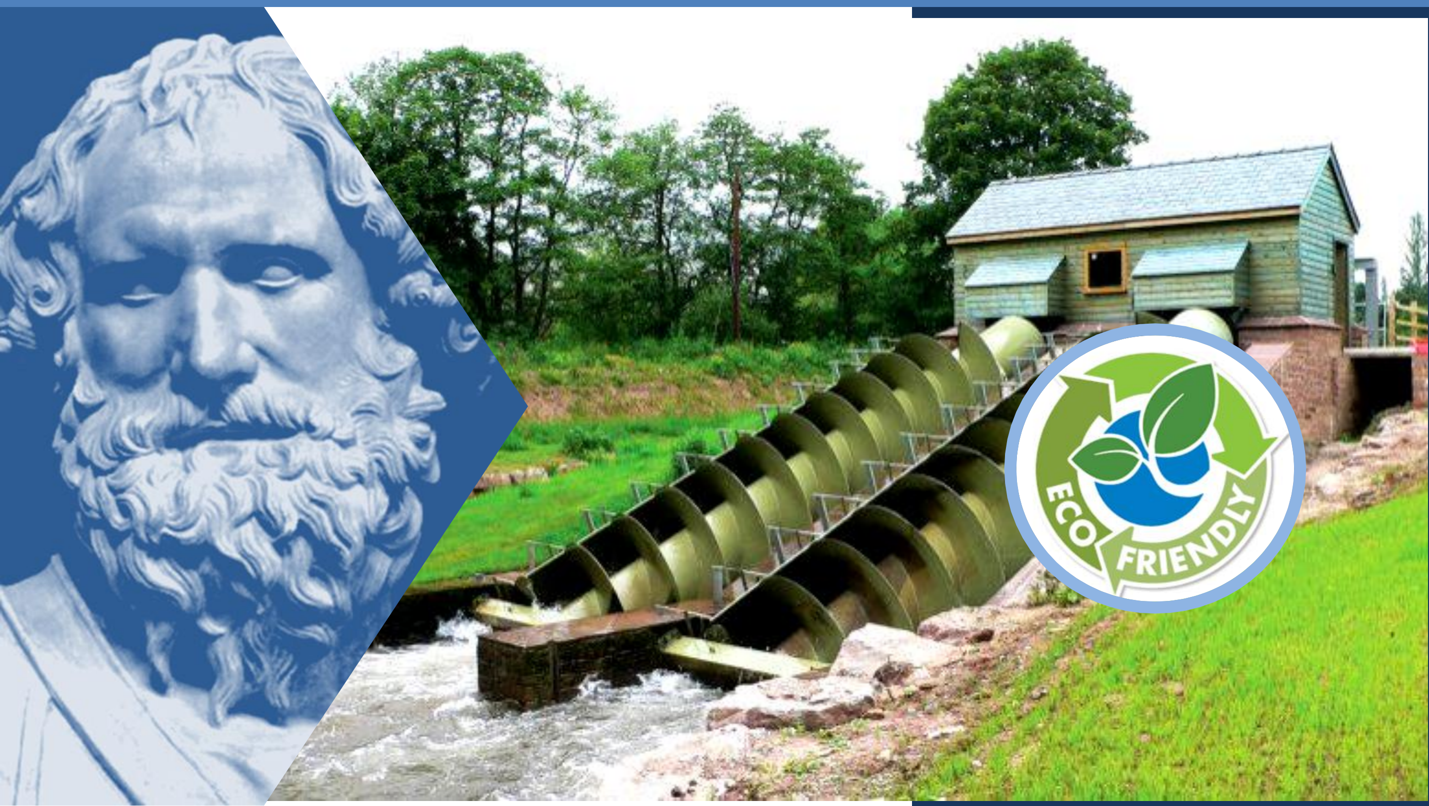
2015

An introductory presentation to the "Archimedean Screw" as a Low Head Hydropower Generator



WATENV

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Contents

***The present booklet is a personal effort of mine to collect data on the Archimedes Screw and by no way represents an original work. All I've done is find the science papers that best describe the hydropower generator in question, copy the interesting pieces and present them in a logical order. For any questions, you can find me at cchar@live.com**

1. Low head Turbines [1]

"Head" refers to the elevation difference between the water levels upstream and downstream of a hydroelectric power plant. As with small hydro, there is not a standard accepted definition of low head hydro. In many jurisdictions, projects with a head of 1.5 to 5 m are considered to be low head. Generally, projects with a head under 1.5 or 2 m are not viable with traditional technology. New technologies are being developed to take advantage of these small water elevation changes, but they generally rely on the kinetic energy in the streamflow as opposed to the potential energy due to hydraulic head.

As with larger hydropower developments, low head hydropower site layouts can vary dramatically from one to another. The development relies on the natural topography of the region in order to take advantage of differences in water elevation to provide head on the plant. This means that there can be tremendous variation in the civil works between sites. Caution must, therefore, be used when making generalizations about low head hydro sites; what applies to one site may not apply to another. However, some broad generalizations can be made.

There are several advantages to low head hydro over other generation types. While not all will apply to a given site, some of the advantages include the following:

• generally smaller impounded reservoir area than for large hydro sites. This reduces both the environmental impact of the projects and associated mitigation costs.

• many low head hydro projects are ROR (run of river) hydro projects. This is thought to reduce both the environmental impact of the projects and the associated mitigation costs.

• there are a large number of existing low head dams and hydraulic structures for flood control and water supply or irrigation. Many of these are suitable for development of low head hydro. This can significantly reduce the capital investment required to develop a hydro station and reduce environmental mitigation and monitoring costs due to reduced environmental impacts.

• diversification of the energy supply is a goal of many governments. Encouraging the development of low head hydro sites can help to meet this goal.

• development of low head hydro sites can also:

1. provide short-term economic benefits for local communities during construction
2. improve water access and navigation in head ponds enhance sport fishing opportunities in head ponds
3. enhance access for resource users to previously inaccessible areas
4. benefit also including income and jobs for community members.

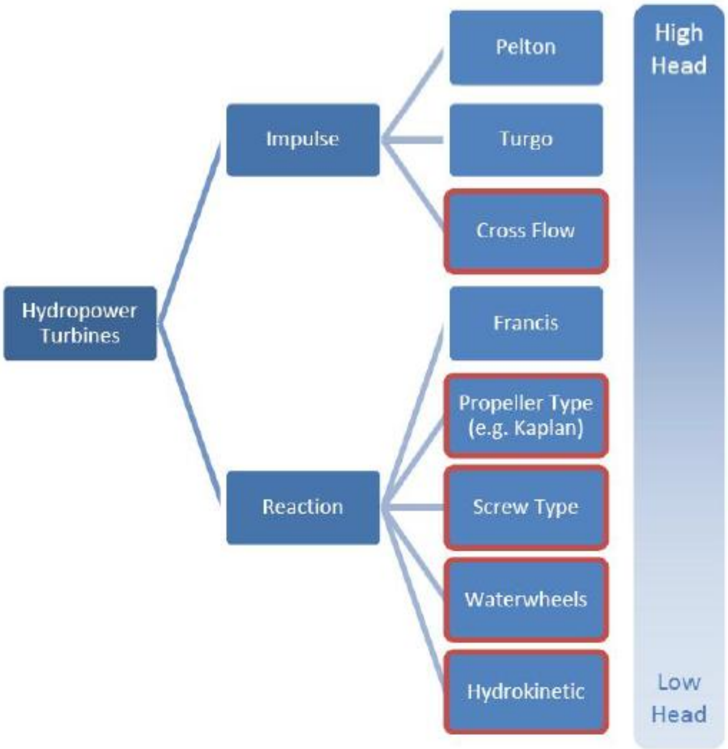


Figure- Classification of turbines

There are also some disadvantages associated with low head hydro developments over other generation types. Again, not all will apply to a given site. Some potential disadvantages include the following:

• Generally, small and low head hydro have limited or no control over when energy is available for generation. Small reservoirs mean that very little water can be stored to be used for generation to follow demand. ROR sites are even more limited; they must generate when water is available with no seasonal storage allowed. Depending on the Power Purchase Agreement, this inability to follow the load can reduce revenues because water cannot be stored for generation during peak demand periods. This in turn would make the project as a whole less economically viable.

• The major disadvantage of low head hydro projects is the project economics. Many of the costs associated with developing a site do not scale down linearly from large to small projects; meaning that on a per megawatt basis, small projects can be far more expensive than large developments.

The size and cost of water conveyance structures and electromechanical equipment required for a hydroelectric project depends largely on the flow rate. The larger electromechanical equipment also requires larger powerhouse facilities. This results in construction costs increasing exponentially as the head decreases, imparting a much larger cost per installed kilowatt to a low head hydro development.

There are, however, some new technologies being developed to circumvent this problem. For example, if a turbine/generator set is placed directly in a stream, with little structural works required, the high cost of the large electromechanical works can be balanced by a reduced need for structural works.

Turbine blades and hydraulic passages are optimized for certain velocities, therefore, for higher flows the turbine dimensions must increase. Not only is the relative cost of the turbine higher at low heads, but the generator cost is also higher. Because low head turbines are associated with high flows and low rotational speeds, the runaway speeds are about 3 times the rated speed, and runaway flows are 2 to 2.5 times the rated flow.

Direct-driven low speed generators with large rotor diameters are subject to high centrifugal forces at such high runaway speeds, resulting in use of more material to resist the internal stress. This means that low head electromechanical equipment gives less power for a unit weight of material and, hence, that generators for low head schemes are generally more expensive.

Another factor that can significantly affect power generation of low head schemes is the relatively high variation in head when the tailwater level rises during periods of high river flows. For a plant with 3 m of head, a rise of 1.5 m in tailwater level significantly reduces the head on the plant. This has a two-fold effect:

• The head available for generation is reduced by 50%.

• The minimum discharge is reduced due to a lack of driving head.

Typically, these factors can combine to result in a 65% loss in power production.

Each low head hydro scheme needs a detailed analysis to find an optimal and most economic solution keeping in view the hydrology, site topography, civil structures, the connected load or grid system, environmental factors, and constraints on transportation.

Very low head hydropower technologies developed within the last few decades. Difficulties in conducting this review arise as the vast majority of sources of information are commercial, only publishing favorable or optimistic data and predictions. Whilst magazine articles and internet blogs discussing most of the technologies exist, they do not present independently verified data.

Many novel hydropower machines exist ranging from patented ideas, operational prototypes, and in the case of just one machine at this time, a commercially established product. What is presented is a selection of the concepts which are considered to potentially be technically and economically viable, covering most of the approaches to exploiting very low head hydropower.

Head, flow and efficiency values provided by the manufacturers are quoted but not verified.

Claims regarding cost or environmental credentials are omitted. This review does not include large free stream kinetic energy converters designed for large scale tidal energy conversion, or ideas that are considered to be technically or economically unviable.

2. Archimedes Screw [1]

The Archimedes Screw is an ancient machine for pumping water from a lower level to a higher one. It is traditionally credited to Archimedes who lived between 287 B.C. and 212 B.C. In recent years, the Archimedes Screw has been installed as a hydropower machine, instead lowering the water and generating power. This turbine consists of a rotating screw supported within a trough by bearings at each end, with a gearbox and generator situated in the control house. The water is lowered within cells which form between the blades and the trough.

Analyses of the geometry and parameters including blade pitch have been conducted for the Archimedes Screw. This is mostly from the perspective of its utilisation as a pump, and its “performance” is based upon the volume of water lifted per rotation. Limited investigation into the efficiency of the Archimedes Screw as a pump has been conducted by initially assuming 100% efficiency from which losses, including leakage, sources of friction and turbulence are deducted. The only work investigating the Archimedes Screw operating as a hydropower machine is a recent Master's dissertation by Harkin (2007).

This work was conducted to investigate the relationship between efficiency and angle of inclination. It was conducted using scale model testing, and secondly a mathematical analysis was conducted, resulting in the power output equation. It is important to note that it has been derived based on an analysis of hydrostatic pressure acting upon the Archimedes Screw. The equation has been derived as an initial attempt to estimate the power output of Archimedes Screws, however the author concludes that whilst the results are of the correct magnitude, the equation requires further development.

To date, the Archimedes Screw is becoming the most commercially successful of the contemporary low head hydropower machines. Dozens of units have been installed in recent times, the main manufacturers being ANDRITZ Atro GmbH (formerly Ritz-Atro GmbH), REHART Group, GESS-CZ s.r.o., Mann Power Consulting Ltd.

The maximum flow rate through an Archimedean screw is determined by the screw diameter. The smallest screws are just 1 meter diameter and can pass 250 liters / second, then they increase in 250 mm steps all of the way up to 5 meters in diameter with a maximum flow rate of around 14.5 m^3/s. The 5 meter maximum is really based on practical delivery restrictions, and in many cases 3 meters is the maximum diameter that can be delivered to a site. If there is more flow available, multiple screws can be installed in parallel.

A series of new Archimedean screw turbines are designed for low heads, in the range of 1 to 10m, with flow rates between 0.1 to 15m^3/s and for inclination angle, between 22 and 40 degrees from the horizontal. For greater heads a cascade of two or more similar energy spiral rotors could give an efficient hydropower solution.

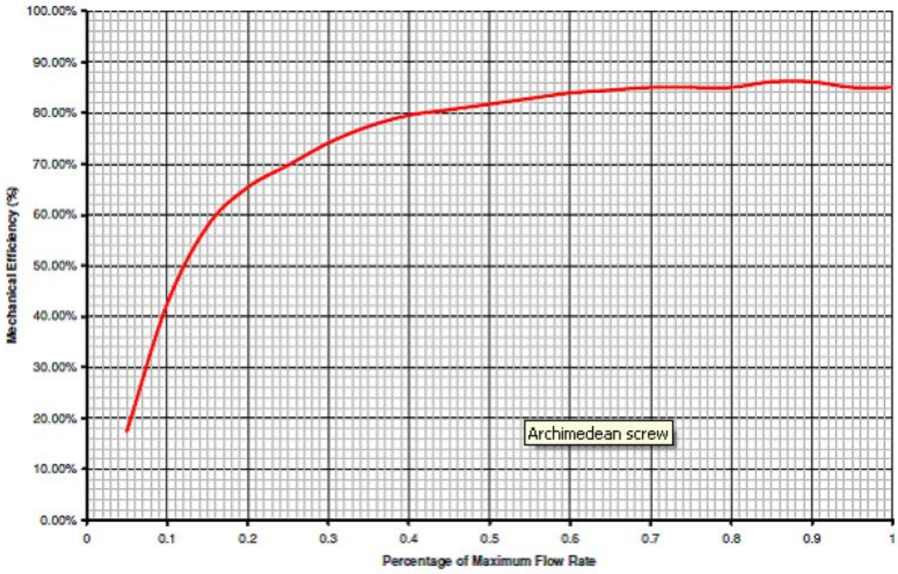
The Archimedean spiral turbine rotors showed the efficiencies between 78 and 83%, making these an interesting alternative for turbines in low head hydropower applications.

Some of manufacturers are claimed that efficiency can be up to 90% for the largest diameter machines. Similar to traditional waterwheels, the filling ratio of the cells is less than one. The machines on the market are currently run at constant speed, the filling ratio increasing with flow rate.

The main parts of an Archimedean screw used as a hydro generator are shown below. The actual screw is below the upper bearing. The helical screw or ‘flights’ are made from rolled flat steel plate that is then welded to a central steel core. Most Archimedean screws have three flights, or three separate helices winding around the central core.

Archimedean screws typically rotate at around 26 rpm, so the top of the screw connects to a gearbox to increase the rotational speed to between 750 and 1500 rpm to make it compatible with standard generators. Even though they rotate relatively slowly Archimedean screws can splash water around, though this is reduced significantly by the use of a splash guard.

Archimedean screws are normally set at an angle of 22 degrees from horizontal, which is the optimum for the most cost-effective installations. There is scope to adjusting the angle slightly if the site requires it (to fit into a particular space for example).



Efficiency of Archimedes screw turbine-Photo credit Renewables first

The best Archimedean screws are variable-speed in operation, which means that the rotational speed of the screw can be increased or decreased depending on the flow rate available in the river. This is much better than having a fixed-speed screw and varying the flow rate through an automated sluice, which creates high head losses and impacts the overall system efficiency. Variable-speed screws are also quieter in operation and don't suffer from "back slap" at the discharge-end of the screw.

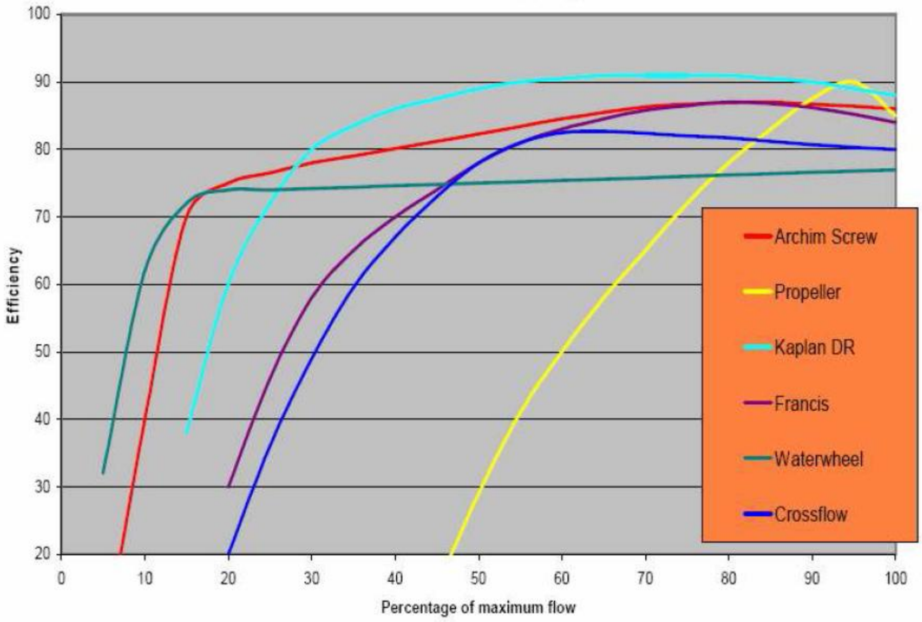
A typical efficiency curve for a good quality variable-speed Archimedean screw is shown on previous figure. This is the mechanical efficiency, so doesn't include the gearbox, generator and inverter losses (these are approximately 15% on in total). It's worth noting that there are some Archimedean screw suppliers that "over sell" the efficiency of screws, so be careful when comparing performance. A lower claimed efficiency may not be because a particular screw is inferior; it could just be that the supplier is more honest!

Good quality Archimedean screws have a design life of 30 years, and this can be extended with a major overhaul which includes re-tipping the screw flights.

A significant advantage of Archimedean screws is their debris tolerance. Due to the relatively large dimensions of the screw's flights and slow rotational speed, relatively large debris can pass through unhindered and without damaging the screw and certainly all small debris such as leaves can pass through without any problems at all. This means that fine screens are not required at the intake to the screw and they can manage with course screens with 100 or 150 mm bar-spacing. This leads to relatively modest amounts of debris build-up on the course screen and removes the requirement for (expensive) automatic intake screen cleaners which are normally required on larger low-head hydropower systems.

The low rotational speed and large flow-passage dimensions of Archimedean screws also allow fish to pass downstream through the screw in relative safety. Archimedean screws are often touted as “fish friendly” hydro turbines, which they undoubtedly are. In non-screw hydro systems this just means well designed intake screens and fish passes / by passes would be required. Note that if upstream fish passage is required at an Archimedean screw site, a fish pass will be required.

The final advantage of the Archimedean Screw is simplified civil engineering works and foundations. Because screws don’t have draft tubes or discharge sumps, it means that the depth of any concrete works on the downstream-side of the screw is relatively shallow, which reduces construction costs. The civil works are also relatively simple, the main part being the load-bearing foundations underneath the upper and lower bearings. In softer ground conditions the load-bearing foundations can be piled.



Source: Source: Eco Evolution, 2009

Advantages of Archimedes screw

• Very cost-effective compared with turbines and water wheels

• Better efficiency with partial loads than comparable water sheels and turbines

• Simple to use, install and maintain

• No complex excavations

• Durable bearings thanks to low speed

• Robust, wear-resistant and reliable

• Fine screen not required; resistant to flotsam and compatible with fish

• Can be used with a head as low as 1 metre and flow as little as 0,1m³/s

• Screws can be coated with the highly wear-resistant "nanoseal" ceramic composite material

Disadvantages of Archimedes screw

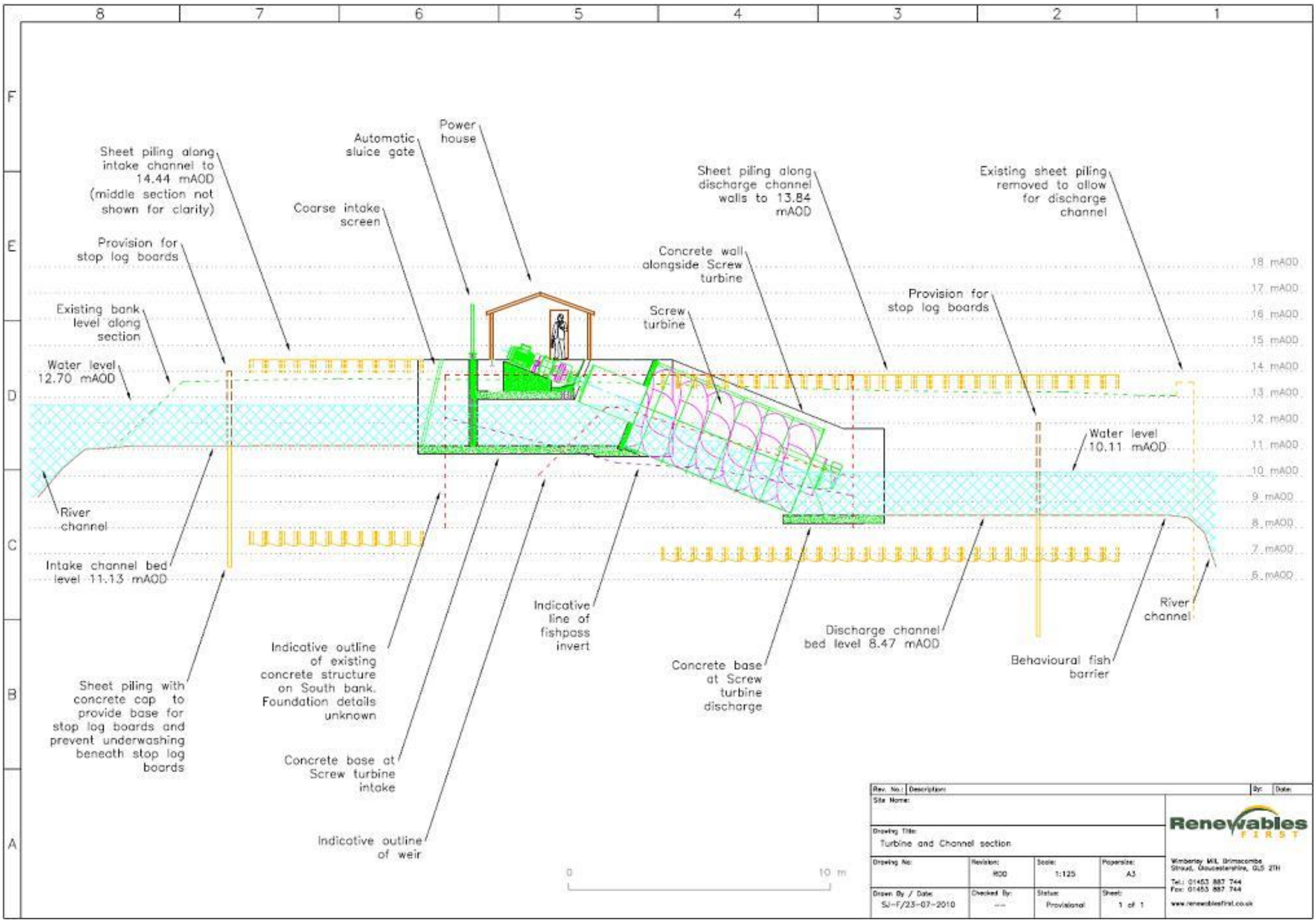
• Change of head during the year and the consequent changes in production

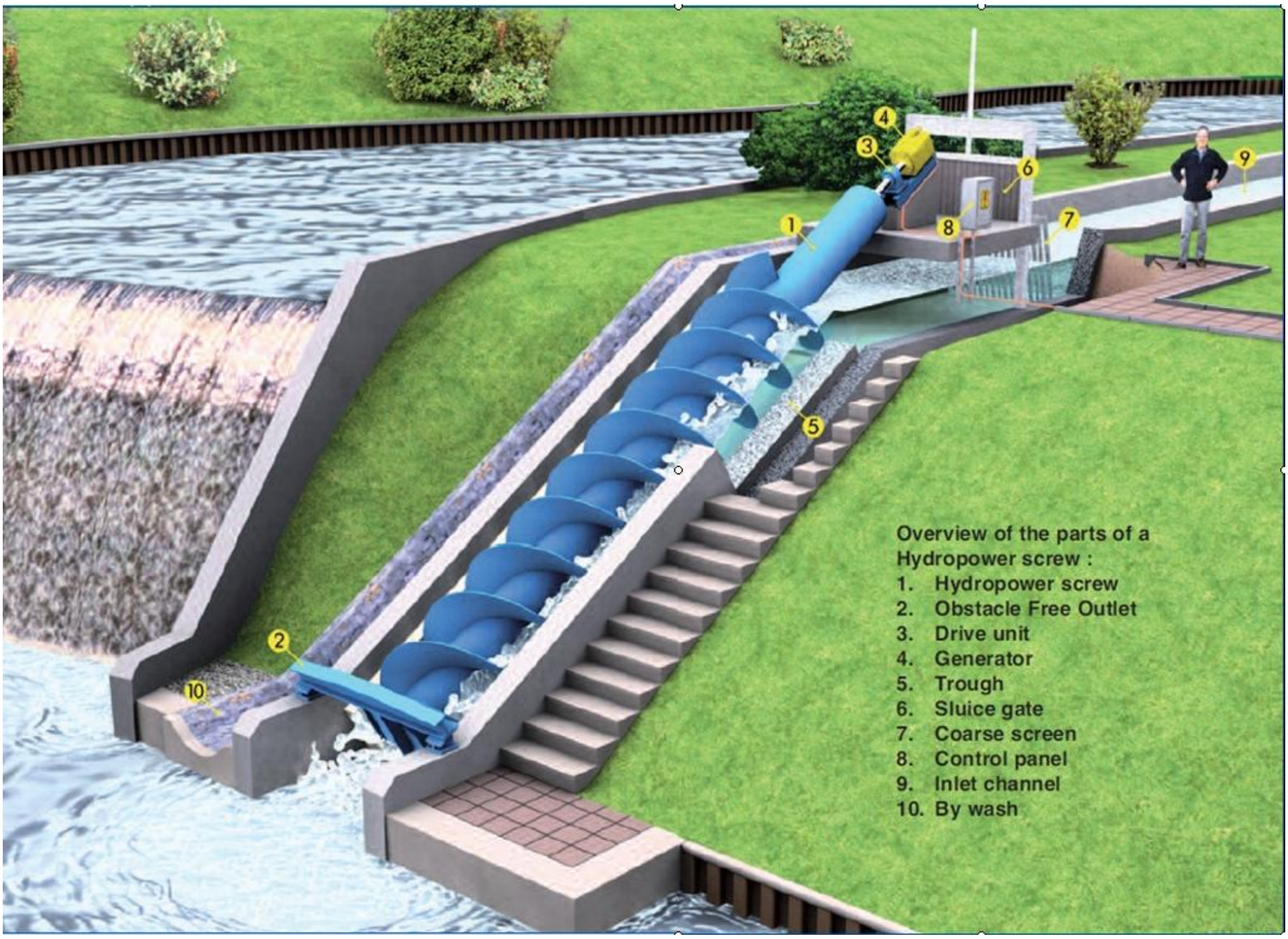
• Requires high flow rates

• Maintenance of lower bearing is difficult

• Low rpm require gearbox and this reduce the efficiency

• For high efficiency Archimedean screws need variable-speed in operation




4
3
6
9
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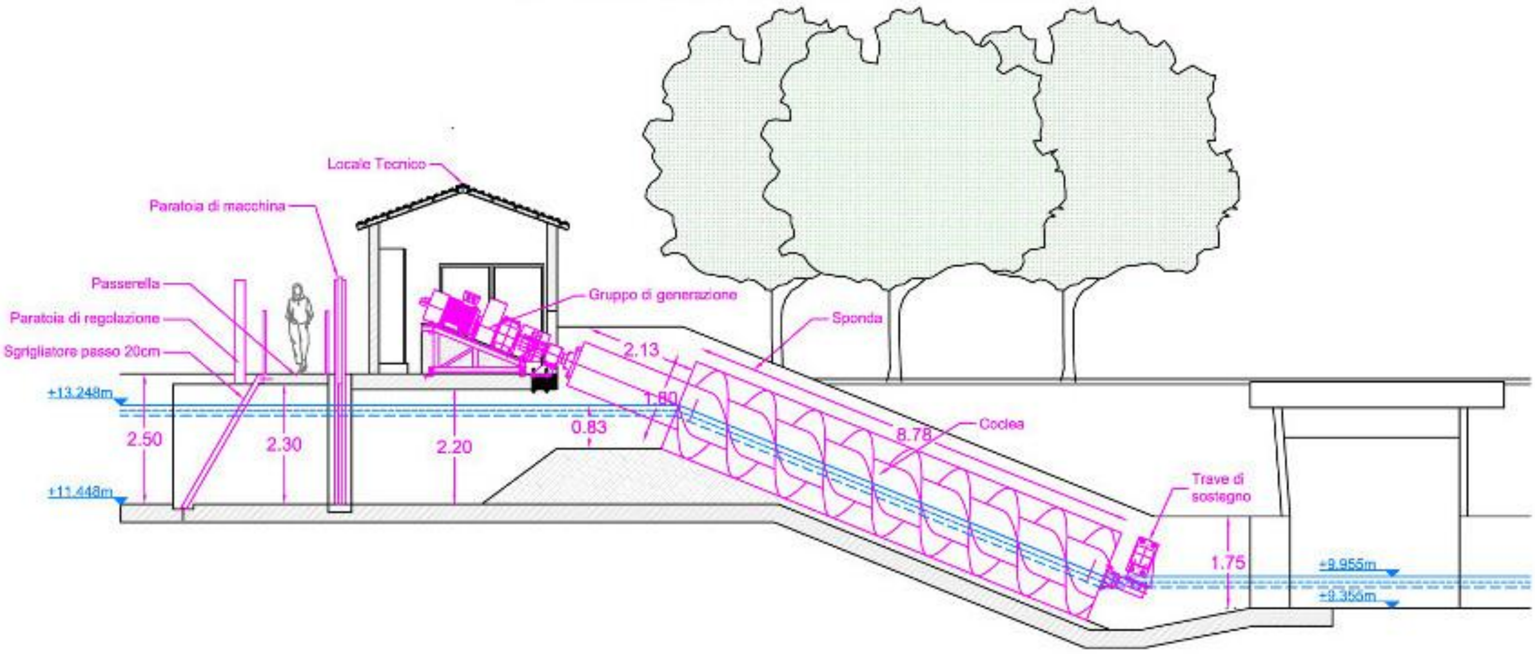
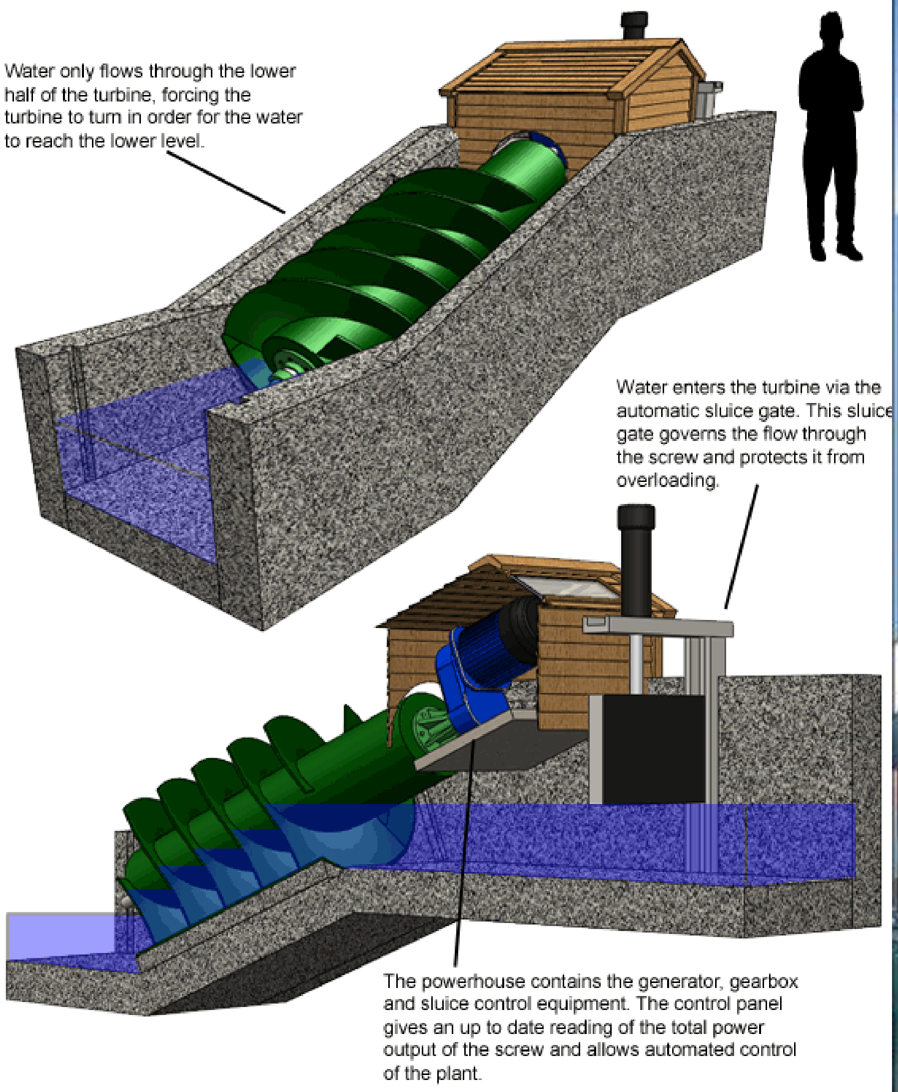
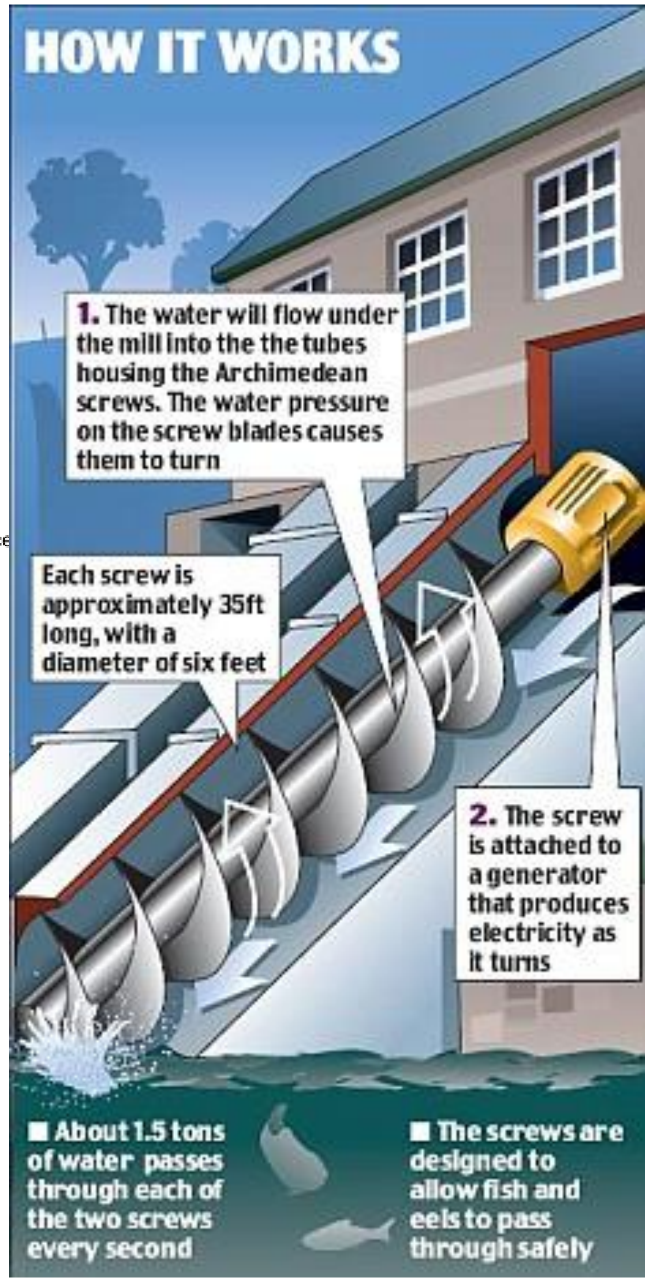
Overview of the parts of a
Hydropower screw :
1. Hydropower screw
2. Obstacle Free Outlet
3. Drive unit
4. Generator
5. Trough
6. Sluice gate
7. Coarse screen
8. Control panel
9. Inlet channel
10. By wash

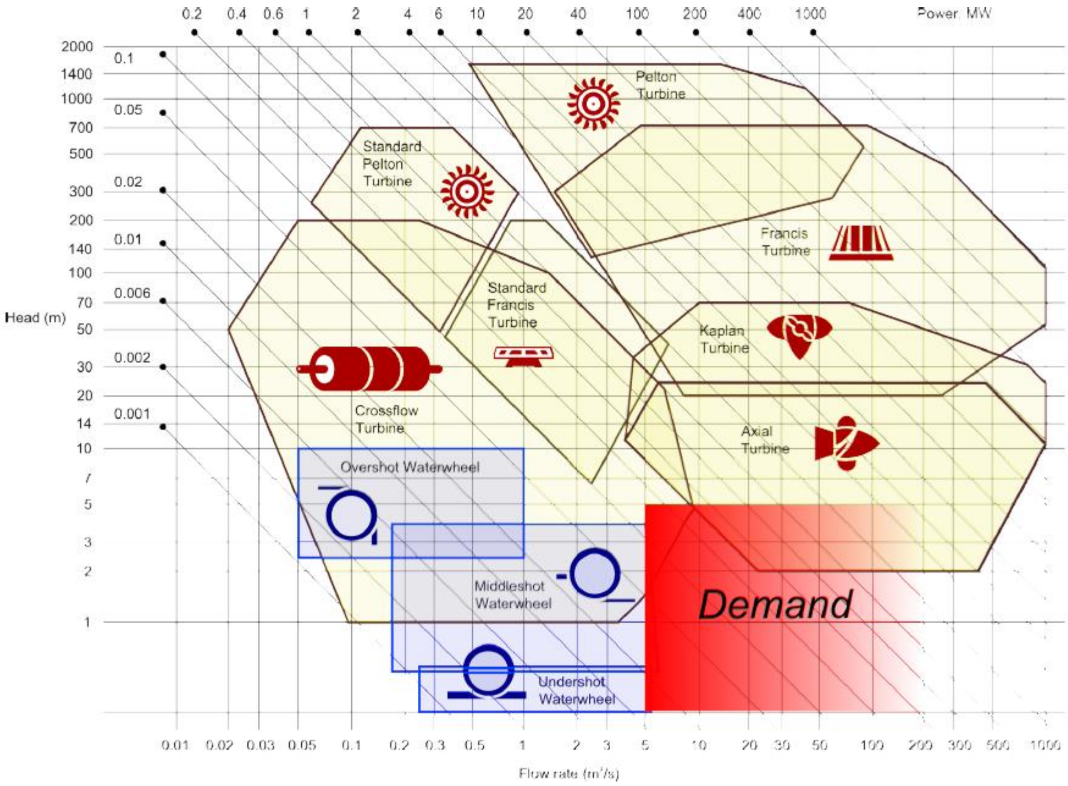

Locale Tecnico
Paratoia di macchina
Passerella
Paratoia di regolazione
Sgrigliatore passo 20cm
Gruppo di generazione
Sponda
+13.248m
+11.448m
2.50
2.30
2.20
2.13
1.80
0.83
8.78
Coclea
Trave di
sostegno
1.75
+9.955m
+9.355m





3. The Archimedean Screw as a Potential Hydropower Generator [2]

In the western world much of the large scale, high output, hydropower sites have now been exploited. Within Europe the focus has now shifted to Small Hydro-Power, which is installations with power outputs beneath 10MW. The European target is to achieve an additional 2.4GWof power generation from Small Hydro-Power plants by 2010, relative to the 2005 generation levels. Within this bracket are sites with 'very low head' which refers to sites where the vertical distance through which flowing waterfalls over structures or terrain is less than 5m. At this point in time, no technology for this bracket satisfactorily meets the economic and ecological requirements required by investors and the authorities. As a result, the Seventh Framework Program's 'Research Priorities for the Renewable Energy sector' set by the European Union includes the development of small turbines for very low heads under 5m as one of its long term targets.



Established hydropower machines and demand for new technology, modified from Giescke and Monsonyi (1998)

To date, the Archimedes Screw is becoming the most commercially successful of the contemporary low head hydropower machines. Dozens of units have been installed in recent times, the main manufacturer being Ritz-Atro, a German manufacturer from which the following information is sourced (Atro 2006). Referring to Figure 3.18, their Archimedes Screws are claimed to be suitable for flow rates up to 5.5 m3=s per unit, and heads between 1m and 10m. Efficiencies are claimed to be up to 90% for the largest diameter machines. Similar to traditional waterwheels, the filling ratio of the cells is less than one. The machines on the market are currently run at constant speed, the filling ratio increasing with flow rate. The manufacturer claims that the Archimedes Screw is driven by the *potential* of the water.

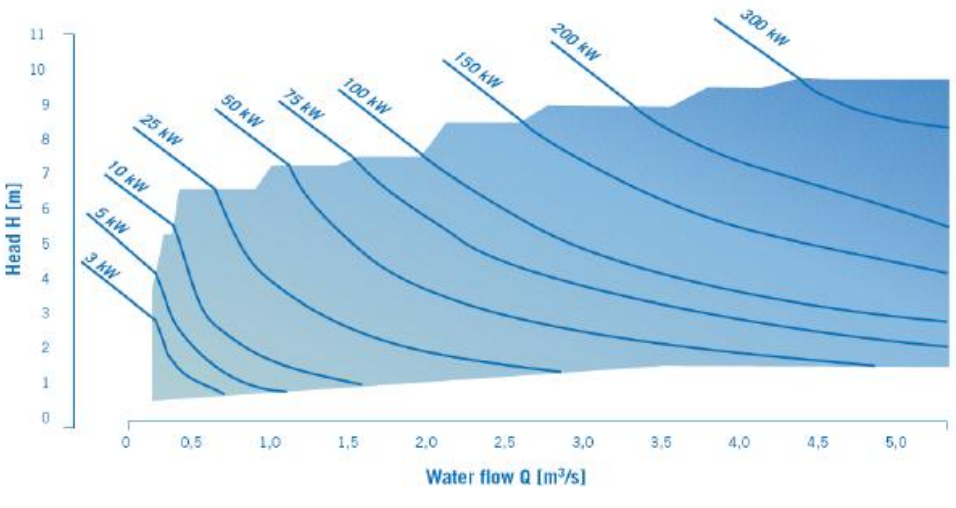


Figure 3.18: *Archimedes Screw* operating range (Atro 2006)

Potential: This term is used to describe machines where the weight of water contained within cells is exploited to drive hydropower machinery by lowering the cells through the available head difference, reducing their potential. The most prominent example of a potential machine is the overshot waterwheel. The concept of potential machines is quite intuitive, the driving force at any point being analogous to the weight experienced when holding a bucket of water. This understanding is however overly simplistic and the fundamental properties of such machines are not recognised in the literature. As a result several machines, such as middleshot waterwheels and Archimedes Screws are at times inaccurately, or at least over simplistically, referred to as being driven by weight or potential.

3.1 Theory of Potential Machines

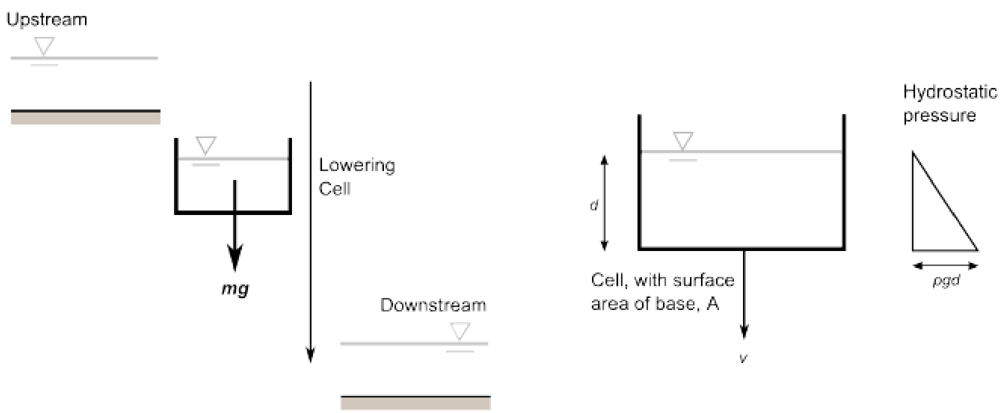


Figure 4.1: Representation of Potential machines: a) weight based analysis b) hydrostatic pressure based analysis

An ideal model of a potential machine is given in Figure 4.1a, showing an individual cell with a horizontal base and vertical walls, containing water, and descending vertically from the upstream to the downstream. Referring to the lowering cell, this is the most intuitive analysis of weight, where the downward force of the water is simply the product of the mass of the water multiplied by the acceleration due to gravity. Although correct, this analysis does not readily make the fundamental properties of such machines clear.

Referring to Figure 4.1b, the weight of the water exerts a force on the cell as a result of the hydrostatic pressure it generates. Hydrostatic pressure is defined as the pressure at a point in a fluid due to the weight of the fluid above it. This is calculated using Equation 5.1, where the hydrostatic pressure is p, the depth of the water in the cell is d, ρ is the density of the fluid and g is the acceleration due to gravity. The hydrostatic pressure is zero at the water's surface and increases linearly to its maximum, ρ x g x d, at the cell's base. The downward force of the water within the cell, F, is the product of the pressure acting on the base of the cell, multiplied by the area of the base. As the cell lowers, and force is exerted over a distance, it is said to do work. The amount of work done per second is the power, P, of the moving cell. This is calculated by multiplying the force by the velocity of the cell, shown by Equation 4.2.

$$p = \rho \times g \times d \qquad (4.1)$$

$$P = F \times v = (\rho \times g \times h \times A) \times v \qquad (4.2)$$

An understanding of the role of hydrostatic pressure and Newton's laws allows the following fundamental properties of true potential machines such as overshot waterwheels to be recognised:

- Potential machines are designed so that the water is contained entirely within cells, not requiring stationary shrouds or casings to retain the water between moving blades. The hydrostatic pressure of the water within the cell acts on all submerged surfaces. Forces exerted by the pressure acting laterally on the opposing sides of the cells cancel. The net force is downwards, resulting from the pressure acting on the base of the cell.
- As the base of the cell is directly beneath the water contained, it is subjected to the maximum hydrostatic pressure that can be generated over its entire surface area. Therefore for a given depth of water, the force exerted on the base of the cell is the maximum available force.
- For a net force downwards to exist, the underside of each cell is surrounded by air at atmospheric pressure. Therefore there is a pressure difference across the base of the cell resulting from the weight of the water on its top surface.
- The movement of the cells of potential machines is vertical, or rather the direction in which work is done is parallel to gravitational acceleration.

3.2 Limitations of the Potential Analysis

The fundamental properties of true potential machines such as the overshot waterwheel have been identified in the previous subsection. Using these as a reference, the departure of machines such middleshot waterwheels and the Archimedes Screw from the potential machine analysis can be analysed. These departures are demonstrated using Figure 4.2 which represents a generic middleshot waterwheel. Initially, a blade at position 'A' does represent a cell exploiting the potential of the water it contains. The cell is horizontal, moves vertically, and the working surface is situated beneath the water and is subjected to the maximum pressure for the given depth.

However, as the blade rotates about the axle, the situation no longer matches the criteria identified for a true potential machine.

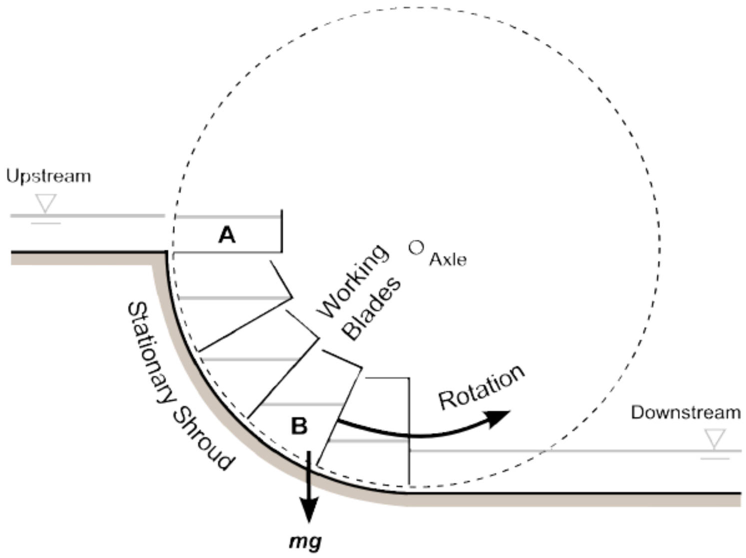


Figure 4.2: Representation of changing operational principle within a middleshot waterwheel

Considering the scenario when the blade is at position 'B':

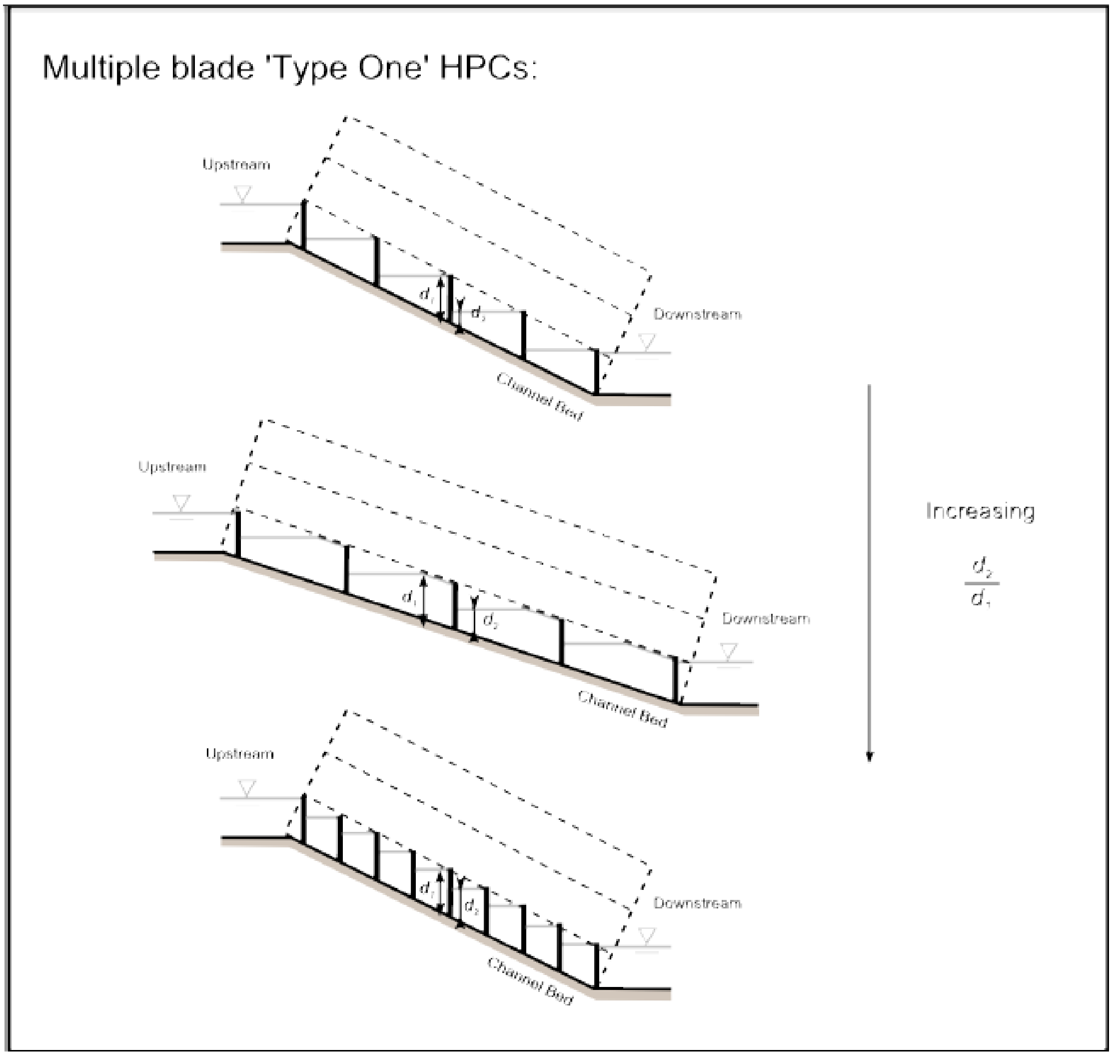
- The water is contained between the shroud and the blade, both of which are subjected to hydrostatic pressure. The blade is mobile and is therefore a working surface which generates power. The shroud however is stationary, so the hydrostatic pressure acting on it does no work, and generates no power.
- The direction of work is not vertical, parallel with gravity, but tangential to the axle of the machine. 'Weight' describes the vertical component of a force exerted by mass as a result of gravity. As the blade rotates beneath the axle and tends towards the vertical, the lateral component of the force exerted by the hydrostatic pressure on the blades becomes dominant.
- The pressure acting on the working surface is not constant as the blade is beside and not beneath the water. It increases with depth from zero at the water's surface, and therefore the maximum force available from a given depth of water is not exerted on the blade.
- The combination of blades and a shroud is not the same as the cells employed by potential machines. The reverse side of the blade becomes partially submersed by the water contained in the previous cell. This creates a counteracting pressure. In the extreme case of middleshot waterwheels with filling ratios of one such as the Aqualienne and Zuppingerrad, this means that the pressure difference across the blades whilst within the shroud is zero as there are no free water surfaces within the cells.

The analysis of a blade at position 'B' on a middleshot waterwheel, where a non-horizontal working surface moves with a lateral component relative to a stationary shroud, applies equally to the blades of an Archimedes screw. It can be appreciated that the generation of pressure, force and power for these machines is significantly more complex than for a true potential machine.

In conclusion, it would be more thorough and exact to describe such machines as being driven by Hydrostatic Pressure, which acts in all directions on all submerged surfaces, as opposed to Potential, which acts only in a vertical direction.

3.3 Application of the Hydrostatic Pressure Converter Theories-`Type one' HPCs

'Type one' HPCs were defined as machines where the working surface extends from the channel bed to the upstream water surface. The ideal theory developed to describe such machines determined that their hydraulic efficiency was fundamentally governed by the ratio of downstream to upstream submerged water depths on the working surface, d_2=d_1. Through the use of experimental testing, this relationship was verified to hold for the peak efficiency, whilst it was also determined that in practice, leakage and turbulent losses became dominant factors at the lower and higher flow rates of any machines' operating range. These three factors need to be considered together when designing any 'type one' HPC installation, as they not only affect the efficiency and ultimately the power output, but also the size and cost. These factors need to be balanced if the design of the installation is to be economically viable.



Consider 'type one' HPCs such as the Archimedes Screw, where the entire head difference is generated over multiple working blades. The literature review indicated that the efficiency of the Archimedes screw operating as a pump was thought to relate solely to the losses of the machine, including leakage and turbulent losses. As a hydropower machine, it's power output was estimated using the equation shown in Appendix A.

Whilst this also approaches the problem through an analysis of hydrostatic pressure, it does not easily lend itself to the understanding of design factors which influence efficiency. Compared to these previous analyses, the 'type one' HPC theory suggests that the efficiency is also a function of the ratio d_2=d_1.

- The d_2=d_1 ratio across the blades of an Archimedes Screw depends the gradient and the number of blades of the installation as shown in Figure 9.1. It also depends on the speed of revolution. Such machines are similar to traditional waterwheels in that they can operate with variable filling ratios. As such, the machines can handle varying flow rates whilst operating at constant speed, determined by the generator and controller. The result of this operational regime is that the filling ratio and thus the d_2=d_1 ratio alters with flow rate. The optimum operational regime would be to run the screw at variable speed, so that the maximum filling and d_2=d_1 ratio can be maintained at all flow rates. The leakage flow through the installation would increase as the length of submersed blade increases with the d_2=d_1 ratio. The greater the number of blades, or rather the turns of the thread, the higher the maximum speed of revolution must be for a given flow capacity. This could create higher turbulent losses at the inlet and outlet of the screw, reducing efficiency at higher flow rates.

- When designing an Archimedes Screw installation, a balance must be sought between performance and cost, which will increase with the d_2=d_1 ratio. This is because additional blades and shallower gradients, which result in longer screws, both require more material and labour to manufacture and install. Equally, maintaining high d_2=d_1 ratios through the use of variable speed generator and control system would increase costs.

$$
\begin{aligned}
&\Lambda_{total} = \\
&n\omega\rho g\sin(\alpha)\left[\left\{\left(r\sqrt{1-\tan^2(\alpha)}\tan(\frac{\pi}{2}-\alpha)\right)\left(\theta_b-\theta_a\right)\right.\right. \\
&\left.+\frac{L}{2n\pi}\left((\pi-\cos^{-1}(\tan(\alpha)))(\theta_b-\theta_a)-\frac{\theta_b^2-\theta_a^2}{2}\right)\right\} \\
&\left\{\frac{-1}{4n^2}\frac{\sqrt{\frac{L^2+4n^2\pi^2R^2}{L^2}}(L^2+4n^2\pi^2r^2)-\sqrt{\frac{L^2+4n^2\pi^2r^2}{L^2}}(L^2+4n^2\pi^2R^2)}{\pi^2\sqrt{\frac{L^2+4n^2\pi^2R^2}{L^2}}\sqrt{\frac{L^2+4n^2\pi^2r^2}{L^2}}}\right\} \\
&+\tan(\frac{\pi}{2}-\alpha)\left(\cos(\theta_a)-\cos(\theta_b)\right)\left\{\frac{L^3}{16n^3\pi^3}\left[\frac{2nr}{L}\sqrt{\frac{L^2+4n^2\pi^2r^2}{L^2}}\pi-\right.\right. \\
&\ln\left(\frac{2n\pi r}{L}+\sqrt{\frac{L^2+4n^2\pi^2r^2}{L^2}}\right)+\ln\left(\frac{2n\pi R}{L}+\sqrt{\frac{L^2+4n^2\pi^2R^2}{L^2}}\right)- \\
&\left.\left.\left.\frac{2nR}{L}\sqrt{\frac{L^2+4n^2\pi^2R^2}{L^2}}\pi\right]\right\}\right]. \qquad (28)
\end{aligned}
$$

Figure 11.1: Archimedes Power output equation by Harkin (2007)

4. Archimedes Screw Design and Optimisation [3]

Presently the dimensioning of a hydropower site equipped with an Archimedes screw is dependent mainly on the experience of the design engineer. This is so because there is no appropriate theory for the optimal hydraulic design of screws used at hydropower sites, especially for the water-inflow conditions. In this paper, a new analytical model for the water inflow is derived, and the optimal values of the inflow parameters are determined to achieve a planned inflow to the screw.

This is a new and growing enterprise in Europe where more than 180 sites are now equipped with Archimedes screws. This is surprising considering that the first installations date back only to the year 1998.

The application of the Archimedes screw for use in hydropower plants to drive machinery was proposed by Radlik in the German Patent No. DE4139134C2 in 1992. Brada (1996) carried out the first experimental verification of the usefulness of a screw for electric-power generation in the years 1993–1995 at the University of Prague.

Presently, the largest screws have outer diameters of no more than 4 m; however, fatigue cracking of the weld of the flights to the central tube may prohibit larger screws. In addition to this high efficiency, the requirement throughout Europe, by law, that hydropower plants be fish friendly has encouraged the use of the Archimedes screw.

The main differences between screw pumps and water-powered screws are the water-transport direction, the inflow conditions, and the outflow conditions. To construct an analytical model of a screw used to generate electric power, consider the cross section of a typical screw used for this purpose shown in Fig. 2.

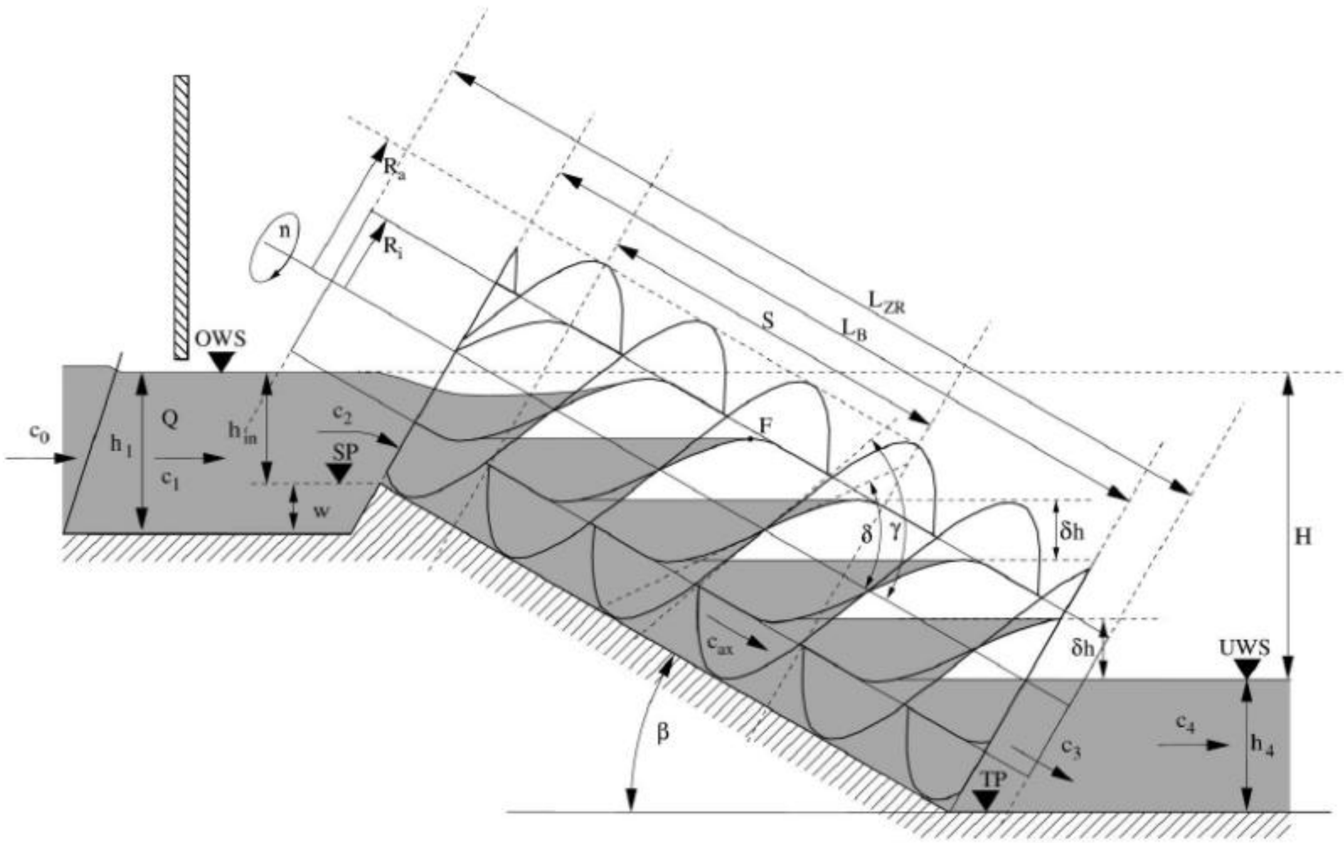


Fig. 2. Cross section of an Archimedes screw used in a hydropower plant

The "chute" of the screw is the region between two successive flights and its inner radius (R_i) and outer radius (R_α). A screw with N flights has N chutes, each with a volume of $\pi(R_\alpha^2 - R_i^2)S/N$ over one pitch-length S of the screw. The term "bucket" is used to refer to one of the maximally connected regions occupied by the trapped water within any one chute when the water is up to the optimal filling point F (Fig. 2). The volume of such a bucket will be denoted by V_B and the volume of all N buckets over one pitch length will be denoted by V_U (=NV_B). Notice that V_U is the volume of water that flows into or out of the screw with each turn of the screw.

The main purpose of this paper is to determine the inflow head h_{in} defined in Fig. 2 to achieve a specified flow Q through the screw when each bucket is optimally filled to volume V_B. To this end, it is assumed that the inflowing water enters through a rectangular channel whose width matches the diameter $2R_\alpha$ of the screw and that the height of the incoming water is Ra. This height condition is to satisfy the well-known fact that to minimize the friction of flowing water in an open rectangular channel the height of the water should be half the width of the channel.

4.1 Discharge of the Screw

The total flow Q to the screw splits into the following five components within the screw as shown in Fig. 3

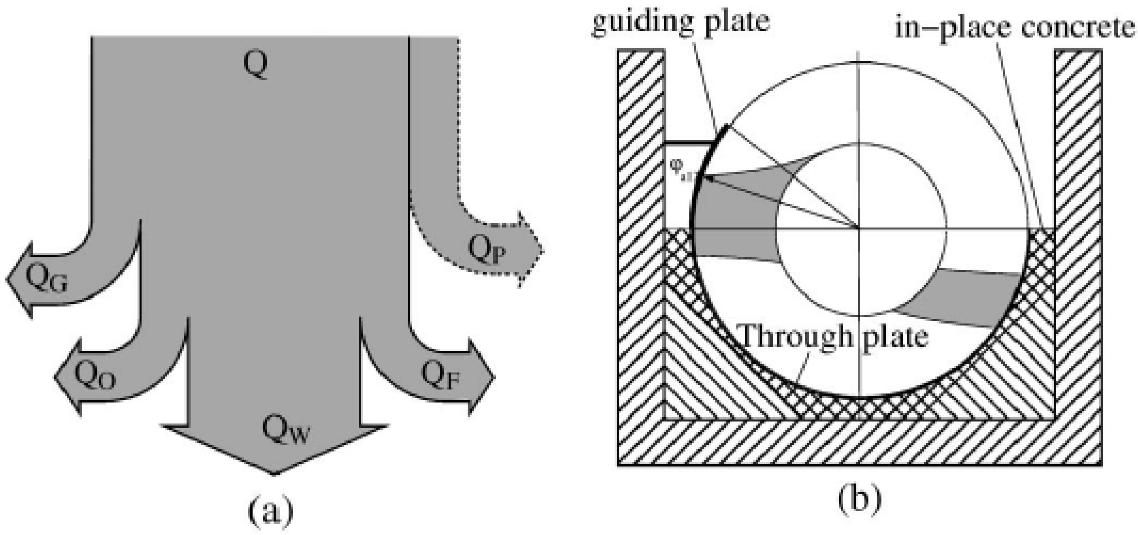


Fig. 3. (a) Balance of the flows; (b) trough and guiding plates

• Q_W: the flow generating the screw torque,

• Q_G: the leakage flow through the gap between the trough and the flights of the screw,

• Q_O: the leakage flow if the screw is filled above the optimal filling point F,

• Q_F: the friction-leakage flow formed by water that adheres to the flights, and

• Q_P: the leakage flow if there is no guiding plate to extend the trough.

The leakage flow Q_P can be eliminated by using a guiding plate as shown in Fig. 3(b), and the friction-leakage flow Q_F can be neglected because its effect is comparatively small. Thus, the total inflow balance equation becomes $Q = Q_W + Q_G + Q_O$. The three components Q_W, Q_G, and Q_O are calculated below.

The volume of water in each bucket remains fixed as the bucket descends because the leakage flows Q_G and Q_O out of each bucket are balanced by the corresponding flows into each bucket from the bucket above it within its chute. The flow Q_W that generates the screw torque is the portion of the total flow Q that fills each bucket up to the optimal filling point F, that is, so that each bucket has volume V_B. The hydraulic energy of the two leakage flows are lost in turbulence, i.e., in heat energy. Only the hydraulic energy of the water of volume V_B in each bucket can be converted into rotational energy of the screw.

The flow Q_W was given by Weisbach (1855) as

$$Q_W = n/60 \times V_U \ [m^3/s] \qquad (1)$$

where n = rotational speed of the screw in min^{-1}. Muysken (1932) suggested the following expression for the maximum rotational speed of a screw:

$$n \leq 50/\ (2R_a)^{2/3}\ [min^{-1}] \qquad (2)$$

Muysken based this expression on experience and experimentation, showing that rotational speeds above this value produce excessive friction losses and centrifugal forces. Most European screw manufacturers recommend and use this rotational speed.

To determine the volume V_U in Eq. (1), the following dimensionless parameters are used in this paper:

$$\rho = R_i/\ R_a \text{ (radius ratio)} \qquad (3)$$

$$\lambda = (S \times \tan\beta)/\ (2\pi Ra) \text{ (pitch ratio)} \qquad (4)$$

$$v_U = (V_U \times \tan\beta)/\ (\pi R_a^2 S) \text{ (volume ratio)} \qquad (5)$$

$$\lambda v_U = (V_U \times \tan\beta)/\ (2\pi^2 R_a^3) \text{ (volume ratio-per-turn ratio)} \qquad (6)$$

The radius ratio ρ must, of course, lie between 0 and 1. This is also true for λ since the maximum pitch S for which a bucket of water can be formed is $2\pi R_a/\ \tan\beta$. It should be noted that v_U depends only on N, ρ, and λ.

From Eqs. (1) and (6) it follows that

$$Q_W = (2\pi^2 R_a^3)/\ \tan\beta \times \lambda v_U \times n/60 \qquad (7)$$

and so for a screw of specified outer radius R_a, angle β, and rotational speed n the flow Q_W is a specified multiple of the volume-per-turn ratio λv_U. A contour map of the values of λv_U for the optimum filling point F as given by Rorres (2000) for a screw with three flights (N = 3) is shown in Fig. 4 for various values of the radius ratio ρ and the pitch ratio λ.

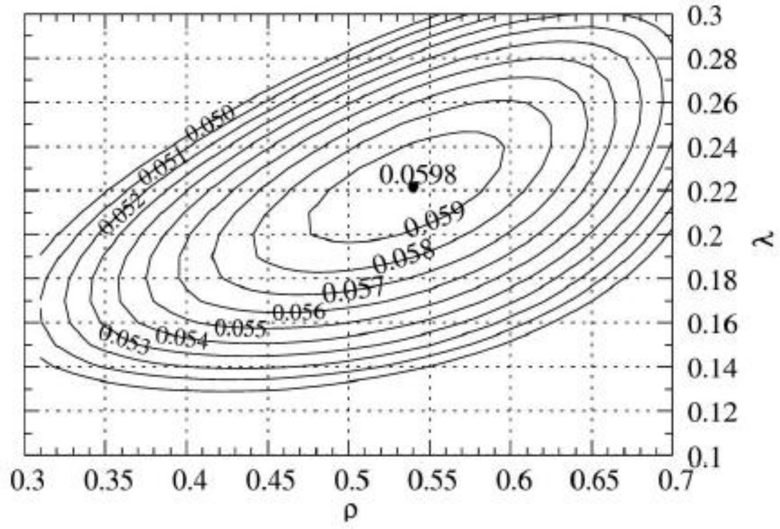


Fig. 4. $\lambda\nu$ values for a screw with three flights ($N = 3$) for the optimum filling point F

4.2 Leakage between the Flights and the Trough

The leakage flow Q_G between the gap of the flights and the trough because of the head difference $\delta h = (S/N)\sin\beta$ between the buckets (Fig. 2) is given by the following equation (Muysken 1932):

$$Q_G = \mu_A \times s_{SP} \times R_a \times (1 + s_{SP}/2R_a) \times \sqrt{(1 + (S/(2\pi R_a)^2))} \times (2\alpha_3/3 + \alpha_4 + 2\alpha_5/3) \times \sqrt{(2gh)} \quad (8)$$

Here, s_{sp} is the maximum gap between the trough and the blades and depends on the bending of the screw, which in turn depends on the screw's weight, the thickness of its materials, and its length between the bearings. This maximum gap is usually estimated by the empirical equation $s_{sp} = 0{,}0045\sqrt{(2R_a)}$ [m] for most screws.

The parameter μ_A in (8) is the contraction discharge coefficient and lies in the range 0.65–1.00, depending on the shape of the edge of the blade. It is set to 1.00 in this paper (maximum gap leakage).

The angles α_3, α_4, and α_5 shown in Fig. 5(a) were first determined graphically by Muysken (1932) and later by Rorres (2000) analytically.

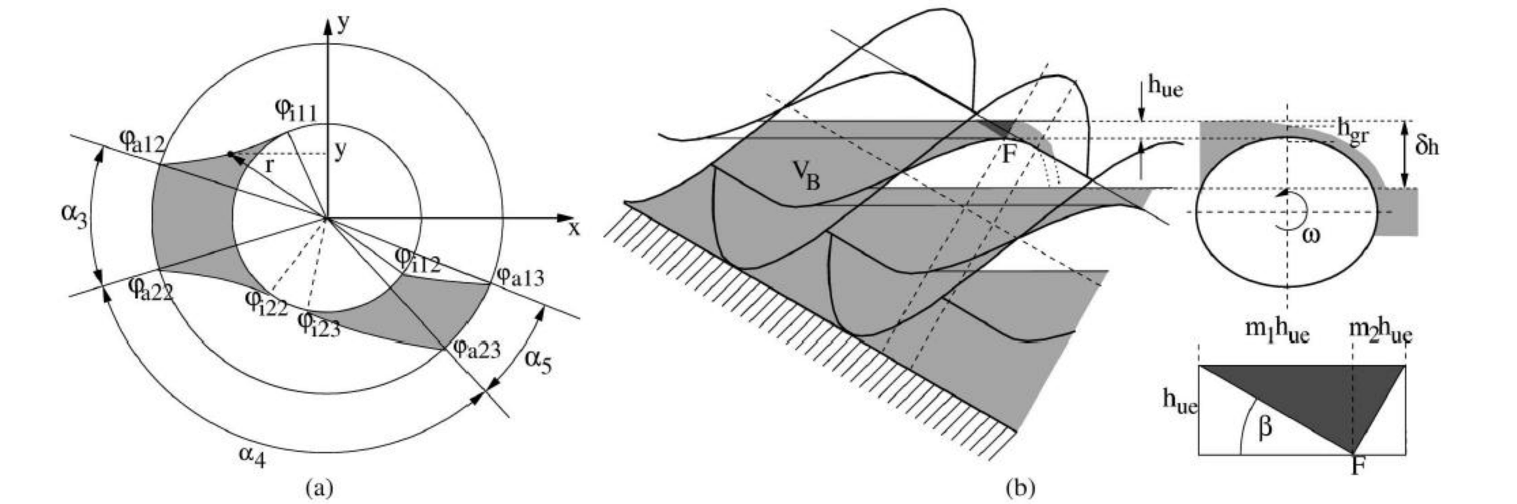


Fig. 5. (a) Definition of the angles determining the leakage current Q_G defined by the water surface in the screw; (b) triangular spillway over the central tube

The overall mechanical efficiency of a screw is determined by the leakage flow Q_G and the screw's mechanical friction losses. Now Eq. (8) shows that the leakage flow does not depend on the rotational speed n. However, the mechanical friction losses decrease as the rotational speed decreases. Consequently, the leakage flow determines the maximum overall achievable efficiency of a screw.

Eq. (8) also shows that for a screw of fixed shape, inclination, and rotational speed, the leakage flow Q_G is of the form $\beta_1 R_a^2 + \beta_2 R_a^{2/3}$ for certain positive constants β_1, and β_2. Eq. (7), on the other hand, shows that Q_W is of the form $\beta_3 R_a^3$ for a certain positive constant β_3. Consequently, the leakage fraction $Q_G = Q_W$ decreases as the outer radius of the screw R_a increases, and so the efficiency of a screw increases as it is scaled up. Calculations by the authors show that this leakage fraction is between 0.02 and 0.06 for typical screw designs in use.

4.3 Leakage from Overflow

The other leakage to be considered is the leakage Q_O across the central tube owing to overfilling of the buckets. Overfilling forms a triangular spillway [Fig. 5(b)] causing a leakage flow. According to Aigner (2008), the expression for a triangular spillway is given by

$$Q_o = 4\mu/15 \times \sqrt{2g} \times (1/\tan\beta + \tan\beta) \times h_{ue}^{5/2} \qquad (9)$$

The value of the weir-flow coefficient μ depends on the shape of the weir and the flow direction and is set to the basic value 0.537 in this paper (maximum overspill leakage). The quantity h_{ue} is the overfall head [Fig. 5(b)]. When the optimal filling point F is attained, there is no spill across the central tube ($h_{ue} = 0$), and therefore $Q_O = 0$ and the bucket is fully filled as shown in Fig. 2.

The water-powered screw has the desirable property that it can handle a flow of up to 120% optimal filling without a significant efficiency loss (Brada 1996b). But the desired design point is to have no overflow leakage, which guarantees the optimal power generation for the screw. To attain this design point, a problem not addressed until now, it is necessary to find the inflow head h_{in} at the inflow to the screw to achieve the optimal filling (Fig. 2). To this end, the following normalized total volume is introduced:

$$v_T = Q/ (\pi R_a^2 c_{ax}) = v_U + Q_G/ (\pi R_a^2 c_{ax}) + Q_O/(\pi R_a^2 c_{ax}) \qquad (10)$$

where c_{ax} the axial-transport speed, is given by

$$c_{ax} = S \times n/60 \text{ [m/s]} \qquad (11)$$

This new dimensionless volume that includes the effect of both leakage currents is an extension of the dimensionless volume v_U given by Rorres (2000). At the optimal filling point the spillover flow Q_O equals zero and Eq. (10) becomes simply

$$v_T = v_U + Q_G/ (\pi R_a^2 c_{ax}) \qquad (12)$$

4.4 Model of the Inflow Head

Fig. 6(a) shows a profile of the fluid level as the water enters from a channel with a flow Q and head h_1, which is the sum of the inflow head h_{in} and the sill height w.

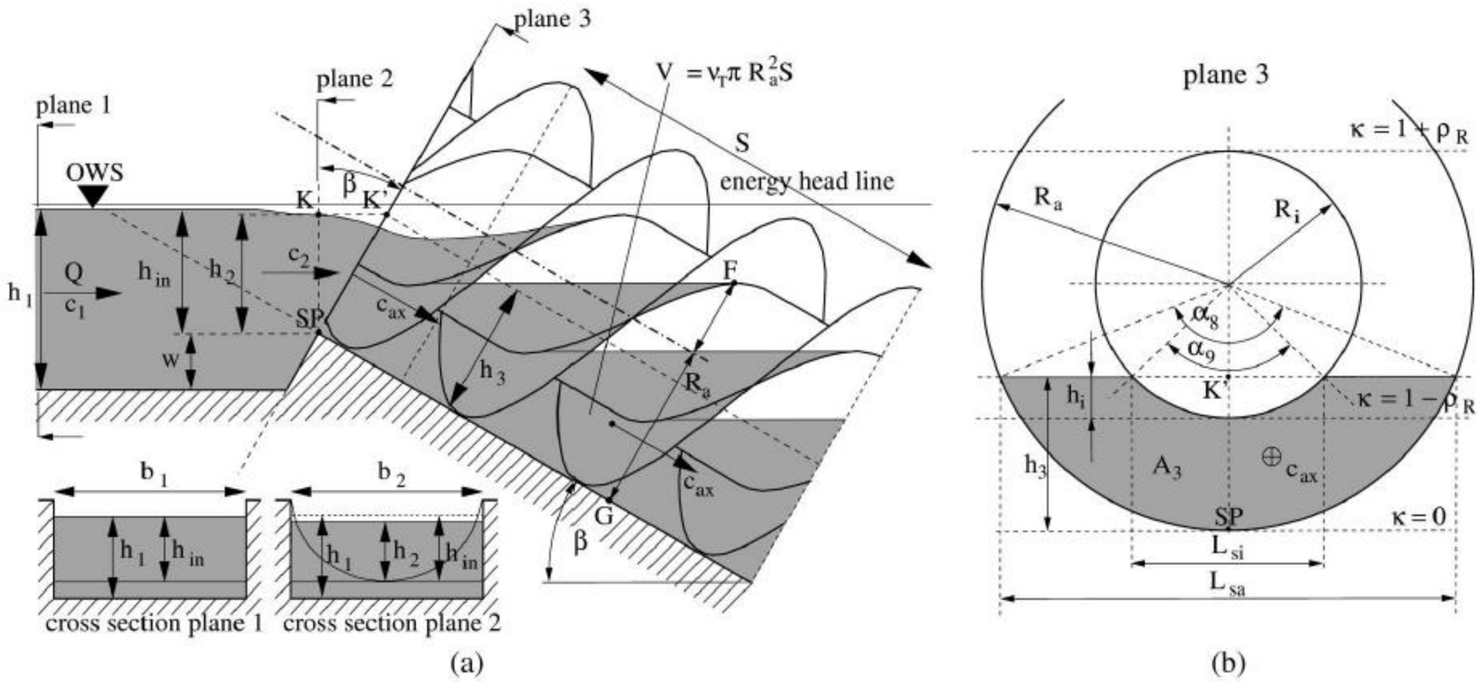


Fig. 6. (a) Inflow cross section of an Archimedes screw; (b) cross section of Plane 3 used to derive the analytical model

The water enters the screw through Plane 2 perpendicular to the water flow with head h_2, which differs slightly from the inflow head h_{in}. To determine the value of h_{in} for a particular flow, first an analytical model for the water depth h_2 is derived. It should be noted that the water level forms a drop-down profile after crossing Plane 2 because the speed of the flow increases in this plane owing to the downward motion of the buckets of water.

Unfortunately, a closed solution for the drop-down profile does not exist since each flight crossing the flow is modifying the drop-down profile. Instead, the assumption is made that under steady-state conditions the complicated flow of the water within the rotating screw can be approximated by a uniform flow of constant height h_3 with the axial speed c_{ax} from Eq. (11). If A_3 is the average cross-sectional area of the water in the screw [Fig. 6(b)], then

$$v_T = A_3/ (\pi R_a^2) \qquad (13)$$

Next, the dependence on v_T on the average height h_3, or equivalently on the dimensionless height $\kappa = h_3/R_a$, is derived. Three cases arise as to how the water level in Fig. 6(b) interacts with the central tube. The first case is for a partially filled circular trough when the water level is below the central tube:

$$v_T = \alpha_8/2\pi - (1-\kappa)/\pi \times \sqrt{1-(1-\kappa)^2} \qquad \text{if } 0 \le \kappa \le (1-\rho) \qquad (14)$$

The second case is when the water level cuts the central tube:

$$v_T = (\alpha_8 - \alpha_9\rho^2)/2\pi - (1-\kappa)/\pi \times [\sqrt{1-(1-\kappa)^2} - \sqrt{\rho-(\rho-\kappa)^2}]$$

$$\text{if } (1+\rho) \le \kappa \le (1-\rho) \qquad (15)$$

The third case is when the water level is above the central tube:

$$v_T = \alpha_8/2\pi - \rho^2 - (1-\kappa)/\pi \times \sqrt{1-(1-\kappa)^2} \qquad \text{if } (1+\rho) \le \kappa \le 2 \qquad (16)$$

(The third case will not arise under the assumption that there is no overflow leakage.) The angles in Eqs. (14)–(16) are given by

$$\alpha_8 = 2\arccos(1-\kappa) \qquad \alpha_9 = 2\arccos((1-\kappa)/\rho) \qquad (17)$$

Numerical methods can be used to determine κ from Eqs. (14)–(16) for a fixed value of v_T . It is interesting to note that Eqs. (14)–(16) depend only on the radius ratio ρ and not on the pitch ratio λ of the screw.

Fig. 7 shows the calculation results for different radius ratios ϱ. Typical values for ϱ used by the manufacturers are $\varrho = 0.3$ and $\varrho = 0.5$. For $\varrho = 0$ the result can be found in standard textbooks on hydraulics, e.g., Bollrich and Preissler (1992).

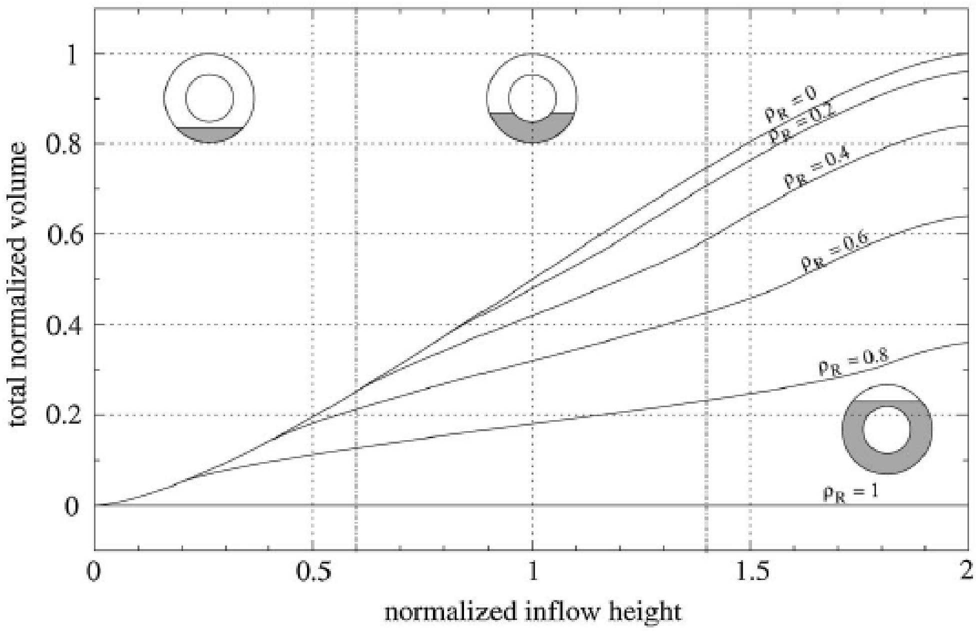


Fig. 7. Calculation results for ν_T *vs*. κ for different ϱ values; three different cross sections shown are for $\varrho = 0.4$

Next, it is assumed that the head h_2 is the projection of the height h3 to the vertical direction; i.e., $h_2 = h_3 \cos \beta$ [Fig. 6(a)].Then the water depth h_{in} can be determined by applying Bernoulli's equation to Plane 1 and Plane 2 [Fig. 6(a)]

$$h_1 + C_1^2/2g = w + h_2 + c_2^2/2g \times (1 + \zeta) \qquad (18)$$

where ζ = hydraulic loss factor; and c_1 and c_2 = flow velocities in Plane 1 and Plane 2, to be determined in the following.

The flow through Plane 1 is now given by $Q = c_1b_1h_1$, and the identical flow through Plane 2 is given by $Q = c_2b_2h_2$, where b_1 and b_2 are the widths of the channels as shown in Fig. 6. For $b_1 = b_2$ the inflow height of a flow Q then becomes,

$$h_{in} = h_1 - w = h_2 + 1/2g \times (Q/(h_2b_2))^2 \times [1 + \zeta - (h_2/h_1)^2] \qquad (19)$$

The only unknown left is the hydraulic loss factor ζ in Eq. (19). It can be estimated by applying Borda-Carnot head loss when the water enters from the rectangular channel (Plane 2) to the tilted circular entrance of the screw (Plane 3). This can be written as,

$$\zeta = (A_3/A_2 - 1)^2 = ((\nu_T\pi R_a)/(\kappa\cos\beta b_2) - 1)^2 \qquad (20)$$

where the facts that $A_2 = \kappa R_a b_2 \cos\beta$ and $A_3 = \nu_T\pi R_a^2$ were used.

4.5 Algorithm for Computing the Inflow Head

From the equations derived so far, the problem of determining the inflow head h_{in} of the incoming water so that the buckets of the screw are filled up to their optimal filling point F (Fig. 2) can be solved. A specific algorithm to accomplish this is the following:

Given: The screw's geometric parameters N, β, Ra, R_i, S, w (Fig. 2) and its rotational speed n in revolutions per minute [Eq. (2)].

Assumptions: The water enters the screw at a height $h_1 = R_a$ [Fig. 6(a)] from a rectangular channel whose width is $b_1 = 2R_a$.

The algorithm is as follows:

1. Determine the radius ratio ρ and the pitch ratio λ from Eqs. (3) and (4). Then, using the algorithm described in Rorres (2000), determine the volume-per-turn ratio $\lambda\nu_U$ for the specified number of blades N. (For N = 3, Fig. 4 can be used to determine $\lambda\nu_U$.)

2. Determine the torque-generating flow Q_W (m^3/s) from Eq. (7).

3. Determine the angles α_3, α_4, and α_5 in Fig. 5 from the algorithm described in Rorres (2000), and then determine the gap leakage flow Q_G from Eq. (8). In this equation the contraction discharge coefficient μ_A is set equal to 1 and the gap width s_{sp} is set equal to 0.0045 $\sqrt{(2Ra)}$ (m).

4. Set the total flow Q equal to $Q_W + Q_G$ (m^3/s), set the axial speed c_{ax} equal to Sn/60 (m/s), and determine the normalized total volume ν_T from Eq. (12).

5. Determine κ from Eqs. (14) and (15), then h2 from h2 = $\kappa R_a \cos\beta$, ζ from Eq. (20), and h_{in} from Eq. (19).

The five steps in the preceding algorithm to determine the

inflow height hin give the following numerical values:

1. ϱ =0.505, λ = 0.1838, and $\lambda\nu_U$ = 0.058.

2. Q_W = 0.2535 m^3/s.

3. α_3 = 0.478 rad, α_4 = 2.338 rad, α_5 = 0.358 rad, μ_A = 1, s_{sp} = 0.0046 m, and Q_G = 0.0137 m^3/s.

4. Q = $Q_W + Q_G$ = 0.2672 m^3/s, c_{ax} = 0.9275 m/s, and ν_T = 0.3327.

5. κ = 0.874, h_2 = 0.397 m, h_3 =0.459 m, ζ = 0.119, and h_{in} = 0.409 m.

Notice that if the total flow Q of the screw is given, steps 4 and 5 of the algorithm allow the determination of the inflow head h_{in}, and this is true whether the screw is optimally filled or not and whether there is any gap or overflow leakage. It also does not depend on how many blades the screw has or what its pitch is. These facts are used in the next section to test the theory with some experimental results.

The following symbols are used in this paper:

A_1 = area of the approach channel (m^2);
A_2 = area of the inflow section (m^2);
A_3 = area of the average flow (m^2);
c_{ax} = axial transport speed in the screw (m/s);
c_0 = flow speed before the debris rake (m/s);
c_1 = flow speed at the approach channel (m/s);
c_2 = flow speed in cross section $K - SP$ (m/s);
c_3 = flow speed at outflow of the screw (m/s);
c_4 = flow speed at tail race (m/s);
g = gravitational constant (m/s^2);
H = head difference (m);
h_{in} = inflow water depth (m);
h_{ue} = overfall head (m);
h_1 = water depth at the inflow channel (m);
h_2 = water depth in cross section $K - SP$ (m);
h_3 = water depth at the tail water (m);
L = length of the central tube (m);
L_B = length of the bladed screw (m);
N = dimensionless number of blades (1);
n = rotational speed (min^{-1});
Q = overall flow (m^3/s);
Q_F = friction-leakage flow (m^3/s);
Q_G = leakage flow through the gap (m^3/s);
Q_O = leakage flow over the central tube (m^3/s);
Q_P = leakage flow without guiding plate (m^3/s);
Q_W = torque-generating flow (m^3/s);
R_a = outer radius of the screw (m);
R_i = outer radius of the screw (m);
S = pitch of the screw (m);
SP = geodetic height above sea level (m);
s_{sp} = gap between the trough and the blades (m);
TP = geodetic height above sea level (m);
V_B = volume of a bucket (m^3);
V_U = volume of N buckets (m^3);
w = sill height (m);
α_3 = leakage angle (rad);
α_4 = leakage angle (rad);
α_5 = leakage angle (rad);
β = inclination of the screw (rad);
δh = head difference between two adjacent buckets (m);
ζ = dimensionless loss factor (1);
κ = dimensionless normalized value of the inflow height (1);
λ = dimensionless normalized pitch (1);
$\lambda\nu_U$ = dimensionless total normalized transported volume per turn (1);
μ_A = dimensionless discharge coefficient (1);
μ_W = dimensionless basic weir discharge coefficient (1);
ν_U = dimensionless normalized volume per turn (1);
ν_T = dimensionless total normalized volume (1); and
ρ = dimensionless radius ratio (1).

5. Gap Flow in Archimedean Screws [4]

Hundreds of Archimedes screw generators (ASGs) have been installed in Europe since the first practical installation by Brada two decades ago. The Archimedes screw generator is particularly advantageous for very low head sites (less than about 6 m) at existing dams and weirs, where they have been found to be an efficient, fish-friendly, aesthetically pleasing solution method of producing zero-emission renewable electricity. One of the most significant sources of losses in practical Archimedes screws is leakage of water between the edges of the rotating screw surfaces and the fixed trough within which the screw rotates. Typically gap flow leakage is modelled using an empirical equation for Archimedes screw pumps, or a model based on quasi-static flow through a gap. It will be theoretically shown that this results in over-prediction of gap flows at high screw rotation speeds in Archimedes screw generators. However, gap leakage should increase with speed in Archimedes screw pumps.

An Archimedes screw (Fig. 1) consists of an inner cylindrical shaft, around which one or more helical surfaces (flights) are wrapped orthogonal to the cylinder surface. The resulting geometry is very much like a conventional screw. The screw sits in (or in some cases has fixed to it) a cylindrical trough. This trough may be a tube that encircles the screw, or it may only extend around the lower half of the screw. When used as a pump, an Archimedes screw is rotated, which traps water between two consecutive flights. One of these bodies of water is called a 'bucket.' It is translated along the length of the screw, from the low end to the high end, as the screw is turned. ASGs operate in reverse: water flows into the top of the turning screw, forming buckets of water that translate down the length of the screw. The hydrostatic pressure that the water exerts on the screw surfaces causes it to turn, lowering the buckets in the process. This rotation can be used to generate electricity by driving a generator from the screw shaft.

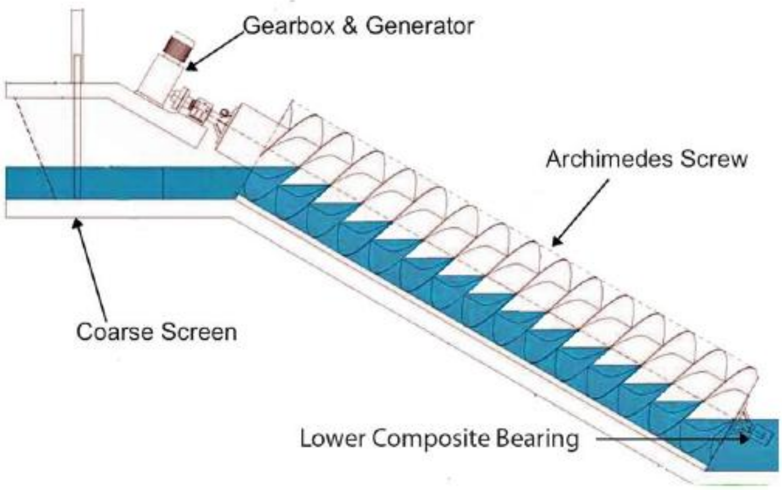


Figure 1. Typical Archimedes screw generator.

Compared to other generation technologies, ASGs are most advantageous at low-head sites (less than about 5 m). They have the potential for maintaining high efficiencies even as head approaches zero, in contrast to most impulse or reaction turbines where efficiency generally drops rapidly once the head recedes below a minimum practical level.

When designing an Archimedes screw for a specific site, it is necessary to optimize over a range of variables, including the screw inner diameter D_i and outer diameter D_o, screw pitch P, number of flights N, overall length L and slope β (Fig. 2).

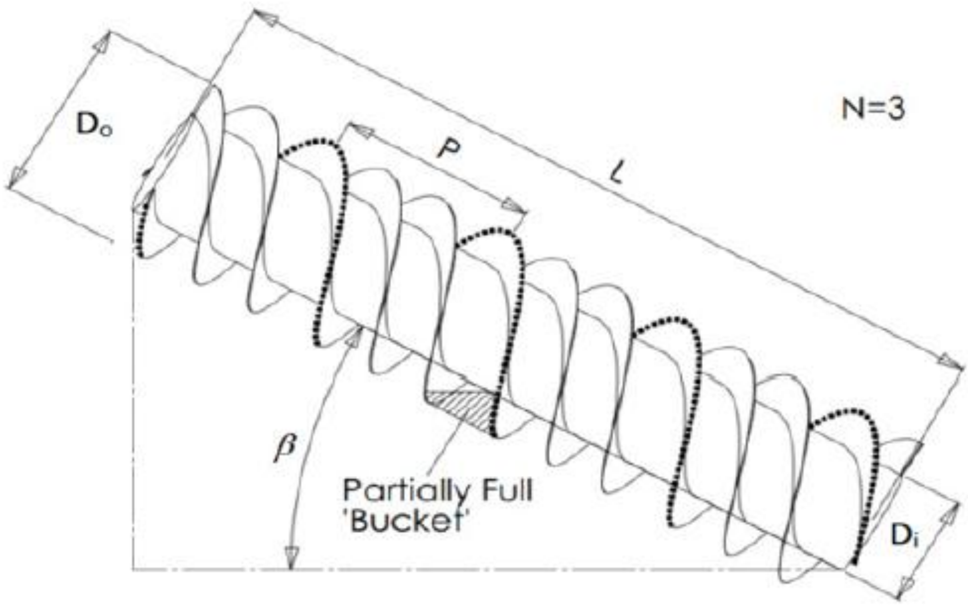


Figure 2. Archimedes screw design variables.

The designer must also select the operating speed of the screw. Optimization is complicated because each of the geometric variables impacts the costs of the final screw.

ASGs have low impacts on fish, which generally pass through an Archimedes screw unharmed. Conversely, fish and small debris can pass through an operating ASG without causing damage to the screw. Kibel investigated the impact of ASGs on fish and found that almost all fish, including eels, trout and salmonoids, passed through the ASG unharmed, and that intake screening was not necessary

5.1 Efficiency

The available power P_{avail} in a water stream is

P_{avail} = ρ x g x h (1)

where ρ is water density, g is the gravitational constant (9.81m/s^2), Q is the volume flow rate of water and h is the available head (or "drop").

This available power can be converted to electricity with very high efficiencies in hydroelectric generation. Gigawatt scale hydroelectric stations can achieve operational efficiencies exceeding 0.9, including all energy conversion losses including hydrodynamic, mechanical and electrical between potential energy in a reservoir and electrical energy delivered to the grid

The mean operational efficiency of the surveyed ASGs was 69%, with maximum efficiencies over 75%. These are very reasonable efficiencies for small scale hydro generation. The surveyed ASGs ranged in capacity from 1 kW to 140 kW, and operated at heads between 1 m and 6 m.

Theoretically, mechanical efficiencies approaching 1.0 should be possible in an Archimedes screw, because the energy conversion involves converting one form of ordered energy to another. However, in a practical system there are several forms of energy loss which reduce mechanical efficiency. The most significant energy loss mechanisms are:

1. Gap leakage: water flowing through the gap between screw and trough does not contribution to power generation

2. Bearing drag: friction in the bearings that support the screw

3. Inlet and outlet losses: energy is lost through water flowing into and out of the screw under non-ideal conditions

4. Viscous drag: energy is lost through shear forces between water and interior screw surfaces.

Losses due to bearing drag, and inlet and outlet losses can be minimized through careful design of the screw and its inlet and outlet. Archimedes screw generators operate most efficiently at relatively low rotation speeds.

The recommended rotational speed of an Archimedes screw pump in revolutions per minute should be,

RPM = 50/ $\sqrt[3]{Do^2}$ (2)

where, Do = the outer diameter of the screw. It is confirmed that most installed European ASGs operate at this speed or less. At higher speeds it is known from experience that friction and centrifugal forces become unacceptably large in many cases. Slower speeds would also be expected to reduce noise and prolong bearing life in an ASG.

Low speed operation keeps bearing friction low. There is also little induced motion in the water within the screw, and in practice Archimedes screws have been analyzed assuming quasi-static conditions in which the water with the screw does not move relative to the screw surfaces. At sites with consistent water supply, the inlet and outlet losses can be minimized by careful design.

However, gap leakage is not minimized by reducing speed, and can therefore still have a significant impact on the mechanical efficiency of the screw.

5.2 Gap Leakage

The volume flow rate of water through the screw Q can be divided into two components,

$Q = Q_b + Q_l$ (3)

where, Q_b is the volume flow rate of water within the buckets of the screw that actively drives the screw. The distribution of quasi-static pressure on the interior surfaces of the screw due to this water results in a net torque that turns the screw. However, a part of the flow Q_l effectively leaks past the screw surfaces. The available energy associated with this water is lost as the water bypasses the screw through the gaps, instead of contributing to turning the screw.

There are several leakage flows, including gap leakage, overflow leakage, and leakage due to water becoming entrained on the screw flights. The latter mechanism is very small in practice, and overflow leakage is zero unless the screw is operating in an over-full state (which is less efficient than operating full). Therefore, all leakage flow will be assumed to be in the form of gap leakage.

If the other loss mechanisms are minimized, the theoretical efficiency n of an Archimedes screw is,

$n = 1 - Q_l / Q$ (4)

Figure 3. A laboratory-scale Archimedes screw. Gap leakage is visible across each flight where the lower bucket water level is below the leakage point, however gap leakage is occurring around the entire wetted perimeter of the flights.

where Q_l is the gap leakage flow rate (or the volume flow rate of water through the gap between the screw and the trough). Leakage can vary from 3 % to 12 % in Archimedes screw pumps and experience suggests leakage occurs at similar rates in ASGs.

5.3 Existing Gap Leakage Models

The most commonly used gap leakage flow model is the empirical formula for Archimedes screw pumps reported by Nagel (1968), which gives the leakage volume flow rate Q_l through the screw, in m^3/s,

$Q_{lN} = 2{,}5x\ G_w \times D_o^{1,5}$ (5)

G_w is the width of the gap between the flight edge and trough in meters, and D_o is the screw diameter in meters. (Note that this equation is not dimensionally consistent.)

Neurnbergk and Rorres utilized a more sophisticated relationship attributed to Muysken to predict gap flow leakage that includes some additional parameters:

$Q_{lR} = C_R \times C_W \times D_o/2 \times (1 + G_W/D_o) \times \sqrt{(1 + (P/(\pi \times D_o))^2} \times (2\alpha_3/3 + \alpha_4 + 2\alpha_5/3) \times \sqrt{(2g\delta h)}$
(6)

C_R is a discharge coefficient (set equal to 1.0), δh is the difference in water level height between adjacent buckets, and α_3, α_4 and α_5 are wetted angles around the gap (in radians). While potentially more accurate than Eq. 5, knowledge of the angular position of intersections between the water planes and the edge of screw (α) is needed, which must be determined from tables or calculated beforehand.

Both models require the gap width, which is not easily measured in practice, since it will usually vary at different locations along the screw. It is often estimated using Nagel's empirical relation,

$GW = 0{,}0045\sqrt{D_o}$ [m] (7)

Neurnbergk and Rorres' model (Eq. 6) requires knowledge of angular water level positions within the screw, which are not easily measured in an operating screw.

Nagel's relations were originally derived for screw pumps under the assumption that the screw would always be operating full. (A screw is full when the water level within the buckets cannot be increased any further without water spilling over the top of the inner cylinder into the next lowest bucket.) This means it cannot be used directly to predict the leakage flow at other fill points

5.4 First Principles Gap Leakage Model

A gap leakage flow model was needed specifically for use in a new Archimedes screw performance model. Since Nagel's model was insufficient, and Neurnbergk and Rorres' model was cast in inconvenient variables, it was decided to derive a gap flow model from first principles.

The model will consider the wetted length of a single rotation of a flight. Note that for an Archimedes screw, the total gap flow through the screw is equal to the gap flow through the gap between a single rotation of a flight and the trough. The gaps are arranged in series along the length of the screw, with similar conditions across each gap, so the flow through a single gap equals the flow out of the screw by gap flow.

The screw will be assumed to be static (as is often assumed when modelling the performance of an Archimedes screw). Water levels in a bucket will remain constant because the water lost to the next lower bucket through gap flow is replaced by gap flow entering the bucket from the bucket above.

The gap can be divided into two flow regions. The first region is the "constant pressure" portion of the gap where both sides of the gap are immersed. The pressure head across all parts of the gap in this region corresponds to δh. At any point in this region, the ideal velocity v through the gap based on Bernoulli's equation is

$$v = \sqrt{2g\delta h} \qquad (8)$$

Multiplying by gap area gives an estimated volume flow rate for this region of

$$Q_1 = G_w \times l_W \times \sqrt{2g\delta h} \qquad (9)$$

where l_w is the length of the constant pressure portion of the gap (Fig. 4), and is determined numerically based on the screw geometry.

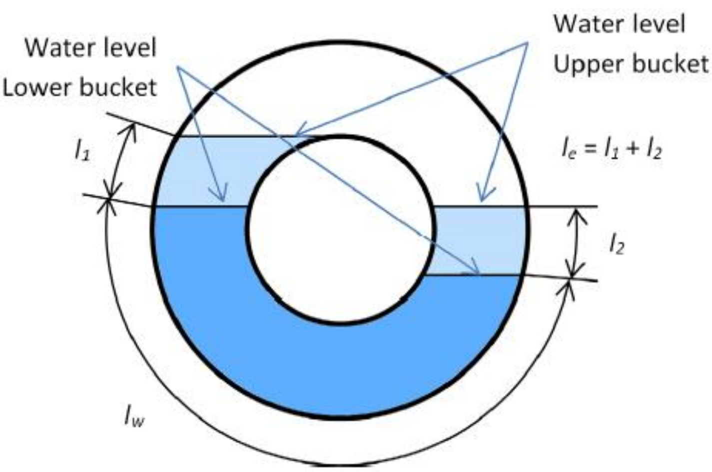


Figure 4. Gap length definitions.

In the second region of the gap, the upstream side is submerged, but the downstream side is above the water level of the lower bucket, and therefore exposed to constant atmospheric pressure. Note that this region consists of two separate parts of the gap, one on each side of the screw, with the "constant pressure" region in between (Fig. 4).

The pressure head will vary along this gap from 0 to *δh.* This variation will be assumed to be linear in order to keep the number of variables in the final formulation to a minimum.

This will introduce a small amount of error into the model, however, since δ*h* is always much smaller than D_o, the degree of this non-linearity is limited. If the total length of this gap region is l_e (Fig. 4), the volume flow rate through this region of the gap Q_2 is found by integrating the local pressure-driven velocity over the gap area:

$$Q_2 = \sqrt{2g} \times \int_0^{\delta h} z^{0,5} \times G_w \times l_e / \delta h \times dz \qquad (10)$$

which simplifies to,

$$Q_2 = G_w \times l_e / 1,5 \times \sqrt{2gh} \qquad (11)$$

Combining Eqns. 9 and 11, noting that δ*h* = (*P*/*N*)sinβ and adding a discharge coefficient *C* to account for minor losses gives a predicted total gap flow of,

$$Q_{IL} = C \times G_W \times (l_w + l_e/1,5) \times \sqrt{(2gP/N \times \sin\beta)} \qquad (12)$$

The value of *C* is one if there are no minor losses in the gap, otherwise it will be less than one. A value of *C* = 0.9 was determined based on measurements of static leakage flow from a small laboratory-scale Archimedes screw. It should be noted that this value of *C* may not be valid for different screws, since minor losses will be strongly dependent on the Reynolds number, smoothness of the trough, and the shape, sharpness and consistency of the flight edges.

5.5 Discussion

Figure 5 shows the predicted gap leakage flow, as a fraction of the total volume flow rate, in a typical Archimedes screw ($P = D_o$, $D_i = D_o/2$, N = 3, $L = 4D_o$) operating in a full state by both models. Gap width is assumed to follow Eqn. 7 and rotation speed is the maximum from Eqn. 2. It is apparent that the two models predict very similar flow rates across a wide range of screw diameters. However, the limitations of Eqn. 5 become apparent when slope is varied. Eqn. 12 accounts for the change in pressure head across the gap as slope varies, while the original Nagel model (Eqn. 5) does not. The Nagel model also cannot account for changes in pitch, inner diameter or fill level within the screw due to its inherent simplicity.

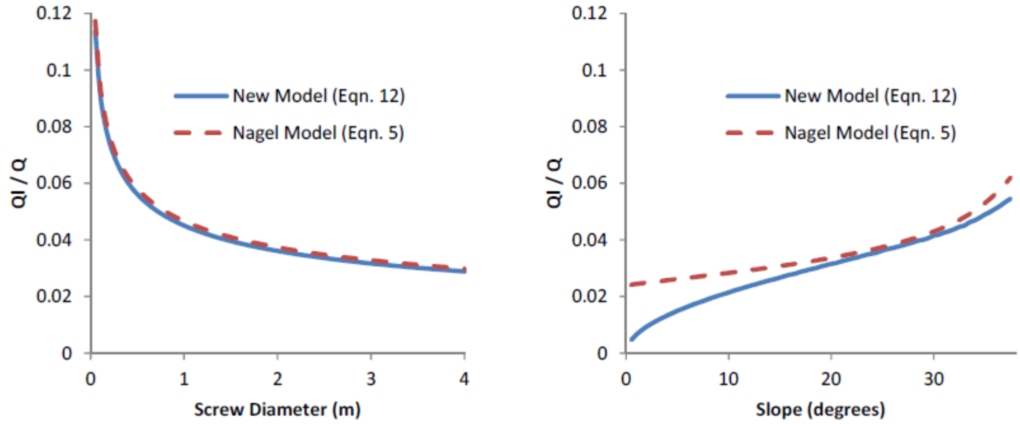


Figure 5. Ratio of leakage flow rate to total flow rate for typical Archimedes screw generator, as a function of screw diameter (left) and slope (right).

Rotation Speed

Neither model attempts to account for the rotation speed of the screw. The previous discussion of theoretical efficiency suggests that efficiency of an Archimedes screw should increase as rotation speed increases. However, in practice this is not the case. The data shows the optimum speed of this ASG for a given Q is relatively low. Operating at higher speeds significantly reduced power output, even though P_{avail} remained constant (at about 2.75 W).

Several mechanisms may be responsible for producing these greater losses at high speeds, including gap leakage, bearing drag, and fluid drag within the screw. Determining the relative importance of each is complicated because none can be directly measured when the screw is operating. Fluid drag within the screw is particularly difficult to quantify.

Decreasing gap flow with increasing rotation speed is only expected in Archimedes screw generators. Archimedes screw pumps would be expected to exhibit increasing gap flow with increasing rotation speed, because viscous drag between the fluid and the trough will be acting in the direction opposite to the water motion, instead of in the same direction.

Enclosed Screws

One suggested approach for minimizing gap leakage losses in an Archimedes screw is to enclose the screw within a tube, leave no gaps between the screw and tube, and rotate the entire screw-and-tube assembly. However, the top of the enclosed screw must be free to turn, so a small gap must be left between the dam or weir that the screw is placed in, and the rotating enclosed screw. This gap will be of similar length to the gap in a conventional screw. One side of this gap will be at atmospheric pressure, so the pressure difference across the gap will be equivalent to local depth at all points along the gap. In a fixed trough screw, the maximum pressure head across the gap is δh, while in an enclosed screw much of the gap will be at greater pressure head than δh, because the geometry of a practical Archimedes screw means δh is a fraction of $D_o/2$. This reasoning suggests the enclosed screw would be expected to have greater leakage than a conventional fixed-trough screw.

It is worth noting that the gap at the top of the enclosed screw remains in a fixed location (unlike the continually translating gaps in a fixed-trough screw), so it may be possible to achieve a narrower gap, or one with greater resistance to flow, and reduce the gap leakage. Another practical issue with enclosed screws is that the bearings must support the entire weight of the screw and the water inside it, whereas in conventional fixed-trough screws much of the water weight is supported by the trough, and bearing loads are also reduced by the buoyancy of the central cylinder.

6. Performance Model of Archimedes Screw Hydro Turbines with Variable Fill Level [10]

Virtually all modeling of Archimedes screws in the literature assumes that the screw is operating in a full condition, with the water level in each bucket as high as possible without any water flowing over the top surface of the inner cylinder into the bucket below. Nuerembergk & Rorres (2013) note that an ASG should operate most efficiently in the **full** condition up to 120% of full capacity with only small decreases in overall efficiency. However, there is little theory or data in the literature for screws operating at **partially full** conditions.

An ASG can be either fixed speed (i.e., using an induction generator phased to the power grid) or variable speed. Both types of systems have been built. Lashofer's (2012) survey of European ASG installations presents cost data for 29 ASGs (17 fixed speed and 12 regulated). The cost per watt of fixed-speed systems was on average less than the cost of the variable-speed systems. Fixed-speed systems can still generate power in a partially full condition even if the available flow is not sufficient to fill the screw at its operating speed. While operating at the full condition may be the most mechanically efficient operating state, it may not be the most economically efficient operating condition, since it will not necessarily lead to the greatest overall energy generation in cases where the available flow varies, or if there is excess flow that could be utilized to produce more power from a given screw.

6.1 Simplified Model of an Archimedes Screw

The primary driving force of an Archimedes screw is the static pressure exerted on the screw surfaces by the water within the screw. Consider the following device as a two-dimensional, non-rotating simplification of an Archimedes screw (Fig.1)

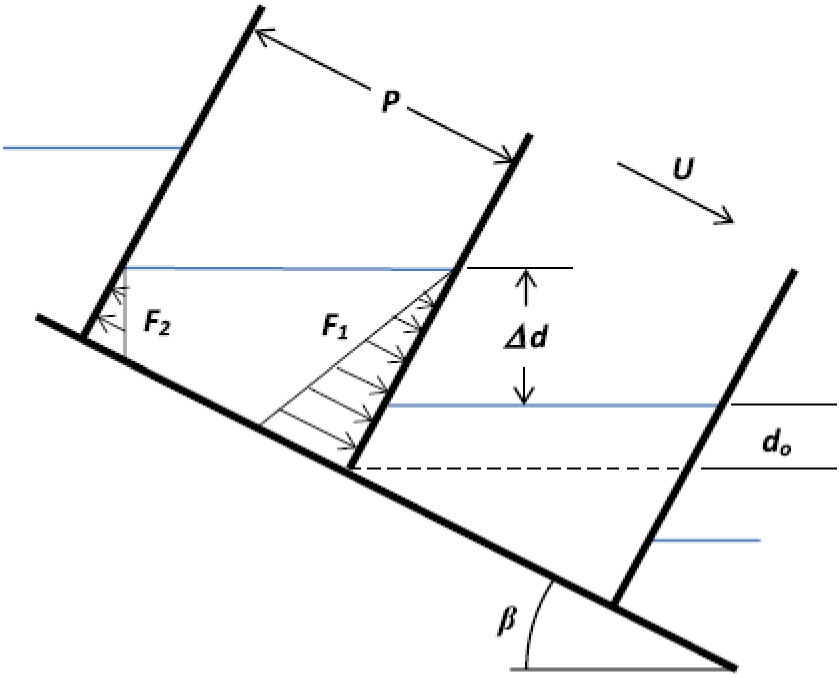


Fig. 1. Simplified model

The base of the device is a straight, fixed surface tilted at angle β to the horizontal. A series of straight vanes are oriented normal to the base surface (so that the vanes are all tilted at angle β from vertical), each separated by a distance P from the vanes ahead and behind. All the vanes translate with constant velocity U along the base surface. There is no gap between the vanes and the base and it will be assumed there is no friction between surfaces. The space between each two adjacent vanes is filled with a volume of water V_b. The volume flow rate of water past any point along the device is then, $Q = (V_b \times U) / P$.

The volume in the bucket can also be expressed as,

$V_b = P/\cos\beta \times (\Delta_d/2 + do)$ (1)

where $\Delta d = P \times \sin\beta$ = height difference between the water surfaces in the buckets on either side a given vane, and d_o = height difference between the water surface and the base of the upstream side vane.

The maximum hydraulic power available in the flow through the device is,

$P_{avail} = \rho \times g \times ((V_b \times U)/P) \times \Delta d$ (2)

where ρ = density of water (typically 1.000 kg/m^3) and g = gravitational constant (9,81 m/s^2).

Next, consider the component of forces acting on the wetted surfaces in the direction of motion that will cause work to be done. The base surface can be ignored because it is unmoving and aligned with the direction of motion. The force per unit width on surfaces 1 and 2 due to hydrostatic pressure can be shown to be,

$F_1 = (\rho \times g)/2\cos\beta \times (d_o + \Delta d)^2$ (3)

$F_2 = (\rho \times g)/2\cos\beta \times d_o^2$ (4)

Note that F_1 acts in the direction of motion and F_2 acts in the opposite direction. The net power is the product of the net force and U

$P_{net} = (\rho \times g)/2\cos\beta \times [(d_o + \Delta d)^2 - d_o^2] \times U$ (5)

The theoretical efficiency of the device is $P_{net} = P_{avail}$. It can be shown algebraically that Eqs. (2) and (5) will give the same values for any combination of input variables. This is also true for cases of very low fill level, where $d_o = 0$ and the water level is less than Δd. This means that the theoretical efficiency of this idealized device is 1.0 for all water depths, slopes, and velocities. This exercise suggests that it is possible for the mechanical efficiency of an ASG to approach 1.0, and that the speed of the screw should not impact efficiency.

Of course, real ASGs are not ideal and the phenomena that have not been included in the model are those that will lead to reduced efficiency. This model assumes an infinitely long device and excludes any end effects. These results suggest [and Nuernbergk & Rorres (2013) show in detail] that maximum efficiency can only be obtained for an Archimedes screw if the inflow conditions are set to minimize energy losses.

The efficiency of a real screw would be expected to decrease with rotation speed because real screws experience bearing and friction losses that increase with speed. Nuernbergk & Rorres (2013) report that Lashofer (2011) confirmed that most installed European ASGs operate at rotational speeds within the operating speed range first suggested by Nagel (1968)

$\omega \leq 5\pi / (3\ (2R_o)^{2/3})$ (6)

where ω = rotation speed in rad/s and Ro = screw radius in meters. At higher speeds, it is known from experience that friction and centrifugal forces become unacceptably large in many cases (Nuernbergk & Rorres 2013). Slower speeds would also be expected to reduce noise and prolong bearing life in an ASG.

Real Archimedes screws must also have a gap between the moving screw flights and the fixed trough. While this is kept as small as possible, some gap must remain to allow for sagging or bending of the screw, thermal expansion, and manufacturing tolerances. This leakage flow will lead to reduced efficiency because the portion of fluid leaking past the screw does not contribute to pressure on the surface. Smaller screws would be expected to have greater efficiency losses due to leakage than larger screws due to scaling effects since gap width as a fraction of screw diameter will decrease as screw size increases.

6.2 Model of a Rotating Screw; Introduction and Variables

The simplified model developed here is too simplified to give guidance to designers attempting to optimize ASGs for specific conditions. The following model was developed to extend the concept of a static pressure driven energy converter from a simplified geometry to one that actually represents an operating Archimedes screw. This model therefore includes rotation speed ω and torque T as parameters.

The screw geometry is defined by the pitch P, number of flights N, outer diameter D_o, and inner diameter D_i (Fig. 2).

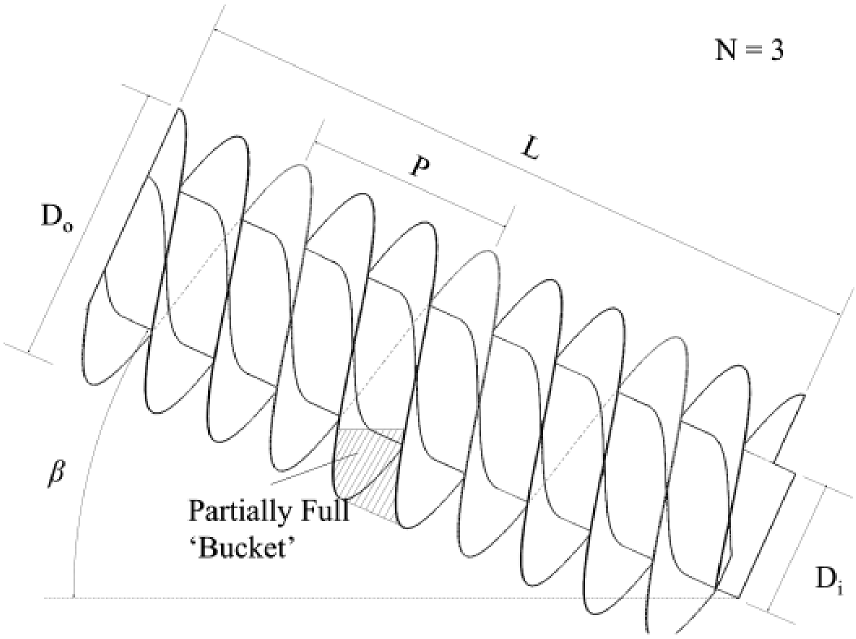


Fig. 2. Rotating screw model geometric variables

The outer and inner radius are defined as $R_o = D_o/2$ and $R_i = D_i/2$. The total length of the screw is L and it is inclined at an angle β to the horizontal (Fig. 2). As in the simplified model, the water volume and torque from a single bucket are considered first. These results are then used with the length and speed of rotation of the screw to calculate total screw power.

The model is defined primarily in a cylindrical coordinate system, where w is aligned with the screw centerline, r is the distance from the centerline, and θ is the angle in the radial plane from an upward normal to the centerline (Fig. 3).

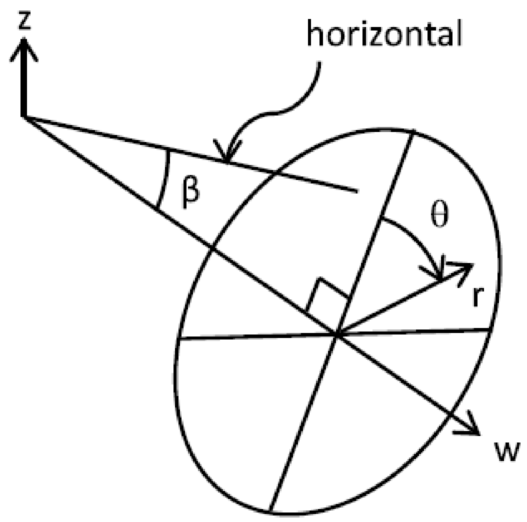


Fig. 3. Rotating screw model coordinate system

The geometry of a screw flight is described by,

$r(w) = r$ (7)

$\theta(w) = 2\pi \times (w/P)$ (8)

where the radius r ranges between the inner radius R_i and R_o. Adjacent flights are offset in w by a distance P=N. It is assumed the screw has a constant speed of rotation ω, in rad/s. A non-dimensional fill factor f is defined that describes how high the water level is within the bucket (Fig. 4).

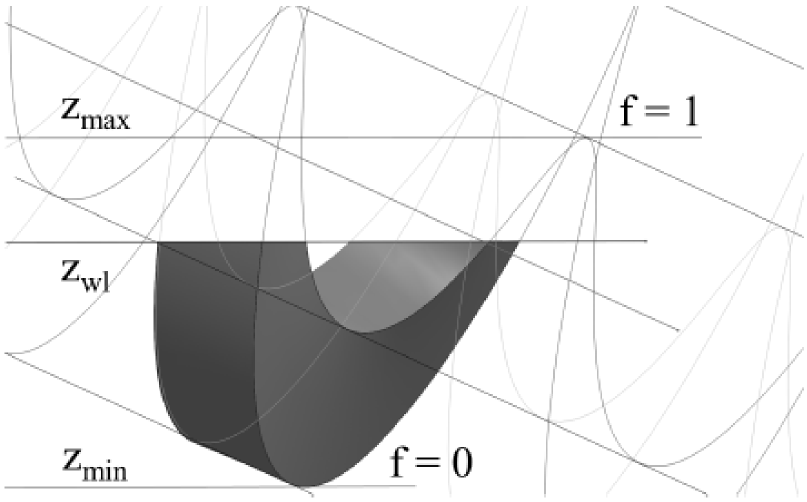


Fig. 4. Fill levels and water level variables

An f of 1 indicates a full bucket, where water is just about to spill over the top surface of the inner cylinder into the next bucket. An f of zero indicates an empty condition, where the water level coincides with the bottom edge of the downstream surface.

A single rotation of two adjacent flights, starting at θ = 0 with the flight radials at the top of the inner cylinder, and ending one rotation later at θ = 2π, will encompass a single bucket of water. Volume and torque are calculated numerically by dividing this single rotation into a series of elements defined in terms of angular position θ and radial position r.

6.3 Bucket Volume

The depth of water in the bucket is defined by the fill factor f. This must be translated into a water surface level relative to the z-axis. The maximum water depth z_{max} occurs when the bucket is completely full, which is when the water surface level coincides with the point θ = 2π, r = R_i on the downstream flight surface. The minimum possible depth z_{min} is defined as the point where the water surface level is at the point θ = π, r = R_o on the downstream flight surface (Fig. 3)

$z_{min} = -R_o \times \cos\beta - P/2 \times \sin\beta$ (9)

$z_{max} = R_i \times \cos\beta - P \times \sin\beta$ (10)

The water surface level for the specified fill factor f is then

$z_{wl} = z_{min} + f \times (z_{max} - z_{min})$ (11)

The downstream and upstream flight surfaces of the bucket are defined as surface 1 and surface 2, respectively, as in the simplified model. It can be shown that the vertical position of a point (r, θ) on surfaces 1 and 2 is given by

$z_1 = r \times \cos\theta \times \cos\beta - (P \times \theta)/2\pi \times \sin\beta$ (12)

$z_2 = r \times \cos\theta \times \cos\beta - [(P \times \theta)/2\pi - P/N] \times \sin\beta$ (13)

These water level relations are used to calculate the volume of water in the bucket by numerical integration. The space between two adjacent flight surfaces is considered for one full rotation. A volume element is defined that connects the same point (r, θ) on the downstream flight surface (surface 1) and the upstream flight surface (surface 2).

- If both points are above the water level, no part of the element is submerged and the element volume dV is zero.
- If both points are below the water level, the entire element is submerged and $dV = (P/N) \times r \times dr \times d\theta$.
- In the intermediate case where the lower point is below water but the upper point is above water, the fraction of the element that contains water is

proportional to the relative vertical distances between the points and the water level,

$$dV = \begin{cases} 0 & z_2 > z_{wl},\ z_1 > z_{wl} \\ (z_{wl} - z_1)/(z_2 - z_1) & z_2 \geq z_{wl},\ z_1 \\ P/N \times r \times dr \times d\theta & z_2 < z_{wl},\ z_1 < z_{wl} \end{cases} \qquad (14)$$

The total volume of water in the bucket is then found by numerically integrating

$$V = \int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} dV$$

6.4 Bucket Torque

The torque applied to the bucket due to static water pressure can be calculated similarly to volume. The water levels and depths of both element ends are calculated as before. The static pressure at a point is a linearly proportional to the depth of the point below the water surface. Pressure p at vertical position z is

$$p = \begin{cases} \rho \times g\ (z_{wl} - z) & z \\ 0 & z \geq z_{wl} \end{cases} \qquad (16)$$

It can be shown that for an element between surfaces 1 and 2 at a specific point (r, θ), the incremental torque dT is,

$$dT = (p_1 - p_2) \times P/2\pi \times r \times dr \times d\theta \qquad (17)$$

and the total torque is obtained by numerically integrating,

$$T = \int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} dT \qquad (18)$$

6.5 Performance

Once the volume and torque of a single bucket have been calculated, the total torque on the entire screw is the torque on a single bucket times the number of buckets along the length of the screw,

$$T_{total} = T \times (L\times N)/P \qquad (19)$$

and power P_{out} is,

$$P_{out} = T_{total} \times \omega \qquad (20)$$

The volume flow rate Q through the screw is calculated based on the volume of water in a single bucket and the speed of bucket translation along the screw axis,

$Q = (N \times V \times \omega)/2\pi$ (21)

and the overall head across the screw is,

$h = L \times \sin\beta$ (22)

The available power in the flow is then,

$P_{avail} = \rho \times g \times h \times Q$ (23)

and the predicted efficiency of the screw is,

$n = P_{out}/ P_{avail} = (T_{total} \times \omega)/(\rho \times g \times h \times Q)$ (24)

6.6 Leakage

Leakage between the flights and trough is one of the largest sources of efficiency losses in ASGs. The most common model in use is the empirical formula reported by Nagel (1968), which gives the leakage volume flow rate Q_l through the screw, in m^3/s,

$Q_l = 2{,}5 \times G_w \times D_o^{1{,}5}$ (25)

where G_w = width of the gap between the flight edge and trough in meters and D_o = screw diameter in meters.

The gap width is not easily measured in practice since it will usually vary at different locations along the screw. Nagel (1968)reported a maximum gap width formula that is often used to estimate gap width,

$G_w = 0{,}0045\sqrt{D_o}$ (26)

with units of both variables in meters.

Nagel's (1968) leakage model (Eq. 25) was originally derived for Archimedes screw pumps under the assumption that the screw would always be operating full (f = 1). It cannot be used in the current model because it does not predict the leakage flow at other fill points. As the fill level changes, the wetted length of the gap will vary nonlinearly, as will the pressure head across the gap. Since leakage has a significant impact on the performance of an ASG, a new leakage model that predicts Q_l across the full range of fill points is proposed.

The volume flow rate of a fluid through a narrow gap takes the general form (Gross 1991),

$Q_l = C \times A \times \sqrt{\Delta P}$ (27)

where ΔP = pressure difference across the gap, A = area of the gap, and C = constant with units of length^{1,5} × mass$^{-0,5}$. The value of ΔP will vary at different points in the gap around the circumference of the screw. Substituting variables appropriate for the Archimedes screw, Eq. (27) becomes,

$$Q_l = C \times G_w \times l_w \times \overline{\sqrt{(P1\text{-}P2)}} \qquad (28)$$

where the expression $\overline{\sqrt{(P1\text{-}P2)}}$ = average root pressure difference around the wetted perimeter of the gap. The length of the wetted gap perimeter is l_w. These values are calculated numerically at the same time as the torque calculations are performed. A value of C = 0.04 m^{1,5} kg$^{-0,5}$ was found based on a measured leakage rate of 0.1 L/s for the stationary laboratory screw detailed below. This value is appropriate for the small screw described, but given the low Reynolds number for this gap, it may not be appropriate for larger Archimedes screws.

When an Archimedes screw is filled beyond its fill point (i.e., $f > 1.0$), overflow leakage will also occur as water flows over the top of the inner cylinder into the cylinder below. To account for this possibility, the overflow leakage model of Aigner (2008), as reported by Neurenbergk and Rorres (2013), was utilized,

$$Q_o = 4/15 \times \mu \times \sqrt{2g} \times (1/\tan\beta + \tan\beta) \times (z_{wl} - z_{max})^{5/2} \qquad (29)$$

where μ = constant equal to 0.537. Application of this equation to the model permits the water level to rise above the fill point, with the penalty of an overflow leakage flow. This implicitly assumes that once the water starts spilling over the top of the cylinder, it imparts no additional torque to the screw.

The total volume flow rate of water through the screw is,

$$Q_t = Q + Q_l + Q_o \qquad (30)$$

When calculating efficiency including leakage effects, Q_t is substituted for Q in Eqs. (23) and (24).

6.7 Laboratory Experiments

A laboratory-scale Archimedes screw was manufactured by Greenbug Energy (Delhi, ON, Canada) to allow controlled testing of an Archimedes screw. Table 1 includes the parameters of this lab screw.

Table 1. Lab Screw Nominal Physical and Simulation Parameters

Parameter	Variable	Value
Slope	β	24.9°
Outer diameter	D_o	0.146 m
Inner diameter	D_i	0.55×0.146 m
Pitch	P	0.146 m
Number of flights	N	3
Screw length	L	4×0.146 m
Head	h	0.25 m
Gap width	G_w	0.762 mm
Fill factor	F	1.00
Rotation speed	ω	10 rad/s
Number of radial elements	$(R_o - R_i)/dr$	400
Number of angular elements	$2\pi/d\theta$	360

The simulation mathematical model used first predicts the volume of water and torque for a single bucket. Fig. 5 shows predicted values for the case study screw. The relationship between volume, fill level, and slope is well-behaved but not linear.

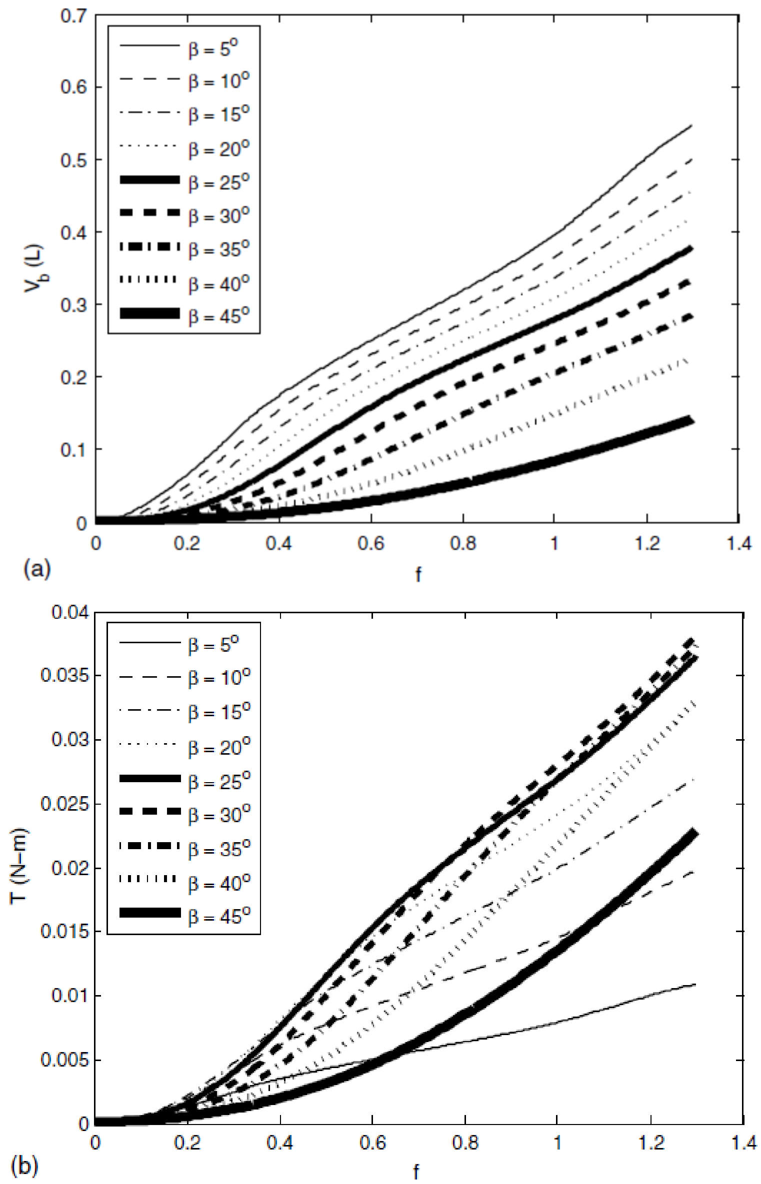


Fig. 5. Bucket volume (a) and torque; (b) as a function of slope and fill level

Figs. 6(a–e) show the predicted efficiency of a screw with the specifications in Table 1, operating across a range of fill levels, with and without gap leakage.

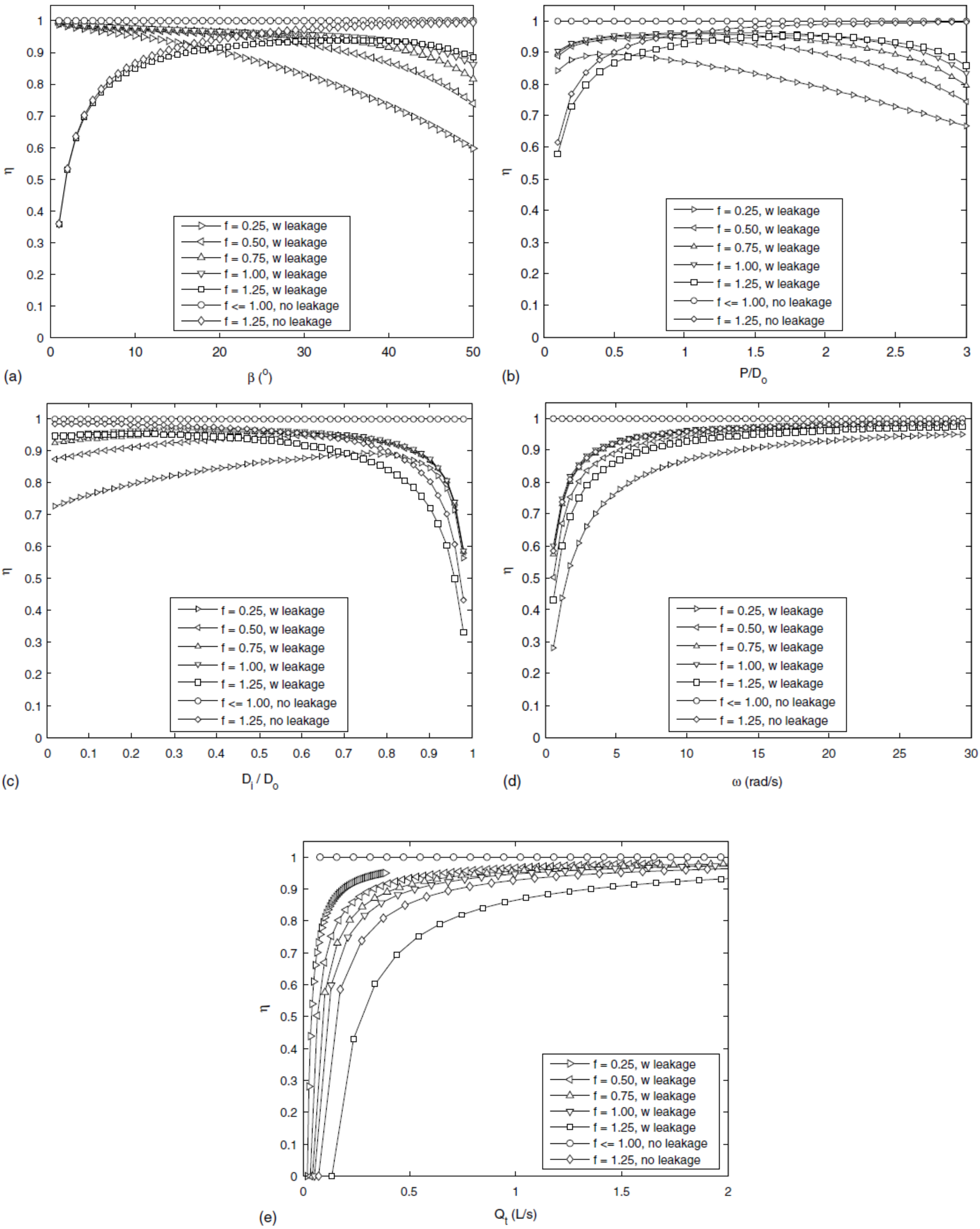


Fig. 6. Predicted efficiency as a function of (a) slope; (b) pitch ratio; (c) diameter ratio; (d) rotation speed; (e) volume flow rate through the screw at several fill levels, with and without gap leakage

Figs. 6(a and b) show predicted efficiency as a function of slope and pitch. The model predicts that with leakage present, for all fill levels, decreasing the slope will increase screw efficiency. This is consistent with the observation that leakage flow rate is driven by the pressure difference, and therefore head difference, between adjacent buckets, which decreases as slope is decreased. Practically, however, a reduction in slope will cause an increase in screw length, increasing the potential for bearing and fluid friction losses in a real screw (and also increasing the cost of the screw).

The ratio of the inner diameter to the outer diameter (D_i/D_o) also impacts efficiency. Rorres (2000) found by theoretical analysis of Archimedes screw geometry that D_i/D_o = 0.54 results in the maximum amount of water per turn of the screw. Rorres' (2000) analysis did not include allowances for leakage. On a practical note, Lashofer et al. (2012) report that almost all installed commercial ASGs have D_i/D_o close to 0.5 and P/D_o close to 1.0. Fig. 6(c) illustrates the impact of varying D_i/D_o on efficiency. The value of D_i/D_o associated with peak efficiency increases as f decreases when leakage is included. At f = 1, the model predicted an optimum D_i/D_o close to the standard 0.5. It also suggests that the efficiency of screws more than about half full is not highly sensitive to D_i/D_o until it approaches 1.0, at which point efficiency declines.

Fig. 6(d) shows the effect of rotation speed on efficiency. When leakage is included, the model predicts greater efficiency can be realized at higher rotational speeds if friction losses remain neglected. This would be anticipated since in this idealized model, leakage flow rate remains constant with speed but the volume of water passing through the screw increases proportionally with speed. In practice, however, efficiency decreases at high speeds as friction losses begin to dominate the system, resulting in the relatively low optimum rotation speed apparent in the experimental data.

Fig. 6(e) gives efficiency as a function of overall volume flow rate (including leakage flows where appropriate). Zero efficiency occurs when the screw is stopped and all flow is in the form of gap leakage. This plot clearly shows the impact of gap leakage. Efficiency increases as Q increases because the magnitude of Q_l stays constant and therefore becomes a smaller and smaller fraction of the total flow through the screw. Higher efficiency is predicted for the lower fill levels because gap leakage is lower (due to shorter gap length and reduced average pressure differences across the gap) and all other losses are neglected.

Leakage effects also influence efficiency with respect to fill level. Once leakage is included, the model predicts efficiency is maximized when the screw is operating slightly more than full. Efficiency decreases both when the screw is overfilled (f > 1.0), and when it is underfilled (f < 1.0). This peak of greatest efficiency near f = 1 holds as other screw parameters are varied. Generally, as f decreases, leakage increases as a proportion of the total flow, and can become the dominant flow at low speeds and low fill levels, leading to significant reductions in efficiency as f approaches zero. The model results suggest that screws operating less than approximately half full will become increasingly less efficient if gap leakage is present. Conversely, operating a screw in a mostly full condition (e.g., f = 0.8 or 0.9) results in only minor efficiency reductions. This illustrates the general observation that Archimedes screw turbines experience only minor losses in efficiency when conditions differ from the design operating conditions, at least until the difference becomes large.

6.8 Comparison to Laboratory Measurements

Fig. 7 shows data from the laboratory screw and corresponding model predictions of power and efficiency as a function of rotation speed. Volume flow rate remained approximately constant at approximately 1.13 L/s during this test, so f decreased as ω increased. The plotted values of f are model predictions, as f is difficult to accurately measure in an operating screw; however, the predicted values were in the same range as visual observations of the operating screw. The model implementation calculated the performance of the screw for all fill levels at a given operating point, and then used the measured Q_t to find the corresponding fill level, and by extension, the values of all other parameters.

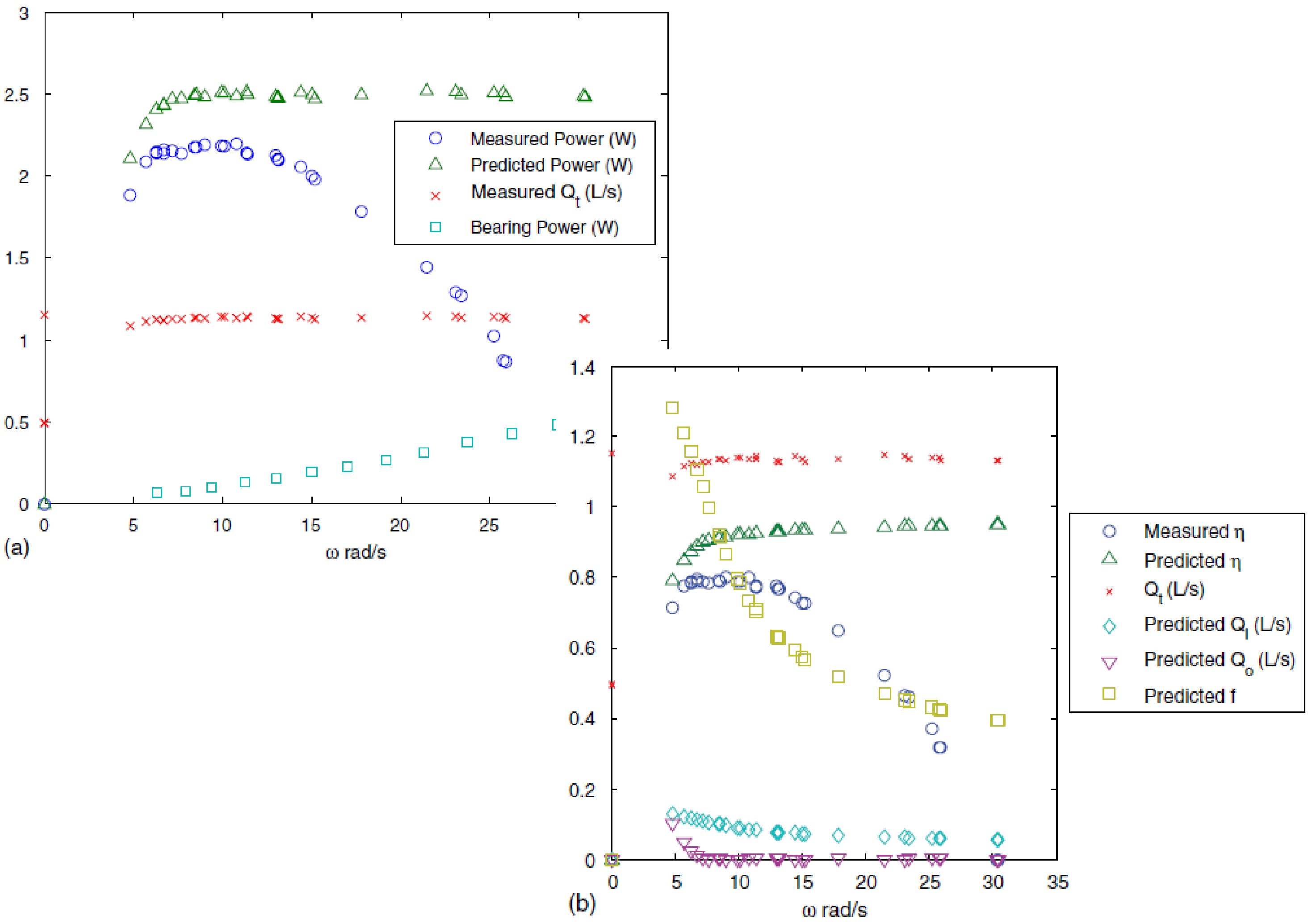


Fig. 7. Comparison of measured and predicted (a) power; (b) efficiency for the laboratory screw

It is believed that the model overpredicts the power and efficiency of the screw at all points partially because it does not include bearing losses. At higher speeds, the increasing difference between the measured and predicted results is believed to be due to a combination of increasing bearing losses and the model not including fluid friction losses that might be expected to increase rapidly as rotation speed increases.

The bearing power data in Fig. 7(a) are the power absorbed by the bearings in the screw without water present, determined based on the rate of rotation-speed decay observed in the dry, unbraked screw after it was initially spun up to a high rotation speed and then monitored as speed decayed due to bearing drag until the screw stopped. It is important to note that loads on bearings in a dry screw will differ from those in an operating water-filled screw, and it was not possible to directly measure bearing drag in a water-filled operational screw. Bearing drag will also change with load on the bearings, so the bearing power data in Fig. 7(a) is only an indication of bearing losses that would occur in an operating screw. The data suggest bearing drag was relatively small at low rotation speeds and increased with rotation speed.

While bearing drag is an important power sink as speed increases, it may not fully account for the difference between measured and predicted power at high speeds. Visual observation of the screw suggests increased fluid motion may occur within the screw as rotation speed increases and this could also be a source of power losses. Unfortunately, it is difficult to quantify fluid drag within an Archimedes screw.

If fluid friction is partly responsible for the difference between measured and predicted power at high rotation speeds, it must be possible to generate sufficient losses by this mechanism. At a rotation speed of approximately 30 rad/s, bearing and fluid friction is consuming all of the available power of approximately 2 W [Fig. 7(a)]. To put this in perspective, 2 W of power is needed to pull a 6-cm square bluff body through water at about 1 m/s, while the lab screw tip speed is 2.2 m/s at $\omega = 30$ rad/s. A correction function could be derived to predict bearing and friction power loss as a function speed. However, many of the variables in the function (such as bearing friction coefficient and effective fluid drag coefficients) are difficult to estimate without experimental data.

Gap leakage is a significant source of uncertainty in the model at low rotation speeds. Fig. 8 shows the predicted efficiency of the laboratory screw assuming the Nagel (1968) leakage model [Eq. (25)], and the new leakage model [Eq. (28)] with several values of C. Recall that if leakage is assumed to be zero, efficiency of 1.0 is predicted for all points. Nagel's model has been included because it is widely used; however, it is only valid for screws operating near a full state and so is not strictly applicable to the majority of this data.

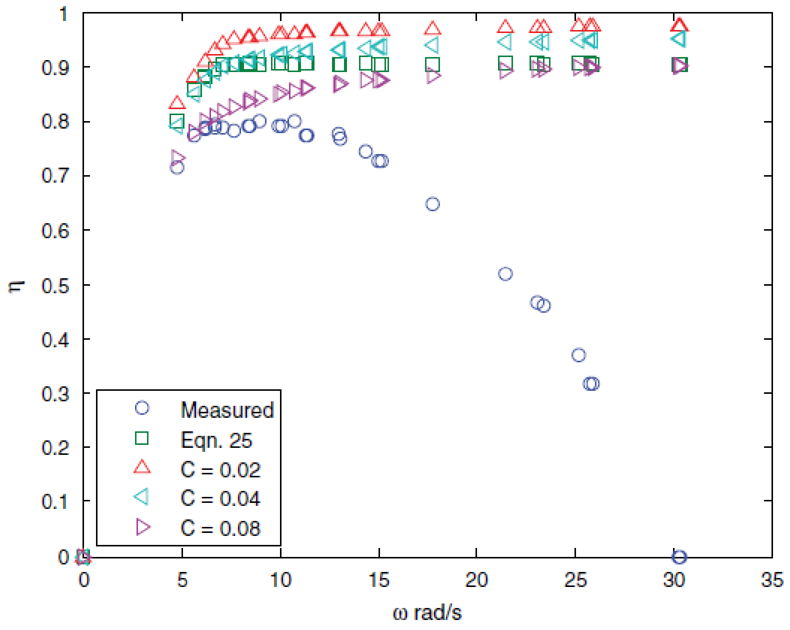


Fig. 8. Predicted power output of laboratory screw with different leakage models

The different leakage models give strikingly different predictions for efficiency (Fig. 8), particularly in the lower speed range where Archimedes screws are most efficient in practice. Predicted Q_l is proportional to C, so this value must be determined as accurately as possible; however, it is difficult to accurately measure leakage in an Archimedes screw. When the screw is turning, it is practically impossible to measure the leakage flow separately from the total flow.

Leakage can be measured in a screw at rest, because then all flow through the screw is by gap leakage (and overflow leakage if water level is high enough). However, even static leakage is difficult to accurately measure: in practice there was up to 20% variation of Ql through the static laboratory screw depending on the angular position of the screw. Even in full-scale ASGs, where this effect would be reduced, there will be different fluid conditions around the gaps in a turning screw than in a static screw. Since the leakage flow cannot be accurately measured, it is particularly difficult to validate leakage models, yet accurate leakage models are necessary to accurately predict the performance of an ASG.

Three mechanisms that reduce efficiency in Archimedes screws have been discussed: gap leakage, bearing losses, and fluid drag within the screw. Experimental results (Fig. 7) showed that efficiency decreases as rotation speed increases in a real screw, and it is important to understand the degree to which each of these mechanisms may lead to efficiency losses. While the relative impact of each mechanism could not be quantified, some conclusions can still be drawn.

Gap leakage is an important source of efficiency reductions even at low rotation speeds, while bearing and fluid losses are predicted to increase with rotation speed.

While current gap flow models are not velocity dependent, it can be argued that gap leakage should decrease as rotational speed increases. Consider the extreme case of water flowing through an ideal screw with no bearing losses or power being generated. It will be turning so fast that the longitudinal water speed within the screw would be approaching the speed that water would flow through the trough without a screw present, which is open channel flow. The speed of this flow is balanced by viscous drag on the trough surface and the pressure gradient is normal to the trough surface. If screw flights were inserted moving at the same speed, there would be no gap flow because the pressure on both sides of the flights would be the same. A fraction of this effect would be apparent at intermediate speeds. At the other extreme of very low speeds, conditions approach the quasi-static state assumed to derive Eq. (28). Under these conditions, the static gap leakage represented by Eq. (28) would be a maximum value, and gap leakage would be expected to decrease as speed increases.

If gap leakage flow does not increase as rotation speed increases, it will be a relatively small loss at high rotation speeds, and energy must also be lost through other mechanisms (such as friction losses). Experimental results [Fig. 7(a)] verified that bearing losses increase with rotation speed, but were not sufficient to determine if bearing losses, or another mechanism such as fluid drag, are the primary power sinks at very high speed.

Notation

The following symbols are used in this paper:

C = constant in leakage equation;
d_o = vertical distance from water level to bottom of upstream vane (m);
D_i = diameter of central cylinder (m);
D_o = screw diameter (m);
F = force (N);
f = nondimensional fill level (-);
G_{wl} = gap width (m);
g = gravitational constant (9.81 m/s^2);
h = head (m);
L = length of screw (m);
l_w = wetted gap length (m);
N = number of flights/number of starts (-);
P = screw pitch (m);
P_{net} = net power (W);
P_{out} = output power/shaft power of screw (W);
P_{avail} = power available in flow (W);
p = pressure (Pa);
Q = volume flow rate assuming no leakage losses (m^3/s);
Q_l = leakage volume flow rate (m^3/s);
Q_o = overflow volume flow rate (m^3/s);
Q_t = total volume flow rate (m^3/s);
R_i = radius of screw inner cylinder (m);
R_o = outer radius of screw (m);
r = radial position (m);
T = torque of a single bucket (Nm);
T_{total} = total torque of entire screw (Nm);
w = location along screw centerline (m);
z = vertical location (m);
Δd = height difference between adjacent buckets (m);
β = slope (angle of screw/trough/surface from horizontal) (rad);
η = efficiency (–);
θ = angular position (rad);
ρ = density of water (1,000 kg/m^3); and
ω = rotation speed (rad/s).

Subscripts

i = inner;
min = minimum;
max = maximum;
o = outer;
total = total;
wl = water surface level;
1 = surface 1 (downstream side of bucket); and
2 = surface 2 (upstream side of bucket).

7. Archimedean Screw Fish Risk Assessment [5]

7.1 Fish trials through Archimedean screws

Fish passage trials performed on Archimedean screws have shown that no significant damage is caused to fish that enter and pass through these turbines. In the UK, this has been demonstrated for salmonids naturally passing down an Archimedean screw and artificially introduced , eels, coarse fish and lampreys. In addition, studies performed in other countries have shown that Archimedean screws do not cause serious damage to fish passing through them.

Table 1: Summary of results from Spah (2001), showing the number that passed through of each species and the lengths of fish affected

Species	No. Tested	Length Range (cm)	No. fish Injured	Injuries
Eel	22	36-58	0	
Grayling	3	20-36	0	
Brown trout	31	8-35	0	
Perch	19	14-18	0	
Chub	63	8-43	5	Scale loss, haematoma
Gudgeon	8	12-14	0	
Bullhead	3	11-14	0	
Dace	1	21	0	
Roach	8	16-21	2	Scale loss, haematoma

The first assessment of fish passage through Archimedes turbines was conducted by Spah (2001). Chub and roach were the only species to suffer any damage; The conclusions of the report were that the damage was most likely due to the leading edge becoming sharpened by stones after prolonged operation.

7.2 Basic hydraulics of Archimedean screws

Archimedean screws work on a relatively simple principle whereby water passes down a long screw, set at an angle of approximately 22 degrees, the gravitational of which causes the screw to turn, generating energy, which is then converted to electricity (see figure 1).

There are four factors that influence the volume of water that can pass down an Archimedean screw and the energy generated. The manner in which each of these variables has an effect is as follows:

1. *Diameter*: all other factors being equal, the larger the screw, the larger the volume of water that it can pass and the greater the quantity of energy that it can generate

2. *Rotational speed*: all other factors being equal, a screw turning at a higher rpm will have a larger volume of water passing through it (per unit time) and will generate a higher quantity of energy

3. *Number of blades*: all other factors being equal, a screw with more blades can handle a greater volume of water and will generate a higher quantity of energy

4. *Pitch*: this is the axial spacing between the blades on the screw and is equal to the diameter of the screw divided by the number of blades.

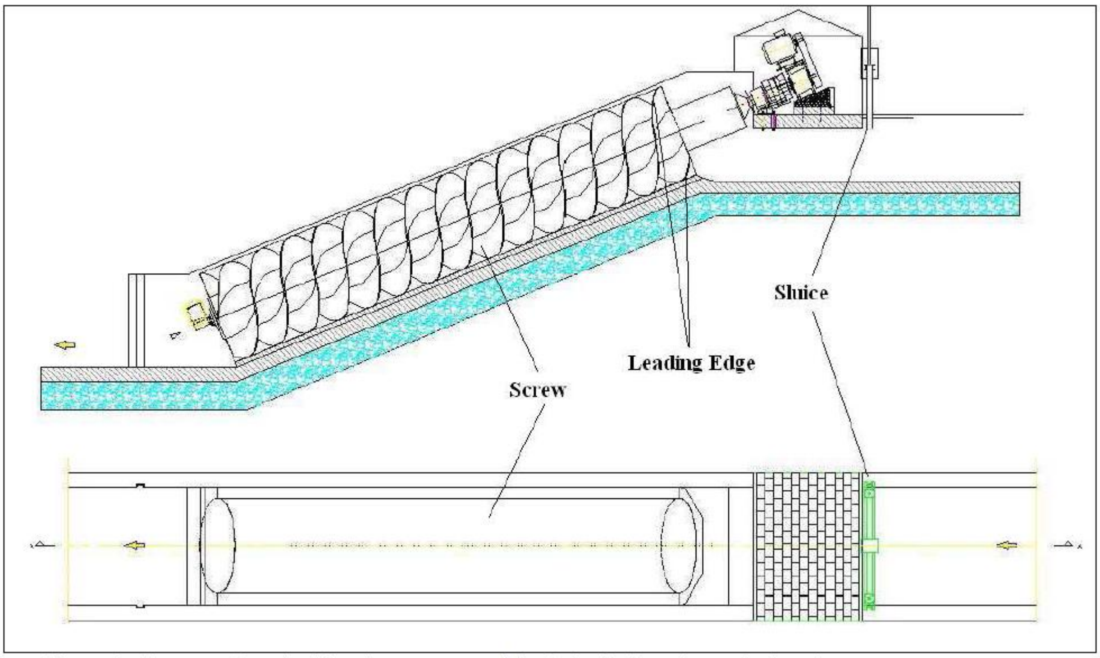


Figure 1: diagram of an Archimedean screw turbine in both elevation and plan view

7.3 Report aims and objectives

While the previous work on Archimedean screws has demonstrated that no significant damage is caused to fish passing through the systems tested, there are concerns being raised about the possible effect on fish of systems of different sizes and with a higher number of blades on fish. Increasingly, proposals include designs for 4 and 5 bladed turbines as they can process more water for a given diameter.

The concerns are being raised by both fisheries specialists within the Environment Agency and angling organisations and can be split into several specific issues, as follows:

1. For screw systems with a higher number of blades there is a higher chance of fish contacting a leading edge. This problem is compounded by the use of smaller screws, which have higher rotational speeds. If this does, as predicted, lead to an increase in the proportion of fish that contact a leading edge, how much of an increase will occur?

2. If more fish contact a leading edge on a screw as they pass into the screw, does this matter? What is the chance of injury occurring?

3. Downstream migration is an important component of the life history and ecology of many fish species. Previous work has demonstrated that fish of several species naturally pass down through Archimedean screws. However it is not known if smaller screws and screws with more blades deter fish from entering and passing downstream. If this is predicted to occur, is it possible to say to what extent?

7.4 The design of Archimedean screws

There is a direct relationship between size and rated rpm for Archimedean screws with smaller machines having a higher average rpm. This relationship is shown for a range of commercially available 3-bladed systems in figure 2

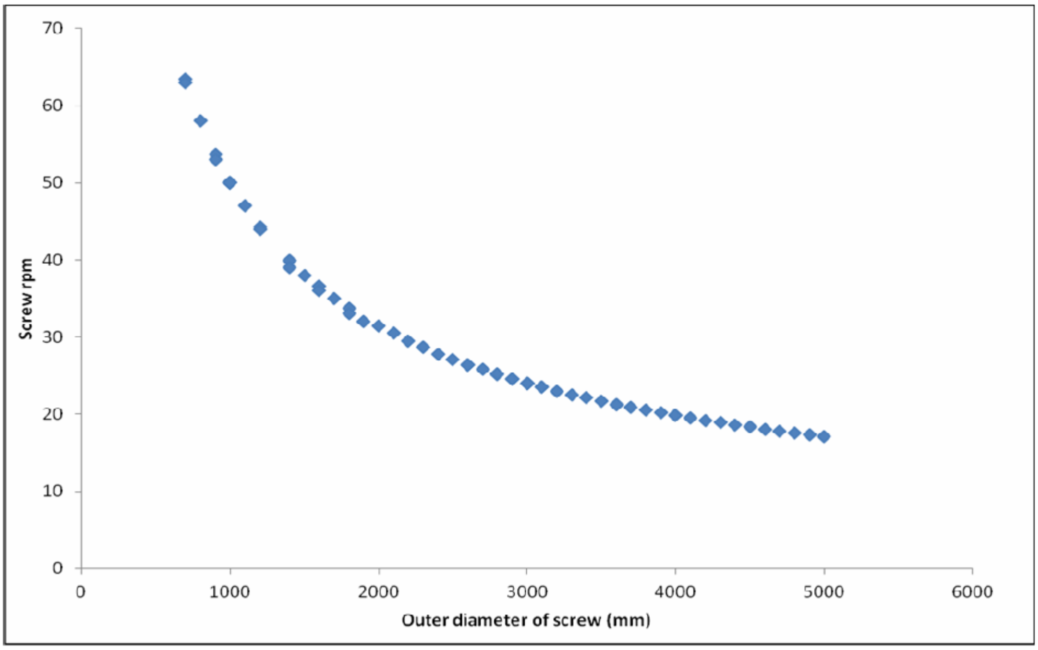


Figure 2: relationship between screw diameter and rotational speed (rpm) for a range of commercially available 3-blade Archimedean screw turbines

As mentioned previously, one of the principle concerns behind this report is the introduction to the UK of 4- and 5-blade screw systems. The first few machines installed in the UK were all 3-blade screws. The relationship shown in figure 2 is extended in figure 3 to include systems with 4 and 5 blades.

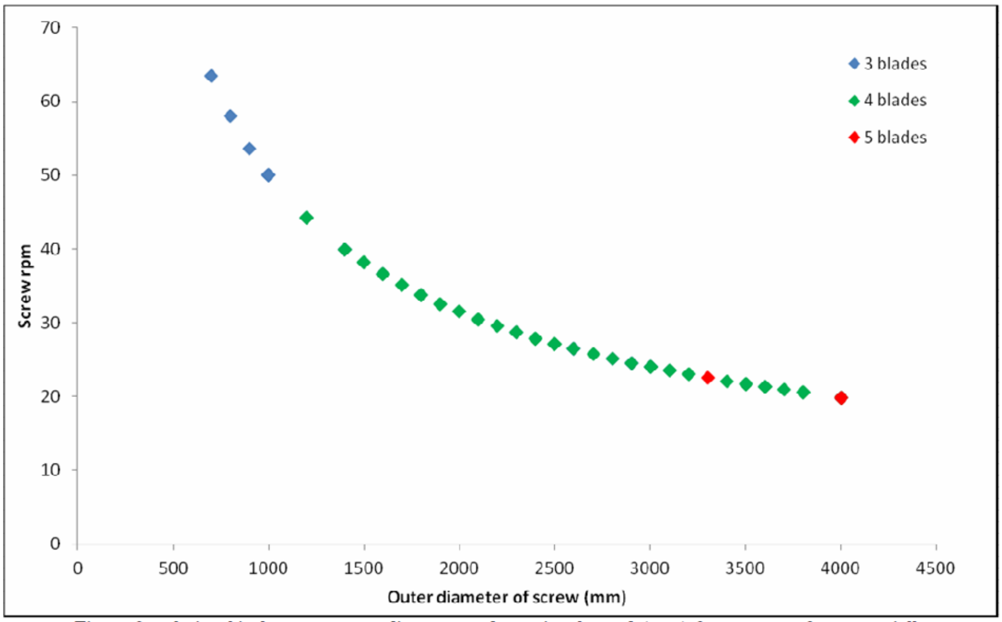


Figure 3: relationship between screw diameter and rotational speed (rpm) for a range of commercially available 3-, 4- and 5-blade Archimedean screw turbine

It is also important to note that there are practical operational limitations on the interaction between screw size and the number of blades. The data shown in figure 2 is a plot of the size and rpm of 3, 4 and 5-blade screws that are actually manufactured. Note that 5-blade screws are not made in the very small sizes of 3-blade or 4-blade screws. The only 5-blade screws being made currently are large systems, > 2 m in diameter.

The only point at which a fish can be contacted by the screw blades is as it crosses the leading edge of the helix. These leading edges are the upstream end of each of the helices that form the screw and are shown in figure 4. As the screw turns, the leading edges scribe a continuous circle at the entrance to the screw, across which fish must pass when they enter the first chamber.

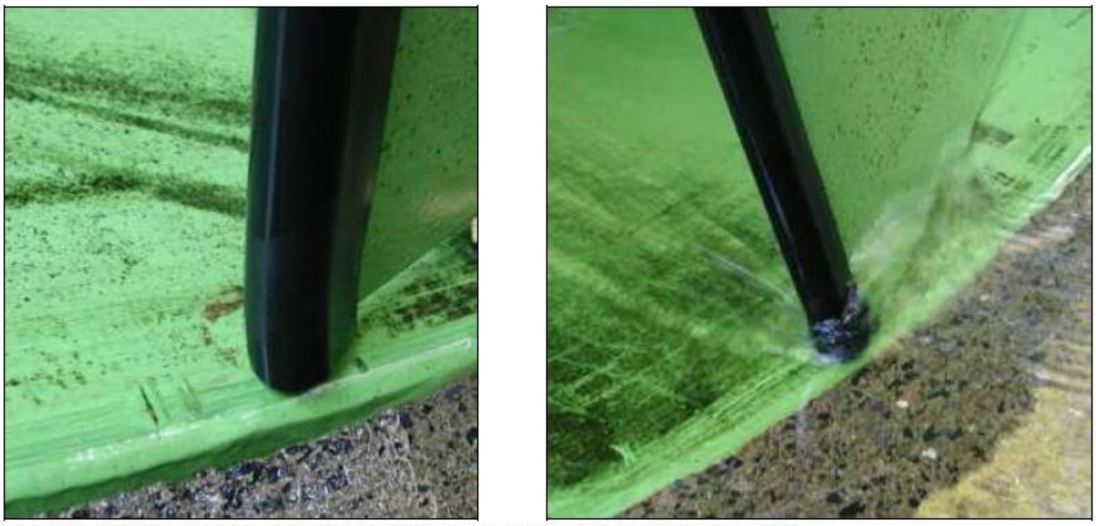


Figure 4: images of the leading edge of a screw, fitted with a rubber bumper

It is obvious that for a screw of a certain size and rpm, the chance of a fish contacting a leading edge is higher for a system with more blades. Equally, a screw turning at a higher rotational speed has a greater chance of contacting a fish with a leading edge. In both cases, a greater number of blades or higher rpm reduces the time between leading edges passing a given point on the circles scribed by the leading edges.

7.5 Probability model development

From a purely mechanistic perspective (ignoring for example any behavioural responses of is to avoid the leading edge of the screw), the probability of a fish contacting one of the leading edges of the screw as it enters is a result of the interplay between various parameters, principally the following:

1. The speed at which the fish enters the screw

The speed at which a fish enters the screw is a direct result of the velocity of the water entering the screw. (It should be noted that as the flow of water into a fixed speed Archimedean screw reduces, the depth of water reduces, while the velocity of water entering the screw remains approximately the same).

2. The size (length) of the fish

A wide variety of fish have been recorded passing down Archimedean screw turbines. Due to their increased length, larger fish will take longer to pass the leading edge of the screw.

3. *The position at which the fish enters the screw, relative to the leading edge*

This refers to the position that a fish enters in the space between the leading edge of the blades. For example, in figure 6 a fish crossing the leading edge circle at point X_1 will have a greater amount of time to cross the leading edge than a fish crossing the leading edge circle at point X_2 as it is further ahead of the approaching leading edge.

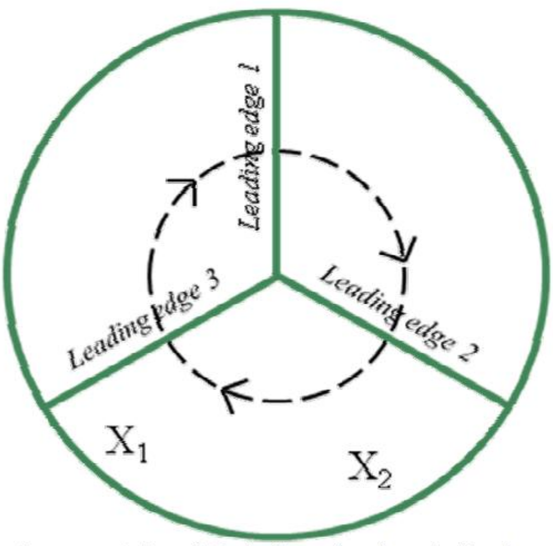


Figure 6: diagrammatic representation of the leading edge of an Archimedean screw and example positions at which two fish could enter the screw

4. The speed of the screw (in rpm)

As seen previously, the design speeds of screws is largely dependent on the size of the screw, larger screws having slower rotational speeds

5. The number of blades on the screw

The greater the number of blades that a screw has, the greater the number of leading edges that could potentially contact a fish passing into the screw.

It is possible to integrate these different parameters into a model that predicts the probability that a fish passing across the leading edge will be struck. For an individual fish this probability (*P*) has a bimodal form; fish either contact a leading edge of the screw or they do not. The model can be described as follows:

T_S *(seconds) = (60/ Un) x (λ/ Λ)*

T_F *(seconds) = L/ V*

P is conditional, such that: if $T_S > T_F$*, P=0 (fish does not contact leading edge)*

if $T_S < T_F$*, P=1 (fish contact leading edge)*

Where:

U = rotational speed of screw (rpm)

n = number of blades in screw

λ = the angle (in degrees) formed by the point at which the fish enters the screw and the next leading edge (see figure 7)

Λ = the angle (in degrees) between leading edges. For 3-bladed screws this is 120^o, for 4-bladed screws this is 90^o and for 5-bladed screws this is 72^o

V = the velocity of the water entering the screw (m/s)

L = the length of the fish (m)

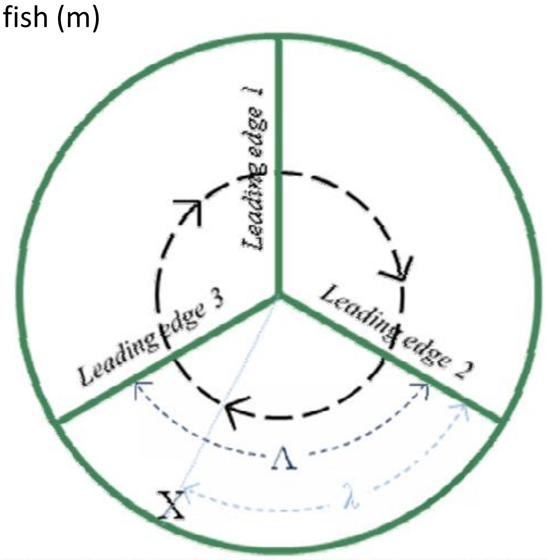


Figure 7: illustrative diagram showing λ and Λ, as described in the model. 'X' is the position at which the fish crosses the circle scribed by the leading edges of the screw

In order for a fish to pass into the screw on any given streamline without contacting a leading edge, it must pass after the sweep of one blade and before the sweep of the next. In the model described here, the two ‘lengths’ are temporal and are essentially the length of time it takes for a fish to cross the leading edge circle and the length of time before the next blade passes.

The rotational speed and the number of blades that the screw has, describe the *temporal* space between two leading edges (which in turn dictates the probability of a fish being struck) and are independent of the size of the screw (although the rpm of a screw is directly related to its size.

7.6 Modelled probability of leading edge contact – empirical corrections

There are several factors that will contribute to lower the probability of fish contacting the leading edge, below that predicted purely by the theoretical model and these are as follows:

- Fish will respond behaviourally in order to avoid being struck by the leading edge

- The movement of the blade through the water pushes water ahead of it, which in turn has a tendency to sweep fish out of the way of the leading edge, particularly smaller fish

- The effect of the push of water sweeping fish out of way is magnified by the fitting of rubber bumpers on the leading edges of screws. This increases the cross-sectional area of the leading edge and hence increases the extent to which the leading edges push water ahead of them.

7.7 Probability of damage occurring

The chance of fish contacting the leading edge increases with the rotational speed of the screw, although not necessarily with the number of blades. This increase in rotational speed is typically also accompanied by a reduction in the size of the screw. In terms of the tip speed of the leading edge, this reduction in size compensates for the increase in rotational speed, such that smaller screws with high rotational speeds have slower maximum tip speeds than larger screws with low rotational speeds.

This relationship means that while the probability of a fish contacting the leading edge is higher for a small screw, the likelihood of physical damage occurring is reduced, due to their lower tip speed. Tip speed has previously been shown to be a key determinant of whether injury occurs when a fish is struck by a turbine blade.

It is important that the risk of damage is placed within the framework of the substantial amount of work already conducted on Archimedean screws and the recommendations and legislation already in place. Screws within the UK are already fitted with rubber extrusions or compressible rubber bumpers in order to reduce the strike force with which they are struck when contacting the leading edge and this already represents a precautionary position, ensuring that any fish that do contact a leading edge are not damaged.

Due to the force of the impact being spread by the compressible bumper or hard rubber bumper, it is highly unlikely that a fish that contacts the leading edge in a screw of any size in the UK will suffer significant damage. This is a critical point, as it means that the issue being considered in this report is not damage *per se*, but the change of delay or prevention of downstream migration.

7.8 Delays to downstream migration

An important consideration is whether fish are delayed in their downstream migration when they reach an Archimedean screw turbine. The diversion of flows through an Archimedean screw (or any other turbine) removes flows from passing over a weir and/or through a depleted reach. As a result, fish may not be as able to use the usual route downstream and instead the primary downstream migration route becomes passage through the Archimedean screw.

However, if fish are reluctant to enter the screw, downstream migration may be delayed. Such delays could occur for a number of reasons, primarily:

- The time gap between the leading edges is too small and fish are dissuaded from entering the screw

- The probability of fish contacting the leading edge is too high and fish are dissuaded from entering the screw. Such a mechanism is reliant on fish exhibiting behavioural risk aversion and a behavioural judgement that prevents them entering the screw.

- The size of the screw and particularly the longitudinal gap between helices (i.e. the chamber size) is too small relative to the size of the fish. Again, this relies on fish making a behavioural judgement prior to entering the screw, however, if a large fish were to enter a small chamber size, there is a higher chance of the fish being damaged by being bumped against the sides of the chamber.

The risk of delays occurring due to a reluctance of fish to enter the Archimedean screw will be reduced if fish are presented with an alternative, viable route downstream such as a by-wash. There is therefore a need to determine recommendations and guidance as to the circumstances under which an alternative route downstream should be provided. This guidance is outlined below, with reference to the primary reasons that may cause fish to delay.

8. Screw Evacuation due to Damage Risk [6]

8.1 Development of existing Hydro Buildings

Exploitation of hydraulic structures is considered in two ways. The first one is the operative exploitation, which aims to make the best use of flow and head to obtain the highest possible energy production and distribution of the most favourable time. The second one is a technical operation designed to provide a normal and trouble-free use of power as long as possible, maintaining safety regulations.

Threats of SHP may occur as a result of flow operation exceeding reliable flows. The operation of hydrological or geological conditions with exceptional features, such as a very thick ice, improper passing of excessive water or ice flows, moving parts' freezing, etc. The passage of excess water by water structures of small power plants, especially passing the great flood water should be carried out in a manner to ensure safety and to minimize the damaging effects of water. Huge congestion and water thrust affect the hydro building as well.

8.2 Ways to evacuate the turbine

Very often a mobile crane is used with the first turbine unit installation . So far turbine installation and removal were made with the use of lifting equipment, gantry cranes or, for smaller turbines hoists embedded in the frames were used. These solutions of disassembly or partial disassembly by crane has its drawbacks, of which the principal one is the time in which the operation is carried out and which may last a few hours. In case of flood, time is a critical parameter and in specific weather conditions driving up a specialized equipment may not be possible. Evacuation system is created in order to pass the flood waters including Q = 1 (so called „century water" – that is one that theoretically occurs once every 100 years).

The main way to prevent damage caused by flood is locating an object on the side channel of the river bank and not on the main channel (mill builders have used such a strategy for a long time). By-pass channels are also used.

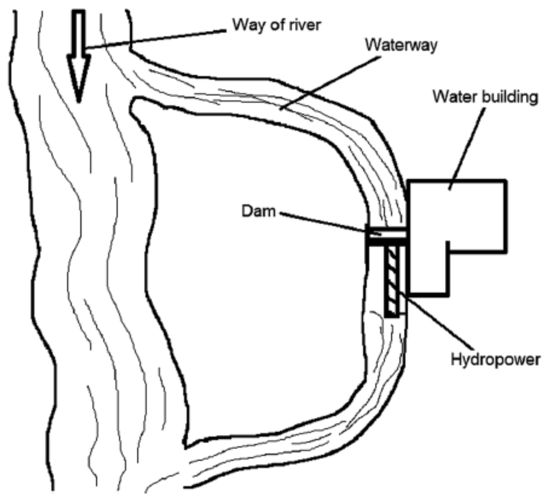


Fig. 2. Diagram of hydraulic engineering structures built on the bypass channel

Evacuation systems are good in situations where a dam or other hydro-technical structure, equipped with only a single path for water are electrified. Then, in the event of water excess, it can be let through easily, quickly and without resistance, protecting the device from damage.

The hydropower plants, driven by an Archimedes screw turbine have now permanently attached turbine unit. In practice, this means that each amount of water greater than the nominal swallowing capacity of turbine is not energetically used. The solution allows moving the turbine inlet up and down generating the energetical use of large water flows, thus significantly improving the SHP technical parameters along with the Archimedes screw turbine. It is estimated that the energy yield from the water additionally introduced onto the turbine can increase the efficiency of the turbine by about 20%, compared to the energy from the water in the nominal swallow capacity of turbine. The problem could be only the rotor flooding. To avoid it, effective decrease is reduced by a half of the diameter of the rotor plus the distance for the low water at large flows.

Separation the turbine from the turbine unit allows the use of larger amounts of water indicated on the rotor. The technical solution also allows rapid evacuation of Archimedes screw turbine on the water surface. As mentioned earlier, none of European manufacturers which are global providers does not have such a range of products (for the Archimedes screw turbines). The example of evacuation system based on the technical concept known from transport systems (e.g. lifts, winches) was used to evacuate self-supporting structure of Archimedes screw turbines. Figure 9 shows the turbine during normal operation. While working, the turbine unit is based on the lower bearer and maneuvering rolls placed on the abutment and the pillar of the dam and this is so called operational phase of the operation of the plant.

Evacuation is carried out by using two mechanical- and hand-driven chain hoists mounted on a crane beam of a steel structure. A scheme of the evacuation way of the turbine above the surface of the water by means of a chain hoist is shown in Figure 10.

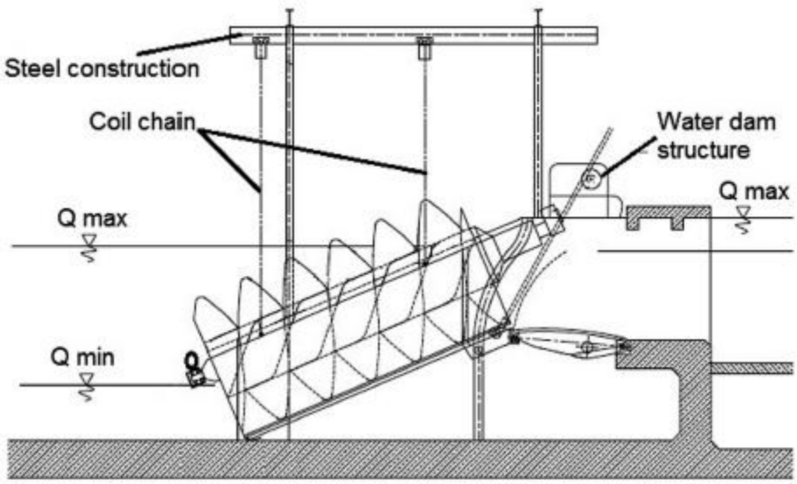


Fig. 9. Archimedes screw turbine scheme of evacuation system at work

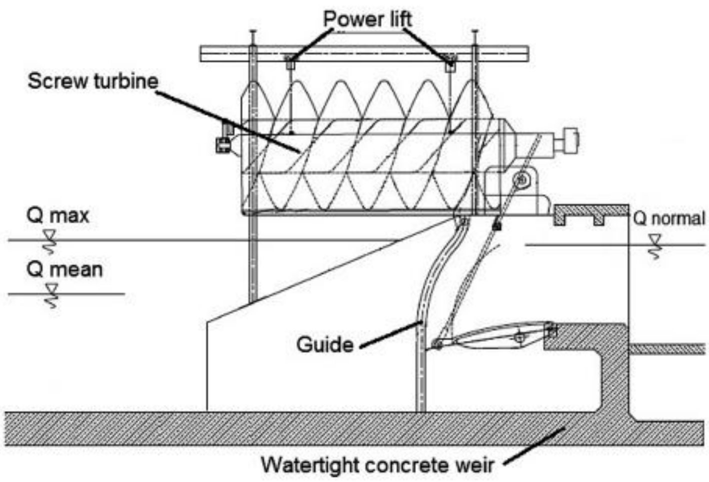


Fig. 10. A scheme of the evacuation way of the turbine above the surface of the water by means of a chain hoist

8.3 Conclusions

It has been pointed out that depending on the type of management section of the river and the present SHP technical solution, there are various ways of quick turbine evacuation in danger, and hence, prevention of damage to the turbine. The concept of separation turbine unit and control system of water flow onto the turbine rotor, while maintaining a constant level of the top water, allows for greater energetic use of the flowing water. Hydroelectric power plants operate at a predictable time and power and are the most reliable source of electrical energy from renewable sources. The advantage of hydroelectric power is the ability to stabilize river flow and prevent flooding. Small hydropower systems have been increasingly used as an alternative energy source so that a small system is installed in small rivers or streams with little environmental effect. In addition, investment in turbines with fish-friendly technology "fish friendliness" allows the use of energy in protected areas, such as "Nature 2000".

9. Application of the Archimedean Screw in Wastewater Treatment Plants and Sewer Systems [7]

Today WWTPs are usually the facilities with the highest energy demand in public ownership. Thus, renewable energy facilities are added in order to reduce the overall demand of energy supply taken from the power grid. Consequently also small hydropower plants are part of this strategy, using an again new identified site for small hydropower implementations. Results show that some sewer structures may be suitable for an implementation of energy recovery or storage facilities, but application is still limited, due to economic reasons, whereas the implementation of an Archimedean screw in the outlet of the WWTP is technically and economically feasible.

It is well known that wastewater treatment plants (WWTPs) are high energy consuming facilities. They account for up to 75 % of the overall energy demand of a municipality. Within the last decades it was established that wastewater should no longer be addressed as waste but as a resource. In this context, the German water association “Emschergenossenschaft” focuses on the production of renewable energy, i.e. energy recovery on WWTPs. In this paper we will present as a case study the WWTP in Bottrop/Germany which treats 1.3 Mio. population equivalent (PE) and has already been technically and energetically optimised by improving the sludge treatment, benchmarking energy usage and replacing high energy consuming devices, respectively. The new integral strategy for plant operation and energy management (cf. Figure 1) also consists of the application of renewable energies. Solar cells are nowadays state of the art and can easily be applied as the WWTP provides large roof and fallow areas. The additional integration of a wind turbine is also analysed. Consequently, the applicability of hydropower is investigated as WWTPs provide a well-defined discharge and, depending on the location, a certain hydraulic head. In general, the outlet structure, i.e. the head to the river is used for energy generation.

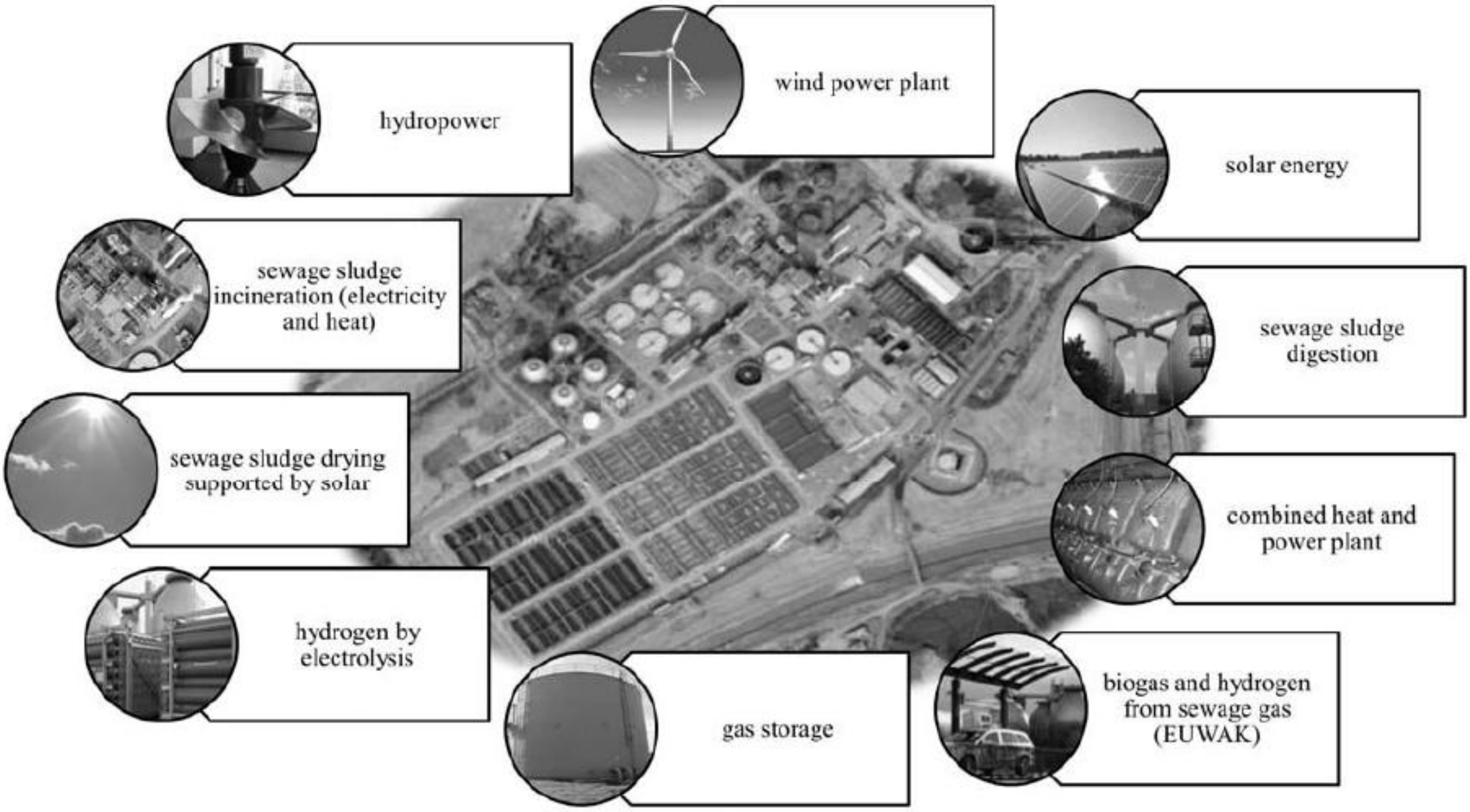


Figure 1. Elements of an integral energy management strategy on WWTPs, the “hybrid power plant Bottrop”

In a second step, hydropower applications in sewer systems should eventually be analysed as sewers provide continuously running and quite often falling water. Moreover no conflicts due to intervention in nature have to be expected as sewers are technical structures. To use the existing heads and a certain space for additional devices, hydropower should be applied within the manholes.

One example is an overshot water wheel in Aachen/Germany. The available wastewater discharge ranges between 30 l/s and 120 l/s; the useable head is 6 m. This results in a theoretical output of 6.7 kW. In a research project, an overshot water wheel was installed crosswise the flow direction, which achieved an average daily output of 65 kWh (4.4 kW). During the operational phase it was found that there was no clogging or entanglements as the water wheel developed a type of "self-cleaning effect" as a result of the rotational movement. The low feed-in compensation made this project not economic.

9.1 Hydropower potential and available technologies

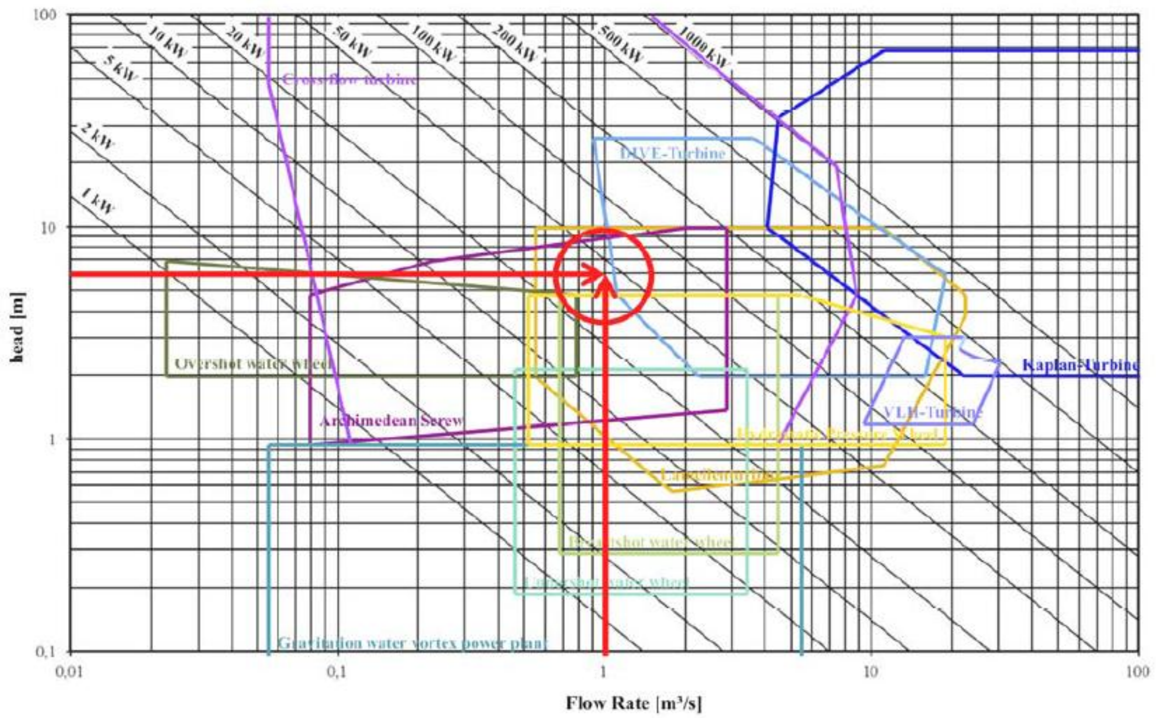


Figure 2. Application ranges of different small hydropower devices (Frehmann, Berger et al. 2013), red arrows mark the intersection for a given head of 6 m and a flow rate of 1 m³/s as an example

In general, hydropower systems can be divided into the following categories: water wheels, turbines and turbine based concepts, Archimedean screw, flow energy converters, mechanical conveyor belts. As flow converters are designed for the use of the kinetic energy of rivers they won’t be presented in further detail.

Since ancient times the *Archimedean screw* is used to lift water or sludge to higher levels. Since several years they are also used vice versa to generate energy in very low-head situations. The falling water moves the helical blades wrapped around the axis which drives via a gearbox the generator. Due to its construction the Archimedean screw has a wide range of high efficiency. The flow rate(maximum 10 m^3/s) determines the diameter of the screw (up to 4 m), whereas the length is determined by the head depending on the angle, where a lower angle increases the efficiency. Disadvantageous can be the required huge dimensions due to the flat angle (< 30 °) as well as the high load due to a massive steel construction. They are nowadays state of the art and produced in different standardized versions.

9.2 Applicability of hydropower in sewer systems

There are serious restrictions on the application of hydropower in sewer systems. In contrast to drinking water distribution networks, in which turbines and further hydropower concepts are already in use for pressure reduction and the simultaneous recovery of energy, untreated wastewater is a far more complex medium. The chemical composition, in particular pH value, oxygen content and temperature result in increased corrosiveness. Therefore, all devices have to be made of resistant materials such as stainless steel or grey cast iron. Additionally, the composition of wastewater causes an explosive atmosphere; thus all components have to be designed accordingly.

The physical composition of the wastewater is the limiting restriction for hydropower in sewer systems. Contraries such as timber or stones can damage parts of the device. Fine contaminants such as sand may have a permanent abrasive effect and result in increased wear. Contaminants consisting of cohesive materials such as cat litter or Benton it can clock the device. Fibrous contents like tissues, ropes or hairs lead to entanglement. Entanglement represents a particularly high risk for propeller-based concepts such as turbines. Therefore, turbines can only be used after preliminary treatment, i.e. screening, which would need energy and the frequent removal of debris. Additionally, propeller based concepts are more sensible to varying discharges and atmospheric influences. To avoid cavitation, an upper reservoir would have to be implemented. This would lead to standing wastewater which causes an explosive and toxic atmosphere. In summary, turbines are not suitable for the application in combined sewer systems. Preference should be given to Archimedean screws, water wheels or trough conveyor concepts, respectively. These devices not under risk of cavitation and, in particular overshot waterwheels and Archimedean Screws, may pass larger contraries due to their design. In any case, the suitability of the technology as well as the boundary conditions such as flow rate, available space, chemical and physical composition of the sewage have to be considered carefully.

9.3 Applicability of hydropower on WWTPs

In general, WWTPs are suitable for hydropower due to mechanically and, depending on the position, biological treated wastewater. In addition, the given discharge is well documented and the technically formed tanks and channels generally allow an implementation. Screens and trash racks are not needed, as the water is already treated. The integration into the energy grid of the WWTP is very simple and expert staff for operation and maintenance is located nearby and available day and night.

The installation of a hydropower unit within the treatment tanks is rather difficult as the treatment must not be disturbed as well as the outflow has to be guaranteed all the time. Gravity flow is usually used within a WWTP; large heads between the tanks can therefore not be expected. Available space is quite rare alongside the tanks. The most likely sight is the outlet structure. The outlet structure provides a certain head for energy recovery and, depending on the construction, some space for an additional device and carries the complete discharge. Depending on the given structure as well as flow rate and head, turbines, water wheels, Archimedean screws or technologies based on conveyor chains may be applied. Turbines should already be planned during the building/construction phase as they need certain concrete structures; if a continuous discharge can be guaranteed they are low in maintenance and contribute constant energy.

9.4 Economics

The economy of a hydro power plant is mainly determined by the yield. If the produced energy will be fed into the energy grid a compensation of 0.12 €/kWh will be obtained (BMU 2011). On the other hand the produced energy may also be used to enhance the self-energy production and thereby minimize energy costs. Energy costs range at the moment at 0.19 €/kWh, but are expected to reach 0.25 € within the next years. Additionally, hydropower improves not only the self-energy production but also the operator’s carbon footprint. Hence it is economically worthwhile to use the produced energy on-site.

Manholes are most often not found near high energy consuming facilities. Thus it is most likely that the produced energy will be fed into the grid. WWTPs on the other hand have a high energy demand. The thereabout produced energy can be used directly and will enhance the self-energy rate. Thus it is mostly likely that hydropower on WWTP is economically more suitable than in sewer systems.

9.5 Case Study WWTP BOTTROP

The water associations Emschergenossenschaft/Lippeverband operate 44 WWTPs. Altogether they provide a theoretical potential of 372 kW for energy recovery at their outlet structures. At eleven sites hydropower is already implemented or will be implemented soon, these sites provide an output of 328 kW, which equals annual savings of 169,000 €. The other sites are technically not suitable due to manifold reasons. Following, the analysis of the WWTP Bottrop is described.

The WWTP in Bottrop was systematically investigated. Concerning a first estimation, four possible sites have been investigated in further detail:

- Digesters
- Outlet of chamber filter presses
- Outlet of primary sedimentation
- Main outlet

It was investigated that sludge treatment is not suitable for hydropower application due to the characteristics of the sludge. Despite the big heads of the digesters of about 30 m the characteristics of the sludge would block any mechanical energy conveyer and thereby disrupt sludge treatment. The outflow of the chamber filter presses could not be used as in-pipe devices are rarely available and the discharge was too little to find an economic solution. Between primary sedimentation and activated sludge tanks a head of approx. 1 m was detected. The mechanically treated sewage is suitable for hydropower units but the design of the basins prevents the installation. Only the outlet structure can be used for hydropower. Depending on the water level in the river there is a net head of 1.40 up to 1.80 m. The average flow rate is 4 m^3/s; within a margin of 40 %. Considering hydrological and constructional restrictions, an Archimedean screw was identified the best solution.

While designing an Archimedean screw within the outlet of a WWTP several restrictions have to be considered. To secure undisturbed treatment, the discharge has to be guaranteed all the time, even at heavy rain events and if the hydropower unit is damaged, respectively. To guarantee the outflow in these cases it is suggested to mount the Archimedean screw in the middle of the channel in combination with a weir of the given head. In operating conditions the weir will damp the water and lead it to the screw; if the screw is not able to use all the water available or is damaged, two bulkheads in the weir on the left and right can be opened to guarantee the rated discharge.

9.6 Case Study EMSCHER Sewer

The Emscher sewer (AKE) is becoming the main intercepting sewer in the river Emscher catchment. In order to collect the wastewater and lead it to the WWTPs by gravity flow over a length of 51 km, it is constructed up to 40 m below the ground. The wastewater is collected in the catchments and then passed to the AKE in more than 30 manholes. The wastewater falls thereby between 5 – 20 m. The dry weather discharges vary between approx. 5 L/s in the smallest catchment up to 1,000 L/s in the largest catchment. Altogether the manholes provide a theoretical potential of 160 kW, which equals a feed-in compensation of 150,000 € a year; following a first estimation, only half of them is worth further consideration.

This manhole hosts a so-called vortex drop structure to transfer the sewage from the given catchment to the AKE. The vortex structure aims to reduce energy and minimise the formation of aerosols. The wastewater is passed through a swirl chamber and through an eddy to a down pipe which ends in a stilling chamber, where the water is calmed down before entering the main sewer. The manhole has an internal diameter of 23 m and an overall height of around 30 m. The inlet from the catchment area is located approx. halfway up. The down pipe has a diameter of 1.6 m and a height of 4.72 m. The gross head is 6 m. The stilling chamber has a base area of 7 x 3 m. As the floor slopes to the outlet, the height increases from 3.05 up to 4.18 m. The flow rate varies between 0.1 L/s at night's minimum up to 1.9 m^3/s at day's maximum. For a first estimation of the potential the mean daily discharge of 1 m^3/s will be considered. In case of rain, the flow rate is approx. three times the mean daily discharge. Thus, hydropower should be designed for the dry weather flow in order to guarantee continuous energy production. With a flow rate of 1 m^3/s and a head of 6 m, the theoretical energy output is 59 kW.

For the installation of a hydropower system there are no constructional restrictions at present, i.e. structural changes are possible as the manhole is not jet under construction. The simplest solution would be the integration of the device in the stilling chamber. An Archimedean screw and water wheels, respectively, are examined. Both systems cannot be installed in the stilling chamber due to their dimensions.

The hydraulic boundary conditions make the integration of a hydropower device difficult. The inlet pipe is more or less rectangular to the AKE. Using the vortex the inflow is ideally directed into the new flow direction. An Archimedean screw or water wheel, respectively, can only be integrated behind the vortex drop structure after the water has been redirected (cf. Figure 4). Thus, the rain weather discharge has to be guaranteed all the time. The entire volume first has to pass through the vortex; the dry weather flow has to be directed to the hydropower device while the rest flows into the main sewer, which requires a bypass solution.

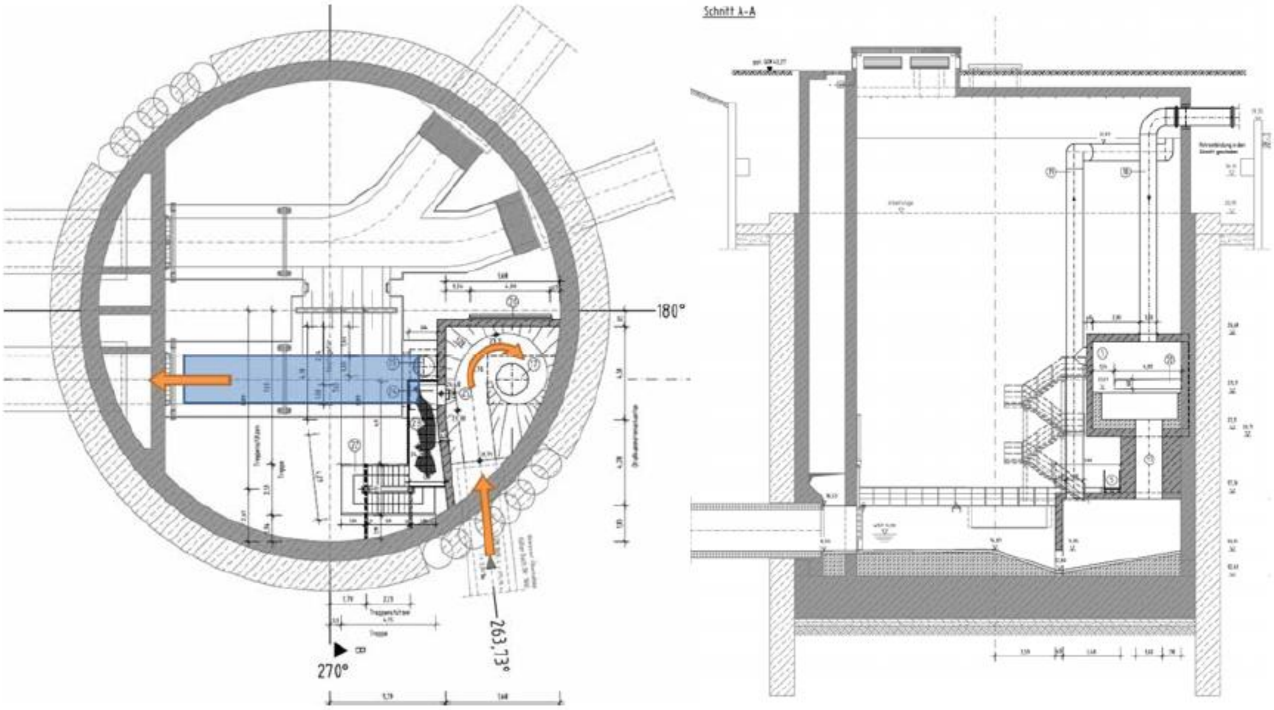


Figure 4. Typical manhole with vortex drop structure of the AKE in top view (left) and site view (right) with flow directions marked by arrows and a possible location for a hydro power device (rectangle)

In addition to the technical feasibility, the cost-effectiveness has to be considered. Considering the given example, the theoretical potential of 59 kW has to be corrected to a defined efficiency (water to wire) of 65 % - owing to the medium - which leads to a remaining potential of approx. 38 kW.

Taking into account the costs of the needed devices and the additional costs due to the corrosive atmosphere and medium, respectively, a cost-effective implementation is not possible. Furthermore, bearing in mind the uncertainties due to the characteristics of untreated sewage and fluctuating discharges there are significant additional risks during operation which have to be considered within the project.

9.7 Outlook

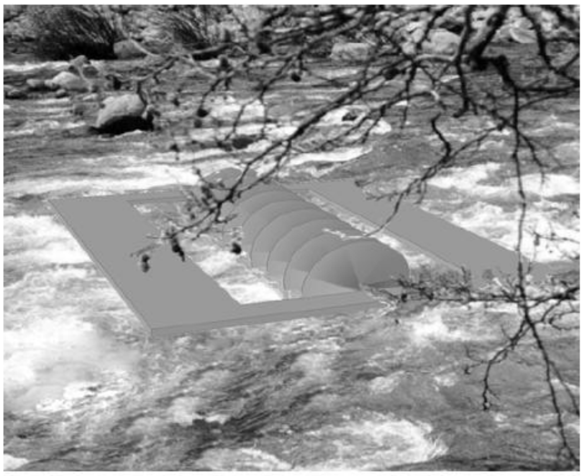
Rising energy prices force operators of sewer systems and WWTPs to intensify the use of existing potentials to increase the self-energy production and energy recovery. Many measures have already been implemented; this paper analyses the applicability of hydropower and in particular systems for low heads in technical sites of the water associations Emschergenossenschaft/Lippeverband.

There are currently no suitable and cost-effective solutions for the use of small hydropower in sewer systems or rather manholes. Turbines are not suitable for use in wastewater and Archimedean screws and water wheels, respectively, can rarely be integrated into the structurally optimised manholes due to their large dimensions. Thus, a hydropower device has to be designed carefully. Usually, it should be designed according to the dry weather discharge to obtain continuous operation; the fluctuations between night's minimum and day's maximum hereby require a broad range of efficiency. In order to ensure optimum flow condition for the device, structural adjustments of the manhole building are indicated. Low efficiencies due to the medium as well as declining feed-in compensation have also to be taken into account.

Concerning WWTPs, different sites have been analysed. As gravity flow is usually used on the plants, it is expected that the most promising site is the outlet structure, as a certain head to the receiving river is usually given to guarantee discharge even at flood conditions. Constructional restrictions determine the applicable technology, it has been described that only few small hydropower techniques are suitable. Out of the large number of small hydropower technologies available, only a few can be used under the difficult conditions in sewer systems and WWTPs, respectively. Thus, in most cases they are not economic. Continuous developments in the fields of hydropower technologies as well as material sciences have to be observed within the next years, as some innovative ideas and prototypes are currently under testing. Moreover, it is anticipated that the continuing increase in energy costs will have a positive impact on the economics of small hydropower systems. If suitable discharges and heads are available, hydropower is a simple and effective way to enhance self-energy production and thereby saving energy costs as well as improving the operator's carbon footprint. Considering the long lifetime of hydropower plants the implementation of hydropower is a foresighted investment.

10. Innovative Floating Archimedean Turbines [8], [9]

Preliminary recent research proved the useful exploitation of a new screw technique and the efficient rediscovering of the old screws, under the form of a second type of AWCT's (Archimedean Water Current Turbines) of horizontal floating cochlear rotors. The AWCT systems, without civil works, are defined as horizontal-axis floating cochlear systems that convert hydro kinetic energy from flowing waters into electricity.



These floating cochlear rotors could harness the unexploited flowing hydraulic potential of natural streams and open channels, coastal and tidal currents as well. Flow speed of around 1.7-2.0 m/s seems that this current flow is sufficiently powerful to drive new well-designed Archimedean spiral power screws and produce valuable electricity.

The possibility of exploiting sea and tidal currents for power generation has given little attention in Mediterranean countries despite the fact that these currents representing a large renewable energy resource could be exploited by "modern old technologies" to provide important levels of electric power. It is also well known that one of the oldest machines still in use is the Archimedes screw. The present paper intends to prove the useful modern rediscovering of some old Archimedean ideas concerning spiral water wheel technologies under the form of new and efficient horizontal-axis Archimedean hydropower turbines.

The conventional method of extracting energy from tidal flows is to place a barrage across an estuary with large tidal range to create a static head or pressure difference, and operate a low head hydroelectric power plant with intermittent, reversing flow. The best-known example of this approach is the installation in "La Rance" River Estuary in France, completed in 1966. The less well-known method of extracting energy from tidal and other sea flows is to convert kinetic energy of moving water directly to mechanical shaft power without otherwise interrupting the natural flow, in a manner analogous to a hydraulic-wind turbine.

In order to be able to utilize the low and very low water head differences and to recover valuable energy, including seawater and tidal potential, different types of water wheels were developed and perfected in the past. Fig.7 gives two different horizontal floating water wheels in Euripus Strait, Greece.

It is technically possible to obtain the optimal exploitation of coastal and continental hydropower potential by considering, that Archimedean screw turbines could be efficient for zero head maritime applications and for various low head sites for different flow rates scales.

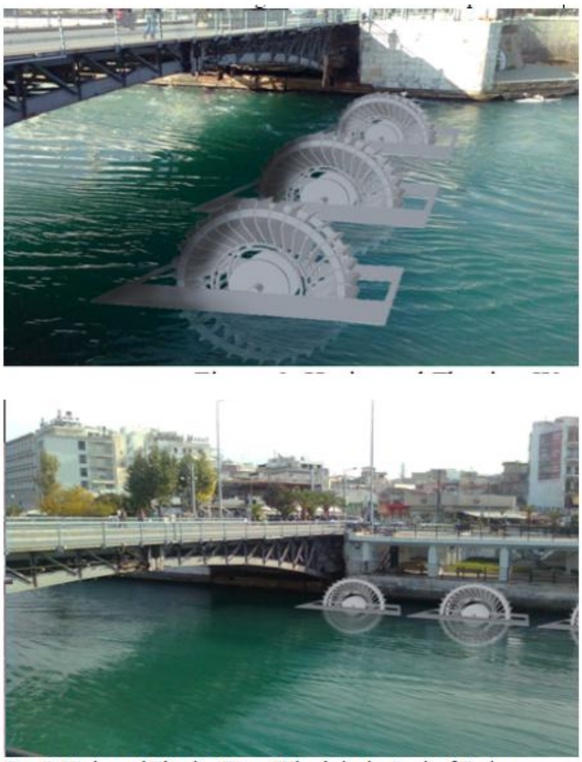


Fig. 7. Horizontal Floating Water Wheels in the Strait of Euripus.

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