Experimental investigation of the factors affecting Archimedes screw generator power output

S. Simmons
University of Guelph
50 Stone Road East
Guelph, Ontario
Canada

K. Songin
University of Guelph
50 Stone Road East
Guelph, Ontario
Canada

W.D. Lubitz
University of Guelph
50 Stone Road East
Guelph, Ontario
Canada

Abstract

Currently there is little literature characterizing the performance of Archimedes screws used as generators. Experimental data is required to validate the existing modelling approaches. This study sought to address this need by gathering a wide range of experimental data for 16 unique laboratory scale Archimedes. Screws varied by the number of flights, flighted section length, inner and outer diameters, and pitch. Measurements were taken at specific, repeatable values of flow rates, rotation speeds and outlet water levels. Certain sets of the lab screws had all but one geometric parameter in common, allowing for a direct comparison of the effect of that parameter on screw performance. A large set of data was collected and used to create a comprehensive database, characterizing screw performance as a function of various design parameters. It was found that the most important factors effecting the power output and efficiency of the screw are, in descending order, the flighted section length, the outlet submergence level, the pitch, the diameter ratio, and the number of blades. The experimental data was also compared to performance predictions by a current performance model. Reasonable consistency and agreement were found between the model and the experimental results, showing that the model is generally valid within a reasonable range of accuracy.

1. Introduction

The Archimedes screw is a technology that has been in use since antiquity, primarily as a pumping device. It was popularized by Archimedes of Syracuse in the 3rd century BCE [1], and some evidence suggests that it may have been used as early as the 7th century BCE in the Assyrian Empire to water the hanging gardens of Babylon [2]. Recently, the technology has found use as an Archimedes Screw Generator (ASG), a microhydro system used for generating electricity at sites with low head and moderate flow rates. The ASG is an emerging approach to hydroelectric power generation that has many advantages when compared to other microhydro technologies, including low environmental impact, low installation and maintenance costs and its ability to operate efficiently in a range of flow and head conditions not suitable for most conventional turbines.

Screw turbines are widely known for being eco-friendly; they are usually retrofitted at old mill flood control dam sites to operate as run-of-river systems. The benefits of this installation technique are that there is no reservoir formed at the upstream end of the installation, and the conditions in the waterway will not change. This reduces negative environmental impacts associated with reservoir flooding, such as loss of wildlife habitat, and the impacts of mercury methylation [3]. ASGs have been in use since antiquity: their simple, robust design has allowed them to perform well throughout the ages. As a hydro turbine, the screw allows for sediment, nutrients, other debris and even fish to pass through the hydropower plant with little to no harm [4]. Conventional turbines require a fine screen to filter any incoming debris to protect the turbine blades from damage. Evidence suggests that this can cause a drastic change in the ecology of riverways downstream of conventional hydroelectric installations [5]. ASGs often operate in a run-of-river configuration, reducing environmental impact due to modified downstream flow regimes. ASGs operate at highest efficiencies in low-head, moderate flow rate environments – up to 80% efficiency can be achieved in these conditions [6], efficiencies which are competitive with other microhydro turbines.

The ASG literature includes models to predict power and efficiency based on flow and geometric parameters, including accounting for major losses. The past few years have seen several attempts to model various aspects of ASG operation, including inlet losses [7], and overall performance [8][9][10]. Nuernbergk (2012) has authored a very good German-language textbook that provides a comprehensive treatment of ASGs as well [11]. Generally, the models described in the literature are mechanistic, parametric models based on a combination of first principles and empirical relationships applied to geometry and operating conditions of specific screw turbines. For example, the Lubitz et al (2014) model predicted the idealized performance of ASGs, and was based off of experimentation with lab-scale screws. This model was later amended to include such losses as internal friction, gap and overflow leakage, filling losses, and losses due to outlet conditions [12]. Recently, there has been interest in using computational fluid dynamics (CFD) simulation to evaluate the performance of an ASG. For example, Dellinger (2016) presented a CFD model of an ASG system, which successfully simulated the entire screw and surrounding civil works [13]. However, there still remain several significant gaps in our collective ASG understanding, particularly within the English literature. All of the current models have been shown to be consistent with available experimental data sets; however, it is arguable that since available data to validate models with was small (generally consisting of data from only one or a few laboratory-scale screws) none of the existing models have been exposed to rigorous testing and validation. More detailed and robust data sets would allow more extensive validation.

1.1. Screw geometry

The Archimedes screw is a set of helicoid planes fastened to a central cylindrical tube. A shaft runs through this assembly, which is then mounted between an upper and lower bearing. The screw is usually housed in a closely-fitted enclosure, often called a trough. There are two ways in which the screw is enclosed: either the blades and a continuous circular trough are fixed together and the entire assembly rotates, or the screw rotates within a fixed trough with a small gap between the two assemblies to allow rotation: the latter configuration is by far the most common. When operating as a generator, water enters the top of the ASG and fills the volumes between adjacent flights of the screw. One of these volumes of water is often called a “bucket” [1]. Fig. 1 illustrates the main geometric parameters of a screw: outer diameter (D_o), inner diameter (D_i), pitch (S), number of flights (N), and the length of the screw’s flighted section (L). The angle of inclination (β) and the rotational speed of the screw (ω) are also key design factors. Fig. 1 also shows a single, mostly full bucket. The variable f is used to indicate non-dimensional fill height within a bucket.

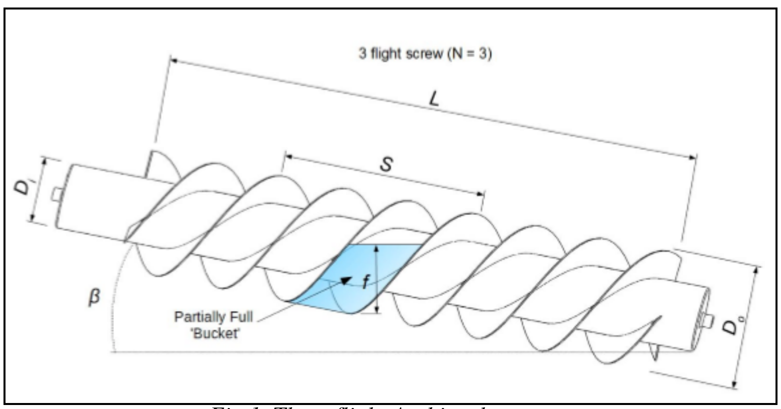

Fig 1. Three flight Archimedes screw.

Calculating the volume of water in a bucket is complicated by the water volume being an inclined helicoid bisected by a horizontal water surface. The calculation of a static bucket volume is discussed by Rorres (2000). At a high the geometric parameters in Fig. 1 can be used to numerically integrate the volume of water within a single bucket.

The main goal when designing a hydroelectric installation is to find an optimum configuration for a specific site that generates the most power from the available flow. In the case of ASG design, the parameters for the diameter ratio, length, inclination angle, number of blades and pitch must be simultaneously optimized in order to find a combination that produces the most hydropower. Waters (2015) used a CFD analysis to compare output torque (and thus, power) of a range of ASGs with varying diameter ratio, length, pitch, and number of blades, and discussed the main impacts that each geometric parameter had on the overall performance of the screw turbine [14]. For each simulation, the outer diameter and rotational speed of the screw remained constant at 0.2 m and 50 RPM, respectively. The head was varied between 0.2 m and 1.2 m, and the results of the simulations were compared to some experimental data, and the idealized performance model of Lubitz, Lyons and Simmons (2014) for confirmation of observed trends. Waters found that the diameter ratio had a large effect on performance. As the diameter ratio decreased (i.e. the inner diameter decreased with a constant outer diameter) the torque increased. This may occur since the smaller inner diameter allows for a larger volume of water in each bucket, leading to higher pressures affecting screw rotation. It was also suggested that there is an optimal value of the screw's length. For a specific site with a defined head, as screw length increases the angle of inclination decreases, and so the longer screw has more distance that it can utilize to extract energy from the water; however, there would also be more frictional losses prevalent in a longer screw. It was also found that as the pitch ratio increased, the torque increased in the screw. This may be because the volume in each individual bucket can increase as the pitch increases – a similar trend to the diameter ratio. Finally, it was found that the number of blades had the least effect on the torque output for the screw. It was found that more flights create less torque, since the thickness of the added blades will occupy some of the volume in the screw that could potentially be used for water. The following section will discuss the experimental setup and procedure, and the results of this testing will be compared to the conclusions of Waters (2015).

2. Methods

2.1. Archimedes screw laboratory

All experiments were conducted in a purpose-built Archimedes screw testing system at the University of Guelph (Fig. 2). The testing system was designed to allow for easy changing of screws and operating parameters, including flow rate and screw rotation speed. Measurements include torque, rotation speed, fill depth and basin depths throughout the system. Table 1 gives the dimensions of the 16 unique laboratory-scale Archimedes screws tested during this study.

In practice, the ASG under study is situated between upper and lower basins. An upstream reservoir is equipped with two 15.2 cm Cipoletti weirs to verify the flow rate through the screw. A variable speed pump moves water from the lower basin to the upstream reservoir, where it then flows into the upper basin and through the screw back into the lower basin. The water flow rate can be precisely controlled by changing pump settings. Volume flow rate through the system is measured by an Omega FTB740 flow meter located within the long, straight return pipe. Since all water pumped to the upstream reservoir must also flow through the screw, the flow rate measured by the meter corresponds to the flow through the screw.

The depths of the water in the upper basin, lower basin, and upstream reservoir were all measured with Keller America pressure-based depth gauges located within stilling wells, and visually verified using rulers. The outlet water depth was adjusted by changing the height of a control weir located in the lower basin between the screw outlet and the pump intake (Fig. 3). The rotational speed of the screw was controlled by a variable frequency drive (VFD) motor. The VFD maintains a constant rotation speed regardless of torque, and so functions as both speed control and energy dissipater. Screw rotational speed is measured with a non-contact magnetic switch-based tachometer that is continuously recorded, and confirmed with a handheld optical tachometer.

Table 1. Dimensions of University of Guelph laboratory-scale Archimedes screws.

Screw	OD (cm)	ID (cm)	S (cm)	L (cm)	N	ID/OD	S/L	L/S
#1	31.58	16.83	44.45	121.92	3	0.53	0.36	2.74
#2	31.62	16.83	31.75	121.92	3	0.53	0.26	3.84
#3	31.67	16.83	25.4	121.92	3	0.53	0.21	4.8
#4	31.69	12.7	31.75	121.92	5	0.4	0.26	3.84
#5	31.66	12.7	31.75	121.92	4	0.4	0.26	3.84
#6	31.62	12.7	31.75	121.92	3	0.4	0.26	3.84
#7	31.62	12.7	31.75	63.5	3	0.4	0.5	2
#8	31.57	12.7	31.75	40.64	3	0.4	0.78	1.28
#9	31.64	10.16	31.75	121.92	3	0.32	0.26	3.84
#10	31.61	10.16	44.77	52.07	4	0.32	0.86	1.16
#11	37.8	16.99	30.2	46.89	4	0.44	0.64	1.55
#12	37.69	16.89	30.4	61.39	4	0.44	0.5	2.02
#13	37.69	16.79	30.51	94.69	4	0.44	0.32	3.1
#14	38.2	16.99	38.3	46.61	4	0.44	0.82	1.22
#15	38.1	16.79	38.2	61.7	4	0.44	0.62	1.62
#16	38.61	16.89	38.3	94.89	4	0.44	0.4	2.48

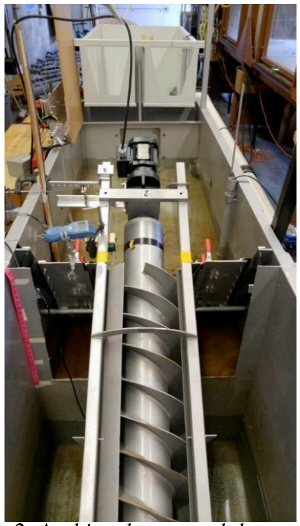

Fig 2. Archimedes screw laboratory.

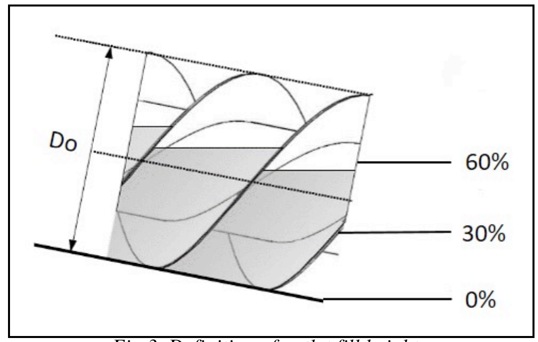

Fig 3. Definition of outlet fill heights.

The water depth within the buckets of the screw were also measured. A static pressure port, in the form of a small flush hole, was put in the bottom of each screw's trough. The static port was directly connected to a pressure sensor (Omegadyne PX309) located adjacent to the port to minimize the length of the pressure line. Static pressure ports were placed in the longitudinal centre of the trough since the buckets in the middle of the screw are always fully formed. It was then assumed that the depth of the measured bucket had a similar volume and depth to the other buckets of the screw.

The measurement of torque about the screw was made using an Omegadyne LC703-25 load cell connected using ball joints to the frame of the ASG and a moment arm. The moment arm from the central axis of the screw turbine to a point perpendicular to the line of action of the load cell was 26.5 cm. The load cell connection to the VFD was the only off-axis support of the screw, so that all moments about the rotation axis were supported through the load cell. If the load cell were removed, the VFD would free to rotate along with the screw. This configuration meant that the screw torque could be computed by multiplying the measured load cell tension force by the moment arm. Load cell alignment, moment arm and the load cell calibration curve were regularly confirmed during testing.

Water depths, rotational speed, flow rate, pressure and torque were all recorded using a National Instruments USB-6009 data acquisition device (DAQ) connected to a computer running a custom-written NI LabView program. All sensors were sampled at a frequency of 1000 Hz (to provide adequate resolution for the rotation speed measurements) and results were time-averaged over a one-minute interval. Mechanical power at the shaft was calculated by multiplying the measured torque and rotational speed. Hydraulic head and then efficiency were calculated using the additional measurements of flow rate and water levels in the upper and lower basins.

2.2. Experimental procedure

Experiments were carried out on each screw to collect data for ranges of flow rate, rotational speed, and screw outlet water level. For each of the 16 screws, data was gathered for their performance under five different flow rates (6, 8, 10, 12, and 14 L/s), six different rotational speeds (20, 30, 40, 50, 60, and 80 RPM), and three different outlet depths (0%, 30%, and 60%). Measured data was compiled in a large database to aid in characterization of the lab-scale ASG performance.

For each test, the angular projection of the screw was used along with the depth measurements in the lower basin to set the outlet conditions described in Fig. 3. The flow rate, fill height, and lower basin height were set to the desired combination, and allowed to reach a stabile state – it usually took two minutes to reach equilibrium. Once stability was reached, each sensor was run and recorded for 60 seconds. Each experimental run yielded approximately 30 measurements, which were then added to the database.

3. Results and discussion

The results of the testing were used to investigate the effect of five main parameters on the performance of the lab-scale ASGs: outlet fill height, flighted length, number of flights, diameter ratio, and screw pitch. The effect of each of each parameter is discussed below.

3.1. Outlet fill height

As mentioned above, the outlet fill height was set to values of 0%, 30%, and 60% of the outlet of the screw filled (shown in Fig. 3). The general trend found in the data was: as the fill height at the outlet increased, the power generated decreased, but the efficiency of the screw increased. This is illustrated in Table 2.

Table 2. Average power and efficiency for all screws with respect to outlet fill height.

Screw	Average Power (W) *Outlet Fill Height*			Average Efficiency *Outlet Fill Height*		
	0%	*30%*	*60%*	*0%*	*30%*	*60%*
1	36.2	35.5	29.4	0.598	0.674	0.653
2	32.5	34.9	30.7	0.578	0.635	0.658
3	34.7	34.2	29.8	0.544	0.603	0.625
4	36.0	35.7	30.2	0.572	0.645	0.653
5	36.6	35.9	29.5	0.579	0.646	0.647
6	36.1	35.1	30.2	0.577	0.636	0.655
7	18.6	18.2	14.2	0.432	0.545	0.545
8	12.7	12.0	8.7	0.385	0.544	0.540
9	36.9	36.0	30.4	0.599	0.671	0.670
10	14.9	15.0	10.5	0.414	0.544	0.515
11	16.8	15.9	9.3	0.455	0.591	0.462
12	21.2	20.4	12.6	0.516	0.653	0.503
13	29.9	28.7	21.9	0.560	0.634	0.574
14	15.8	14.3	6.4	0.461	0.613	0.327
15	22.4	21.4	13.7	0.604	0.754	0.642
16	32.0	29.9	20.8	0.623	0.710	0.573

The decrease in power as the outlet fill height increases can be explained as a result of decreasing head – or a decrease in available power. As the outlet fill height increases, and the inlet fill height remains constant, the power available in the water decreases, and the number of buckets available to turn the screw is limited.

Though the screw is generating less power at the outlet, it is doing it more efficiently. The efficiencies for this analysis were calculated using the available hydraulic head, taking the decrease of head into account. This increase in efficiency could be explained by a backpressure that develops at the outlet of the screw – when the screw empties with a 0% outlet fill height, the lower buckets are less full because they tend to spill out of the screw. It was noticed that the 60% outlet fill height screws did not exhibit this response. Since it was shown that the 60% outlet condition consistently provides screws with the maximum power efficiency, and best represent real life ASTs, only the 60% outlet fill heights will be analysed for the remainder of this paper.

3.2. Length

Three sets of three screws with all parameters in common but their flighted length were used to compare the length of the screw to power production. Theoretically, longer screws should be able to generate more power, more efficiently, since more buckets of water can form and there are more faces to interact with the water to generate power. There is a trade-off though: as the length increase, so do the frictional losses in the system. So, it is suggested that individual flow rate and head conditions will have their own optimal screw length.

The three sets of screws available for the length analysis are as follows – with each screw set geometrically identical except for their differing flighted lengths.

- Screws 6, 7, and 8 have three flights, a 31.75 cm pitch, and a diameter ratio of 0.40
- Screws 11, 12, and 13 have four flights, a 30.4 cm pitch, and a diameter ratio of 0.44
- Screws 14, 15, and 16 have four flights, a 38.3 cm pitch, and a diameter ratio of 0.44

For each of these sets, the average power and efficiency for the tested flow rates and rotational speeds across each screw was compared to the changing length. The results of these tests are shown in the plots of Fig. 4.

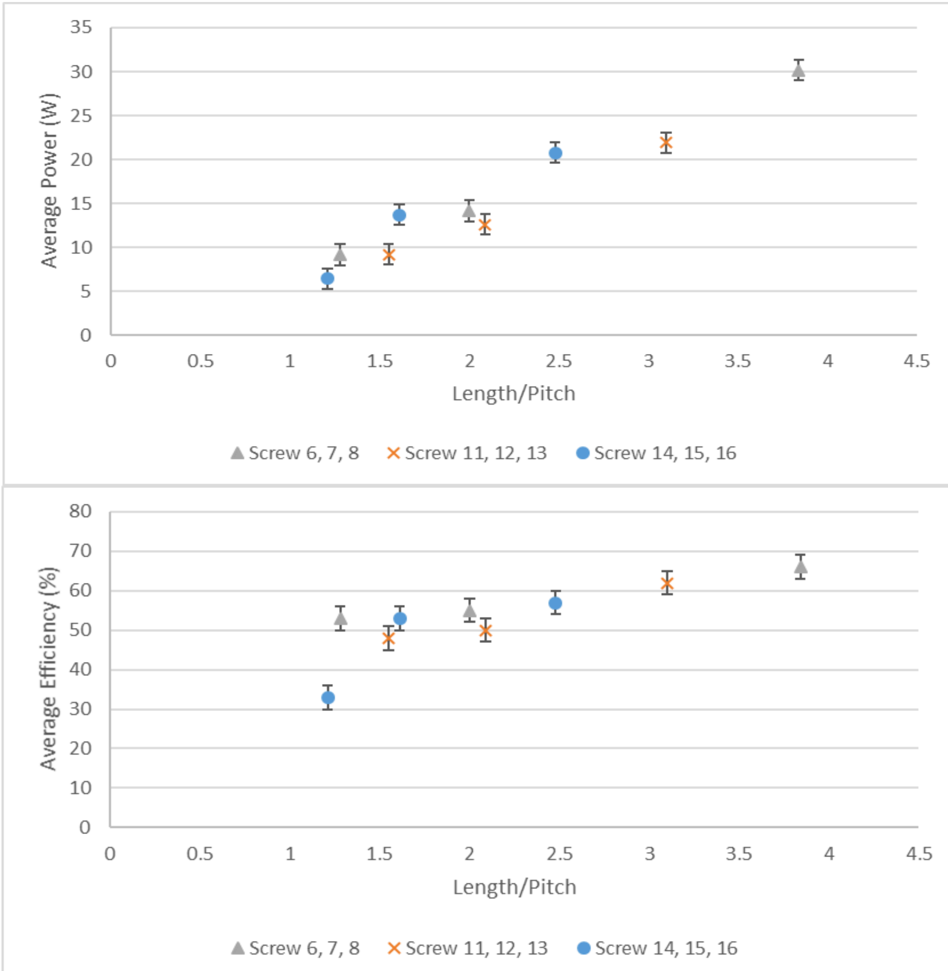

Fig 4. Power (top) and efficiency (bottom) as a function of flow and rotation speed for screw sets

The results of the testing seemed to agree with the theory discussed above. It was found that, on average, an increase in length of 20 cm, in the laboratory-scale screws, corresponded to an increase of 5 W in power, and 3.5% in overall efficiency. It was found that, at high rotational speeds and low flow rates, the power and efficiency in the screw approaches 0 W and 0% efficiency, and in some cases, reached negative values. This phenomenon can occur when the turning of the screw is mainly supported by the VFD; in this case, the water in the screw is essentially being pumped by the screw, and not aiding in the ASG's rotation.

The previous section showed that as the water at the outlet increases, and shortens the effective length of the screw, less power was produced, and this result was repeated here. As the screw was physically shortened, less power was created - in this case, however, the efficiency of the screws generally increased with length. In this data set, efficiency began to level off at different points for each screw. This may be an indication of the maximum length conditions of the screw, as the frictional losses start to show a more significant effect on power generation. Frictional losses from the water interacting with the screw and the weight on the bearings are just a few potential losses that are introduced with increased screw lengths. To further define potential losses in the screw, the number of flights in the screw is compared.

3.3. Number of flights

To reiterate, the number of flights in a screw is the number of independent helical surfaces in the screw, or the number of surfaces that are "started" at the ends of the screw. Since the number of flights (or blades) in the screw directly relates to the number of buckets in the screw, it was theorized that an increase in number of flights (and thusly the number of buckets) should correspond to an increase in power production and efficiency. Since a screw with more flights has more overall turns, each bucket is smaller in volume, however there is an increase in surface area that is in contact with water. Rorres (2000) predicted greater power generation with an increased number of flights; the prediction was based on the more flighted screw's ability to hold a higher over volume of water, because a narrower inclined helicoid volume can have a higher fill level without overflowing [1]. Although, in a real screw, the additional surfaces would also be expected to produce larger power losses due to internal friction between the water and the rotating surfaces. As well, there will be a noticeable loss of available volume for water to fill due to the thickness of the material of any additional blades. A set of three screws were used to compare the effect of the number of flights on power production in ASGs. Screws 4, 5, and 6 are geometrically identical except that they have five, four, and three flights, respectively. Fig. 5 demonstrates the power and efficiency contours for the screw set at the range of rotational speed and flow rate values repeated for the database.

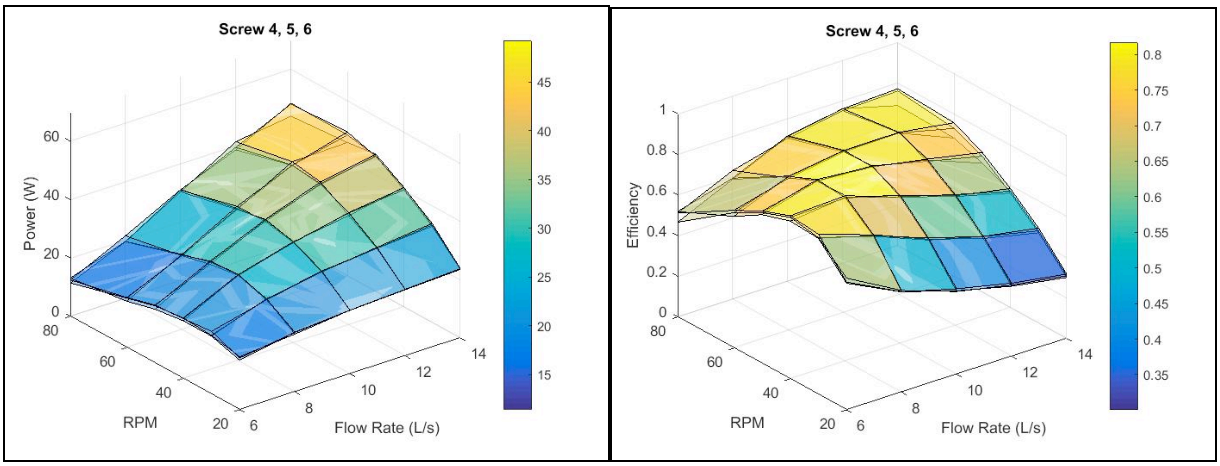

Figure 5. Power (left) and efficiency (right) as a function of flow and rotation speed for screws 4, 5, and 6

Experimentation suggested that a change in the number of flights of and ASG does not have a significant impact on screw performance. The power and efficiency of each of the three screws remained nearly constant across the range of tested flow rates and rotational speeds. Each screw occupies a different layer in the contour plots shown above; however, the layers are nearly indistinguishable across each screw because the results are so similar. The average power and efficiency of screws 4, 5, and 6 were all around 30 ± 1.4 W and 65 ± 3.1%, respectively, matching well within experimental uncertainty. A significant difference in power and efficiency was only noticeable at maximum values of power and efficiency, where three flights was associated with slightly lower values when compared to screws 4 and 5.

Overall, the number of flights did not have a significant effect on power production in the laboratory-scale ASGs. It is believed that any increase in power production due to the addition of flights was approximately offset by added frictional losses that were much more noticeable in the small-scale setting. Potential power losses associated with additional flights include: inlet impact losses, larger bearing losses, and more internal fluid friction losses. It is suggested that the number of flights may have a more significant impact in larger-scale ASGs, since large-scale systems with higher flow rates should have proportionally lower internal fluid friction losses, when compared to the laboratory-scale system.

3.4. Diameter ratio

The diameter ratio of an ASG refers to the ratio of inner diameter over the outer diameter (D_i/D_o). Screws 2, 6, and 9 were geometrically identical except for their inner diameter (and therefore, diameter ratio). Screw 2 had the largest inner diameter, with a diameter ratio of 0.53, followed by screw 6 which had a ratio of 0.40, and Screw 9 with a 0.32 ratio – these dimensions are shown in Table 3 along with the average power and average efficiency of each screw. It was theorized that a larger outer diameter and a smaller inner diameter would correspond to higher power production, since it would allow for a larger volume of water to form in the screw's buckets.

Table 3. Average power and efficiency for Screws 2, 6, and 9.

S	OD	ID	S	N	ID/OD	L	Power (W)	Efficiency (%)
#2	31.62	16.83	31.75	3	0.53	121.92	27.98 ± 1.29	67 ± 3.1
#6	31.62	12.70	31.75	3	0.40	121.92	27.61± 1.39	67 ± 3.1
#9	31.64	10.16	31.75	3	0.32	121.92	27.86 ± 1.40	69 ± 3.2

The table showed an increasing efficiency with decreasing diameter ratio – or increasing bucket size. Fig. 6 shows the contour layers of the three screws, comparing the rotational speed and flow rate to the power production and efficiency, respectively. It can be seen in the figures that the contour layers are indistinguishable on the power plot – the power production values for each test match for each screw within experimental uncertainty, similarly to the results of the number of flights, above. Interestingly, the efficiency was distinguishable still; it appears that the best efficiency for the set of screws, in descending order, is Screw 9, Screw 6, and Screw 2. It should be noted that the flow condition of 14 L/s was not included in the figure because the data for Screw 2 was incomplete for this case.

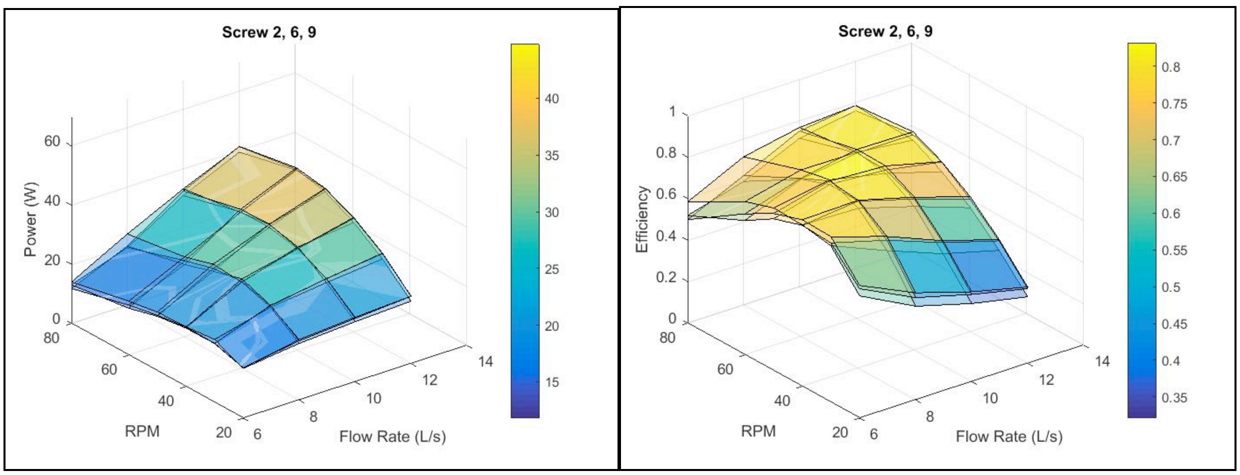

Figure 6. Contour power and efficiency for Screws 2, 6, and 9.

The average power and efficiency did not show a particular trend, and generally tended to match across the different diameter ratios within experimental uncertainty. Nonetheless, it should be noted that a screw with a smaller diameter ratio (i.e. Screw 9 in these trials) will form larger buckets of water, reducing contact between the water in the screw and the central rotating shaft of the screw. At flow rates below 8 L/s and rotational speeds above 50 RPM, the water in the buckets of Screw 9 did not contact the central rotating cylinder of the screw, and only interacted with the blades and trough of the system. It is suggested that, in low flow situations, a smaller diameter ratio would reduce frictional power losses by eliminating the frictional loss due to the interaction of water and the central shaft of the screw. However, within experimental uncertainty, the effect of the diameter ratio on ASG performance was found to be very minimal.

3.5. Pitch

To compare the last parameter, a set of screws that were geometrically identical except for their pitch, were tested. Screw 1, 2, and 3 were all 121.9 cm long, with 3 flights, a diameter ratio of 0.53, and pitches of 44.5 cm, 31.8 cm, and 24.5 cm, respectively. The pitch is a measurement of the distance between each turn of a screw's flights taken lengthwise down the screw. As such, a larger pitch corresponds to larger individual buckets in the screw, and a smaller pitch means the screw will have a larger quantity of smaller buckets. Theoretically, the screw with the smaller pitch, and thusly more buckets, should produce the most power, however, with every new bucket introduced in the system, there will be additional internal fluid friction losses. Table 4 illustrates the results of the testing for this set of screws.

Table 4. Average power and efficiency for Screws 1, 2 and 3.

Screw	OD	ID	S	N	ID/OD	L	Power	Efficiency (%)
#1	31.58	16.83	**44.45**	3.00	0.53	121.92	26.43 ± 1.36	66± 3.1
#2	31.62	16.83	**31.75**	3.00	0.53	121.92	27.98 ± 1.39	67 ± 3.1
#3	31.67	16.83	**25.40**	3.00	0.53	121.92	27.37 ± 1.34	63 ± 2.9

The data showed that Screw 2 was the most powerful and efficient ASG tested in this group; it had a pitch that was finer than Screw 1, and coarser than Screw 3. The reason that Screw 2 performed the best is because it offers a good trade-off between the benefits of a coarse and fine pitch. The coarse pitched screw has fewer buckets, but each bucket holds a larger volume of water. A larger volume of water will have a higher static pressure to aid in power production, but, even though each bucket is larger in a coarse pitch screw, there is less overall surface area interacting with the buckets to convert the pressure into a rotational force. The fine-pitched screw will have narrower buckets that may fill up to a higher point before overflowing, as Rorres (2000) noted, and will have more contact surfaces; but this also allows for frictional forces to take a more significant effect on performance.

Based on the results of this laboratory-scale testing, it is suggested that a pitch ratio (the ratio between the pitch and the outer diameter) of 1.0, as Screw 2 has, seems to be the best option for ASG power production and efficiency since it gives the advantages of reduced internal fluid friction losses, and a sufficient number of buckets to produce power.

4. Conclusions

This study sought to create and analyse a database for the power production of 16 unique laboratory-scale ASGs under consistent operating conditions of flow rates, rotational speeds, and outlet fill heights. The flow, rotational speed, and outlet fill heights were held at repeatable values for the range of screws to allow for direct comparisons of the effect of the outlet fill height, screw length, number of flights, diameter ratio, and the pitch on power generation, and screw efficiency.

It was found that the length of the screw had the largest effect on screw performance; an increase in screw length corresponded to and increase in power and efficiency within the screw geometries tested. The way in which the power and efficiency results plateaued, however, suggests that there is a point in which longer screws introduce more losses than benefits. At this point, the internal friction losses in the screw start to dominate, and the performance of the screw is worsened. The next most important parameter was found to be the outlet fill height. As the fill height increased to account for 60% of the outlet of the screw submerged, the power decreased, but the efficiency increased. The decrease in power was due to the decrease in head as the outlet filled up, and thusly a decrease in available power. The screw was generating less power more efficiently in these cases, suggesting that a longer screw should be designed for sites, so that the outlet of the screw is submerged to about 60%.

The pitch of the screw was the last parameter to show significant impact outside of experimental uncertainty. A balance was found for the pitch in order to optimize screw performance. The screw pitch must be set to be coarse enough to reduce internal fluid friction losses, but not so coarse as to have too few buckets for power generation. The diameter ratio results suggested that as the ratio decreased (i.e. internal diameter decreased) the power production and efficiency would increase; however, the trend in the data are within the realm of experimental uncertainty, and cannot be accepted directly as proof of this relationship. The number of flights in the screw had similar results to the diameter ratio testing. There was no definitive relationship that the results suggested until the screw was put under high flow conditions and high rotational speeds. Under these conditions, the data showed that the three-flighted screw underperformed compared to the similarly performing four and five flighted screw.

Altogether, this study carried out a significant amount of testing over a wide range of unique lab-scale ASGs. The results of the testing suggest that the ideal ASG will have the right combination of outlet fill height, length, number of flights, diameter ratio, and screw pitch so that it will be long enough to have the largest screw buckets possible while still having enough buckets available to successfully turn the screw.

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The Authors

Scott Simmons graduated with a Bachelor of Engineering in the co-op Mechanical Engineering program at the University of Guelph (Canada). He worked as a mechanical designer throughout his work terms; designing gauging systems and components for automated machine cells. He is currently a Masters of Applied Science (MASc) candidate in the School of Engineering at the University of Guelph. His work has a focus on modelling the power output and efficiency of Archimedes screw generators. He has been involved with the Archimedes screw generator group since 2012, having served as a research assistant during his undergraduate studies.

Kathleen Songin graduated with a Bachelor of Science in Physics, with a minor in Mathematical Science, from the University of Guelph (Canada) with Honours. During this time, she worked in the Mathematics department, developing a simulation model called the compuGUT. She also worked under the Physics department, analysing depth selective Mossbauer spectroscopy. She has recently completed a Masters of Applied Science (MASc) in Environmental Engineering at the University of Guelph. She completed her thesis on an experimental investigation of Archimedes Screw Turbines (AST), studying the effect of rotation on water level in ASTs. In addition, comparing the power output and efficiency for a variety of laboratory-sized screws at fixed speeds and incoming flow rates.

William Lubitz is an Associate Professor in the School of Engineering at the University of Guelph (Canada). His primary research interests are microhydro power systems, assessment of wind and solar energy resources and modelling of microclimates in greenhouses and other passive solar structures. His research group has been studying Archimedes screw generators since 2011.