro-hydro now offers similar potential to most developing countries, with applications in village lighting, mechanized food processing, and the supply of power to small-scale industrial activities.

This book is part of the effort to realize this great potential. It will help local manufacturers and rural development engineers to select a pump and convert it for use as a turbine for a micro-hydro scheme.

The development of this technology is the fruit of collaborative efforts between ITDG staff and colleagues and friends in several countries where the field work was carried out. It is an excellent example of what can be achieved by matching the resources of UK research institutions to needs in developing countries.

Take-up of the technology outlined in this book is now freely available to individuals, communities and organizations worldwide, helping to fulfil ITDG's strategic aim of wide dissemination of appropriate technologies, which can be manufactured locally at affordable cost. ITDG continues to widen the availability of appropriate technologies through its programme of training courses in micro-hydro and integrated rural energy. Specific courses are also held on induction generators and local manufacture of electronic controllers. For details, please write to ITDG.

ITDG Energy Unit
Intermediate Technology Development Group (ITDG)
Myson House, Railway Terrace, Rugby CV21 3HT, UK

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Thanks are also due to the agents and authors of the Fabulous Furry Freak Brothers for permission to reproduce the cartoon strip.

Arthur Williams
Nottingham, 1995

INTRODUCTION

This User's Guide has been written to help those who wish to install micro-hydro schemes in remote areas, using a standard pump unit as a low-cost alternative to a conventional turbine. The methods and types of equipment described in this guide are for isolated schemes (not connected to a grid supply) generating between 200 W and 30 kW.

The information and advice given are based on research and development work carried out at Nottingham Trent University with the Intermediate Technology Development Group (ITDG). The research concentrated on the use of pumps and motors as turbine and generator units. This guide complements the book by Nigel Smith, *Motors as Generators for Micro-hydro Power,* also published by IT Publications.

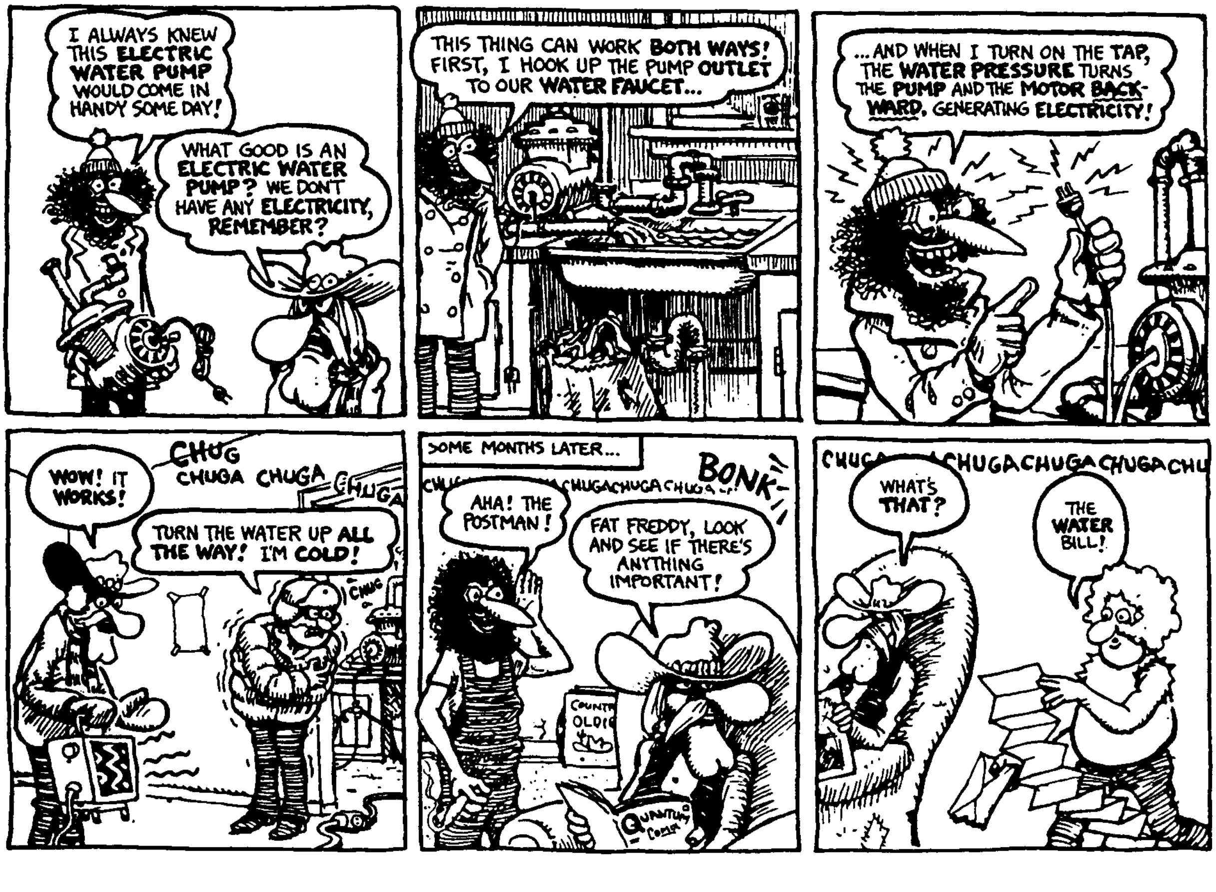
The writer draws on practical experience with pump-as-turbine schemes in the UK, India, Nepal and especially those installed by the Aga Khan Rural Support Programme in northern areas of Pakistan.

The first part of the guide covers general information on pumps for use as turbines. Typical applications are described in Chapter 1; reasons for using a pump as turbine in Chapter 2; the selection of pump type in Chapter 3. An explanation of the performance characteristics of pumps as turbines is given in Chapter 4.

Before selecting a pump as turbine for a particular site, other aspects of the micro-hydro scheme need to be considered. Civil engineering design is covered in Chapter 5 with electrical design in Chapter 6. The methods for selecting a pump as turbine for a specific site are covered in Chapter 7. Once the turbine is on site, tests and adjustments may be made to improve the performance, as detailed in the final chapter.

It has been assumed that the reader has an understanding of the basic concepts of head and flow rate, and of pump or turbine design and operation. The Guide has been written to take the reader through all the necessary design steps in a clear and simple way. It gives straightforward examples to show how to do the calculations.


THE FABULOUS FURRY
FREAK BROTHERS
by SHELTON, SHERIDAN, & MAVRIDES 1980
MAIL
FINAL NOTICE
THE ELECTRIC COMPANY
WHAT HAPPENED TO THE ELECTRICITY? THERE'S NO POWER!
CLICK CLICK
CLICK YANK
CLICK PULL
CLICK
MAYBE IT HAS SOMETHING TO DO WITH THIS ELECTRICITY BILL FOR $487.53!
HOW'RE WE GONNA RUN THE HEATER? HOW'RE WE GONNA RUN THE STOVE? HOW'RE WE GONNA RUN THE REFRIGERATOR?
IT'S IMPOSSIBLE TO LIVE WITHOUT ELECTRICITY!
YES AND NO! IT IS IMPOSSIBLE TO LIVE WITHOUT ELECTRICITY, BUT IT IS POSSIBLE TO GENERATE ELECTRICITY BY USING NATURAL WATER POWER!
YEAH, BUT WE'RE IN THE MIDDLE OF THE CITY!


I ALWAYS KNEW THIS ELECTRIC WATER PUMP WOULD COME IN HANDY SOME DAY!
WHAT GOOD IS AN ELECTRIC WATER PUMP? WE DON'T HAVE ANY ELECTRICITY, REMEMBER?
THIS THING CAN WORK BOTH WAYS! FIRST, I HOOK UP THE PUMP OUTLET TO OUR WATER FAUCET...
...AND WHEN I TURN ON THE TAP, THE WATER PRESSURE TURNS THE PUMP AND THE MOTOR BACKWARD, GENERATING ELECTRICITY!
CHUG CHUGA CHUGA CHUGA
WOW! IT WORKS!
TURN THE WATER UP ALL THE WAY! I'M COLD!
CHUG
SOME MONTHS LATER...
BONK
AHA! THE POSTMAN!
FAT FREDDY, LOOK AND SEE IF THERE'S ANYTHING IMPORTANT!
WHAT'S THAT?
THE WATER BILL!

1. APPLICATIONS FOR PUMPS AS TURBINES

Common applications for pumps as turbines are given below.

Village schemes, mainly for household lighting
In many village schemes in developing countries, where the main electric load is evening lighting, a pump as turbine (PAT) is suitable. During the daytime the generator can also be used to power equipment that will benefit the economy of the village, e.g. a circular saw, crop dryer or sewing machines. In this type of scheme, the PAT will be designed to run using the flow available at the driest time of the year.

Sometimes, where only lighting is needed, the water supplying the micro-hydro scheme can be used during the daytime for irrigation or for running a water mill for grinding corn.

Electricity for remote farms
Pumps as turbines have been successfully used for supplying electricity to a remote farm in the UK. Since the output of the generator is limited, some care has to be taken in switching on appliances, not to overload the system. Spare power, not used in general appliances, can be used for background heating.

Battery charging and other intermittent load applications
In Sri Lanka, many households depend on lead-acid batteries for their electricity. These are usually taken to the nearest town for re-charging. A PAT scheme could be used to provide power for battery charging more locally. In this case, it is not essential that the power is available continuously, and it is possible to run a PAT intermittently. Other applications for an intermittently operating PAT are for running refrigerators for ice-making or vaccine storage, or for running electrically heated crop dryers.

Water pumping
Pumps as turbines can also be used for pumping water for domestic use, where a farm or village is situated above the main stream level. In this case the PAT is connected directly to a high-head centrifugal or positive displacement pump, which pumps a small quantity of water to a high head. This type of scheme has been used in Tanzania.

2. WHY USE A PUMP AS A TURBINE?

Standard pump units when operated in reverse have a number of advantages over conventional turbines for micro-hydro power generation. Pumps are mass-produced, and as a result, have advantages for micro-hydro compared with purpose-made turbines. The main advantages are as follows:

- Integral pump and motor can be purchased for use as a turbine and generator set
- Available for a wide range of heads and flows
- Available in a large number of standard sizes
- Low cost
- Short delivery time
- Spare parts such as seals and bearings are easily available
- Easy installation - uses standard pipe fittings

There are several practical benefits of being able to use a direct drive pump as turbine (PAT), i.e. one in which the pump shaft is connected directly to the generator, as explained in the next section.

Pump suppliers usually stock a number of different pumps designed to be suitable for a wide range of heads and flows. The actual range of heads and flows for which a PAT is suitable is explained in a later section.

The simplicity of the PAT means that it does have certain limitations when compared with more expensive types of turbine. The main limitation is that the range of flow rates over which a particular unit can operate is much less than for a conventional turbine. Some ways of overcoming this limitation are covered at the end of this chapter.

Using a Direct Drive Pump as Turbine

One of the advantages of using a PAT instead of a conventional turbine is the opportunity to avoid a belt drive. However, in some circumstances there are advantages to fitting a belt drive to a PAT. The advantages of using a direct drive arrangement are summarized below.

- Very low friction loss in drive (saving up to 5% of output power).
- Easier installation - PAT and generator come as one unit.
- Lower cost - no pulleys, smaller baseplate.
- Lower cost (in the case of a 'mono-bloc' design) because of simpler construction, fewer bearings, etc.
- Longer bearing life - no sideways forces on bearings.
- Less maintenance - no need to adjust belt tension or replace belts.

The use of combined pump-motor units is recommended for micro-hydro schemes that are to be used only for the production of electricity, and where the simplest installation possible is required. There are, however, some limitations to using such integral units, as listed below:

- Turbine speed is fixed to speed of generator - thus reducing the range of flow rates when matching the PAT performance to the site conditions.
- Limited choice of generators available for a particular PAT.
- No possibility of connecting mechanical loads directly to the PAT.

Suitable Range of Site Heads and Flows

Standard centrifugal pumps are manufactured in a large number of sizes, to cover a wide range of heads and flows. Given the right conditions, pumps as turbines can be used over the range normally covered by multi-jet Pelton turbines, crossflow turbines and small Francis turbines. However, for high head, low flow applications, a Pelton turbine is likely to be more efficient than a pump, and no more expensive.

The chart in Fig. 1 shows the range of heads and flows over which various turbine options may be used. The range of Pelton and crossflow turbines shown is based on information from the range of turbines manufactured in Nepal, and is compared with the range of standard centrifugal pumps running with a four-pole (approx. 1500 rpm) generator. The range of PATs can be extended by using either a two-pole (approx. 3000 rpm) or a six-pole (approx. 1000 rpm) generator, as shown in Fig 2. This range of pumps as turbines is based on standard centrifugal pumps produced by a major UK manufacturer.

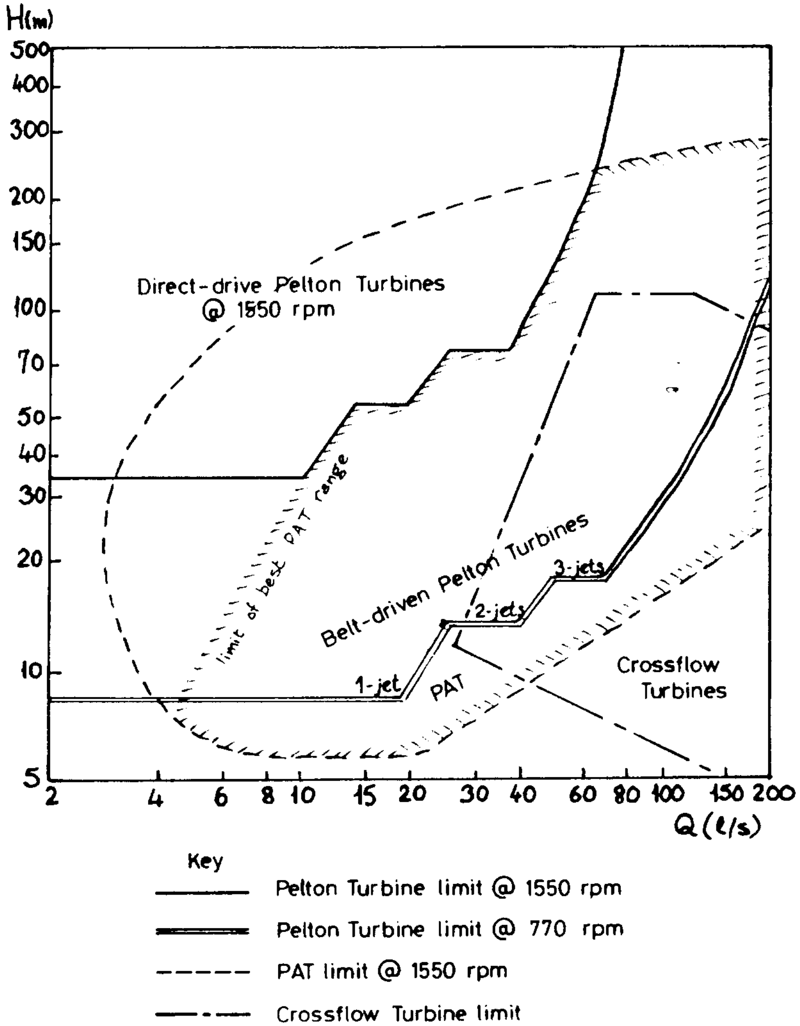


Fig. 1. Head-flow ranges for various turbine options.

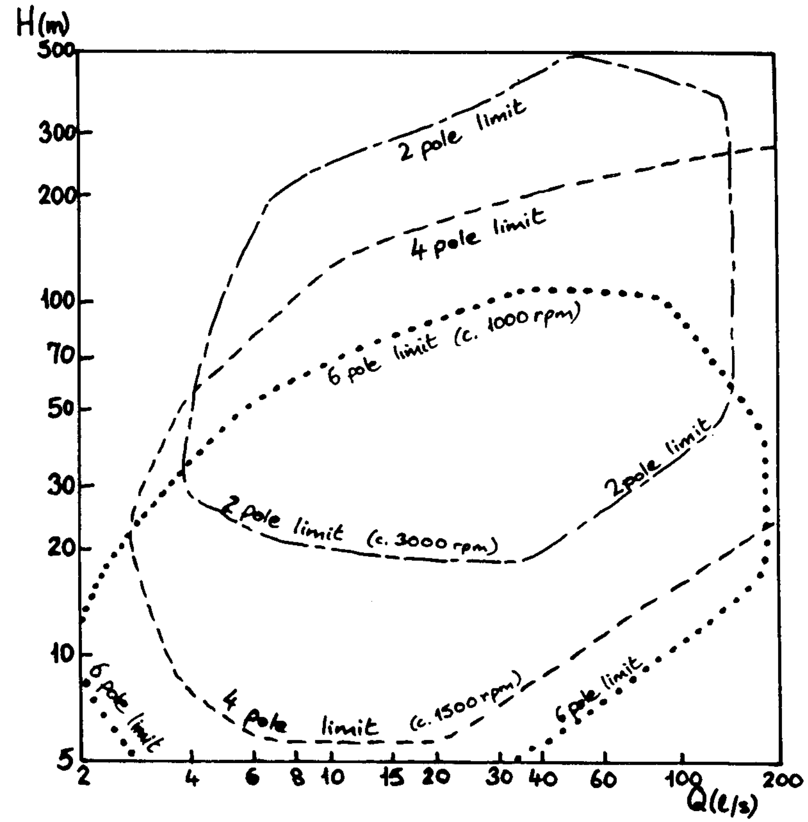


Fig. 2. Head-flow ranges for direct drive pumps as turbines.

The use of a pump as turbine has greatest advantage, in terms of cost and simplicity, for sites where the alternative would be either a crossflow turbine, running at relatively low flow, or a multi-jet Pelton turbine. For these applications, shown by the hatched area on Fig. 2, a crossflow turbine would normally be very large compared with an equivalent PAT. Very small crossflow turbines are more expensive to manufacture than larger ones because of the difficulty of fabricating the runner. Therefore, a crossflow installation would require a large turbine running at slower speed than an equivalent PAT, resulting in the need for a belt drive to power a standard generator. A Pelton turbine for this application would require three or four jets, resulting in a complicated arrangement for the casing and nozzles, although it would be more flexible than a PAT for running with a range of flow rates. A small Francis turbine could also be used in this range, but would be even more expensive than a crossflow turbine.

What dictates the use of a pump as turbine is that it requires a fixed flow rate and is therefore suitable for sites where there is a sufficient supply of water throughout the year. Long term water storage is not generally an option for a micro-hydro scheme because of the high cost of constructing a reservoir.

Overcoming the Limitations of Using a Pump as Turbine

A purpose-built water turbine is generally fitted with a variable guide vane (or vanes) or a spear valve, which allows the machine to run efficiently with a wide range of flow rates. When a standard centrifugal pump is used as a turbine, no such adjustment is possible. However, once installed, a pump as turbine that is well matched to the site conditions will operate close to maximum efficiency.

If the flow rate falls a little below that required for maximum efficiency, power can still be generated - but less power will be obtained. This is explained in more detail in Appendix A. Another option for dealing with low flow rates is to use intermittent operation. By using a special intake and a small storage tank it is possible for a PAT to run intermittently. The special intake consists of a syphon arrangement, which is described in Appendix C.

If the flow rate increases, it is not possible to generate more power using only one pump. A second pump could be installed but the additional cost of installing more than one unit may outweigh the advantage of buying a pump instead of a conventional turbine. Appendix B gives more details of parallel operation of PATs.

When a direct drive electric pump is used, the turbine and generator must run at the same speed. This can limit the range of flows over which the pump can run. Care must be taken to avoid overloading (either electrical or mechanical) of the generator. The electrical output of an induction generator should normally be limited to 80% of the rated power output as a motor.

3. CHOICE OF PUMP TYPE

This chapter describes the main types of pumps and explains which of these are suitable for use as turbines. It also gives suggestions for checking the manufacturing quality of a pump before purchase, and gives important information relating to the choice of pump speed, size and seal type.

Pump Types and Suitability

Investigations of various pumps as turbines, suggest that there are several types that are suitable for micro-hydro, running in reverse with an induction generator, as listed below.

Centrifugal pumps	End-suction	most suitable
	In-line	less efficient
	Double suction	less efficient
	With round casing	inefficient
Axial flow pumps (for low-head sites)		small sizes not available
Self-priming pumps		only suitable if valve is removed
Submersible pumps	Dry-motor, jacket cooled	suitable
	Dry-motor, fin cooled	unsuitable
	Wet-motor, borehole type	unlikely to be suitable
Positive displacement pumps	(e.g. gear pumps, mono pumps, piston pumps)	All unsuitable

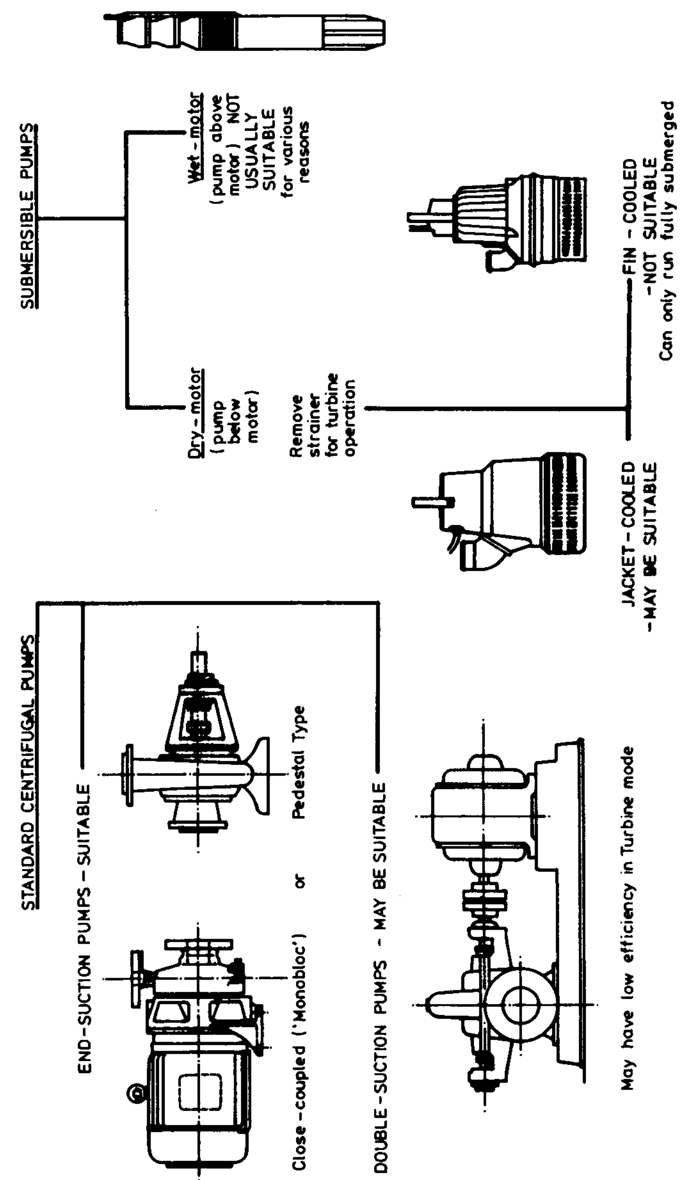


Fig. 3. Pumps suitable for use as turbines.

It is wisest to stick with standard centrifugal pumps, which are the most easily available and cheapest type. The list above and the chart in Fig. 3 give a summary of various types of pump, together with comments about their suitability, for cases where a pump may already be available.

Standard centrifugal pumps are widely used for water supply and irrigation, are of relatively simple design, and easy to maintain. They also cover a wide range of heads and flows. They are sometimes referred to as 'end-suction' pumps, as they have a single suction pipe that is at right-angles to the outlet pipe.

A centrifugal pump may have its own bearings, or may be integral with an induction motor that contains the necessary bearings. Integral units are sometimes referred to as 'monobloc' pumps. On smaller sizes of end-suction pump, the impeller is screwed on to the shaft. However, the impeller will not unscrew in turbine mode, unless a solid object causes the impeller to stick, because the torque is normally applied in the same direction as in pump mode.

Some small centrifugal pumps do not have a spiral volute, but a simple round casing with an angled outlet pipe (see Fig. 4). Although this type of unit may have a high pump efficiency, it has been found to run inefficiently as a turbine.

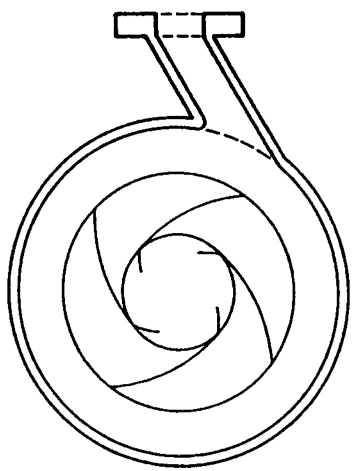


Fig. 4. Pump volute with no spiral.

Dry-motor submersible pumps are widely used for construction site drainage and for pumping from open wells. The motor is integral with the pump, and in the type suitable for micro-hydro use, it is cooled by the pumped water flowing through a jacket on the motor housing. However, some pumps of this type have rubber linings on the diffuser parts, which may prevent the impeller from running in reverse.

Wet-motor submersible borehole pumps are of a more specialized design but may be suitable for use as turbines. They usually contain a non-return valve, and may also incorporate a thrust bearing designed only for pump operation.

Dry-motor submersible pumps with fin-cooling are not suitable for use as turbines, as they will overheat unless submerged below water level.

Choice of Pump Quality

In less industrialized countries it is important to check the quality of the pump to be used as a turbine. In many countries, there are small workshops making centrifugal pumps to designs copied from the larger manufacturers. Many of these pumps are of low quality, and are not recommended for use as turbines as they will have poor performance and will not last long when used as micro-hydro turbines.

To check the quality of a pump, carry out the following five inspections:

1. Impeller eye clearance. Inspect the opening at the front of the impeller that fits to the suction inlet of the pump (known as the impeller eye, see Fig. 5) to see how closely it has been machined to match the casing. Poor quality pumps often have no machining at the impeller eye, leaving a large clearance, which results in very high leakage losses, causing significant inefficiency when running as a turbine.

2. Casting quality. Check for fins, etc. in the inside of the pump. Avoid pumps in which the insides of the castings are very rough as these will be inefficient.

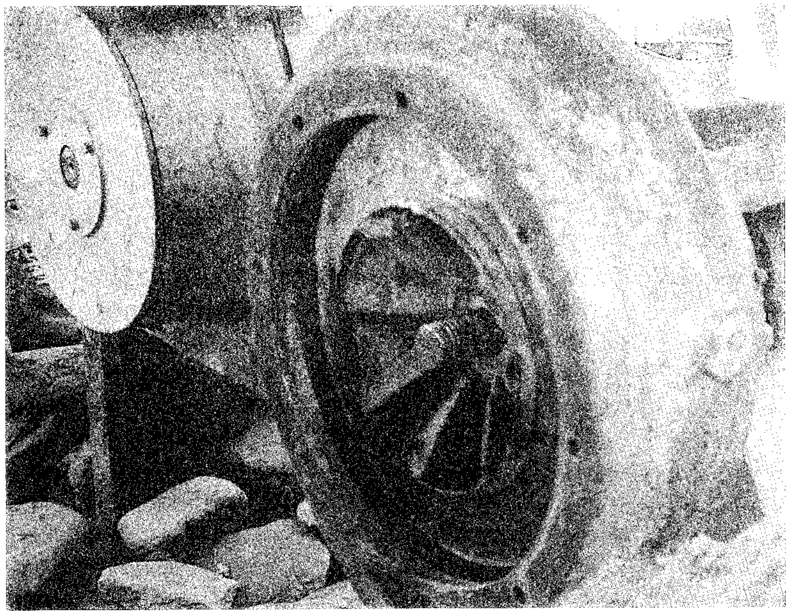


Fig. 5. Impeller eye of poor quality pump.

3. Impeller material. Some small centrifugal pumps have the impeller made from carbon-impregnated nylon. This material is satisfactory for use in a PAT, as long as there are not many hard particles in the water. Other types of plastic should be avoided. Pumps with a bronze or cast iron impeller should be suitable.

4. Shaft material. If the shaft is easily marked by scratching with a knife, then the pump should not be used since it is probably made of mild steel which will corrode and wear, putting more strain on the bearings, etc.

5. Bearing quality. Tests have shown that the axial thrust in turbine mode is lower than in pump mode. However, bearing wear was found to be a problem on belt-driven pumps as turbines installed in Northern India, and in Pakistan. On direct drive units there is less likely to be a problem, since there is much less radial load on the bearings.

Choice of Pump Seal Type

Both mechanical seals and stuffing glands have been found to operate satisfactorily in turbine mode. Stuffing glands are used in many developing countries because of their low cost, and the availability of replacement material. Mechanical seals are preferred if they can be obtained easily from the pump manufacturer, since they require less maintenance (as long as they are correctly fitted) and cause less resistance to the rotation of the pump shaft.

If a PAT is fitted with gland packing then it is important that there is a water-thrower disc to prevent leakage of water directly into the bearing housing. If such a disc is not fitted, one can be made from sheet metal and secured to the shaft with a jubilee clip, as illustrated in Fig. 6. Care must be taken with certain types of seal, which incorporate a spring, to ensure that the spring will operate effectively with reverse rotation.

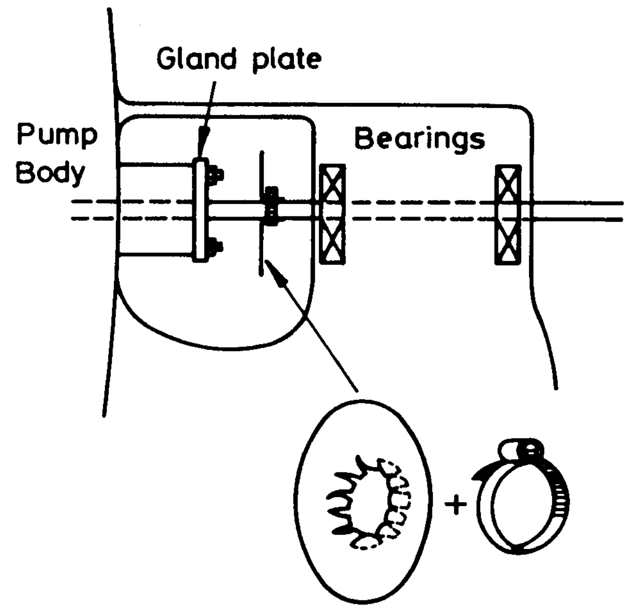


Fig. 6. Water thrower disc for pump with gland packing.

Choice of Pump Speed

The choice of pump speed will depend on the head and flow rate at the site. This is explained in the next section of this chapter. For many sites there may be pumps of different speeds that are suitable. A higher speed pump will be more compact and cheaper than a lower speed equivalent but it will wear more rapidly. Bearings and seals will need to be replaced more often and the impeller itself may become worn if a high speed is used. A low speed pump will be more expensive to buy but will last longer and need less maintenance.

Understanding Standard Pump Sizes

Pump manufacturers often use a series of three numbers to define the size of standard centrifugal pumps. These numbers represent the nominal diameters of:

- suction (inlet) pipe
- discharge (outlet) pipe
- impeller

The smallest dimension refers to the discharge diameter and the largest dimension refers to the impeller diameter. For example, a pump designated 3"×2"×8" will have a discharge diameter of 2", an impeller diameter of 8" and a suction pipe diameter of 3". This pump might also be designated 2"×3"×8" or in metric sizes 80-50-200 (all dimensions in mm).

Standard metric sizes for the pipe connections are: 25,32,50,65,80,100,125,150 and 200 mm.

Some manufacturers give only two numbers. These do not always refer to the same dimensions. For imperial sizes the most common notation is discharge diameter followed by suction diameter (e.g. 2"×3"). For metric sizes, the most common notation is suction diameter in mm followed by impeller diameter in this case given in cm (e.g. 50-20)

The nominal impeller diameter refers to a typical impeller size for the pump casing (e.g. 8"). The actual impeller diameter may vary from, say, 6½" to 8¼". It is important when using a pump as a turbine that the impeller is no less than 90% of the nominal diameter.

Using the same example, the minimum impeller diameter for a nominal 8" pump is:

$$\frac{90}{100} \times 8" = 7.2"$$

The reason for this is that the efficiency is likely to be reduced, particularly in turbine mode. Pumps with 'cut-down' impellers should therefore be avoided when making the initial selection of a PAT.

4. PUMP AND TURBINE PERFORMANCE CURVES

Understanding Pump as Pump Performance Curves

Before looking at your pump as a turbine, you need to understand it as a pump. The main tool for this is the performance curve, which shows how the head and flow delivered by the pump are related. Fig. 8 shows a typical head-flow curve for a pump. As the flow delivered by the pump increases, the delivery head decreases. This curve is often available from the pump manufacturer.

The other piece of information that you need to know for your pump is the point at which it works most efficiently. This is called the *best efficiency point.* The pump efficiency, plotted against the flow rate, is shown in Fig. 7. The maximum value of efficiency varies according to the type and size of pump, but is typically 40% to 80%. The best efficiency point (bep) occurs at a particular value of flow rate.

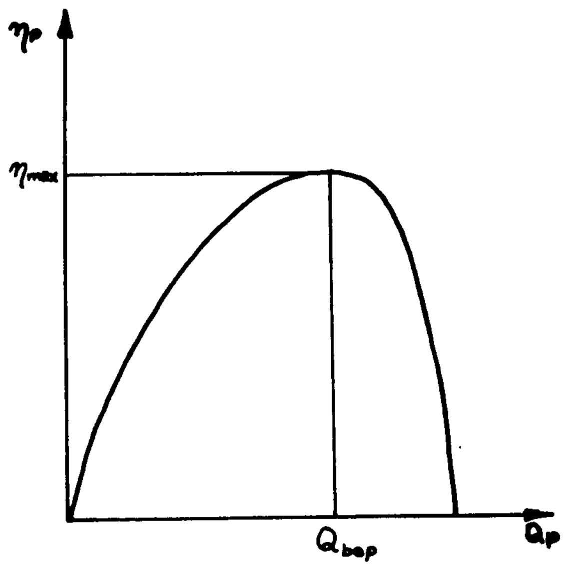


Fig. 7. Pump efficiency curve.

The efficiency values can be shown on the head-flow curve, as shown in Fig. 8. Information from pump manufacturers is sometimes shown in this way.

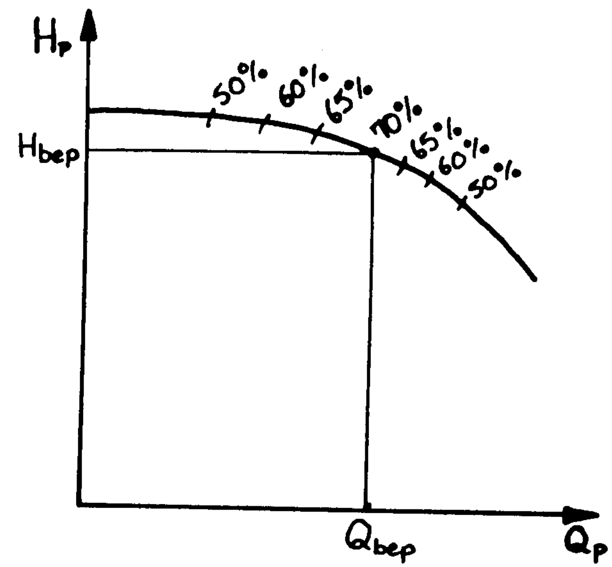


Fig. 8. Pump head and flow, with efficiency values shown.

If you have no efficiency data for the pump, but do have a curve showing input power against flow rate, then it is possible to calculate the values at the best efficiency point. The relationship between head, flow-rate, input power and efficiency is given by the following equation:

$$Efficiency(\eta) = \frac{H \times Q \times 9.81}{P_{in}} \times 100 \qquad (1)$$

where: H is head (m)
Q is flow rate (l/s)
P_{in} is mechanical input power (W)
9.81 is acceleration due to gravity (m/s^2)
η is pump efficiency as a percentage.

The steps for calculating the value of maximum efficiency are as follows:

1. Use the head-flow curve to obtain the head and flow rate at best efficiency point (bep)
2. Use this flow rate on the power input-flow curve to get P_{in}
3. Put these values in equation (1) to obtain the efficiency.

Note that, especially for pumps with integral motors, the power curve may show *electrical* power consumption rather than *mechanical* input power. In this case, use Appendix D to estimate the efficiency of the motor. Then use the following equation to calculate P_{in}:

$$P_{in} = P_{elec} \times \frac{\eta_{motor}(\%)}{100} \qquad (2)$$

where: P_{in} is mechanical input power (W)
P_{elec} is the electrical power consumption of the motor (W)
η_{motor} is motor efficiency as a percentage.

Example 1: *Finding pump best efficiency conditions.*

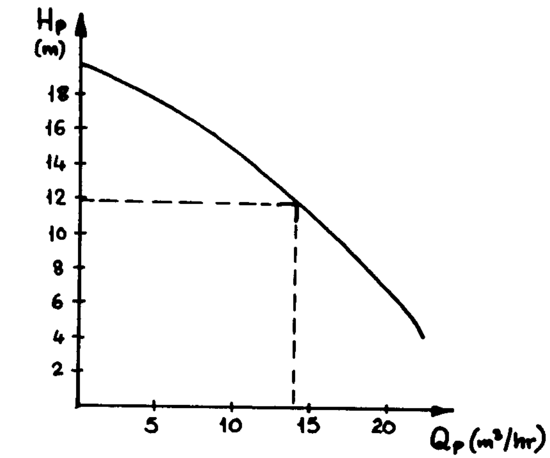
The manufacturer of a 65-40-200 (2.5" × 1.5" × 8") pump gives the head-flow curve and electrical power input curve as shown below in Figs. 9a and 9b. The flow at best efficiency is 14 m^3/hr, which can be converted to 3.89 l/s by dividing by 3.6, the conversion factor given in Appendix E. The head at best efficiency is 11.8 m.

The motor is rated at 1.5 hp (1.1 kW), 1450 rpm, for operation on a 3-phase, 50 Hz supply. According to the table in Appendix D, this size of motor has a maximum efficiency of around 75%. The value of electrical power consumed, for the best efficiency point, can be found from Fig. 9b. At a flow rate of 14 m^3/hr, the power is 1050 W. This is P_{elec}. Using equation (2):

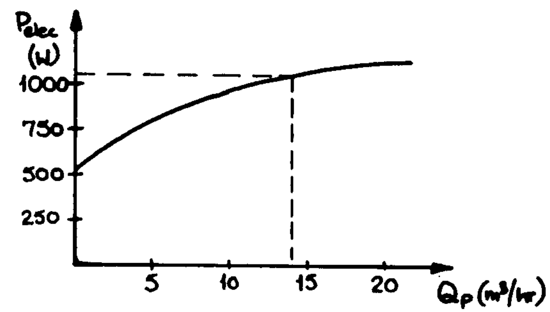
$$P_{in} = P_{elec} \times \frac{\eta_{motor}(\%)}{100} = 1050 \times \frac{75}{100} = 788\ W$$

The pump best efficiency is therefore, from equation (5):

$$\eta = \frac{H \times Q \times 9.81}{P_{in}} \times 100 = \frac{11.8 \times 3.89 \times 9.81}{788} \times 100$$
$$= 57\%$$



(a) Head and flow, with best efficiency point.



(b) Electrical power consumption

Fig. 9. Manufacturer's pump curves.

Obtaining the Best Efficiency Point with Limited Data

If the best efficiency point is not known, but you have a power curve, calculate the efficiency using equation (1) as above, for a number of different flow rates. By a trial and error method, obtain the maximum efficiency. The head and flow corresponding to the maximum efficiency will define the best efficiency point.

Sometimes, no curve is available that shows either input power or electrical power consumption. In this case, some information may be obtained from the pump name plate. The data given on the pump name plate may consist of a single value for head and for flow (which is not always the head and flow for best efficiency pump operation) or a range of heads and flows. One approximation for the best efficiency conditions can be made by using:

$$Q_{bep} = 0.75Q_{\max}; \quad H_{bep} = 0.75H_{\max} \tag{3}$$

A useful check can be made on these estimates by an alternative method, which is based on physical measurements of some parts of the pump. This method is detailed in Appendix F.

Understanding Pump as Turbine Performance Curves

The performance curve for the turbine shows how the head is related to the flow through the turbine (see Fig. 10). For turbine operation, the flow increases with increasing head. The single curve shown is for the normal operating speed, i.e. that determined during detailed design.

It is also possible to plot the curve showing the head and flow available at the site (see Fig. 11). This is the head available at the turbine and is equal to the vertical height between the intake from the stream and the turbine outlet, less the frictional head loss in the penstock. The intersection of the turbine performance curve and the site curve in Fig. 11 gives the head and flow at which the turbine will actually operate. This is known as the operating point.

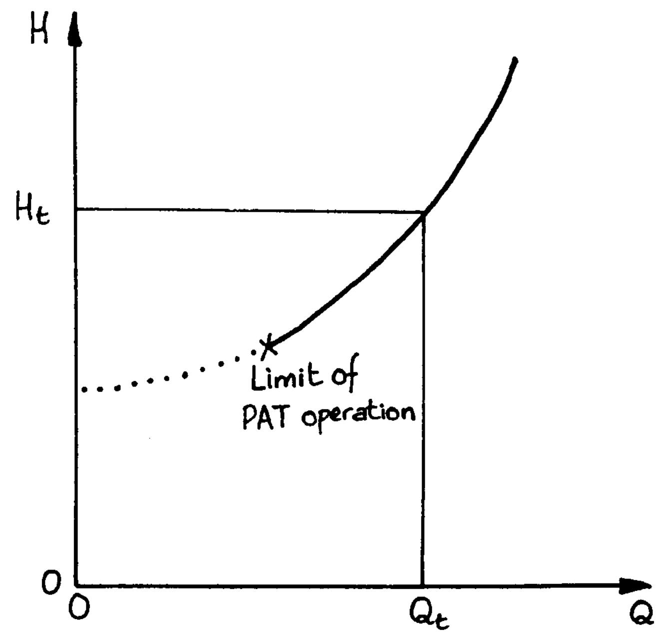


Fig. 10. Pump as turbine head and flow.

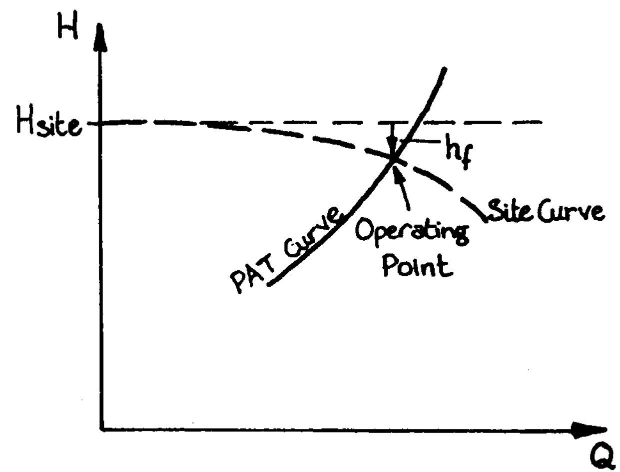


Fig. 11. Turbine curve and site curve.

The speed of the turbine will vary according to the load that is put on it, and there is a different head-flow curve for each speed. Three such curves are shown in Fig. 12. The middle curve, labelled N=100% is for the normal operating speed (the same as in Fig. 10). The curves labelled N=130% and N=80% are for speeds 30% higher and 20% lower than normal operating speed. Note that for each speed, the operating point is given by the intersection of the turbine curve with the site curve.

If a load, which is higher than design load, is put on the turbine, the speed goes down. For the pump shown in Fig. 12, this causes a slight increase in flow rate, which is usually the case for centrifugal pumps running as turbines. When the load on the turbine is reduced, the speed increases. If there is no load, the speed of the turbine increases to a maximum, which is known as *runaway.* The curve of maximum speeds is also shown on Fig. 12 (labelled N=max). In the case illustrated, the actual speed at runaway is (by extrapolation) approximately 140% of normal operating speed.

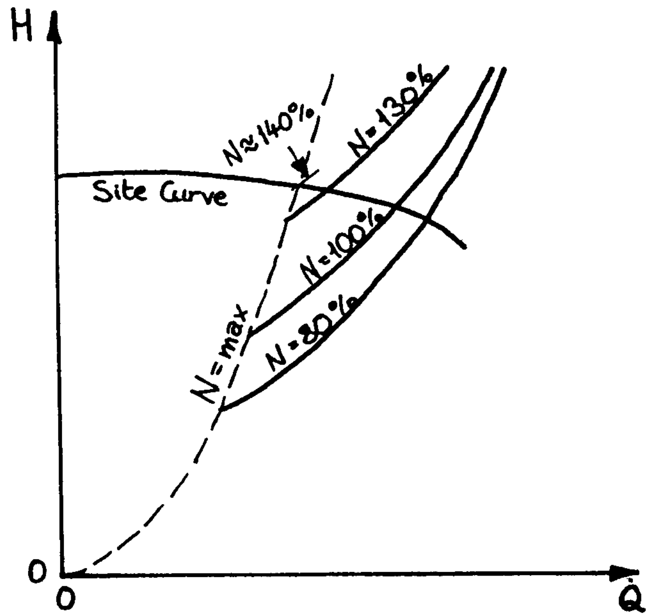


Fig. 12. Turbine head and flow at different speeds.

5. DESIGN OF THE CIVIL WORKS

This chapter gives information on the design of the civil works that are associated with a micro-hydro scheme. The subject is too large to be covered in this one chapter, so these are some suggestions for designs that are particularly appropriate when using a pump as turbine (PAT).

Intake Requirements

The term intake refers to the structure at the point where the stream water is taken into the micro-hydro scheme. This requires the same elements for a PAT scheme as for any run-of-river hydropower plant. What is different about a PAT scheme is the relatively small size of the impeller channels, and the type of seals, which require careful attention to be given to the intake design. The intake structure is designed to carry out four separate functions:

1. Collecting the water from a stream to divert it into the penstock.
2. Separating stones, sand and silt from the water.
3. Preventing floating particles (grass, leaves, peat, etc.) from entering the penstock and blocking the turbine.
4. Providing an overflow to take away excess water. The intake must be designed to take away the largest flood flow that can be anticipated, otherwise the whole structure may be washed out during flood conditions. The penstock must be keyed into the intake structure and must be protected from falling debris during flood conditions.

There are many suitable designs of intake, which can be found in other books (see Further Reading list). For a PAT installation the main requirements are that no floating items large enough to stick in the impeller channels can be allowed to enter the penstock. For example, if the impeller channels are 10 mm wide, the wires on the trashrack should be at most 6 mm apart.

The intake must also collect any hard silt that is likely to catch in the pump seal. This must then be flushed away from time to time. If silt enters the pump, the life of either a mechanical seal or gland packing will be severely shortened. One example of an intake design found to be suitable for a small PAT installation is shown in Fig. 13.

(1) Concrete retaining wall with overflow.
(2) Drainage pipe to flush away silt, with plunger and bung.
(3) Smooth concrete base - maximum 10% slope.
(4) Concrete skirts to prevent undercutting.
(5) Cylindrical wire mesh trashrack - top is above water level in order to act as a vent.
(6) Buried plastic penstock.

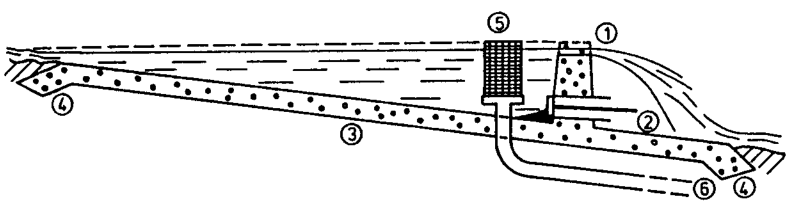


Fig. 13. An intake suitable for a small PAT system.

Design of Penstock

The penstock is often the most expensive part of a micro-hydro scheme. It is therefore important to design the penstock for minimum cost, without compromising the engineering specifications required. The choice of material will depend on what is available locally, the pressure requirement, the difficulties of transport and the technology available for making joints in the pipe. The choice of pipe diameter will determine the head loss and will depend on the expected flow rate.

Steel pipe is usually more readily available than plastic pipe, but is heavier to transport. Plastic pipe suitable for use as a penstock is usually of either uPVC or MDPE. MDPE pipe is more flexible than uPVC and may be available in longer lengths, thus requiring fewer joints. uPVC pipe is usually cheaper, and is available in 6 metre lengths only. These may be joined by heat welding or by using pipe with collars and O-ring seals. Both types of plastic pipe should be buried to prevent degradation due to exposure to sunlight. Before burying the penstock, it is advisable to check the joints for leaks.

The pipe must be able to withstand the pressure that can occur due to sudden blockages in the penstock, which result in pressure surges known as water-hammer. MDPE and uPVC pipe ratings are designed to withstand these surge pressures, but for steel pipe it is necessary to calculate the safe wall thickness, as shown in Appendix G.

It is also advisable to install a vent pipe near the intake. This is a small vertical pipe connected to the upper side of the penstock and is open to the atmosphere.

Since the PAT is a fixed flow unit, the penstock diameter can be chosen to match the required flow, and the head loss calculated accurately. The head loss in the penstock should never be more than one-third of the gross head, otherwise an increase in flow will result in a decrease in available power. It is important to take into account that the PAT selected may take a larger flow rate than anticipated. This can result in a much larger head loss than originally calculated.

From the author's experience, the most economical design for the penstock is to allow a head loss of between 10% and 20% of gross head. Appendix H shows how to select the most suitable size of penstock, and how to calculate the head loss.

At the PAT inlet a reducer is usually required because the turbine flow is greater than the pump flow, and the penstock is therefore of a larger diameter than the outlet pipe required when pumping.

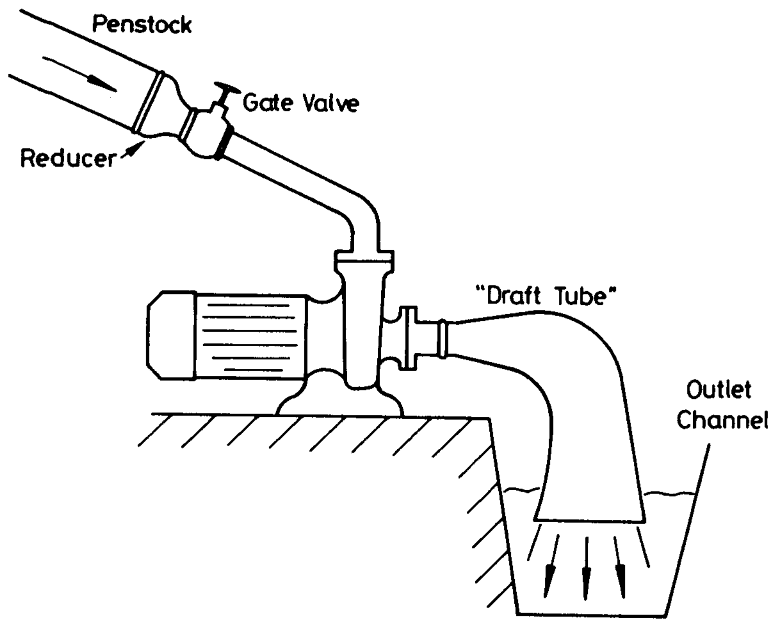


Fig. 14. Pipework arrangement for PAT.

Valves

It is usual practice to put a valve at the bottom of the penstock, before the water enters the turbine. This may not be necessary where the intake is close to the turbine and can be fitted with a sluice. The valve must be of a type that cannot be closed quickly, otherwise large pressure surges can occur in the penstock. It is therefore recommended to use a gate valve. The cost of the gate valve can be reduced by inserting it immediately before the PAT, where the pipe diameter has been reduced from the diameter of the main penstock. This also has the advantage that the water flow will be cut off more evenly as the valve is closed.

Fig. 14 shows a recommended layout for the PAT and adjoining pipework.

Pump as Turbine Outlet

At the outlet from the PAT, it is advisable to use an expanding pipe, known as a draft tube, or a reducer in reverse to increase the diameter of the pipe where it discharges into the outlet channel. The water in the outlet channel is at atmospheric pressure, and therefore the channel must be large enough and with a great enough slope to take away the full flow from the PAT. If a pipe is used to return the water from the outlet channel to the stream, this pipe must be at least twice the diameter of the penstock to take the water away without causing the outlet channel to overflow.

6. DESIGN OF THE ELECTRICAL SYSTEM

Various options exist for the design of the electrical system to be installed with a PAT. Choices must be made concerning the type of generator, number of phases and the type of control to be used.

Choice of Generator Type and Number of Phases

The generator may be a conventional synchronous generator or an induction generator. Many pumps are available with an induction motor directly coupled, which can be used as a self-excited generator. This type of unit is likely to be less expensive than a separate PAT and synchronous generator. Induction generators are usually more easy to obtain, especially in less industrialized countries. The rotor is stronger than in a synchronous generator, and cannot be damaged even if the generator load is disconnected and the turbine speeds up to runaway. Another advantage of an induction generator is that it cannot be burnt out through overloading, since it simply loses excitation and stops generating under these conditions.

For smaller systems (less than 15 kW), it is recommended to use a single-phase distribution system, unless three-phase power is required for motors. However, for long transmission lines, the use of three-phase distribution can cut the cost of the cable by up to 70%. Where the generator is more than 500 metres from the main loads, three-phase transmission should be considered.

Single-phase transmission has the advantage of not needing the loads to be split into three equal parts, which may be difficult to achieve. Single-phase synchronous generators are available up to 15 kW, but single-phase induction machines are not normally available for outputs greater than 4 kW. An alternative is to use a three-phase induction generator with a single-phase output, using the 'C-2C' connection as shown in Fig 15. Here the load is connected to one phase, while the currents in the generator are balanced (for one particular value of load). Even for powers of less than 4 kW, a three-phase generator connected in this manner may be more appropriate than a single-phase unit. A single-phase machine is likely to be more expensive, but less efficient than the equivalent three-phase machine. Note that for this connection, the

induction machine must be connected for 220/240 V delta, if this is the single-phase voltage required.

The size of the capacitors determines the voltage and speed at which the generator will run. As an example, a 1.1 kW generator needs capacitors of 25 + 50 µF. If the capacitance is increased, the generator will run at a lower speed or higher voltage. Further information on the sizing of the capacitors is given in the handbook by N Smith, *Motors as Generators for Micro-hydro Power.*

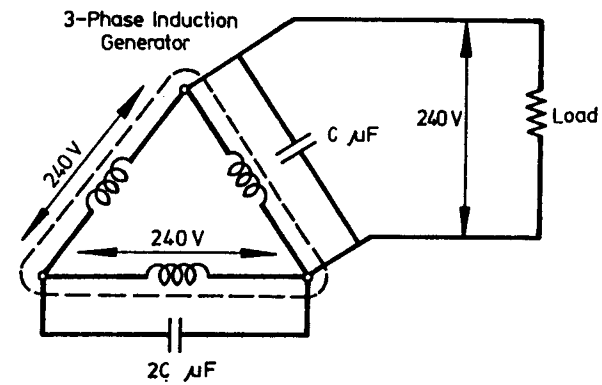


Fig. 15. Single-phase output from a three-phase induction generator.

Generator Speeds

Synchronous generators run at a fixed speed which is known as synchronous speed. This can be calculated from the electrical frequency and the number of 'poles':

$$N_s = \frac{120}{p} \times f \qquad (4)$$

where p is the number of poles
f is the electrical frequency (usually 50 or 60 Hz)
N_s is the synchronous speed (rpm)

Induction motors run at a speed slightly lower than synchronous speed.

The motor speed, N_m, is given by the equation:

$$N_m = \frac{120}{p} \times f(1-s) \qquad (5)$$

where s is the small fraction (0.02 - 0.05) by which the motor speed is lower than synchronous speed, which is known as slip.

When an induction motor is converted for use as a generator, it runs slightly above synchronous speed:

$$N_{gen} = \frac{120}{p} \times f(1+s) \qquad (6)$$

The equations given above can be combined to give a relationship between generator and motor speed:-

$$N_{gen} = \frac{240f}{p} - N_m \qquad (7)$$

Example 2: *Calculation of induction generator speed.*

A 4-pole induction motor running from a 50 Hz supply has a rated speed of 1455 rpm. The synchronous speed is:

$$N_s = \frac{120}{4} \times 50 = 1500\ rpm$$

The rated motor speed is 45 rpm less than N_s. As a generator it will run at 45 rpm greater than N_s, i.e. at 1545 rpm

Or, using equation (4):

$$N_{gen} = \frac{240f}{p} - N_m = 3000 - 1455 = 1545\ rpm$$

Standard centrifugal pumps are normally offered with the option of a 2-pole or 4-pole motor. 6-pole motors may also be available but would need to be ordered specially and are much more expensive. Typical induction generator running speeds are shown in Table 2.

Poles	50 Hz motor	50 Hz generator	60 Hz motor	60 Hz generator
2	2900	3100	3500	3700
4	1450	1550	1750	1850
6	950	1050	1150	1250

Table 2. Typical induction motor and generator running speeds (rpm).

Control of the Electrical Output

The simplest micro-hydro electric systems are those which have a fixed electrical load. This type of system is commonly used in developing countries, where the generator is often used to supply lights for a few hours each evening. All the lights come on at the same time and are switched off at the same time, so that the load on the turbine is constant, and the speed and voltage remain stable.

The voltage and frequency produced by the generator need to be controlled in order to safeguard the equipment which is being supplied. The type of equipment connected to a village micro-hydro scheme in a developing country is unlikely to be damaged if the voltage is kept within the range -25% to +10%, and the frequency from rated value up to 5% above rated value. If the loads are only resistive (filament lights, heaters, etc) then the frequency can be allowed to vary much more. Fixed load systems are cheap and easy to install but have the disadvantage of being inflexible.

Apart from the fixed load option, mentioned above, there are four ways to control a micro-hydro generator. Of these, mechanical governing is not suitable for a PAT system because there is no flow control in the turbine. The other three options are described below.

1. Manual governing of the turbine

This requires someone to adjust the gate valve according to changes in the generator voltage and frequency. With an induction generator, since there is no danger from overloads, this type of control may be used.

2. Electronic Load Control

This type of control keeps the load on the generator constant by diverting spare capacity into 'ballast' loads, in order to allow other loads to be switched on and off. This type of control can be used for a synchronous generator or for an induction generator. The synchronous generator requires an automatic voltage regulator (AVR) as well as an electronic load controller (ELC), while an induction generator requires a single controller known as an induction generator controller (IGC).

The ballast loads must be purely resistive, and can be used for space heating, electric drying or water heating. These ballast loads may need to be situated next to the controller in order to eliminate interference to medium and long wave radio reception.

The induction generator controller has been given thorough field trials at sites in the UK and in Nepal. Although it is a relatively 'high-tech' piece of equipment, most of the components are available throughout the world. Local engineers in Nepal, Sri Lanka and Indonesia have been given training and are now manufacturing the controller in their own workshops.

3. Manual Governing of the Electrical Load

This type of control requires an operator, who observes the generator voltage (perhaps through the brightness of a light bulb). He or she switches on or off a set of ballast loads, e.g. water heating elements, in order to keep the voltage more or less constant. Manual governing of the electrical load is particularly suitable for small village systems where the generator is used only during the evenings to supply electric lights. It has been implemented succesfully at a village in northern Pakistan.

7. SELECTING A PUMP AS TURBINE FOR A PARTICULAR SITE

This chapter gives procedures for selecting a pump as turbine to match a particular site, using either performance calculations or turbine testing.

Matching a Pump as Turbine to Site Conditions

In selecting your site, you choose a particular set of head and flow conditions. The flow rate is normally determined by the minimum flow rate, i.e. the flow that is available throughout the year. The head is determined by the vertical height between the intake from the stream and the turbine outlet, less the head loss in the penstock for this particular flow rate. A pump needs to be selected for which the head and flow, at the turbine best efficiency point, are as close as possible to the site conditions.

This section gives the calculations needed to get the turbine head and flow at best efficiency point for a particular pump. The running conditions in terms of head and flow, for best efficiency as a turbine, are very different from the rated pump output, although the PAT efficiency will be approximately the same as for pump operation. Friction and leakage losses, within a centrifugal pump, result in a reduction of head and flow from the theoretical maximum. The head and flow required, when running as a turbine, will be greater than the theoretical values, in order to make up for the losses. The following equations are given in the literature to predict turbine head and flow for constant speed:

$$Q_t = \frac{Q_{bep}}{\eta_{max}}; \quad H_t = \frac{H_{bep}}{\eta_{max}}; \quad \eta_t = \eta_{max} \tag{8}$$

where Q_{bep} is the flow rate at pump best efficiency point (bep)
H_{bep} is the head at pump bep
η_{max} is the pump maximum efficiency
and Q_t is the flow rate at turbine best efficiency point (bep)
H_t is the head at turbine bep
η_t is the turbine maximum efficiency.

These equations imply that the ratios Q_t/Q_{bep} and H_t/H_{bep} are equal, but experimental results show that the head ratio is usually greater than the flow ratio between turbine and pump modes. The prediction can be improved by using different powers of η_{max} for the head and flow ratios, following a method proposed by KR Sharma of Kirloskar Co., India. If the turbine speed is the same as the pump speed, these equations are:

$$Q_t = \frac{Q_{bep}}{\eta_{max}^{0.8}}; \qquad H_t = \frac{H_{bep}}{\eta_{max}^{1.2}} \qquad (9)$$

The following example shows how to calculate the head and flow needed by the turbine when the turbine speed is the same as the pump speed.

Example 3: *Calculation of turbine best efficiency point (at pump speed).*

The manufacturer of a particular pump gives curves that show that as a pump its maximum efficiency is 62% when delivering 20 l/s at a head of 16 m at 1500 rpm. The pump is required for use as a turbine, driving a synchronous generator at 1500 rpm. The turbine performance at best efficiency predicted from equations (9) will be:

$$Q_t = \frac{Q_{bep}}{\eta_{max}^{0.8}} = \frac{20}{0.62^{0.8}} = \frac{20}{0.682} = 29.3\ l/s;$$

$$H_t = \frac{H_{bep}}{\eta_{max}^{1.2}} = \frac{16}{0.62^{1.2}} = \frac{16}{0.563} = 28.4\ \text{m}.$$

Often the turbine speed will not be the same as the rated pump speed and it is necessary to use additional equations to take into account different running speeds of turbine and pump. Before presenting the equation it is necessary to explain the 'Affinity Laws'.

The Affinity Laws relate the head, flow and power of a pump or turbine to its speed:

Flow (Q) is proportional to speed (N)
Head (H) is proportional to N^2
Power (P) is proportional to N^3

These relationships can be use particularly for calculating the running conditions at best efficiency point. The equations for head and flow are:

$$Q_t(\text{at } N = N_t) = \frac{N_t}{N_p} \times Q_t(\text{at } N = N_p) \qquad (10)$$

$$H_t(\text{at } N = N_t) = \left(\frac{N_t}{N_p}\right)^2 \times H_t(\text{at } N = N_p) \qquad (11)$$

where N_p is the rated pump speed
N_t is the turbine running speed

Substituting these equations into equations (9) gives:

$$Q_t = \frac{N_t}{N_p} \times \frac{Q_{bep}}{\eta_{max}^{0.8}}; \quad H_t = \left(\frac{N_t}{N_p}\right)^2 \times \frac{H_{bep}}{\eta_{max}^{1.2}} \qquad (12)$$

An example of carrying out this calculation is given on the next page. It must be stressed that, although this method is more accurate than the equations normally given in the literature (8) it is still only approximate. The actual values of Q_t and H_t may be as much as ±20% of the predicted value for the bep. This may or may not have a significant effect on the PAT output, depending on the performance characteristics. It is therefore recommended that, wherever possible, after initial selection, the pump is tested as a turbine to find out what power will be produced at the available head and flow. The method for testing is described in the next section.

Example 4: Calculation of turbine best efficiency point at 1550 rpm.

The head available at a particular site is 26 m, and the flow is 7 l/s. It has been suggested that the pump looked at in Example 3 could be used as a turbine for this site.

The induction motor is to be used as a generator directly driven from the turbine. The turbine speed is therefore fixed by the generator speed, which can be calculated in the manner given in Example 2 (chapter 6). From the pump speed of 1450 rpm, the turbine speed is calculated to be 1550 rpm. Using the equations above (12), the predicted best efficiency conditions for turbine operation are:

$$Q_t = \frac{N_t}{N_p} \times \frac{Q_{bep}}{\eta_{max}^{0.8}} = \frac{1550}{1450} \times \frac{3.89}{0.57^{0.8}} = 6.52\ l/s;$$

$$H_t = \left(\frac{N_t}{N_p}\right)^2 \times \frac{H_{bep}}{\eta_{max}^{1.2}} = \left(\frac{1550}{1450}\right)^2 \times \frac{11.8}{0.57^{1.2}} = 26.5\ \text{m}.$$

These values of head and flow are close to the site conditions, and the pump is therefore suitable.

Test Procedure

In order to be certain of the performance of a pump as turbine (PAT) it is necessary to test it over a range of heads and flows. A feed pump will be required which must be capable of producing a greater head and flow than the predicted PAT best efficiency point (bep) conditions. The feed pump must have approximately four times the power rating of the PAT. An accurate pressure gauge will be required to measure the head across the PAT and the flow rate can be found by one of two methods, as described below.

(i) Using a measuring tank

The flow rate may be measured by timing the PAT discharge into a known volume, as shown in Fig. 16. For this test the tank must have a volume (in litres) of at least 20 x Q_t, where Q_t is in l/s. The value of head across the PAT is calculated from the pressure gauge reading, H_1, converted to a head of water (see Appendix E), added to H_2, the height between the centre line of the PAT outlet and the bottom of the discharge pipe.

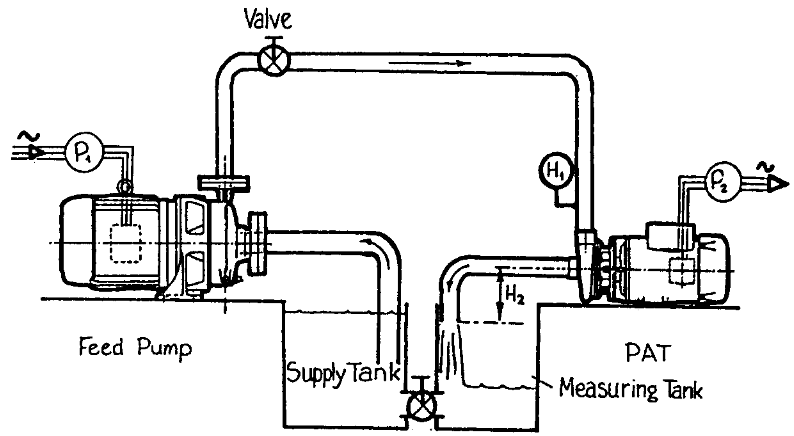


Fig. 16. PAT test set-up, using a measuring tank.

(ii) Using a flow meter

For this method, the value of Q_t is obtained from an orifice plate, venturi meter or other accurate flow measuring device. The value of head, H_t, is $(H_1 + H_2)$, where H_1 is obtained as in (i) above, and H_2 is the height between the centre line of the PAT outlet and the water level in the tank. The layout of the test equipment is shown in Fig. 17.

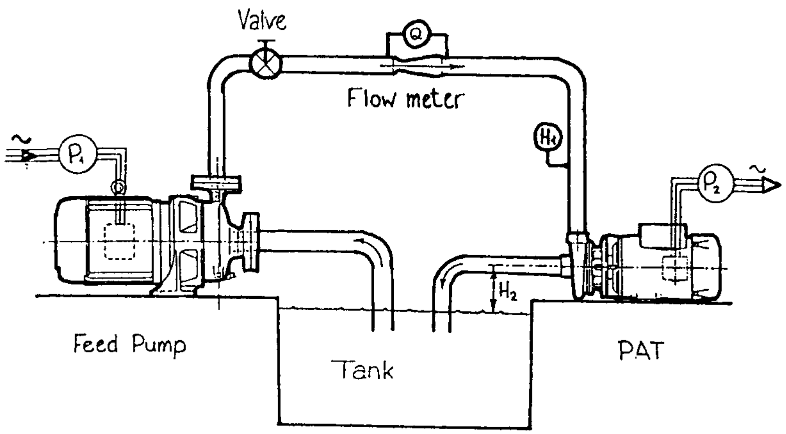


Fig. 17. PAT test set-up, using a flow meter.

Checking Generator Power Output

Care must be taken to ensure that the induction machine is not overloaded when running as a generator. It is recommended to de-rate an induction motor by 20% when running as a generator because of the likely imbalance in the currents flowing in each phase. The maximum electrical power output of the generator is therefore 80% of the rated motor power. This is the case for either C-2C connected generators or generators connected to 3-phase loads.

When running at rated voltage, the efficiency of a generator is normally around 2% lower than its rated motor efficiency. The electrical output of the generator will be approximately:

$$P_{elec} = \eta_{gen} \times P_{out}$$

where η_{gen} is the generator efficiency
P_{out} is the turbine output power ($= 9.81 \times \eta_t H_t Q_t$)

Example 5: *Checking generator power output.*

The pump as turbine in Example 4 is connected to a 1.1 kW, three-phase induction machine.

The expected turbine output power is:

$$P_{out} = \eta_t H_t Q_t \times 9.81$$
$$= 0.57 \times 26.5 \times 6.52 \times 9.81 = 966 \text{ W.}$$

The motor data does not include an efficiency value. This can be estimated using the values in Appendix D. The motor efficiency is given as 75%, and the generator efficiency is therefore 75-2 = 73%. The electrical output is therefore:

$$P_{elec} = 0.73 \times 966 = 705\ W$$

This compares with the maximum recommended generator output, which is 80% of rated motor power:

$$P_{gen.max} = 0.80 \times 1100 = 880\ W$$

The electrical power output is much lower than the recommended maximum and therefore the 1.1 kW machine is large enough for use as a generator with this turbine.

8. OPERATION OF A PUMP AS TURBINE

Operation of a Pump as Turbine at Site

Once a PAT has been selected for a particular site, the head, flow and power output are fixed within quite close limits, as shown in Fig. 18. The site curve represents the net head at the turbine, taking into account losses in the penstock, which are proportional to Q^2. A small variation in speed, and hence frequency, can be achieved by adjusting the value of the generator excitation capacitance, as noted in Chapter 6.

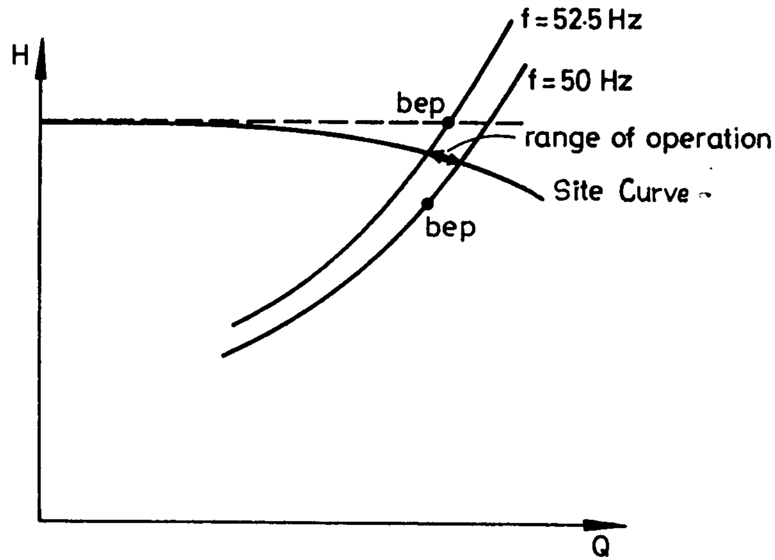


Fig. 18. PAT performance curves, showing limited operating range.

To illustrate the effects of selecting a PAT that does not really match the site conditions, Fig. 19 shows four possible PAT performance curves which could apply. The head and flow available for a particular site (Q_d,H_d) are shown by the small square (the subscript *d* is used to indicate the design conditions).

If the PAT selected for the site has its best efficiency point at (Q_d,H_d) then the actual running conditions will be:

$Q = 100\%\ Q_d$ $\qquad \eta = 100\%\ \eta_{max}$,
$H = 100\%\ H_d$ $\qquad P_{out} = 100\%\ P_d$.

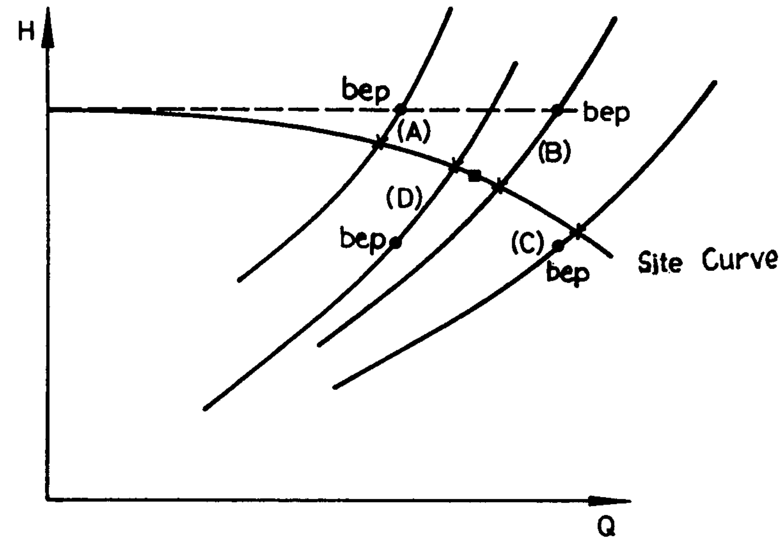


Fig. 19. Four PAT options for a particular site.

The effects of selecting each of the four PATs with performance curves A,B,C,D is given in Table 3. These figures are based on data for a typical radial-flow PAT and allowing for a frequency variation of 50-52.5 Hz (or 60 - 63 Hz).

PAT label	A	B	C	D
P/P_d	82%	98%	105%	94%
Q/Q_d	75%	102%	125%	94%
η/η_d	98%	96%	99%	97%

Table 3. Effects of pump as turbine selection.

For case A, the PAT flow rate is much lower than the design flow rate, and the power output will therefore be much lower than for the other cases. Case B results in the head and flow being close to the desired values, but there will be a drop in efficiency. In case C, the flow rate required is relatively large and there may not be enough water to keep the generator running during times of drought. When there is sufficient flow, the PAT output power will be greater than required, which means that a larger generator will be needed.

Of the four cases, the most suitable is D, because the head and flow are close to the desired head and flow, and there will be less drop in efficiency compared with case B. This is due to the shape of the efficiency curve in turbine mode (see Fig. 20). The efficiency of a PAT at 80% of bep flow will normally be lower than at 120% of bep flow.

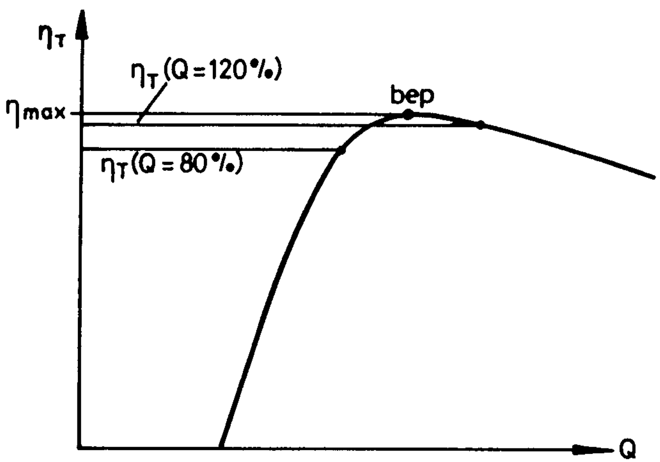


Fig. 20. Typical pump as turbine efficiency curve.

It is useful to note that, where there is a relatively large loss in the penstock, the effects of a poor turbine performance prediction are reduced, because a lower flow rate will be compensated by a larger net head at the turbine. This is the case in the above example, where the head loss in the penstock has been taken as 20% at the desired operating point. Where a site has been designed with a low friction loss in the penstock, the effects of poor turbine prediction on the power output, flow required and turbine efficiency will be more than in this example.

Adjustments to a Pump as Turbine after Installation

If a PAT has been installed without accurate testing and fails to produce the output power required, then it is usually possible to find out in what way the actual performance differs from the required performance. Knowing this, it may be possible to improve the match between the PAT performance and the site conditions. For most standard centrifugal pumps the turbine operation is similar to the curves shown in Fig 18. The various options, and some methods for improving the performance are listed below.

1. PAT flow is less than predicted

In this case the performance is similar to A in Fig. 19, and there is little that can be done to improve the PAT performance. The user must either be satisfied with the output being lower than expected, or obtain a PAT that runs efficiently at a higher flow rate.

2. PAT runs at expected flow rate, but power output is down

In this case, there are three possible reasons for the poor performance. A simple test can be used to find out what is happening. Increase the running speed of the PAT, and measure the power output while keeping the voltage as nearly as possible constant. For a system with an induction generator, the speed can be increased by decreasing the generator capacitance.

If the power output increases slightly, then the PAT is similar to case D in Fig. 19. In this case, the PAT can be run with the reduced value of capacitance, and the generator speed can be allowed to increase by up to 10%.

If the power output remains the same, then the PAT maximum efficiency is lower than expected. Some improvement may be obtained by dismantling the pump and cleaning the casing and impeller to give better surface finishes.

If the power output decreases, then the PAT is similar to case B in Fig. 19, where the head required by the PAT is greater than that available. Some improvement in performance may be obtained by turning down the outside diameter of the impeller. This is best done in steps of perhaps 5%, and then re-tested. The impeller should in any case not be reduced in diameter by more than 10%.

3. PAT flow greater than predicted, but power output the same
In this case, the PAT operation is similar to case C in Fig. 19. The speed of the turbine should be kept as low as possible, to reduce the head and flow requirement. However, the electrical frequency should not be less than 98% of the rated value. The performance may be improved by inserting a conical sleeve into the inlet side of the PAT casing (the pump outlet). This will reduce the flow area at the PAT inlet, and therefore reduce the flow requirement. On some sites it may be possible to divert more water into the intake of the scheme so that it can run at larger flow rate than the original design.

APPENDIX A:
Pump as Turbine Operation at Reduced Flow

Once a PAT has been selected and installed at a particular site, it will run at a fixed flow rate and head, at the point where the site net head matches with the PAT head. If the flow rate at the site falls below this value, then the intake will draw air, as shown in Fig. A1, and the pump will run at a lower head until the penstock refills itself.

As an example, on a site with a gross head of 63 m, the PATs will operate down to a head of around 35 m before flow surges in the penstock cause the system to become unstable. At this head the flow can be as little as 75% of the normal flow, but the power output will have fallen to around 25% of normal, because of the drop in efficiency.

However, some flexibility of the operating conditions can be achieved as long as air does not become trapped in the penstock, which causes flow surges or even a complete break in the flow of water.

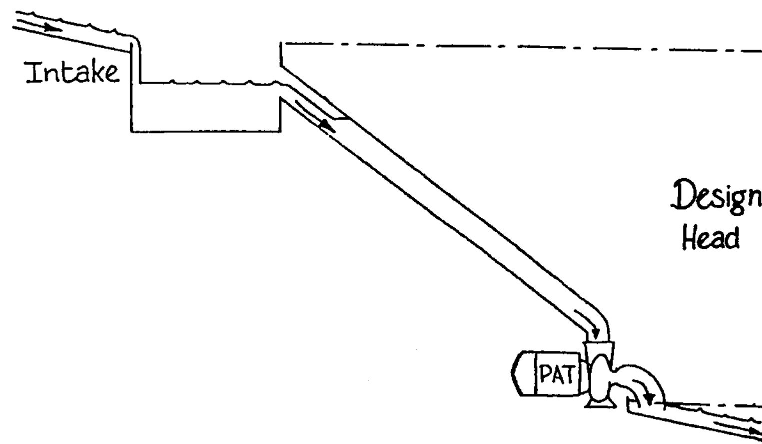


Fig. A1. Operation of a PAT at reduced flow

APPENDIX B: Parallel Operation of Pumps as Turbines

A pump as turbine runs efficiently at one value of flow rate, once the head has been fixed by the site conditions and the speed fixed by the generator. When the flow is much greater than the usual operating flow, two pumps may be used in parallel in order to make use of the additional power available. When setting up a system with two PATs it is possible to use two identical machines or to use two pumps of different sizes, most likely one using twice the flow rate of the other. Used separately, these can be for low and medium flow rates, or used together for a large flow rate.

The advantages and disadvantages are listed below in Table A1.

OPTION	ADVANTAGES	DISADVANTAGES
2 identical PATs	Easy maintenance - units use same parts and can be interchanged	Only two flow rate options: at normal flow rate or twice normal flow rate (see Note 2)
2 PATs with 2:1 flow ratio	Wide power range: 3, 2 or 1 times normal flow rate (see Note 1)	Needs careful selection of PATs

Table A1. Advantages of parallel PAT options.

Note 1. Operation of two PATs in parallel is only worthwhile where the head losses in the penstock are low. In general, it is not worthwhile to use a combination of 2:1 PATs together, if the penstock head loss for minimum flow is greater than 2%.

Fig. A2 illustrates the effect of the pipe loss on the operating points of two PATs running in parallel, one designed for minimum flow, and one for twice minimum flow. The calculation on page 48 shows that the output from both pumps in parallel may be barely greater than the output of the large pump on its own.

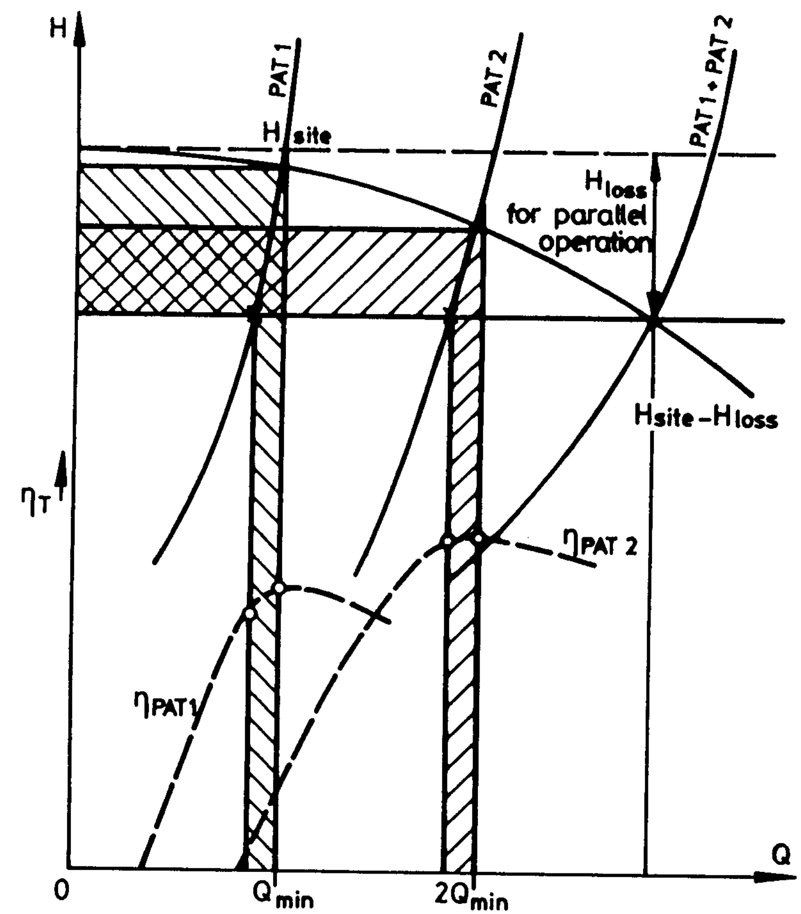


Fig. A2. Parallel operation of PATs.

Calculation:
If the head loss at minimum flow is 3%, then at twice minimum flow the loss is 12%, because the pipe losses are proportional to Q^2. The larger PAT must therefore be designed to run at 88% of site head. The pumps selected for a particular site are:

PAT1: 48.5 m, 4 l/s, 55%, 1.05 kW.
PAT2: 44 m, 8 l/s, 65%, **2.24 kW**.

Running both together, the additional head loss will cause them to run at around:

PAT1: 38 m, 3.6 l/s, 52%, 0.70 kW.
PAT2: 38 m, 7.5 l/s, 64%, 1.79 kW.
... Total Power output ...**2.49 kW**.

There are also two options for selecting the generator(s) when PATs are run in parallel. Either a single generator can be driven from one or both of the PATs, or each PAT can be connected to its own generator. The advantages and disadvantages of these options are shown in Table A2.

OPTION	ADVANTAGES	DISADVANTAGES
One generator	Lower cost, simpler to connect electrically; Large generator may be useful for motor loads (see Note 2)	Needs a complicated drive arrangement with belts or clutches; Poor efficiency when generating low power
Two generators: one for each PAT	Can remove one unit for maintenance without cutting off supply	Need to ensure correct sharing of electrical load between generators; More expensive than a single generator

Table A2. Advantages of one or two generators for parallel PATs.

Note 2. A larger generator has higher inertia (i.e. acts as a flywheel), which will assist in the starting of motors.

APPENDIX C:
Syphon Intake for Intermittent Operation

An option for dealing with low flow rates is to use a syphon intake and a small reservoir. When the flow rates are low, the system will run intermittently. This system can also cope when higher flow rates are available since then it will operate continuously. A scheme using this type of syphon intake has been installed by Dulas Engineering at a remote site in Wales.

For this type of scheme, the PAT is selected to operate at a flow rate several times greater than the minimum flow rate. Figure A3 shows the syphon intake arrangement. When the reservoir is full, the syphon operates and water flows through the penstock to the turbine until the reservoir is empty. The syphon then breaks the flow to the turbine until the reservoir is full again.

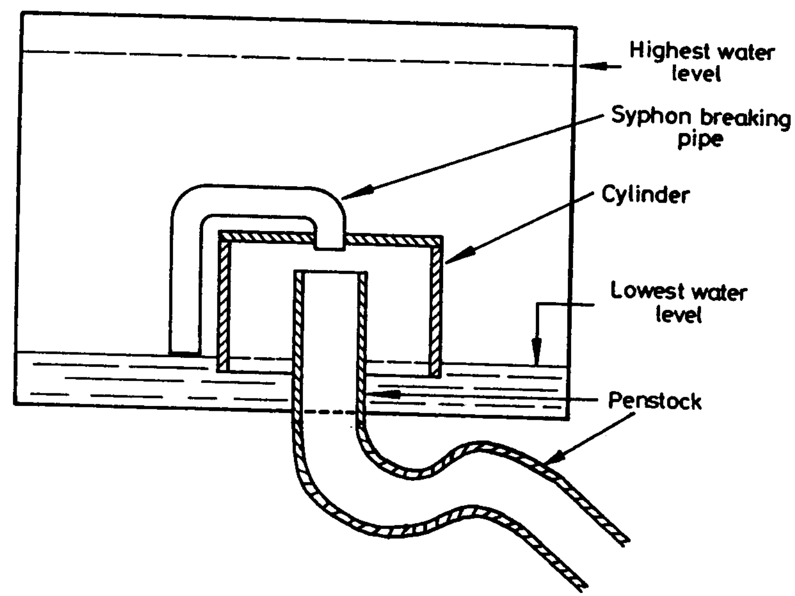


Fig. A3. A syphon intake for intermittent operation.

APPENDIX D: Typical Efficiencies of Induction Motors

The table below gives efficiency values for three-phase and single-phase induction motors for a range of different motor power ratings. These power ratings are those normally given on the motor name plate. The values given are for typical induction motors from a reputable European manufacturer. For motors manufactured in less industrialized countries, the efficiencies may be up to 5% lower.

Three-phase induction motors:

Rated Power (kW)	**0.55**	**1.1**	**2.2**	**4.0**	**11.0**
Motor Efficiency	70%	75%	79%	82%	87%

Single-phase induction motors:

Rated Power (kW)	**0.55**	**1.1**	**2.2 (2-pole)**	**2.2 (4-pole)**
Motor Efficiency	66%	68%	70%	75%

Table A3. Typical efficiency values for induction motors.

Note 1. When the motor is used as an induction generator, efficiency for the same voltage and frequency (240 V, 50 Hz) is likely to be approximately 2% less than the motor efficiency.

Note 2. The efficiency values for single-phase motors are quoted for the capacitor-start, induction run type.

APPENDIX E: Unit Conversion for Head and Flow

The tables below give factors for the most commonly used units of pressure (or head) and volume flow-rate. To convert from a unit on the left-hand side of the table to a unit at the top of the table, multiply by the figure in the appropriate row and column. (Hg is the symbol used for mercury and H_2O for water)

	mm. Hg	kN/m²	ft.H_2O	in.Hg	psi	m. H_2O	bar
mm. Hg	1.00	0.133	.0446	.0394	.0193	.0136	.00133
kN/m (kPa)	7.50	1.00	0.335	0.295	0.145	0.102	0.001
ft.H_2O	22.42	2.989	1.00	0.883	0.434	0.305	0.0299
in.Hg	25.40	3.386	1.133	1.00	0.491	0.345	0.0339
psi	51.72	6.895	2.304	2.037	1.00	0.703	0.0689
m. H_2O	73.6	9.81	3.281	2.897	1.422	1.00	0.0981
bar	750.1	100.0	33.46	29.53	14.50	10.19	1.00
atm	759.9	101.3	33.89	29.92	14.69	10.33	1.013

Table A4. Conversion factors for head (pressure).

	US.gpm	Imp.gpm	m³/h	l/s	ft³/s
US.gpm	1.00	0.8325	0.227	0.06308	0.00223
Imp.gpm	1.201	1.00	0.273	0.07577	0.00267
m³/hr	4.403	3.666	1.00	0.2777	0.00981
l/s	15.85	13.20	3.60	1.00	0.0353
ft³/s(cusec)	449.0	374.4	102.0	28.32	1.00
m³/s(cumec)	15850	13200	3600	1000	35.31

Table A5. Conversion factors for flow-rate.

APPENDIX F: Estimating Pump Performance from Physical Measurements

Professor Z. Samani of New Mexico State University took data from a large number of pumps and formulated equations to estimate the pump performance.

The measurements which you will need from your pump are:

- internal diameter of pump outlet, D_{out}, in metres (m)
- outside diameter of pump impeller, D_{imp}, in metres (m)

From the pump name plate obtain:

- pump speed, N_p, in revolutions per minute (rpm)
- motor power, P_m, in kilowatts (kW)

The flow rate at best efficiency point, Q_{bep} (l/s), is related to the diameter of the pump outlet, D_{out}, the pump operating speed, N_p, and the impeller diameter, D_{imp}, by the following equation:

$$Q_{bep} = 8.6 \times N_p \times D_{imp} \times (D_{out})^2 \qquad \text{(A1)}$$

The maximum efficiency of the pump can be estimated from the flow rate using the chart in Fig. A4. This chart is based on the work of H. H. Anderson.

The chart plots the following equation:

$$\eta_{max} = 0.9 - (13.2 \times Q_{bep})^{-0.32} \qquad \text{(A2)}$$

where η_{max} is the maximum pump efficiency (as a number rather than as a percentage).

An estimate for the pump head at best efficiency point, H_{bep} (m), can be made using the power rating of the motor, P_m (kW). The equation required is:

$$H_{bep} = 92 \times \frac{P_m \times \eta_{max}}{Q_{bep}} \qquad \text{(A3)}$$

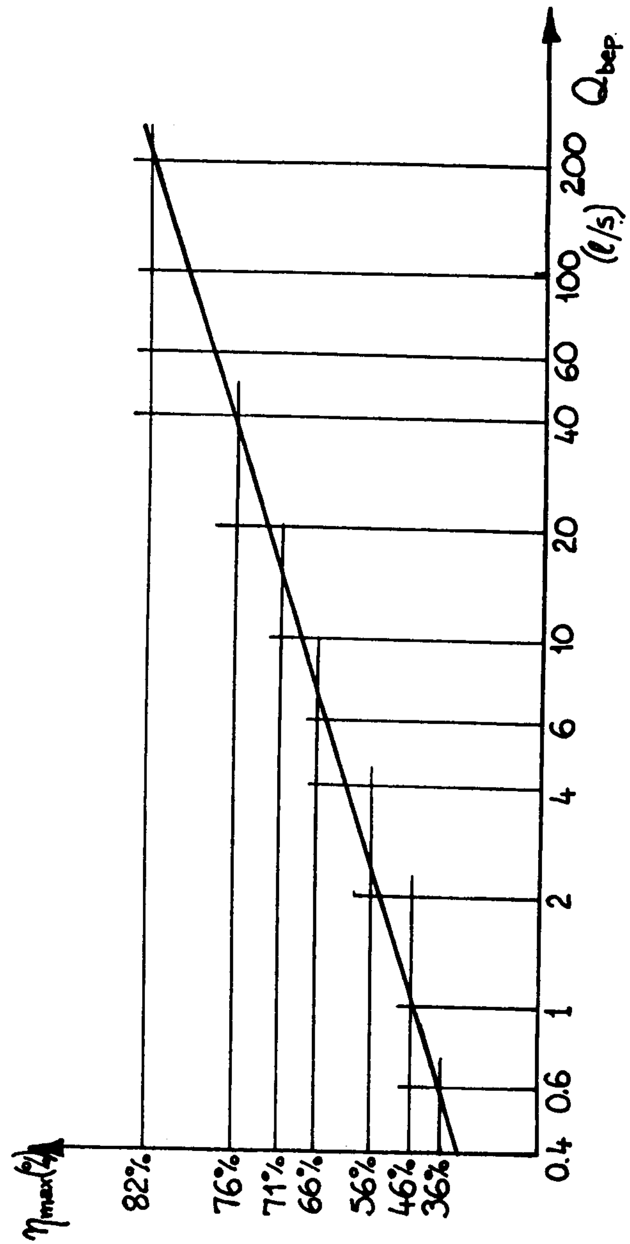


Fig. A4. Pump maximum efficiency related to flow rate

A more accurate value for the maximum efficiency is obtained if the 'specific speed' is taken into account. The specific speed, N_s, is a number which related to the shape of the pump, and is calculated using the following equation:

$$N_s = N_p \times \frac{\sqrt{Q_{bep}}}{H_{bep}^{0.75}} \qquad \text{(A4)}$$

The original value for the maximum efficiency can now be corrected using the following equation, which is also plotted in Fig. A5:

$$\eta_{max}(corrected) = \eta_{max} + 0.04 - 0.29(0.32 - \log_{10}(0.0015N_s))^2$$

This corrected value of efficiency can now be used to recalculate the head, H_{bep}, using equation A3. These values of H_{bep}, Q_{bep}, and η_{max} can be used to make a prediction of the turbine performance of the pump, using the procedure in Chapter 7.

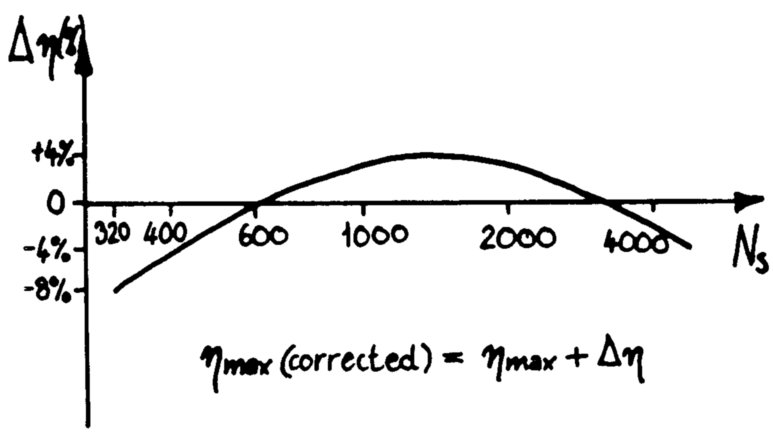


Fig. A5. Correction for pump maximum efficiency.

APPENDIX G: Calculating Safe Wall Thickness for Steel Pipe

The safe wall thickness depends on the surge pressure which may occur. The maximum surge pressure in a steel pipe is given by:

$$h_{surge} = \frac{0.18Q}{D_{pipe}^2}$$

where Q is flow-rate in l/s and D_{pipe} is pipe diameter in m.

The calculation of the safe wall thickness for steel pipe has an allowance of 1.5 mm included to allow for corrosion. This does not need to be included if the pipe has been effectively treated to prevent rusting. The equation to calculate the wall thickness t_{min} (mm) is:

$$t_{min} = 1.5 + \frac{D_{pipe} \times (H_{site} + h_{surge})}{30}$$

Example A1: Calculation of pipe thickness for mild steel penstock.

A mild steel penstock of diameter 150 mm carries a flow of 35 l/s from a site head of 15 m. The maximum surge pressure which can occur is:

$$h_{surge} = \frac{0.18 \times 35}{0.15^2} = 280 \text{ m.}$$

Since the pipe is mild steel, the corrosion factor must remain and the minimum wall thickness is:

$$t_{min} = 1.5 + \frac{0.15 \times (15 + 280)}{30} = 1.5 + 1.5 = 3.0 \text{ mm.}$$

APPENDIX H: Selecting Penstock Diameter and Calculating Head Loss

The head loss in the penstock depends mainly on the flow rate through the pipe, its length and its diameter. The author has devised a simple method for calculating the head loss and hence selecting the pipe diameter.

First, choose a pipe diameter (D_{pipe}) for which the flow rate in litres/second is closest to the value in the second row of Table A6. This table is suitable for uPVC, MDPE and good quality steel pipe[1]. It gives values of flow rate (Q_{table}) for a range of standard pipe diameters.

D_{pipe} (mm)	75	100	125	150	175	200	250	300	350
Q_{table} (l/s)	3.5	7.6	13.7	22.2	33.4	47.5	85.5	138	210

Table A6. Flow rates for various standard pipe sizes.

After choosing a pipe diameter, check the value of percentage head loss using the formula:

$$\frac{h_f}{H_{site}}(\%) = \frac{L_{pipe}}{H_{site}} \times \left(\frac{Q}{Q_{table}}\right)^2 \qquad \text{(A5)}$$

where h_f is the head loss
H_{site} is the gross site head
L_{pipe} is the *length* of the pipe in m
Q is the flow rate through the turbine in l/s
Q_{table} is the flow rate given in Table A6.

[1] Q_{table} is the value of flow rate which produces a head loss of 1% per unit length of pipe for a roughness of 0.06 mm, as found in Dept. of the Environment, 'Tables for the hydraulic design of pipes', HMSO, London, 1977

If the head loss, h_f, is between 10% and 20% of the gross head, then the pipe is suitable. Otherwise, choose the next size of pipe and repeat the calculation to find the head loss, as shown in the example below.

***Example A2:** Calculation of penstock head loss.*

A scheme with a gross head of 12 m has a penstock of total length 50 m carrying a flow of 35 l/s. From the table above, the flow rate given for a 175 mm pipe is 33.4 l/s. For 35 l/s the percentage head loss will be:-

$$\frac{h_f}{H_{site}}(\%) = \frac{50}{12} \times \left(\frac{35}{33.4}\right)^2 = 4.58\ \%$$

This head loss is relatively small, so it may be possible to use a pipe of 150 mm diameter. Q_{table} is now 22.2 l/s and the percentage head loss is:

$$\frac{h_f}{H_{site}}(\%) = \frac{50}{12} \times \left(\frac{35}{22.2}\right)^2 = 10.3\ \%$$

In this case, the head loss is only just greater than 10%, so it would be worth seeing if it is possible to use the next smaller diameter of pipe.

For 125 mm diameter, Q_{table} is 13.7 l/s and the head loss is:

$$\frac{h_f}{H_{site}}(\%) = \frac{50}{12} \times \left(\frac{35}{13.7}\right)^2 = 27\ \%$$

This is an unacceptably large head loss, and therefore the 150 mm diameter pipe should be used.

The actual head loss can be calculated from the percentage head loss:

$$h_f = \frac{10.3}{100} \times 12 = 1.24\ \text{m}$$

Further Reading

Micro-hydropower Sourcebook, by A R Inversin, NRECA, New York, 1986.

Micro-Hydro Power: a design guide, by A B Harvey and A P Brown, IT Publications, London, 1989.

Micro-hydropower: a guide for development workers, by P Fraenkel et al, IT Publications, London, 1991.

Motors as Generators for Micro-hydro Power, by N P A Smith, IT Publications, London, 1994.

Centrifugal Pumps, by H H Anderson, Trade & Technical Press, Morden, UK, 3rd Edition, 1980.

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