

The breastshot waterwheel: design and model tests

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Breastshot waterwheels—that is, waterwheels where the water enters the wheel approximately at the level of the axis—were in widespread use in England and Germany during the nineteenth and early twentieth century. Although this type of wheel even today has the potential for the economical and environmentally acceptable exploitation of small hydropower with low heads from 1.5 to 2.5 m, very little is known about its performance characteristics. In order to assess the breastshot waterwheel for hydropower generation, a study of design methods and a series of model tests were conducted at Queen's University Belfast. Sample calculations for a 4 m diameter wheel are given to explain the design principles. Tests on a 1:4 scale, 1 m diameter model gave efficiencies of 78.5% over a broad range of flows. Based on these measurements and observations, improved geometries for in- and outflow were developed, resulting in maximum efficiencies of 87.3%. An initial ecological assessment indicated that waterwheels may have a significantly reduced ecological impact when compared with turbines. The breastshot waterwheel was found to be an efficient and ecologically acceptable hydraulic energy converter with the potential for further development.

1. INTRODUCTION

The waterwheel is one of the oldest hydraulic machines known to humankind and has been in use since antiquity. Originally built of wood, the availability of new materials, namely wrought iron, and the increasing demand for mechanical power during the industrial revolution led, in combination with the development of hydraulic engineering, to the rational design of waterwheels, resulting in much increased performance and efficiency. Three distinct types of waterwheels evolved: the overshot, the breastshot and the undershot wheel.^{1,2} Overshot wheels were investigated quite thoroughly and were found to have efficiencies of more than 85% for a broad range of flow rates from 0.2 to $1.0Q/Q_{\max}$.³ Only one measurement of an undershot or Zuppinger wheel is known to the authors. Researchers from the Technical University of Stuttgart in 1977 measured the efficiencies of a 42 kW Zuppinger wheel which was built in 1886 and had been in continuous operation since. The measurements showed efficiencies of 71–77%.⁴ The breastshot wheel however, which was particularly popular in Britain, was never investigated.

Breastshot waterwheels (Fig. 1) can be broadly defined as those

where the water enters the wheel at axis level. A typical waterwheel installation would consist of a weir in the main river, which generates the head difference, an intake structure, the mill race, the waterwheel itself and finally the tailrace which leads the water extracted from the river back to its source. Today, such an installation would be called a run-of-river power station. Occasionally, millponds were built to store water overnight for usage during the day. It is estimated that in England in 1850 there were 25–30 000 waterwheels in operation.⁷ In Germany, there were 33 500 waterwheels registered for commercial use as late as 1925.² Sample counts by the author suggest that 60–70% of these wheels employed head differences below 2.5 m—that is, they were designed as undershot or breastshot wheels. Although waterwheels were then clearly regarded as efficient and cost-effective machines, practically all waterwheels, and with them the knowledge about technology and design, have disappeared today. The remains of many water wheel installations—that is, weirs, intakes and mill races—do however still exist. Currently, it is estimated that there is a potential of 600–1000 MW of unused small hydro power with low heads in the UK and around 500 MW in Germany.^{8,9}

A review by the German Umweltbundesamt (Environment Agency) of small hydropower as renewable energy source concluded that low-head microhydro power ($H < 2.5$ m, $P < 100$ kW) is uneconomical and has a negative ecological impact.¹⁰ With installation costs of £7–10 000/kW installed capacity, standard turbines (Kaplan or Ossberger turbines) do not constitute a cost-effective solution. The high speed of the turbine leads to external damage to fish passing through the hydropower installation, while the low pressures can cause internal damage. In addition, the safe downstream passage of juvenile fish is impeded. Waterwheels were however not even mentioned in the report. It can be expected that waterwheels will have a very low environmental impact due to their low speed, and due to the fact that they operate at atmospheric pressures.

Recently, some small companies have started to build overshot and Zuppinger waterwheels again (see www.bega-wasserkraft.de; www.hydrowatt.de; www.waterwheelfactory.com). In particular, overshot wheels, which can be used for head differences of 3 m and more, have proved to be commercially viable energy converters with a very low environmental impact due to their large cell sizes and slow

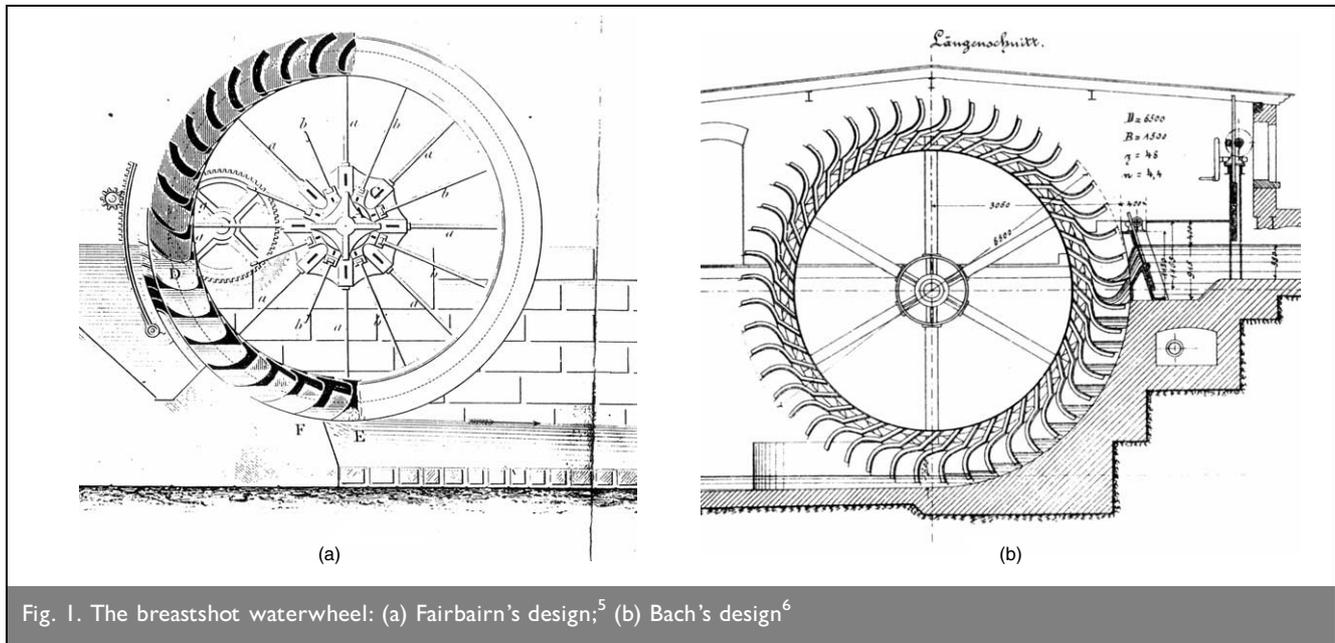


Fig. 1. The breastshot waterwheel: (a) Fairbairn's design;⁵ (b) Bach's design⁶

speeds. The exploitation of sites with head differences of 1.0–3 m is however still problematic since the undershot wheels, which are occasionally built, are only marginally economical. Breastshot wheels, which have not been built for nearly 100 years, would however be considerably smaller in diameter than undershot wheels and run faster, resulting in a more economical design. They could also be built in steel which offers considerable advantages in strength, longevity, operation and efficiency over wood (the thickness of the wooden blades constitutes a blockage of the inflow which reduces the efficiency by an estimated 3–5%⁶). The quality of wood required for a waterwheel thus means that virtually no price advantage over steel is gained. The breastshot waterwheel may therefore offer an attractive and economical solution for low head hydropower sites.

2. DESIGN OF BREASTSHOT WATERWHEELS

2.1. General

The design of waterwheels aims at guiding the water into and out of the wheel with a minimum of losses, so as to maximise its potential and, to a much smaller degree—its kinetic energy. In the case of a breastshot waterwheel, the losses at exit and entrance are therefore minimised. The most advanced design method was developed by the German engineer Carl von Bach (1886).^{11–13} The design starts with some general parameters, known from experience, moves on to determine the cell shape according to exit conditions and then leads to the design of the inflow details.

2.2. Design parameters

A waterwheel has to be designed for a given head difference H and a flow rate Q . The diameter D of a breastshot wheel can be taken as twice the head difference H or slightly more. Bach recommends a value of $D = H + 3.5$ m. This however leads to quite a large wheel diameter for smaller values of H ; in this report the diameter is taken as $2H$ as indicated in Fig. 1 in order to design a smaller and thus more economical wheel. At design speed, each cell will only be partially filled to avoid turbulent losses and overflowing, and to allow for easy exit of the water.

The filling ratio ϵ can be taken as 0.4 (large variation in flow) to 0.5 (smaller variation in flow); it is assumed that a larger filling ratio would result in a build-up of flow resistance in the cell, leading to increased losses at the inflow.¹⁰

In order to illustrate the design method, a waterwheel of 4.0 m diameter with a head difference of 2.0 m and a flow rate of 0.5 m³/s will be designed using the method and recommendations given by Bach.¹¹ The tangential velocity v_t should be within 1.5–2 m/s and was taken as 1.8 m/s. This results in a wheel speed of 8.6 rpm. Fig. 2(a) shows the side elevation of a breast wheel with the relevant design parameters.

The depth d of the cells can be calculated as a function of the head difference and the diameter with the following empirical equations

1a	$d = 0.4 \sqrt[3]{\frac{D}{H}} \text{ to } 0.5 \sqrt[3]{\frac{D}{H}}$
1b	$d = 0.4 \sqrt[3]{\frac{4.0}{2.0}} \text{ to } 0.5 \sqrt[3]{\frac{4.0}{2.0}} = 0.50 \text{ to } 0.63 \text{ (m)}$

A depth d of 0.60 m was chosen for the model. The width B of the wheel follows

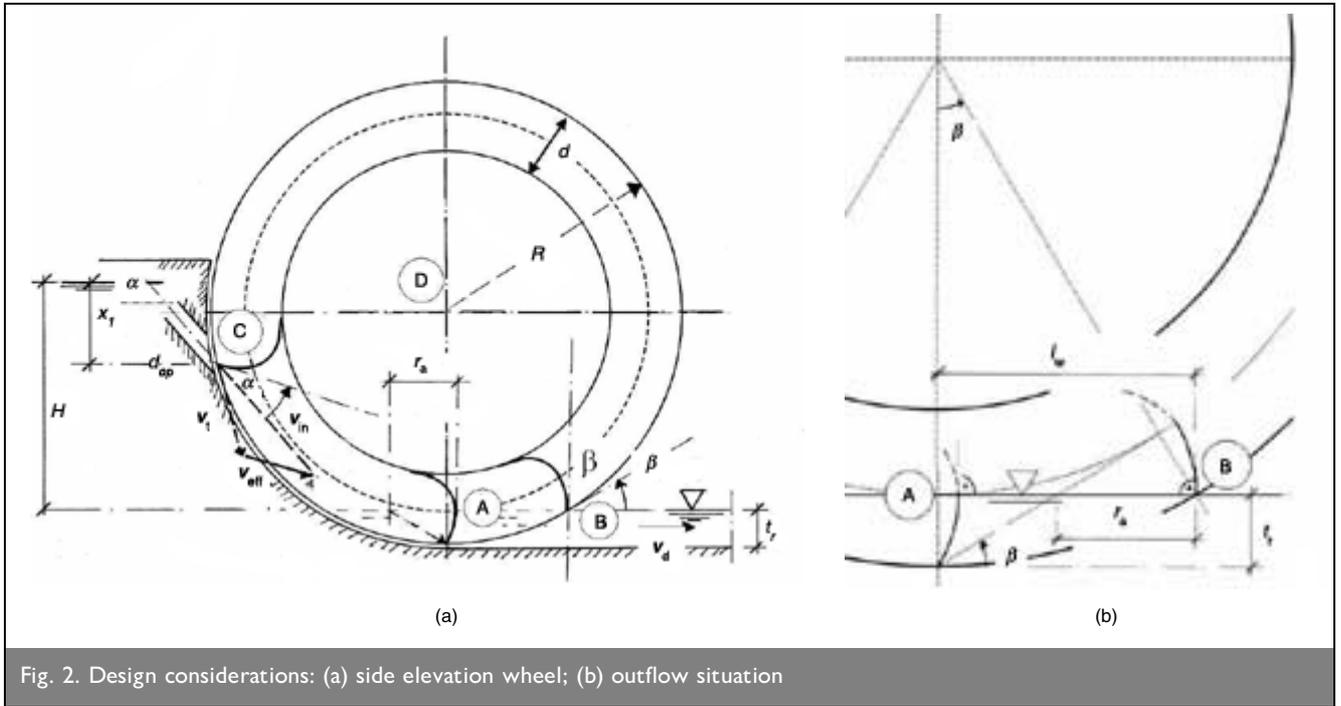
2	$B = \frac{Q}{v_t \epsilon d} = \frac{0.5}{1.8 \times 0.45 \times 0.60} = 1.03 \text{ m, say } 1.00 \text{ (m)}$
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2.3. Design of downstream situation

At the outflow, the water must move with the tangential speed of the wheel of 1.8 m/s. The submerged depth at the exit t_r can be determined by

3	$t_r = \frac{Q}{B v_t} = \frac{0.5}{1.00 \times 1.8} = 0.278 \text{ (m)}$
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Figure 2(b) shows the situation at the exit. In order to avoid losses at the exit of the blade out of the water, the blade should be curved in such a way so that the angle of intersection of blade and water surface is always normal to the water surface



from point A to point B along the wetted length l_w . The radius r_a of the lower section of the blade is given by

4a	$\beta = \arcsin \frac{R - t_r}{R} = \arcsin \frac{2.0 - 0.278}{2.0} = 30.6^\circ$
4b	$r_a = \frac{t_r}{\sin \beta} = 0.546 \text{ (m)}$

The blade follows this radius from the outer edge to the point of maximum submergence (i.e. $R - t_r$). It is then curved with a

smaller radius r_b to approach the maximum depth d of the cell tangentially as shown in Fig. 3(a).

2.4. Design of inflow

The downstream condition determines the geometry of the blade. The inflow detail then has to be designed in a way so as to minimise losses during inflow. The inflow is usually channelled through a slot-type arrangement called *coulisse*; this is illustrated in Fig. 2(a) as a single inflow channel. The inflowing water enters the cell at a depth x_1 below the upstream water level at point C. In order to maximise the use of the

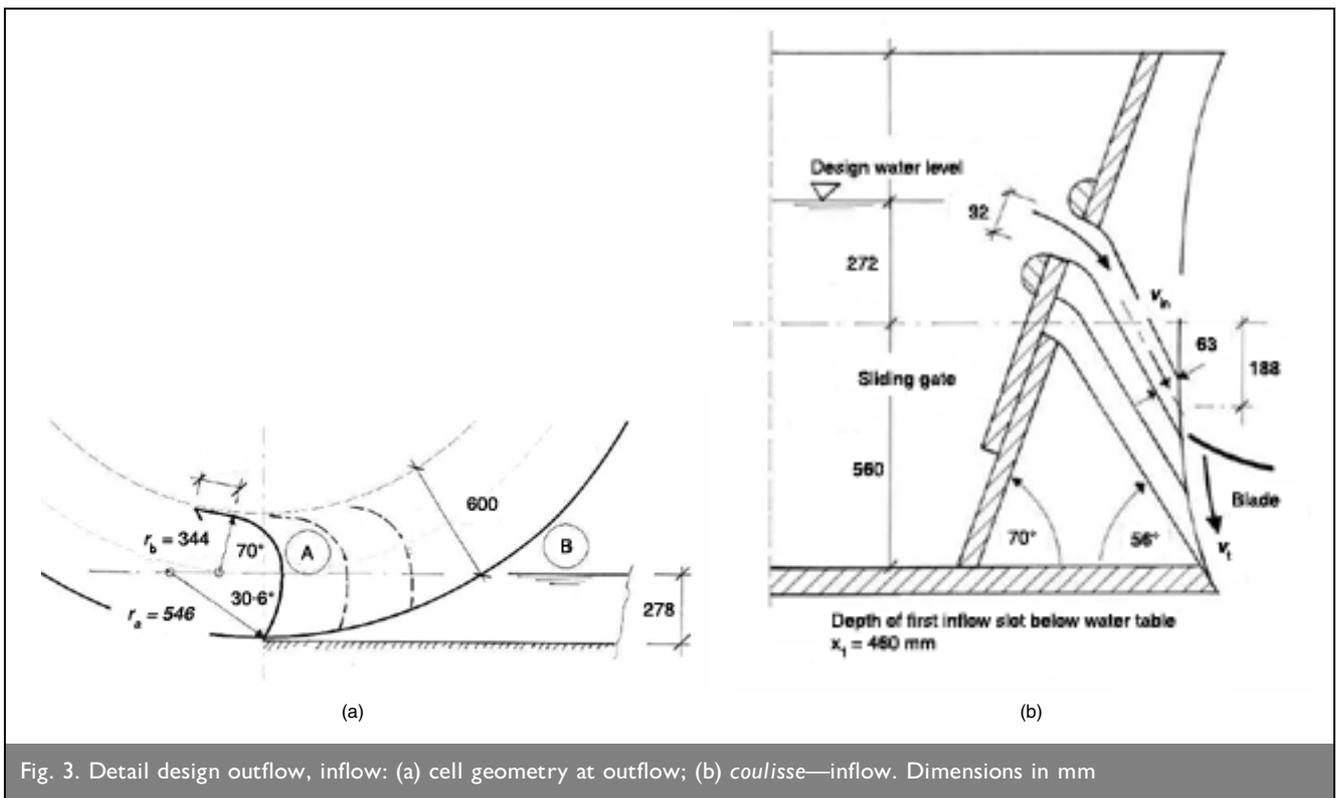


Fig. 3. Detail design outflow, inflow: (a) cell geometry at outflow; (b) *coulisse*—inflow. Dimensions in mm

kinetic energy of the water, the inflow velocity v_{in} should be approximately twice the tangential velocity v_t of the wheel. It is important to avoid turbulent losses at the inflow generated by the water jet hitting the cell blade at an inclined angle. The inflow vector v_{in} enters the cell with the angle α so that the effective inflow vector v_{eff} , which constitutes the vector sum of the inflow vector v_{in} and the tangential velocity vector of the wheel blade v_t , is parallel to the blade. The angle α under which the water enters the cell is usually chosen so that $\tan \alpha = 0.5$ or $\alpha = 26^\circ$ (sometimes up to 30°). The larger α becomes, the faster the wheel or the slower the water has to move. The entry velocity v_{in} of the water can now be determined by

$$5 \quad v_{in} = \frac{v_t \sin \beta}{\sin(90 - \beta - \alpha)} = \frac{1.80 \times \sin 59.4}{\sin(90 - 30.6 - 26.3)} = 2.852 \text{ (m/s)}$$

With this velocity, the depth of the inflow underneath the upstream water level can be calculated, assuming a loss factor of 0.1

$$6 \quad x_1 = 1.10 \times \frac{2.852^2}{2g} = 0.456 \text{ (m)}$$

Simplifying, an inflow detail with one opening can now be designed with a loss factor of 0.08 for the inflow. Assuming that the upstream water level is 0.278 m above the axis of the wheel, so that the head difference is exactly 2.00 m, the angle ϕ of the inflow channel with the horizontal is

$$7 \quad \phi = \beta + \alpha - \arcsin \frac{0.456 - 0.278}{2.00} = 30.6 + 26.3 - 5.11 = 51.8^\circ$$

With an inflow velocity of 2.852 m/s, the required theoretical depth d_{op} for a single inflow opening as shown in Fig. 2(a) can be determined by

$$8 \quad d_{op} = \frac{Q}{0.92 \times v_{in} \times B} = \frac{0.50}{0.92 \times 2.852 \times 1.0} = 0.191 \text{ (m)}$$

It should be noted that this is the width of the opening close to the water wheel; the entry width for the inflow channel must be wider since the flow velocity at that point will be smaller. In the actual wheel, the inflow is divided into three channels in

order to accommodate varying flow rates. The inflow detail can be seen in Figs 3(b) and 5(a).

3. MODEL TESTS

3.1. Experimental set-up

For the model tests, a 1:4 scale model of the water wheel designed in the previous section was built and tested at Queen's University Belfast. For the model, Froude scaling was assumed. The wheel had the following details

- diameter $D = 1.00 \text{ m}$
- head difference $H = 0.50 \text{ m}$
- design flow $Q = 0.015 \text{ m}^3/\text{s}$
- speed $n = 17 \text{ rpm}$
- tangential velocity $v_t = 0.9 \text{ m/s}$
- depth of cells $d = 0.15 \text{ m}$

All other dimensions were scaled accordingly. The wheel had a width of 0.25 m and was set in a Perspex channel of 2.50 m length. The supply water was pumped into a 1 m long PVC channel with a sharp crested weir at the outflow. This channel was used to measure the inflowing volume Q as shown in Fig. 4. The inflow detail had three slots, with widths of 15 mm each. The centreline of first slot (inflow point into the wheel) was located at the depth $x_1 = 115 \text{ mm}$ below the upstream design water level, the other two slots accordingly below. The inflow angle was chosen as 57° , slightly larger than the design value for practical reasons. This resulted in an inflow angle $\alpha \approx 30^\circ$. The speed of the wheel n was timed with a stopwatch over five revolutions. The friction force F was measured with a 195 mm diameter friction brake, using a spring balance (PESOLA 80050) of 50 kg capacity with an accuracy of 0.3%. The spring balance was checked before and after the tests against calibrated weights and found to work within the accuracy limits given by the manufacturer. The inflow detail and the wheel with friction brake are shown in Fig. 5. In accordance with the standard definition of the efficiency of hydropower installations, the available hydraulic energy was defined as the product of head difference and flow rate (the kinetic energy at the inflow will be very small).¹⁴ The head difference H was determined as the difference between upstream and downstream water levels. The

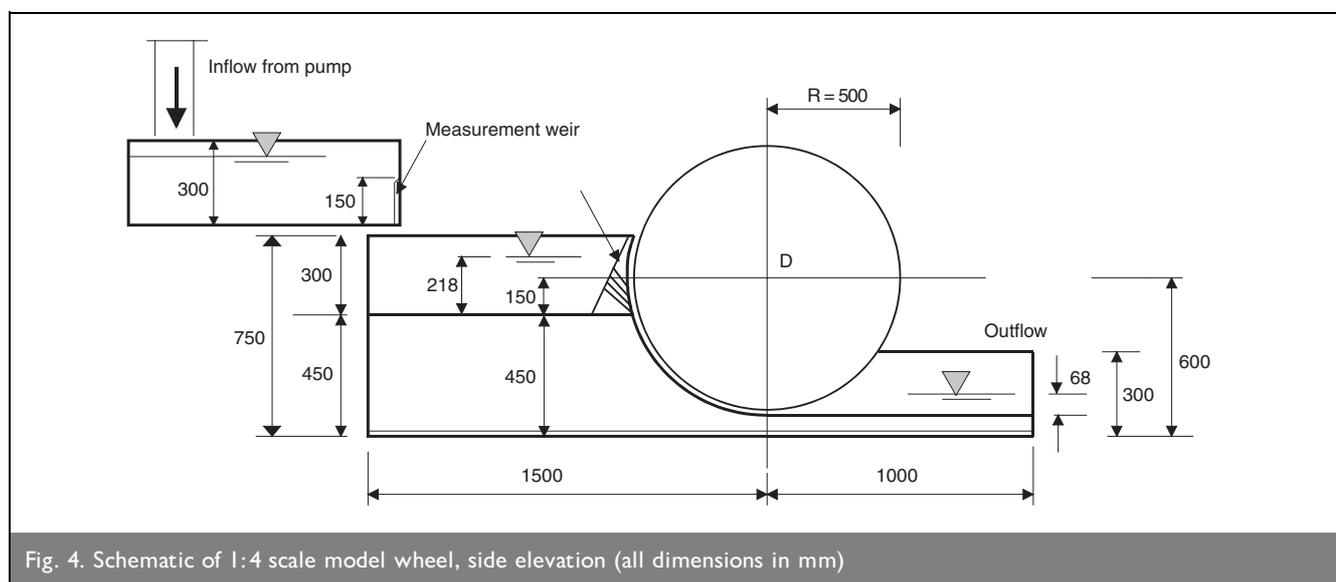
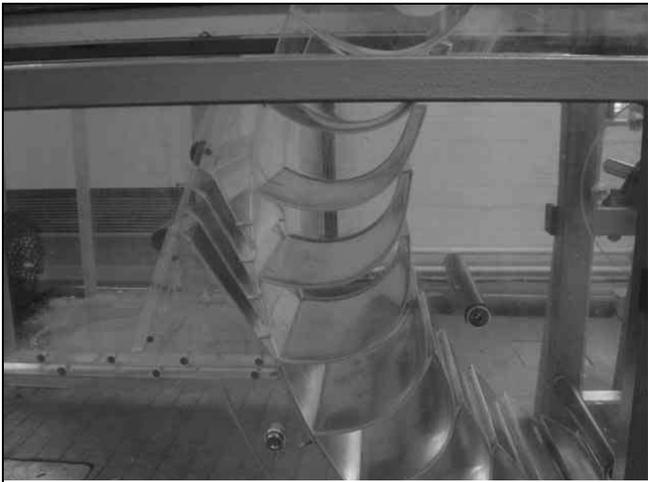
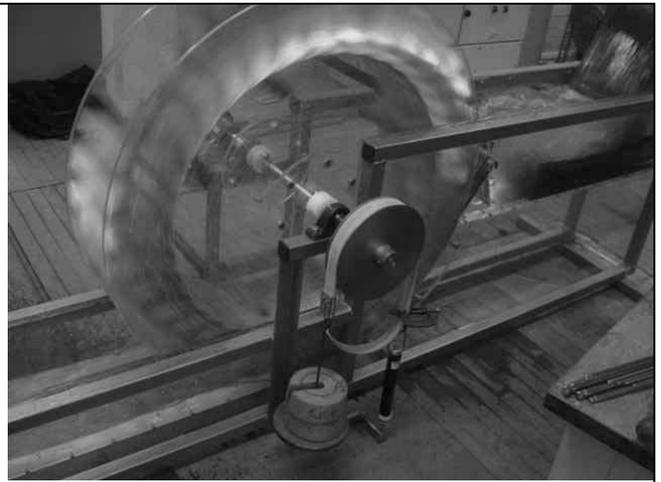


Fig. 4. Schematic of 1:4 scale model wheel, side elevation (all dimensions in mm)



(a)



(b)

Fig. 5. Model wheel: (a) inflow detail; (b) side view of wheel with friction brake

shaft efficiency η could then be calculated using the following formulae, whereby 0.0975 m is the lever arm of the friction force.

9a	$P_{in} = Q \times H$
9b	$P_{out} = F \times 0.0975 \times \frac{2\pi n}{60}$
9c	$\eta = \frac{P_{out}}{P_{in}}$

The outflowing water moves with the tangential speed of the wheel and is therefore substantially faster than the water in the inflow channel. The kinetic energy of the outflowing water thus constitutes that part of the energy which flows through the system without conducting work.

3.2. Experimental results—series 1

The tests were conducted with flow rates of 2.70, 5.25, 7.55, 8.63 and 10.77 l/s, weights of 77.5 to 405 N, and wheel speeds from 6 to 34 rpm. The inflow opening was adjusted to the flow rate. During the experiments it was observed that the design flow rate of 15 l/s could not be reached. This was attributed to the losses at the inflow, which appeared to be larger than assumed in the design calculations. The very fast water jet generated by the coulisse inflow created very strong turbulent motion of the water inside the cells; the horizontal water surfaces in the cells shown in Fig. 1(b) are therefore quite misleading. At the outflow, a 'wave' formed where the blades exit the water, indicating that the horizontal outflow does not allow for the water to move away from the waterwheel. The test results were analysed, and power output and efficiency plotted against wheel speed, peripheral speed and flow rate.

Figure 6 shows that the power output is a function of speed. The design speed of 17 rpm corresponds well with the observed speed for maximum power out of 15–18 rpm.

In Fig. 7(a) the efficiency is shown as a function of the flow rate. Maximum efficiencies reach 0.78 for $0.18 \leq Q/Q_{max} \leq 0.7$. The efficiency curve is nearly horizontal in this area; the waterwheel can therefore operate at high

efficiencies for a broad range of flows without active control elements as would be required in turbines. Fig. 7(b) finally shows the efficiency as a function of the ratio of tangential speed of the wheel and inflow speed. The inflow speed varied slightly between the different flow rates for two reasons.

- There were slight differences in up- and downstream water depth and subsequently head difference.
- The centre of inflow dropped down with the opening of first two and then three inflow slots, increasing the inflow velocity of the water.

It can be seen that maximum power out is achieved for $v_t/v_i = 0.5$ to 0.7, corresponding reasonably well with theory.

3.3. Experimental results—series 2

The inflow as recommended¹¹ could not absorb the design flow rate due to losses at the inflow (confined flow causing a build-up of water level), and due to losses at the outflow. In order to increase the flow rate, to reduce losses and to achieve a simpler

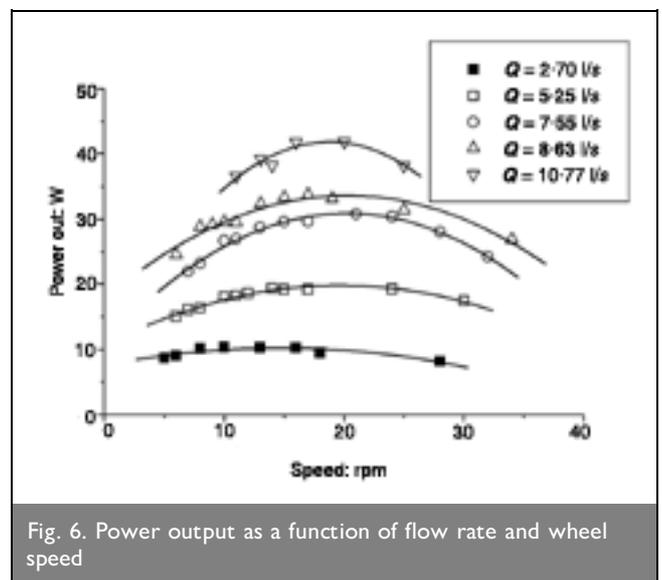


Fig. 6. Power output as a function of flow rate and wheel speed

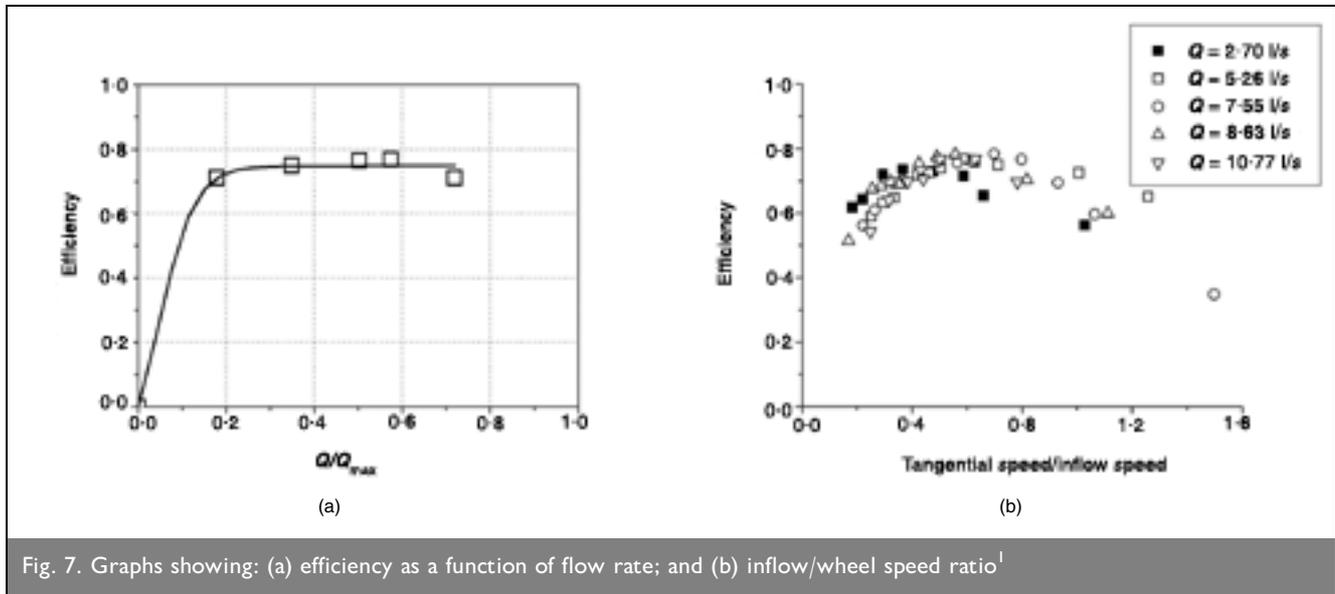


Fig. 7. Graphs showing: (a) efficiency as a function of flow rate; and (b) inflow/wheel speed ratio¹

design, an overflow-type inflow was developed from the original coulisse-type inflow, see Fig. 8(a).

Figure 8(b) shows the efficiency plotted against the flow rate. The maximum efficiency reaches 81.5%—that is, 3% more than the original geometry. The largest flow was measured as 85% of the design flow, which constitutes a 15% increase. The speed for maximum power output was again 16–18 rpm, coinciding well with the design assumption and giving a ratio of tangential to inflow speed of 0.6, similar to the previous experiments. The inflow of the water appeared to be smoother, without the very fast jet generated by the coulisse-type inflow.

3.4. Experimental results—series 3

The wave-like surface of the water at the outflow from the wheel implied that the waterwheel was pushing the water along the tail race channel. Assuming a Manning’s coefficient of 0.002 for Perspex, it was calculated that the water would move with a velocity of 0.85 m/s along a channel with a slope of 6/1000. The tailrace channel was adjusted accordingly. This

seems reasonable since a natural channel would also have a gradient.

Figure 9(a) shows the efficiency as a function of the relative speed. Again, the maximum efficiencies are reached for speed ratios between 0.5 and 0.6. The maximum flow rate was 0.98 Q_{max} , so that the design flow rate could be reached with this arrangement. For flow ratios between 0.25 Q_{max} and 0.98 Q_{max} , the theoretical inflow speed with the weir arrangement only varies by 19%. A waterwheel with a weir inflow can therefore be operated at near maximum efficiency with constant speed for all flow rates. Fig. 9(b) shows the efficiency against flow rate curve. The high efficiencies of 86–87% are maintained over a wide range of flows. The loss in head for the 1m long tailrace was 6 mm, corresponding to 1.2% of the total head. The efficiency gained was 87.3 – 81.5 = 5.8%. An inclined tailrace which carries the water away under its own weight appears therefore essential for an efficient waterwheel.

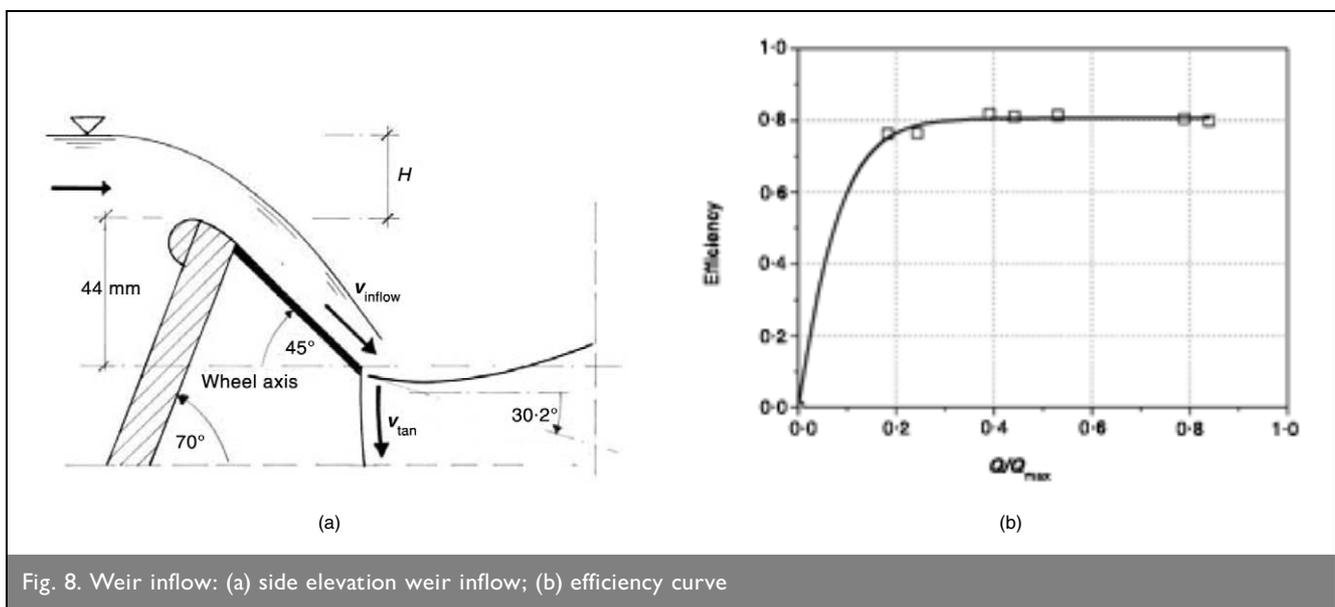


Fig. 8. Weir inflow: (a) side elevation weir inflow; (b) efficiency curve

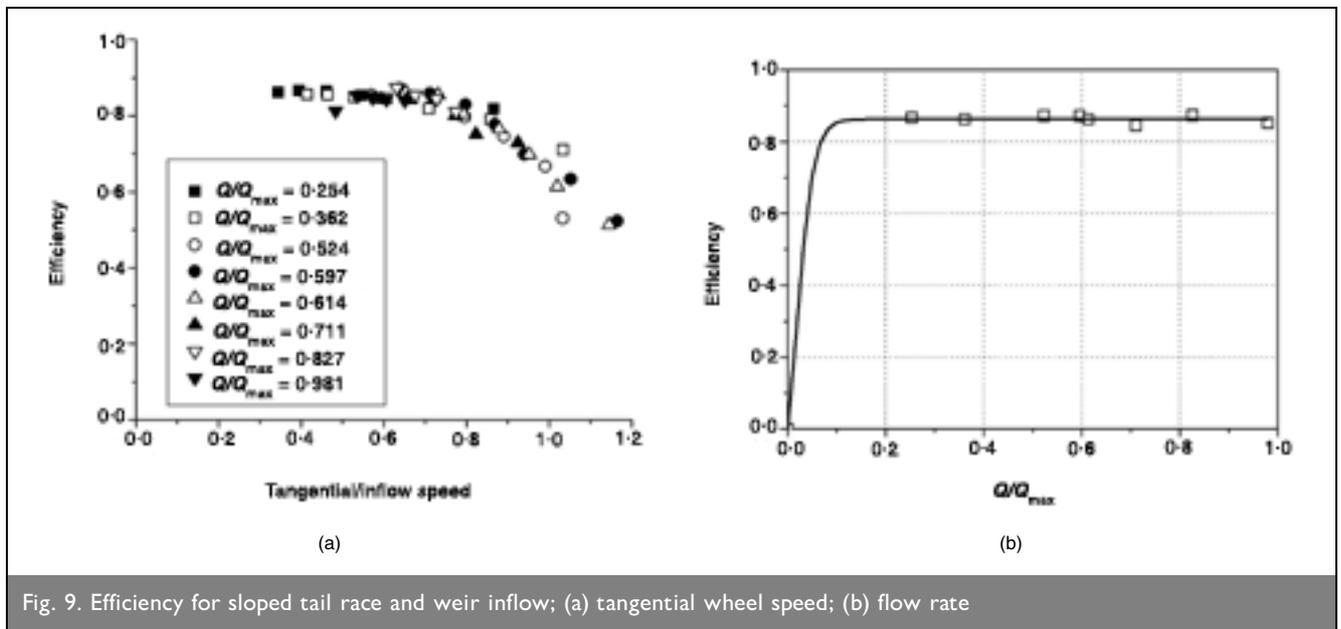


Fig. 9. Efficiency for sloped tail race and weir inflow; (a) tangential wheel speed; (b) flow rate

3.5. Scale effects

The performance of overshot waterwheels both at model scale, with a 1.00 m diameter model, and at full scale (3.60 m diameter) was investigated in 1935³. It was found that scale effects were negligible for flow rates up to the design flow. A further increase of the flow resulted in higher model scale efficiencies compared with full scale. This result seems reasonable since the main driving force of the wheel is the potential energy of the water. Only when very turbulent in- and outflow conditions prevail, will scale effects be expected. Since the inflow into the breastshot waterwheel is quite similar to that of an overshot wheel, scale effects should be small in the model tests reported here.

4. DISCUSSION

The experiments described in this paper show that the breastshot waterwheel is an efficient energy converter for very small head differences. The broad range of flow rates shown in Fig. 9(b) also shows that the breastshot wheel is well suited for sites with highly variable flows. The water wheel was designed following the method outlined in the literature. Initial measurements showed that the required upstream water level above the inflow was up to 14% higher than the theoretically predicted value; at the same time the maximum flow rate was only 70% of the design flow. Both factors point to the fact that the inflow detail as designed provides a higher flow resistance than assumed in the design calculations.

In Bach's design method,¹¹ the inflow detail is designed for the first (of three) slots only open. This is justified by saying that this means that parallel inflow happens even for small flow volumes. For larger volumes, the effective entry angle will be larger, so that the water hits the blade from above. Designing the inflow for the design volume would mean that at small flow rates the inflowing water hits the curved blades from slightly underneath, generating additional losses. In the experiment however, the inflowing water jet appeared to be too fast, shooting along the blade and generating a turbulent motion of the water in the cell which can be assumed to cause energy losses.

An overflow-type inflow reduced losses and led to a simpler design for the inflow. The weir inflow had an angle of 45°, which still allowed for a near parallel inflow of the water but reduced the speed and subsequent turbulence inside the cells. The inclined tailrace finally allowed the water to run off with the tangential speed of the wheel. This combination appears to be close to the optimum. A weir inflow also has the practical advantage that it is less liable to blockage, and simpler to build.

The design method used¹¹ used can be considered as being applicable, with the modifications mentioned before (inflow, tailrace). Some details (tailrace slope required) were not mentioned in the textbooks, and some loss factors were assumed to be too low. Since the design method was based on assumptions not verified by measurements, such errors can be considered unavoidable. The inflow itself should not be designed as a coulisse inflow, but as a weir inflow with an angle of 45–50° with the horizontal. This results in effective entry angles of 0–15° (45° inflow) to 7–20° (50° inflow angle). The condition that the water enters the cell parallel to the cell blade cannot be satisfied—it would require larger inflow velocities. These can only be obtained by moving the point of entry downwards, so that a higher proportion of the available potential energy is transformed into kinetic energy. This constitutes a disadvantage since only a part of the kinetic energy (theoretically 50% in the case of a water jet acting on a flat plate) can be extracted as mechanical power, whereas the potential energy can theoretically be used to 100%. For a given flow rate Q and the resulting potential energy $E_{pot} = H \times Q = 0.50 \times Q$ the percentages of kinetic energy for both inflow configurations can be estimated as follows

Coulisse

10a	$E_{kin} = Q \frac{v^2}{2g} = Q \frac{1.5^2}{19.62} = 0.115Q \equiv 0.23E_{pot}$
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Weir

$$E_{\text{kin}} = Q \frac{1 \cdot 08^2}{19 \cdot 62} = 0 \cdot 057 Q \equiv 0 \cdot 114 E_{\text{pot}}$$

The weir has a head difference of approximately 0.067 m between the water level in the inflow channel and the water surface in each cell. This results in a reduced inflow velocity of 1.08 m/s. Equations 10(a) and 10(b) shows that, for a weir inflow, the percentage of the potential energy which is transformed into kinetic energy is significantly smaller than in the coulisse inflow. This arrangement can therefore be expected to have a higher overall efficiency. The effective entry angles of 0–20° appear to be an acceptable compromise between the demand to utilise as much potential energy as possible, and the design aim to minimise losses at the inflow.

5. DEVELOPMENT POTENTIAL

5.1. Electricity generation

Today, waterwheels would in most cases be used to generate electricity. For the design of a waterwheel installation, it can be assumed that the waterwheel is designed to operate at nominal capacity for 5000–6000 h per year. This value is determined by taking the total amount of electricity generated in a year and dividing it by the rating of the power generator. The main disadvantage of waterwheels for electricity generation is their slow speed. Standard a.c. generators require speeds in excess of 600 rpm to generate a 50 Hz current. This in turn requires a gearbox with a transmission ratio of approximately 1:60. In recent installations of over- and undershot wheels, the costs for power transmission made up 25% (undershot) to 45% (overshot) of the total cost. The author is currently developing alternative solutions for the power train. This includes a synchronous belt drive cascade to replace the gearbox, in combination with a slow speed generator. This power train is expected to generate additional losses of 8–9% (compared with 2–3% for the gearbox). It would however reduce the cost of the power train from approximately £1500/kW to £400/kW installed capacity, leading to overall estimated reduced costs for a breastshot wheel installation of £3100–3500/kW installed capacity. A waterwheel installation would operate parallel to the grid, so that the grid can be used as a variable load. In order to put the power generation costs of a waterwheel's installation into context, the costs were compared with wind and solar power. For this comparison, the following assumptions were made.

- The payback period is 15 years.
- The annual maintenance costs are 4% for wind power, 1% for hydro power and 0.5% for solar power.
- The annual operational time at nominal capacity is 5500 h for hydropower, 1325 h for wind power (average for 2002 in Germany) and 800 h for solar power.

This leads to costs of 4.2p/kWh for microhydro power, 6p/kWh for wind energy and 42p/kWh for solar energy. The costs for wind power stated here are taken for a 600 kW installation, excluding grid connection. It can be seen that when compared with other renewable energy sources, microhydro power can be considered as an economical development.

5.2. Ecological impact

Downstream migration is a common behavioural feature of fish

and an essential element of the life cycle of diadromous species, which have to move between marine and freshwater habitats for spawning. Migration facilities for fish however are usually designed to attract them in the tailwater and to enable their upstream movement, while downstream migrants typically pass the spillway or the power house. Studies of fish mortality and injuries during the passage through Kaplan turbines (commonly used at low head dams) revealed mortalities of eels (*Anguilla anguilla*) and various salmonid species of 5–25%¹⁵ and 11–14%¹⁶ respectively, and serious injuries of other fish species between 3.7% (perch *Perca fluviatilis*) and 50% (common bream *Abramis brama*).¹⁵ Mechanical damage resulting from the rotating blades have been identified as a paramount mortality factor.¹⁷

Considering the fact, that all larger European rivers are heavily fragmented by hydropower dams, similar losses at each turbine constitute significant threats for the survival of migratory species. Accordingly, substantial research has been conducted on the ecological impact of turbines and numerous fish screens have been designed to prevent fish from passage through turbines. All these kinds of screens, bypasses and wire bars are more or less costly and require permanent additional effort for operation and cleaning.¹⁷

In contrast to turbines, fish moving through a waterwheel will not touch any rotating parts and thus be transported downstream in moderate flows and moderately filled cells. Although no information about the ecological impact of waterwheels is available, it appears that the slow speed of the wheel in combination with the low velocities of the water, which is well within the swimming performance of small fish, and the fact that the waterwheel operates under atmospheric pressures should cause only little environmental impact. However, further experimental and mark/recapture studies should be performed on the design of inflow structures and wheel blades to enable a most gentle fish passage without lowering the power efficiency. A protective rack has to be provided to prevent the ingress of large pieces of floating debris and larger fish into the wheel, but expensive fish screens are not necessary.

5.3. Outlook

Although in Europe waterwheels are considered for electricity generation only, they may also be of interest as a mechanical power source in developing countries. Developments in this field would however require close cooperation with interested development agencies and local users in order to consider local requirements and conditions.

Breast shot waterwheels were originally built for power ratings of up to 16 kW/m width.¹² The model tests do however imply that the specific capacity of the wheel (m³/s/m width) can be increased significantly with a moderate loss in efficiency. The construction costs can possibly also be reduced by, for example, improving the geometry of the blades. In combination with the development of a more economical power train, this could lead to the design of an economical and ecologically acceptable power converter for low-head hydro power sources with specific capacities of up to 30 kW/m or overall capacities of up to 120 kW. Further research is however required in order to achieve this goal.

6. CONCLUSIONS

Breastshot wheels were in widespread use in the nineteenth and early twentieth century. There is however only little design information and no information about their performance characteristics available. A literature review showed that the most advanced design method was developed at the end of the nineteenth Century by the German engineer Carl von Bach. In order to illustrate this method, a 4 m diameter wheel was designed. A 1:4 model of the wheel was subsequently built and tested in order to assess the viability of the design method, and the performance of the water wheel. Although breastshot wheels were very popular in the nineteenth and early twentieth century, the experiments described in this paper constitute, to the authors' knowledge, the first performance tests of breastshot wheels. Initial tests with the model wheel showed efficiencies of 78% over a wide range of flows from $0.18 < Q/Q_{\max} < 0.7$. At the outflow a wave-like water surface was visible, indicating additional losses. The inflow detail was found to generate higher losses than anticipated in the design method. This led to a reduced capacity of 70% of the design capacity, and possibly a reduction in efficiency. The inflow detail was redesigned as an overflow. In combination with an incline of the tailrace this led to efficiencies of 87% for $0.25 < Q/Q_{\max} < 0.98$. An initial ecological assessment showed that waterwheels can be expected to have a significantly reduced impact on the fish population when compared with turbines. Further development potential both in terms of improved performance and ecological impact exists. Breast shot waterwheels may therefore offer an economic and environmentally acceptable energy converter for low head differences of 1.5 to 2.5 m and flow rates of 0.5–2.5 m/s.

REFERENCES

- MÜLLER G. and KAUPPERT K. Old water mills—Britain's new source of energy? *Proceedings of the Institution of Civil Engineers—Civil Engineering*, 2002, 150, No. 4, 178–186.
- MÜLLER G. and KAUPPERT K. Performance characteristics of water wheels. *IAHR Journal of Hydraulic Research* (paper 2454, in press).
- MEERWARTH K. D. *Experimentelle und theoretische Untersuchungen am überschlächtigen Wasserrad* (Experimental and theoretical investigation of an overshot waterwheel). PhD thesis, Technical University of Stuttgart, 1935.
- NEUMAYER H., REMPP W., RUPPERT J., SCHWÖRER R. *Untersuchungen am Wasserrad-Triebwerk der Kunstmühle W. Seifried KG, Waldkirch-Br.* (Investigation of a waterwheel power plant at the flour mill W. Seifried KG, Waldkirch/Breisach). University of Stuttgart, 1979, technical report.
- FAIRBAIRN W. *Treatise on Mills and Mill-Works, Part 1.* (3rd edn). Longmans, Green & Co., London, 1874
- MÜLLER W. *Die eisernen Wasserräder: Atlas* (the waterwheels: technical drawings). Veit, Leipzig, 1899.
- MCUGIGAN D. *Small Scale Water Power*. Wheaton, Exeter, 1978.
- GORING O. Powering up the River Thames. *International Water Power and Dam Construction*, 2000, No. 11, 34–35.
- KÖNIG F. and JEHL C. *Bau von Wasserkraftanlagen*, 3. Aufl., C. F. Müller Verlag, Karlsruhe, 1997.
- BUNGE T. (ed.) *Wasserkraftanlagen als erneuerbare Energiequelle—rechtliche und ökologische Aspekte* (Hydro power installations as renewable energy sources—legal and ecological aspects). *Umweltbundesamt: Text 01/01*, 2001.
- VON BACH C. *Die Wasseräder* (The waterwheels). Konrad Wittwer Verlag, Stuttgart, 1886.
- MÜLLER W. *Die eisernen Wasserräder, Erster Teil: Die Zellenräder und Zweiter Teil: Die Schaufelräder* (The iron waterwheels, Part 1: the cell wheels and Part 2: the blade wheels). Veit & Comp., Leipzig, 1899.
- MÜLLER W. *Die Wasserräder* (The waterwheels). Reprint of the 2nd edn, Moritz Schäfer, Detmold, 1991.
- NOVAK P., MOFFAT A. I. B., NALLURI C. and NARAYANAN R. *Hydraulic Structures* (3rd edn). Spon, London, 2001.
- HADDERING R. H. and BAKKER H. D. Fish mortality due to passage through hydroelectric power stations on the Meuse and Vecht Rivers. In *Fish Migration and Fish Bypasses* (JUNGWIRTH M., SCHMUTZ S. and WEISS S. (eds)). Fishing News Books, Blackwell, Oxford, 1998, pp. 315–328.
- COUTANT C. C. and WHITNEY R. R. Fish behavior in relation to passages through hydropower turbines: a review. *Transactions of the American Fish Society*, 2000, 129, 351–380.
- CLAY C. H. *Design of Fishways and Other Fish Facilities* (2nd edn). CRC Press, Boca Raton, 1995.

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