Numerical and experimental study of an Archimedean Screw Generator

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Abstract. Finding new, safe and renewable energy is becoming more and more of a priority with global warming. One solution that is gaining popularity is the Archimedean Screw Generator (ASG). This kind of hydroelectric plant allows transforming potential energy of a fluid into mechanical energy and is convenient for low-head hydraulic sites. As it is a new and growing technology, there are few references dealing with their design and performance optimization. The present contribution proposes to investigate experimentally and numerically the ASG performances. The experimental study is performed for various flow conditions and a laboratory scale screw device installed at the fluid mechanics laboratory of the INSA of Strasbourg. The first results show that the screw efficiencies are higher than 80\% for various hydraulic conditions. In order to study the structure of 3D turbulent flows and energy losses in a screw, the 3D Navier Stokes equations are solved with the k-\omega SST turbulence model. The exact geometry of the laboratory-scale screw was used in these simulations. Interestingly, the modeled values of efficiency are in fairly good agreement with experimental results while any friction coefficient is involved.

1. Introduction
Finding new, safe and renewable energy is becoming more and more of a priority with global warming. Micro-hydroelectricity has a large unexploited potential in Europe as there are thousands of unused mills and weirs. The generation of renewable energy with Archimedes Screw Generators (ASG) transforming potential energy of fluid into mechanical energy is a growing technology suitable for low-head hydraulic sites [1]. Among the main advantages of the ASG is that high efficiencies can be maintained despite fluctuations in inflows discharges. An important point is its robustness: sediments and small debris can pass through an ASG without damaging it. The ASG is also assumed as fish friendly [2, 3].
For design and optimization purposes, some theoretical models trying to link the ASG performance to its geometrical parameters and flow features have been proposed [4–7]. The theoretical values of screw torque and efficiencies found in [7, 8] are in fairly good agreement with experimental values. However, this last model and the previous ones rely on the calibration of a hydraulic roughness and gap leakage coefficients. In the present contribution a new approach based on the numerical resolution of the 3D Navier Stokes equations with the $k-\omega$ SST turbulence model is introduced (see also [8]). As demonstrated, the strength of this method is to allow studying accurately the ASG performance only with usual turbulent closure models. The numerical simulations indeed enable to determine pressure and velocity fields where it is very difficult to measure them experimentally. The integration of pressure field on the screw surface enables the calculation of the torque and hence its efficiency.

2. Definition and operating principle of an ASG

An ASG consists in a screw rotating in an open and fix trough. The potential energy of the fluid flowing into the plant is transformed in mechanical energy thanks to the rotation of the screw. This mechanical energy is then transformed in electricity using a generator. The power provided by the ASG is given by:

\[ P_{ASG} = \rho g Q H \eta_{ASG} \]  

with $P_{ASG}$ the power in Watt, $\rho$ the density of water in $kgm^{-3}$, $Q$ the flowrate in $m^3s^{-1}$, $g$ the gravitational constant (9.81 $ms^{-2}$), $H$ the geodetic head in m and $\eta_{ASG}$ the efficiency of the whole system.

The ASG efficiency is directly linked to the generator’s efficiency and to the different hydraulic losses present in the plant. These main head losses, which result in power losses, are due to the leakages and frictional forces induced by the fluid viscosity. Hence, it is important to minimize these losses in order to obtain an optimal efficiency.

The main features of an ASG are the head $H$, the total flowrate flowing through the plant $Q$, the rotational speed of the screw $n$ and the ASG efficiency $\eta_{ASG}$. The geometrical parameters of the screw shown in figure 1 are:

- Outer radius: $R_a$
- Inner radius: $R_i$
- Pitch of the screw: $S$
- Total length: $L$
- Threaded length: $L_b$
- Number of blades: $N$
- Screw inclination: $\beta$

As Rorres [9], we define the volume of water trapped between two successive blades. This volume is named "bucket" and is equal to $V_B$. Then, we define the optimal filling point of the screw that is reached when the filling level in a bucket is at the limit of overflowing in the next lower bucket. Figure 2 shows a screw at the optimal filling point with the different buckets. When the water level is below the optimal filling point, the screw is operating in under-filling. Conversely, when the water level is above this point, the screw is in over-filling.

We define now the leakages present in ASG. The first one, is the leakage due to the gap between the blades and the trough $Q_f$. The second one, is the leakage $Q_{over}$ due to the flow over the screw’s core that appears when the screw operates in over-filling.

By neglecting the two flow leakages, the flowrate flowing through the ASG is equal to the volume of water evacuated in one turn of the screw multiplied by its rotational speed. Hence, we define the nominal flow $Q_{nom}$ by:

\[ Q_{nom} = N V_B \frac{n}{60} \]  

(2)
with $Q_{nom}$ in $m^3 s^{-1}$, $N$ the number of blades, $V_B$ the bucket volume in $m^3$, and $n$ the rotational speed of the screw in $min^{-1}$. The rotational speed of the screw $n_{nom}$ is then equal to:

$$n_{nom} = \frac{Q_{nom} \cdot 60}{N \cdot V_B} \quad (3)$$

The total flowrate $Q$ flowing in the screw is the sum of the nominal flow plus the both leakage. Then, we have:

$$Q = Q_{nom} + Q_l + Q_{over} \quad (4)$$

The mechanical power provided by the screw is equal to the product of the torque and the rotational speed of the screw:

$$P_{mec} = C \cdot \omega \quad (5)$$

with $C$ the torque provided by the screw in $Nm$ and $\omega$ the rotational speed of the screw in $rad s^{-1}$.

The turbine efficiency $\eta$ is then equal to:

$$\eta = \frac{P_{mec}}{P_{hyd}} \quad (6)$$

with $P_{hyd}$ the power provided by the fluid in $Watt$.

All the following experimental and numerical efficiencies are determined by equation 6.

3. Experimental device
An experimental device with laboratory-scale Archimedean screw has been elaborated to experimentally investigate its performance. This device enables to establish the turbine efficiency under a wide range of flow conditions and geometrical parameters. Thus, it is possible to modify the flowrate $Q$, the inclination of the screw $\beta$, the rotational speed $n$ and the downstream water level $h_{out}$. It should be noted that the upstream water level $h_{in}$ depends on rotational speed of the screw and flowrate. All these parameters are represented in table 1 and figure 4.

The device is installed in a 0.75 $m$ width and 5 $m$ length with outlet and outlet pipes of 0.15 $m$ diameter. The Archimedean screw, manufactured in Acrylonitrile Butadiene Styrene (ABS) with a 3D printer, is supported by two sealed ball bearings. Water is supplied at the
The fluid then flows through the screw and drives it in rotation thanks to the pressure exerted on the blades. Eventually, water escapes at the outlet located behind the adjustable weir that allows to control the downstream water level $h_{out}$. A magnetic flowmeter is used to measure discharge in the inlet pipe with a measuring accuracy of $\pm 0.5\%$. The water levels $h_{in}$, controlled downstream by ASG suction capacity and inflow discharge, and $h_{out}$ controlled downstream by the weir (cf. figure 4) are measured with dial gauges.

The gap between the screw and the through was measured with slip gauges at both ends of the screw for different radial positions. A maximal gap of 0.001 m and a minimal of 0.0006 m were found.

The screw axis is directly coupled to a DC motor (48 V, 418 W) which allows to control the rotation speed of the turbine. This motor is also used as measurement device. The rotational speed is thus proportional to the voltage and torque to electrical current. The torque delivered by the motor is defined by $C_{\text{motor}} = k_{\text{torque}}I$ and its rotational speed by $n = k_{\text{speed}}U$ with $k_{\text{torque}}$ the motor’s torque constant, $k_{\text{speed}}$ the motor’s speed constant, $I$ the electric current at the motor’s terminals and $U$ the voltage at the motor’s terminals. It is important to note that the values of $k_{\text{speed}}$ and $k_{\text{torque}}$ given by the constructor are independent of the motor efficiency in our range of rotational speed.
The torque provided by the screw is equal to:

\[ C_{\text{screw}} = C_{\text{motor}} + C_{\text{friction}} \]  

with \( C_{\text{friction}} \) the torque induced by the friction in bearings. This torque was determined experimentally by measuring the motor torque necessary for rotating the screw for different rotational speeds (cf. [7]). It appears that this torque increases linearly with the rotational speed as shown in the following equation:

\[ C_{\text{friction}}(n) = 0.000171 n + 0.046065 \]

As we use preloaded bearings, we expect that equation 8 does not change significantly with a screw partially or completely filled.

The power delivered by the screw is then determined by:

\[ P_{\text{screw}} = C_{\text{screw}} \omega_{\text{screw}} \]

with \( P_{\text{screw}} \) the power in W, \( C_{\text{screw}} \) the torque in Nm and \( \omega_{\text{screw}} \) the rotational speed in rad s\(^{-1}\).

Eventually, the screw efficiency is defined by:

\[ \eta = \frac{P_{\text{screw}}}{\rho g Q H} \]

with \( H = h_{\text{in}} - h_{\text{out}} \). As \( H \) is the geodetic head, it is thus assumed that the screw does not recover the kinetic energy of the fluid but only the potential one.

For each acquisition corresponding to one flow condition and one rotational speed, 1500 instantaneous values of rotational speed and torque are recorded in 15 s. An average value of rotational speed and torque is then obtained. These values are used to determine the screw efficiency.

4. Computational Modelling

In order to simulate the flow occurring within an Archimedean Screw, it is necessary to solve the 3D Navier-Stokes equations. To perform that, we use the open source CFD software OpenFoam (2.3.1). In our simulation case, we have to model an incompressible, turbulent and multiphase flow. Moreover, to take into account the rotation of the screw, it is necessary to use a dynamic mesh.

4.1. Turbulence and free-surface modelling

Most turbine applications including ASGs involve turbulent motions. It is thus necessary to take into account the turbulence phenomena to have a realistic flow simulation. There are several numerical methods, more or less accurate and time consuming, for turbulence simulations. The most commonly used methods are the Direct Numerical Simulation (DNS), the Large Eddy Simulation (LES) and the Reynolds Averaged Navier-Stokes (RANS). In this case, we choose to solve the Reynolds averaged Navier-Stokes equations (RANS) with a \( \text{SST} k-\omega \) turbulence model for closure. Smaller and larger eddies are all modelled with a Boussinesq eddy viscosity assumption. According to [10], this turbulence model is commonly used in CFD engineering and have wide applicability, is accurate, simple and economical to run.

It is necessary to accurately determine the evolution of free surface flow occurring within an ASG. The Volume Of Fluid (VOF) introduced by [11] is a powerful method for incompressible flows with two non-miscible phases like water and air. It uses cell’s filling to determine the time evolution of flow in a meshed area. This method was chosen because it is widely used and validated in different multiphase flows problems ([12] and [13]).
4.2. Meshing and boundary conditions

The computational mesh must be built very carefully in order to reach realistic and robust results. In this study, the mesh is built with OpenFOAM’s utilities. Thus, the main domain is created with blockMesh utility whereas mesh refinement around the screw, the trough and the weir are achieved using snappyHexMesh with 3D .stl files coming from the experimental device. Hence both numerical and experimental ASG have the same dimensions that are exposed in table 1. In this CFD study, the screw’s inclination is set at $\beta = 24^\circ$. Although the gap $s_{sp}$ between the trough and the screw is comprised between 0.001 m and 0.0006 m, we used the maximal gap. We then have in the CFD simulation $s_{sp} = 0.001$ m.

![Computational domain](image)

**Figure 5.** Computational domain

The computational domain, represented in figure 5, is composed of 5 million cells. To best simulate the gap leakage, a particular attention is paid for mesh refinement near the trough. A first refinement is achieved along the inner trough surface. The mesh located between the trough and the screw blade is then refined a second time. Eventually, we obtain three cells in the gap region which corresponds at approximately 0.3 mm width cells. Remark that a slight supplementary refinement in the gap region implies a huge increase in total mesh size, hence calculation costs. This model however gives fairly good results as presented in the following.

The screw rotation is simulated using sliding mesh with Arbitrary Mesh Interface (AMI). Thus, the main domain is divided into a fixed part and a rotating one. Both areas are separated by AMI surfaces. The rotating domain contains the screw and the trough. As we want to simulate an Archimedean screw turbine with fixed trough, we impose a zero velocity condition on the trough’s surface. Eventually, the rotational speed of the rotating part is then equal to the rotational speed of the screw. This means that the rotational speed of the screw is imposed and independent of the flow conditions.

The upstream boundary condition is a discharge and the downstream one is a pressure condition equivalent to a fixed water level.

In order to save computational time, the initial condition consists in a half water filling of upstream and downstream tank, air in the screw part. The initial upstream and downstream water levels are take from experimental results.
4.3. Turbine modelling

The turbine efficiency is determined by equations 9 and 10. We then have to determine, using CFD, the torque provided by the screw \( C_{\text{screw}} \) and the head \( H \). The values of the rotational speed \( \omega_{\text{screw}} \), the flowrate \( Q \) and of the flow properties are defined previously in the simulation parameters (cf. §4.2).

The torque computation is achieved with the OpenFOAM utility Forces. It is a post-processing tool that allows computing force, moment, lift and drag data. This tool determines the torque induced by the pressure exerted by the fluid on the screw \( C_p \) and the torque due to the fluid viscosity \( C_v \). The first one is a generator torque whereas the second one is a brake torque.

The elementary force induced by the fluid pressure is equal to:

\[
\overrightarrow{dF}_p = dA.P_f.\vec{n}
\]  

with \( dA \) an elementary surface in \( m^2 \) and \( P_f \) the pressure of the fluid exerted on \( dA \) in \( N.m^{-2} \) and \( \vec{n} \) the surface normal vector. The elementary moment is then obtained by vector product. The utility then integrates the elementary torque on the whole screw’s surface to obtain the total pressure torque.

The shear stress due to viscosity of the fluid is defined by:

\[
\tau = \rho.(\nu_t + \nu).\frac{\partial U}{\partial n}
\]

with \( \rho \) the fluid density in \( kg.m^{-3} \), \( \tau \) the shear stress in \( N.m^{-2} \), \( \nu \) the kinematic viscosity of the fluid in \( m^2.s^{-1} \), \( \nu_t \) the kinematic turbulent viscosity in \( m^2.s^{-1} \) and \( U \) the fluid velocity in \( m.s^{-1} \). A wall function is used to determine \( \nu_t \) and special attention is paid to the value of \( Y^+ \) on the whole screw’s surface. Indeed, in order to have the first node in the log-law layer, we always have values of \( Y^+ \) ranged between 30 and 200 as suggested in [14]. The utility Forces integrates the shear stress on the whole surface and then compute the brake torque \( C_v \).

Eventually, the total torque provided by the screw projected on its axis is equal to:

\[
C_{\text{screw}} = C_p - C_v
\]

The head \( H \) is determined, as defined previously, by \( H = h_{\text{in}} - h_{\text{out}} \). The value of \( h_{\text{out}} \) is specified in the initial conditions thanks to the outlet boundary condition. The value of \( h_{\text{in}} \) depends on the flow conditions and on the rotational speed of the screw. The free surface upstream the screw is given by the contour filter in Paraview trough a value of \( \alpha = 0.5 \). The value of \( h_{\text{in}} \) is then measured at the same location as in the experimental device.

Once the numerical simulation is converged, the values of \( C_{\text{screw}} \) and \( h_{\text{in}} \) are measured and then averaged over 6 ASG revolutions.

4.4. Convergence criteria

In order to establish whether or not the numerical simulation is converged, it is necessary to define convergence criteria. The first criteria deals with the mass balance averaged on six screw rotations that must be lower than 1 percent. Indeed the rotation of the last blade and the water volume of the last bucket create waves, hence oscillations of outlet discharge. Indeed, the mass balance applied on the whole system must tend towards zero. As the torque \( C_{\text{screw}} \) and the head \( h_{\text{in}} \) are necessary to determine the screw efficiency, the latest convergence criteria relates to the stabilization of these physical values also at 1 percent relative error around a mean value for 6 ASG revolutions. Eventually, we consider that the calculation has converged when all three criteria are respected.
5. Results and discussion

To study the ASG performance and to evaluate the results provided by the numerical simulations, two sets of measurements are done. Each time, the numerical results are confronted to experimentation. The first set is done with variable flow rates and fixed rotational speed. The second one is completed with constant flow rate and variable rotational speeds. It was shown that downstream water level $h_{out}$ influences directly the screw performance [15]. That’s why, all the following results are performed at optimal downstream level.

Figure 6 shows the water free surface in the ASG for a flow rate of $Q = 2.8 \, l.s^{-1}$ and a rotational speed of $n = 70 \, rpm$. In this case, we can see that the screw is operating in over-filling. Thus, the phenomenon of over-filling leakage is visible for this simulation (cf. figure 6). This phenomenon appears when the flow rate flowing in the screw is too high for a given rotational speed. An interesting point is that for the same flow conditions, the over-filling leakage occurs at the same locations in experiments and simulations. It is also possible to see the gap leakage on the simulation (cf. figure 6). Eventually, the visualization of the water free surface on figure 6 shows that the numerical simulation takes into account the different leakage losses.

Figure 7 exposes the theoretical torque provided by the screw depending on its radial position for $Q = 1.8 \, l.s^{-1}$ and $n = 70 \, rpm$. The screw torque $C_{\text{screw}}$ oscillates around an average value of 0.29 N.m with several peaks from which periodic patterns can be observed. Interestingly, a radial periodicity of $\Delta \theta = 360/N = 120^\circ$ induced by the three blades ($N = 3$) can be shown. Visualization of the screw torque evolution (figure 7) demonstrates that there are non negligible fluctuations. These fluctuations are mainly induced by the fact that the first blade and the last one are not subjected to the same fluid pressure depending on the circumferential position of the screw. From several simulations it appears that the amplitude of torque fluctuations increases when the screw filling decreases - not presented here. Moreover, a difference between maximal and average torque greater than 7% was found for low filling. Eventually, torque fluctuations are not negligible especially for short screws whose length is on the order of the outer diameter. The torque provided by the screw can be smoothed by increasing the screw length or the number of blades.
Figure 7. Torque evolution depending on the circumferential position of the screw for $Q = 1.8 \text{l.s}^{-1}$, $\beta = 24^\circ$ and $n = 70 \text{rpm}$.

Figure 8. Experimental and modeled efficiency depending on $Q/Q_{\text{nom}}$ for $\beta = 24^\circ$ and $n = 70 \text{rpm}$.

Figure 9. Experimental and modeled efficiency depending on $n/n_{\text{nom}}$ for $\beta = 24^\circ$ and $Q = 2.8 \text{l.s}^{-1}$.

Figure 8 represents the evolution of screw efficiency in function of $Q/Q_{\text{nom}}$ for $\beta = 24^\circ$ and $n = 70 \text{rpm}$. The maximal efficiency obtained is slightly above 80%; this is in close agreement with the results found in the literature [5, 16, 17]. These results demonstrate that the ASG is able to keep high efficiency for a $\pm 20\%$ variation of discharge around the optimal value. For highest value of flow rate, screw efficiency decreases because of the leakage over the screw core. It is assumed that the efficiency decreases below $Q/Q_{\text{nom}} = 0.8$ because of the gap leakage which becomes more important from this point. We notice close agreement between numerical and experimental values of efficiency. For too low flow rate, simulation overestimates the ASG efficiency. This may be due to an underestimation of the gap leakage and it shows the importance to refine the mesh between blades and trough.
Figure 9 shows the screw efficiency depending on \( n/n_{\text{nom}} \) for \( \beta = 24^\circ \) and \( Q = 2.8 \text{ L.s}^{-1} \). Results demonstrate that there is only one optimal speed which corresponds approximately here to \( n/n_{\text{nom}} = 1.05 \). Performance degradation for low speed is due to the over-filling leakage. Conversely, when the rotational speed is too high, the screw is under-filled. Gap leakage and frictions forces will then be predominant. For low speed, we notice close agreement between simulations and experimentation. It is particularly interesting to note that the CFD give almost the same optimal value of \( n/n_{\text{nom}} \). For highest values of speed, screw efficiency is overestimated. It is assumed that friction forces and gap leakage are both underestimated by the CFD.

6. Conclusion
The main goal of this paper was to develop a new methodology based on CFD for predicting ASG performance. A laboratory scale screw allowed testing screw performance for different functioning points was presented. The complex flows occurring within this screw has been modelled thank to 3D simulations with a dynamic mesh technique. We show first that gap leakage phenomenom and over filling leakage can be simulated with the CFD approach. The comparison between numerical and experimental efficiencies is fairly good with a relative error in the order of a few percent around the optimal functioning point. There is still room for progress for severe under-filling flow conditions especially for gap leakage calculation. The approach is yet validated for a large range of flow conditions and paves the way for testing various screw geometries which would be too expensive to test experimentally and particularly for real scale screws.

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