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Review and evaluation of Archimedes screw pump design guidance

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Scott Simmons, Lian Miller, and William David Lubitz†
School of Engineering, University of Guelph
50 Stone Rd E, ON, N1G 2W1, Canada

Abstract

The Archimedes screw (also termed "hydrodynamic screw" or "Archimedean screw") has been used for a variety of industrial applications since roughly 700 BCE. Its most common historical implementations were for land drainage/reclamation, irrigation, and to convey mixed media (i.e., granular solids, suspended solids, etc.). The screw has also found use as a hydroelectric generator since the earlier 1990s. Due to its robust design, screw pumps are still commonly used for land drainage and wastewater conveyance. Design methods for Archimedes screw pumps (ASPs) are not well documented in the literature. The leading text offering engineering design guidance for ASPs was compiled by Nagel (in 1968) and offers mostly empirical design guidance based on data and experimentation that usually do not appear to be further documented in the literature. Most design techniques are based off a paper presented by Muysken in 1932. The paper uses many simplifications and empirical models based on undocumented experiments, so there is a need to evaluate and update modelling techniques to determine optimized design of ASPs using modern computational techniques. This study investigates the literature of Archimedes screw pumps, and presents and summarizes current modelling techniques. Experimental methods and data are presented and compared to current performance prediction models from the literature. Results are analysed and used to suggest areas for further research and improvement in current engineering design guidance.

1. Introduction

Archimedes screws (also termed "hydrodynamic screw" or "Egyptian screw") have been use in various applications for more than two millennia. When implemented for pumping applications, it is often called an Archimedes screw pump (or ASP). Archimedes screws are a helical array of one or more blades wrapped around a central cylindrical tube. The technology is very similar to an auger, however when implemented as a pump, it is common for the inner cylinder to be proportionally larger (on the order of half the total diameter) (Figure 1). The screw may be operated as a pump or a hydroelectric generator; in either implementation, volumes of water are trapped in the interstitial spaces between the blades forming what is commonly called a "bucket" of water [1].

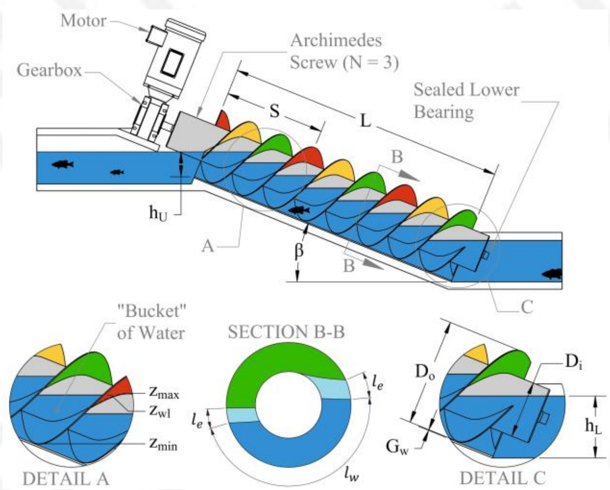

Figure 1: Archimedes screw geometry [2].

† Associate Professor, corresponding author, wlubitz@uoguelph.ca

Figure 1 demonstrates the geometric and operational parameters of an Archimedes screw for use as a pump. Screw geometric parameters include the outer diameter (D_o), inner diameter (D_i), pitch (S), number of blades (N), flighted length (L), gap width (G_w), and inclination angle (β). Operating parameters include the upper- (h_U) and lower water levels (h_L), bucket water level (z_{wl}), rotation speed (Ω), and flow rate (Q).

While operating as a pump, a motor is usually used to supply rotational mechanical power about the screw axis of rotation. The screw turns and draws water into the low end of the screw, forming a fully enclosed bucket volume. The rotation of the screw causes the volume of water to translate upward along the screw, and discharge at the top end of the screw. Archimedes screws may also be used in the reverse direction for hydropower generation, by converting fluid pressure into rotational mechanical energy. A few screw installations make use of the screw's ability to function reversibly for both pumping and generating electricity. The Olympic white water canoeing course in Stockton-on-Tees (UK) has four parallel screws that pump water to supply high flow rate to the course during competition. When high flows are not required, the screws are used to generate electricity [3].

Due to their simple, robust design, screw pumps can move fluids, granular solids, and fluids with suspended solids (i.e., wastewater) and aquatic fauna (i.e., fish, eel, etc.) safely. Studies have shown that Archimedes screws allow safe passage of fish and other aquatic wildlife in either their hydropower generation [4]–[7] or pumping configuration [8], [9]. As such, they are an ideal pumping scheme for land reclamation, wastewater treatment plants, and aquaculture farms [10].

2. Historical Context

Evidence suggests screw pump technology had been used for irrigation as early as the 7th century BCE in the Neo-Assyrian Empire of Sennacherib (704-681 BCE) [11]. During Sennacherib's reign, screw pumps were used to irrigate the gardens in the Assyrian capital of Nineveh. Some historians have postulated that these may have been the Hanging Gardens of Babylon [11]. The Neo-Assyrian Empire likely introduced the screw pump as an agricultural technology through diplomatic relations or after its annexation of Egypt in 671 BCE [12].

Archimedes of Syracuse – famous Greek mathematician, philosopher, astronomer, physicist, and namesake of the technology – might have learned about screw pump technology, rediscovered, or reinvented it during his studies in Alexandria circa 287 BCE [1]. Archimedes popularized the device. It is said he implemented the device: to irrigate the Nile delta [11], as the first documented naval bilge pump [13], and to operate as a ship propulsion device [14]. In 212 BCE, Syracuse and the majority of Sicily was annexed by the Roman Empire. Shortly after the conquest of Egypt, the technology seems to have been modified by the Romans to more resemble the present-day Archimedes screw pump [15]. Screw pumps subsequently found widespread use throughout the Mediterranean, particularly for draining Roman mines during the Roman Imperial period (circa 27 BCE to circa 5th century CE) [16]. After the end of this period, further use of the screw pump appears to be undocumented in the western world for many centuries.

Screw pumps likely continued operation in ore mines across the Iberian Peninsula during the Visigothic and Muslim regimes, and after the Reconquista [17]. There is evidence in some early Islamic literature that screw pumps found continued use in the Middle Ages [16]. The famous poetic collection known as the *Maqamat al-Hariri* [18] was later illustrated by artists like Yahya ibn Mahmud al-Wāsiṭī; one of his illustrations depicts an Archimedes screw pump irrigating some gardens [19]. Evidence suggests screw pumps in the mines of modern day Aljustrel (Portugal) were operated and improved during the transitions of power in the Iberian Peninsula [20]. Evidence suggests that Portuguese traders introduced a crank-driven Archimedes screw to Japan in 1637 CE [17], suggesting that the technology may have found global use (Figure 2).

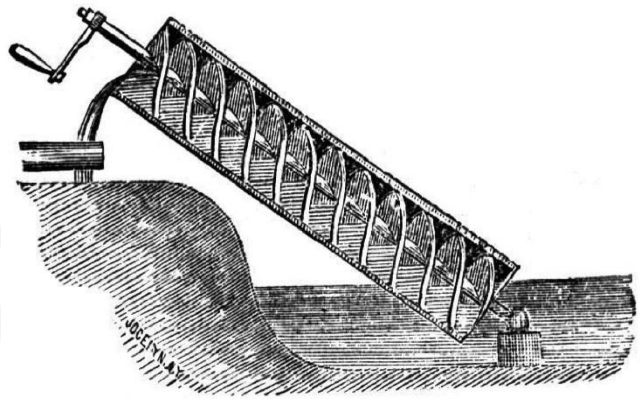

Figure 2: hand crank Archimedes screw pump [1].

Screw pumps are well-documented in surviving Renaissance literature [21]–[25]. During the Dutch Golden Age (late 1500s to late 1600s), Archimedes screws were implemented in polder mills for land drainage [26], [27] due to their ability to move large volumes of water across a low head [15]. A polder mill consists of a screw pump powered by a windmill. Early screw pump polder mills could pump a large volume of water to a head of roughly 1.5 metres; modern steel screw pumps can bridge heights of 4 to 5 metres [28]. In about 1930 CE, in the Netherlands alone, there were roughly 300 screw pumps in use for lowland drainage and flood control; the vast majority were wind-powered [15]. Due to its widely adopted use in the Netherlands, Dutch manufacturing companies lead the way in the manufacturing and development of screw pumps [15].

The Archimedes screw found other uses in the early 19th century. In 1829, Austrian engineer Josef Ressel planned to use a modified Archimedes screw to develop a propulsion mechanism for ships. The design was modified by English inventor Francis Smith and in 1839 a ship named "Archimedes" was launched using an early form of the modern-day propellor [15]. The screw pump was not commonly used during the establishment of public water supply systems in the early 19th century due to its low delivery head. Other technologies, such as the piston pump, were available to support larger delivery heads. Notably though, with modern materials, the screw pump has become a popular option for wastewater treatment facilities [15].

In the modern context, the Archimedes screw has found applications as a water- [29] and wastewater pump [30], a conveyor for grains [31], [32], a fish ladder [33]–[35], a drive mechanism for amphibious vehicles [36], an injector for plastic moulding [37], land reclamation [28], heart valve replacements [38], and for hydropower generation [10]. The remainder of this paper focusses on the Archimedes screw pump for use in water- and wastewater conveyance.

3. Design Details and Modelling

Likely due to the age of the technology, screw pump design is empirically-based and screw pumps are largely designed by experience [14]. Documentation of screw pump design and performance is very limited in the literature. This study documents the state of the art for the Archimedes screw pump technology – it should be noted that the vast majority of research and development was performed more than 40 years ago in the previous century. For example, one of the pioneering works for screw pump design was written by Muysken as a Dutch-language article in 1932; it served as the foundational mathematical work for screw pumps. The paper suggests some empirical design guidelines that are often reiterated in the literature [1], [15], [39]. Early modern experimentation was conducted by Addison [29] to address the lack of information on screw pump performance; little has been added to the English-language literature since Addison's experiments in 1929.

The Screw Pump Handbook [39] seems to be the most complete design manual for screw pumps; it was published in an English translation and used many of the principles discussed by Addison (1929) and Muysken (1932). The Handbook documents detailed design instructions based on practical experience, and suggested heuristic relationships to simplify design work. The work seems to be based on a large set of experiential data; however, none of the data is presented.

There have only been a few notable English language publications since the Screw Pump Handbook in 1968. Rorres [1] presented a mathematical analysis that optimized screw pumped design to achieve the maximum filling of the screw. The optimization used a few idealised assumptions that made the results less practical (i.e., assuming infinitely thin blades). In practise, the blade thickness limits screw efficiency – as a hydropower generator, the optimum number of blades for an Archimedes screw seems to be between 3 and 5 [40].

There is little literature on screw pumps in the past few decades. Monserrat et al. [41] published an article that explored the cost-savings associated with a variable inclination screw pump. The only other contribution to the English-language literature for Archimedes screw pump performance and design was a laboratory experiment on a small screw pump by our research group at the University of Guelph [42] to address the lack of literature and data on ASPs, and to examine parallels between the screw generators and pumps.

3.1 Flow Rate (Q) and Gap Leakage Flow Rate (Q_g)

Nagel and Radlik (1988) noted at their time of publication that maximum discharge rates of installed screw pump stations were on the order of 3 m^3/s [15]. However, a diesel-powered screw pumping station in Kinderdijk (Netherlands) was built more recently with four parallel, large screw pumps that are able to move roughly 4.2 m^3/s of water each [43]. In the reverse orientation, the largest active Archimedes screw generator powerplant uses a screw that converts roughly 14.5 m^3/s of flow into electrical power [44]. It is suggested that screw pumps could be made at this size, though in some cases multiple, smaller screws would be installed instead. Parallel-screw operation was historically beneficial for pumping stations that were required to have variable discharge rates; screws could be turned on or off to meet flow or drainage demands. Modern variable speed drive systems allow for screws to deliver variable flow rates simply by increasing or decreasing rotation speed. In cases where a high flow demand is required across a low head, it still may be advantageous to use multiple smaller-diameter screws to meet installation requirements since very short screws are subjected to proportionally higher hydraulic losses. Ultimately, the use of an individual screw or parallel screws at a pumping station are site-specific.

Since the delivery flow rate of screw pumps are often the most important design criteria, a reliable flow rate model is integral to the design of a screw pump station. Muysken (1932) presented the following formula to determine the delivery rate of a screw pump [45]:

$$Q = q \cdot \Omega \cdot D_o^3 \tag{1}$$

where the flow rate is dependent on the rotation speed Ω (rev/min), outer diameter D_o (m), and a dimensionless geometric parameter (q). The parameter q is determined based on the geometry and fill level of water in the screw buckets.

The dimensionless geometric flow parameter (q) is a function of screw design variables and numerical integration of the screw's geometry. Effectively, the discharge (Q) of the screw is calculated as product of the volume of water in a bucket of the screw, the number of blades of the screw, and the rotation speed of the screw. Simplifying that expression allows for Eqn. 1. Effectively, the dimensionless coefficient (q) represents the dimensionless volume of water per pitch of the screw.

The value for q was calculated by Muysken (1932) using integration across polar coordinates. Using Figure 3, the water level in the bucket was estimated by the angles M-O-P (denoted α_1) and Q-O-R (denoted α_2). The angular estimates are shown overlayed on Section B-B of Figure 1. The water levels of Figure 1 were generated with computer aided design (CAD) software; the inaccuracies incurred by Muysken's assumption that the water level follows the angles are noticeable in the figure.

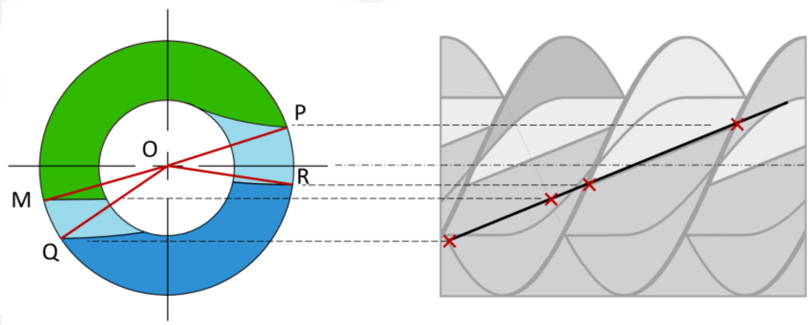

Figure 3: Muysken's characterization of bucket water level (adapted from [45]) overlayed on CAD representation from Figure 1.

Muysken (1932) discretized the blade's cross-section with radial disks of thickness z and presented the calculation of q as

$$q = \frac{\pi\,\sigma}{360 \cdot z} \sum \frac{r}{D_o} \cdot (\alpha_1 + \alpha_2) \tag{2}$$

where the flow parameter is a function of the diameter ratio (σ), the thickness of each discretized disk (z), the radial location of the discretized disk (r), the outer diameter (D_o), and the water level angles α_1 and α_2.

Since it is a difficult calculation to perform, Muysken (1932) provided a lookup table that could be used to determine q based on the diameter ratio $\sigma = D_i / D_o$ (ranging from $\sigma = 0.4$ to 0.65) and the inclination angle ($\beta = 22°$ to 40°). The diameter ratio is the ratio of the inner diameter to the outer diameter; along with the pitch ratio (ratio of pitch to diameter), it is a valuable dimensionless parameter to characterize the geometry of Archimedes screws. For intermediate values, linear interpolation is suggested across this range [45]. The lookup table was expanded by Nagel and Radlik (1988), a representation of it is shown in Table 1; note that the table is only valid for three-bladed ($N = 3$) screw pumps.

Table 1: q-values for three-bladed Archimedes screw pumps [45].

D_i / D_o	$\beta = 22°$	$\beta = 26°$	$\beta = 30°$	$\beta = 33°$	$\beta = 35°$	$\beta = 37°$	$\beta = 40°$
0.4	0.00507	0.00460	0.00393	0.00354	0.00324	0.00295	0.00247
0.45	0.00503	0.00460	0.00405	0.00365	0.00334	0.00304	0.00255
0.5	0.00500	0.00460	0.00417	0.00376	0.00343	0.00313	0.00262
0.55	0.00479	0.00436	0.00406	0.00366	0.00335	0.00309	0.00259
0.6	0.00457	0.00417	0.00395	0.00356	0.00326	0.00299	0.00250
0.65	0.00435	0.00381	0.00383	0.00347	0.00315	0.00287	0.00242

Comparisons to experimental measurements showed that actual flow rates usually exceeded calculated values by 10 to 12% [45] and in some cases up to 30% [46]. Experiments later conducted by Nagel (1959) agreed with these conclusions, showing exceedance of actual measurements by 11 to 18% of the calculated flow rates. So, an updated flow rate model was suggested based on a nominal exceedance in flow capacity of 15% [15]:

$$Q = 1.15 \cdot q \cdot \Omega \cdot D_o^3 \quad (3)$$

As the relationship suggests, the flow rate calculation provides approximate results with a wide range of associated uncertainties. Therefore, this relationship should be a focus of model improvements.

Screw pumps must have a small gap between the rotating screw and the fixed enclosing trough to allow the screw to turn freely. Some water will leak through this gap during screw operation, reducing pumping efficiency. The rate of volumetric leakage between the trough and the screw blades has also been modelled to help with performance predictions. The gap leakage rate (Q_g) depends on the width of the gap (G_w). Nagel (1968) suggested the gap width could be calculated with respect to the outer diameter of the screw [39]:

$$G_w = 0.0045\sqrt{D_o} \quad (4)$$

the gap leakage flow rate may then be calculated using the following empirical relationship [39].

$$Q_g = 2.5 \cdot G_w \cdot D_o \cdot \sqrt{D_o} \quad (5)$$

In both equations, all lengths are in units of meters, and Q_g has units of m^3/s. This model is dependent on an empirical leakage loss constant of 2.5 and the geometry of the screw; it does not properly account for the rotation speed or the fill height of water in the screw buckets. This is another area of screw pump design that will need improvement. It is noted that the leakage loss in a wastewater pumping station is expected to be minimized due to gap-reducing deposits present in the effluent during operation [15].

3.2 *Screw Size (D_o, D_i, and S)*

Pump stations are often designed to deliver a specified desired flow rate. So, the relationship shown in Eq. 1 is often rearranged to determine the required size of screw to supply the desired flow rate [39]:

$$D_o = \left(\frac{Q}{q \cdot \Omega}\right)^{1/3} \quad (6)$$

It is noted that the geometric flow parameter q is dependent on the outer diameter, inner diameter, pitch, and inclination angle, so sizing the outer diameter may be an iterative process [15]. Nagel and Radlik suggest using the nominal flow rate (Eq. 1) and not the adjusted flow rate (Eq. 3) when sizing the screw so that the pumping station may supply more than its rated delivery flow rate if required [15].

The inner diameter and pitch of the screw must be known to use Eq. 6. Screw pumps are often described by their diameter ($\delta = D_i / D_o$) and pitch ratios ($\sigma = S / D_o$). It is common for screws to have diameter ratios between 0.45 and 0.55 and pitch ratios between 1 and 1.2 [39]. While it was suggested empirically that the optimal diameter ratio should be between 0.45 and 0.5 [15], a theoretical analysis of Archimedes screw pumps suggested the optimal diameter ratio may be closer to approximately $\delta = 0.54$ for screws with any size or number of blades [1]. It is suggested that the pitch ratio should be maintained at $\sigma = 1$, using an heuristic model, to prevent any additional costs of manufacturing [39]. Recent Archimedes screw generator (ASG) experiments suggest this value corresponds to good performance [47]; though ASGs operate in the reverse direction, this seems to support Nagel's (1968) claims.

Rorres (2000) found the theoretical optimal diameter ratio as a balance of geometric parameters. The pitch ratio (S/D_o) and diameter ratio (D_i/D_o) were compared to a "volume-per-turn" ratio, which was effectively the theoretical, non-dimensional volume of water in each of the screw's buckets. Therefore, the optimal pitch and radius ratios were simplified to a three-degree of freedom problem. The "volume-per-turn" ratio was defined as the "volume ratio" (v) divided by the number of blades (N). The volume ratio is

$$v = \frac{4 \cdot V_b}{\pi D_o^2 S} \quad (7)$$

where v is the volume ratio and V_b is the volume of water in the bucket (found using numerical integration). Rorres' (2000) analysis showed that the pitch ratio was consistently optimal when $S = D_o$. When considering the optimal diameter ratio, the denominator of Eq. 7 could therefore be considered a constant value given that the pitch (S) was always roughly equal to the outer diameter (D_o). Considering that the cross section of a bucket is similar to an annular cylinder (cf. Figure 1, Section B-B), the numerical integration to solve the bucket volume depended on the height of water in the bucket, the thickness of the annular cylinder ($t = (D_o - D_i)/2$), and the width of the bucket (which was dependent on the pitch). The height of water in the bucket increases as the inner diameter of the screw increases. However, the thickness of the annular region of the bucket will decrease as the inner diameter increases. Therefore, a balance exists in which the inner diameter is small enough to maximize the annular thickness and large enough to maximize the water height in the bucket. As mentioned above, this balance was found to roughly be at $\delta = 0.54$ [1].

For reference, the screw shown in the diagram of Figure 1 has a pitch ratio of approximately $\sigma = 1$, and the diameter ratio is close to $\delta = 0.5$.

3.3 Number of blades (N)

Theoretically, it was shown that the addition of blades increases the efficiency of screw pumps [1]; however, that conclusion was drawn after making a few simplifying assumptions, including neglecting the thickness of the screw blades and friction losses between water and adjacent blade surfaces. Nagel and Radlik (1988) suggest that screw pumps should be installed with 2 or 3 blades, and that 1-bladed screw pumps have many added inefficiencies [15]. Experimentation on ASGs has suggested that screws operate best with 3, 4, or 5 blades but the increase in operational efficiency with the addition of blades may be counteracted by an increased cost of manufacturing. It was suggested that screw generators should be designed with either 3 or 4 blades on a site-specific basis [47], and this is likely the case with screw pumps as well.

3.4 Inclination Angle (β)

Muysken (1932) suggested the most favourable inclination angle for screw pump design was 26° – a value corresponding to the highest mechanical efficiency in his studies [45]. However, it was noted that Muysken's research focussed on the use of screw pumps for dewatering and land reclamation – applications that have low delivery head requirements. At sites that require a large delivery head, it may be beneficial to install screw pumps at steeper inclination angles to save material costs even though inefficiencies increase [15]. The plot below shows the change in flow rate with respect to the inclination angle as a percentage of the flow rate associated with an inclination angle of $\beta = 30°$ (Figure 4).

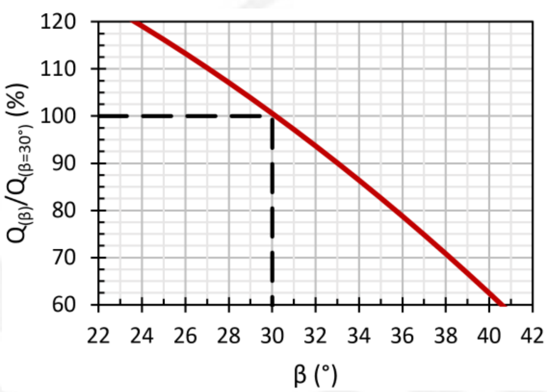

Figure 4: approximate flow behaviour with respect to inclination angle (adapted from [15]).

The considerations for steep and shallow inclination angles generally relate to cost, geometry, and hydraulic effects.

In general, screw pumps with steeper inclination angles are shorter and less expensive to manufacture and install. There are, in turn, less cost in civil infrastructure and screw components, as well as a smaller pumping station footprint for steeply inclined screws. A shorter screw also has less surface area for hydraulic friction to act on its blades, and less buckets along its length, so friction loss is proportionally smaller. However, since there are less screw buckets along the shorter length, the impacts of the inlet and outlet losses are proportionally higher, leading to an increased variation in torque and a proportional increase in hydraulic losses. The variation in torque manifests itself as consistent oscillations in torque as the screw buckets' drain and fill at the outlet and inlet during operation. The steeper inclination also causes the screw's buckets to support less volume of water; so, a steeper screw may do a similar amount of work to pump a proportionally smaller flow.

Screw pumps with shallow inclination angles are longer and allow for a higher volume of water in their buckets and can thusly support higher flows while operating at lower rotation speeds. Due to their length, the inlet and outlet losses are proportionally smaller and the system's torque oscillations have lower magnitudes. However, a shallower inclination angle also increases the wetted area of water in the screw's buckets. That, coupled with the fact that a shallower inclination requires the screw to be longer, makes the hydraulic friction loss proportionally much higher in longer screws. Since the screw is longer, the cost of manufacturing and installation proportionally increases. As well, the footprint of a less-inclined screw pump plant is proportionally larger than a steeply inclined screw pump plant.

It is suggested that the inclination angle should be determined on a site-specific basis to balance the cost of manufacturing and the benefits of increased mechanical efficiency.

3.5 Rotation Speed (Ω)

Perhaps the most well-known contribution of Muysken's research was his empirically determined relationship for the maximum recommended rotation speed of a screw pump. The "Muysken limit", as it is sometimes called, is

$$\Omega = \frac{50}{\sqrt[3]{D_o^2}} \tag{8}$$

where Ω is the maximum recommended rotation speed (rev/min) and D_o (m) is the outer diameter of the screw (cf. Figure 1). Based on previous research, the authors suggest that Eqn. 8 becomes increasingly inaccurate for proportionally small screw pumps (i.e., $D_o \approx 0.4$ m or less) [47]. In fact, Muysken (1932) plotted his relationship in comparison to experimental data and previously developed speed selection relationships (Figure 4); the unsuitability of all relationships for small screws are evident in all of the curves asymptotic approaches to the vertical axis.

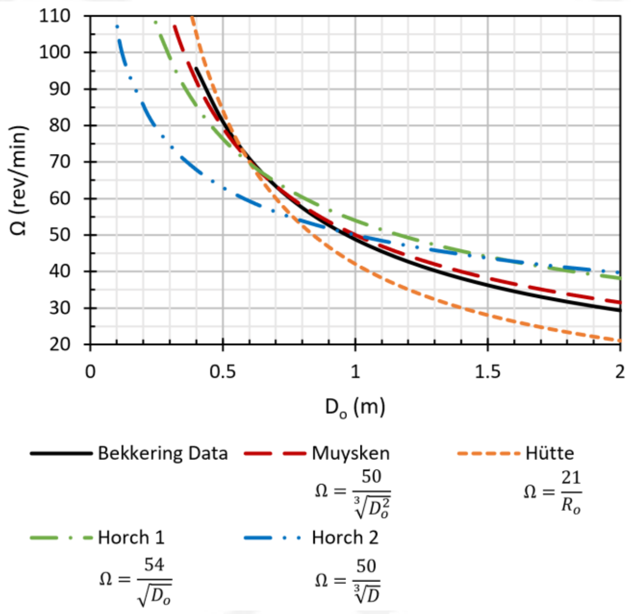

Figure 5: suggested speed as a function of screw outer diameter (adapted from [45]). The plot shows relationships from various sources in the literature [45], [46], [48].

The Muysken relationship is the most commonly implemented; it is suggested to be appropriate for use between 18 and 92 rev/min [15]. Recent experimentation with a laboratory-scale screw (D_o = 0.316) showed that rotation speeds between 30 to 60 RPM yielded optimum efficiencies under a variety of conditions. This suggests that the relationships shown in Figure 5 are only appropriate for industrial sized pumping applications.

3.6 Water Levels (h_U and h_L)

Modern steel screw pumps can bridge a height of H = 4 to 5 metres [28] with efficiencies similar to other pumping technologies. The overall delivery head at a screw pump station is the distance from the upper (receiving basin) water surface to the lower (supply basin) water surface (cf. Figure 1). The upper and lower water level are often described in non-dimensional terms of upper (ψ_U) and lower (ψ_L) screw submergence levels. The lower submergence level may be calculated as:

$$\psi_L = \frac{h_L}{D_o \cos\beta} \tag{9}$$

The lower submergence level is a critical parameter in screw pump performance. More recent models suggest the optimal lower submergence level for ASPs may be calculated as [49]:

$$\psi_L' = \frac{D_i + D_o}{2D_o}\sqrt{1 - \left(\frac{S\tan\beta}{\pi D_i}\right)^2} \tag{10}$$

The optimal lower submergence level is dependent on the geometry and operating parameters of a screw pump, but in screws with common configurations it may be between 50% and 80%. Recent experimentation seems to show agreement with this range of values [42].

The upper submergence level may be calculated by substituting h_L for h_U in Eq. 9. The upper water level is often $\psi_U = 0$ in wastewater applications and will only have significant impacts on screw pump performance at proportionally high levels (i.e., $\psi_U > 0.6$) [42]. However, if the upper water level significantly exceeds this level, efficiencies begin to decline rapidly as water from the upper basin starts to flow backwards through and over the screw.

If the upper and lower water level are known, the overall delivery head of the pump station may be determined with Eq. 11.

$$H = h_g + h_U - h_L \tag{11}$$

In Eq. 11, h_g is the geometric (or centre-to-centre) head of the screw; it is calculated as:

$$h_g = L\sin\beta \tag{12}$$

3.7 Flighted Length (L)

The flighted length of an Archimedes screw pump may be determined iteratively since it is dependent on the inclination angle. Lower inclination angles often correspond to higher flow rates and efficiencies, but also correspond to a longer screw; thus, increasing the cost of the installation. The opposite can be said for steeper inclination angles. In practice, a balance between manufacturing cost and performance is required as discussed in Section 3.4.

For a given inclination angle, Eq. 12 may be rearranged to determine the required length of the flighted section of the screw (Eq. 13).

$$L = \frac{h_g}{\sin\beta} = \frac{H - h_U + h_L}{\sin\beta} \tag{13}$$

3.8 Mechanical Efficiency (η)

Practical pump efficiencies range from 65% (small diameter screws) to 75% (large diameter screws) [50]. It is suggested that with careful design and site-specific optimization, screw pumps may operate at up to approximately 85% percent efficiency including electrical system losses, based on precedent observed in screw generator operation [51]–[53]. The wide range in efficiency estimates is largely due to the lack of documentation and guidelines for screw pump development, and represents another important area for further investigation.

4. Worked Example

In this section, the current state-of-the-art modelling techniques will be implemented. An example has been adapted from Nagel and Radlik (1988) to show one way in which the empirical and heuristic models may be used to design a simple screw pump.

Example: a pumping station is required for dewatering a polder. It is required to pump a maximum discharge of 240 L/s to a height of 1.2 m. Design the screw pump and determine its maximum operating speed.

To start, the nominal flow equation (Eqn. 1) can be rearranged to solve for the diameter of the installation (Eqn. 6). Note that the flow correction coefficient ("1.15" in Eqn. 3) is not applied in this equation so the screw pump can over-supply if required; this allows the system to be more robust.

The dimensionless flow coefficient (q) must be determined first to solve for the diameter. A value for q may be taken from the lookup table (Table 1). To use the lookup table, a value for the inclination angle, number of blades, and the diameter ratio must first be selected. An inclination angle of β = 26° was selected since Muysken suggests it is ideal for dewatering pump stations. Rorres' optimal theoretical value of diameter ratio (σ = 0.54) was selected. Finally, the number of blades was set to N = 3 since it is the most common orientation for screw pumps. Using the lookup table, we can linearly interpolate between values of q = 0.0046 and 0.00436, yielding a value of q = 0.00441.

β =	26°
$\sigma = D_i / D_o$ =	0.54
N =	3
q =	0.00441

Before the outer diameter is calculated, a value for the rotation speed must be selected. According to Muysken, the maximum rotation speed is dependent on the outer diameter of the screw pump (cf. Eqn. 8). So, Eqn. 8 can be rearranged and substituted into the diameter equation (Eqn. 6) to solve for the diameter with respect to only the dimensionless flow coefficient and the required nominal flow rate:

$$D_o = \left(\frac{Q}{50 \cdot q}\right)^{3/7}$$

Performing the calculation yields and outer diameter of D_o = 1.037 m. Now, the maximum rotation speed may be calculated using Eqn. 8; yielding a maximum speed of 48.8 rev/min.

D_o =	1.037 m
Ω =	48.8 rev/min

It is a good idea to check that the dimensionless geometric flow parameter, rotation speed, and outer diameter actually yield the required discharge by using Eqn. 1. Performing the calculation, a nominal flow rate of Q = 240 L/s was determined, showing that the values are reasonable.

Q =	240 L/s

Next, the length of the screw will be determined to address the required head of 1.2 m. For a dewatering station, we can set the upper water level to h_U = 0 m; though this requires the screw to do more work than necessary to pump the water to the desire height, it does eliminate the need for a backflow valve/control system.

h_U =	0 m

Next, the optimal lower water level was calculated. Using an interpretation Muysken's relationship (Eqn. 10), the optimal lower submergence level can be determined. However, values for the screw pitch must first be determined – Rorres' suggestion that the optimal pitch ratio is roughly 1 was used. The inner diameter was also a requirement of the calculation. The inner diameter was calculated quickly since the outer diameter and diameter ratio were already known. Note that the inner diameter is often rounded to the nearest available standard steel tube size.

$\delta = S / D_o =$	1
$S =$	1.037 m
$D_i =$	0.560 m

Eqn. 9 was then rearranged such that the dimensional form of the lower water level could be determined.

$$h_L = \psi_L \cdot D_o \cos\beta$$

The optimal lower submergence and corresponding lower water level were found to be:

$\psi_L =$	73.75%
$h_L =$	0.6875 m

The head equation (Eqn, 11) was then rearranged to solve for the geometric head. The geometric head is a function of the screw's length, so once it was known, it could be rearranged to determine the length the screw must be to meet the head requirement. The length was found to be:

$L =$	4.306 m

So, to support a required discharge of 240 L/s over a height of 1.2 m, the following screw was designed using current, empirically based modelling techniques.

$D_o =$	1.037 m
$D_i =$	0.560 m
$S =$	1.037 m
$\beta =$	26°
$L =$	4.306 m
$N =$	3
$\Omega =$	48.8 rev/min

In fact, based on Muysken (1932) and Nagel's (1968) analysis, it is likely that this screw may actually be able to support a maximum discharge of up to 15% higher than calculated, e.g.

$$Q = 1.15 \cdot q \cdot \Omega \cdot D_o^3 = 0.276\ m^3/\text{s}$$

Depending on the application, having the capability to provide more flow than necessary may be helpful. However, an optimal design should have more certainty so that costs and energy expenditure can be minimized. The empirical and heuristic models presented in this section are a good start to screw pump design, but there is substantial room for improvement in model accuracy.

5. Conclusions

The main goal of this study was to review the current state-of-the-art design guidelines for Archimedes screw pumps. The authors' research will continue to address the lack of performance data in the English-language literature of ASPs. Data may then be used to make informed updates and design suggestions, and to develop mathematical models to optimize Archimedes screw pump design in subsequent research.

In a similar research program investigating Archimedes screw generators for hydropower production, the authors conducted laboratory experiments, field experiments on operating Archimedes screw powerplants, and used that data to validate a computational fluid dynamic (CFD) model; the CFD model was then used to accurately approximate screw generator performance for any geometry or size of screw installation, under any operating conditions [47]. This allowed for better characterization of screw generator power production without having to build or prototype a screw for every data point. The authors propose to use a similar process to better characterize Archimedes screw pump performance.

The data gathered from this research project will be used to develop more accurate and robustly evaluated design tools and mathematical models, with the overarching goal of using those models to improve the design of Archimedes screw pumps. Design optimization will be crucial in the near future. Archimedes screw pumps are often used for dewatering low-lying or flood-prone areas; in a climate impacted future dewatering technologies will be integral for draining low-lying regions.

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