Lancaster University
Faculty of Science and Technology
Department of Engineering

Low Head Hydropower

A performance improvement study for Archimedes Screw turbines using mathematical modelling

Ilias K. Kotronis

MSc by research Thesis

Supervisor: Professor George A. Aggidis

Lancaster, UK, September 2016

Declaration

The author declares that this thesis has not been previously submitted for award of a higher degree to this or any University, and that the contents, except where otherwise stated, are the author's own work.

Signed:

Date:

Abstract

The aim of the present MSc by research is to study the low head hydro power technologies, focusing on the use of Archimedes Screw, as hydro turbine, for electricity production and suggest potential design improvements with proper use and installation guidelines, intending to increase the produced energy per annum. The novelty introduced by the present research is the development of a methodology for the overall estimation of the performance for a hydro power station using Archimedes Screw turbine by identifying the loss mechanisms and minimize the losses produced from each one. As improvement was considered not only the nominal efficiency increase, but also the increase of the efficiency in relatively high and low flows and the exploitation of an extended range of flow and head, aiming in any case to maximize the produced electricity.

The method used was the estimation of the produced power and the power losses calculated by a mathematical model which was developed in Matlab, taking into consideration the geometrical values of the turbine, the flow, the water level upstream and downstream, the speed and the efficiency of the secondary devices (gearbox etc). A serious effort for a CFD simulation was made. Unfortunately, the large number of elements and the large simulation time made in combination with the limited available time and resources made impossible the extraction of any results.

Since no test rig was available and also there was no time for significant experimental work during one year duration MSc, manufacture's data for efficiency and hydrological conditions were used from a small hydro power station with Archimedes Screw turbine to be able to validate the model. The same station was used as a case study for this research, for which were suggested potential improvements.

As a result, the estimated turbine's efficiency curve has a percentage variation from the curve provided by the manufacturer between -5.65 % and 3.47 %, and the turbine's power curve between -0.46 % and 7.59 %. As a conclusion, it seems that the model can predict the efficiency and power with an acceptable accuracy for a preliminary level study of such a turbine. In addition, the maximum efficiency of a well designed and installed Archimedes Screw turbine seems very promising. However, as a free surface device it has a number of parameters with effect in the creation of losses which are limited in the known impulse and reaction hydro turbines.

Keywords

Archimedes Screw, Low Head Hydropower, Numerical Modelling, Hydraulic Machinery, Hydro Turbines, Renewable Energy

Acknowledgments

To begin with, I feel obliged to thank my academic supervisor Professor George Aggidis for his help and support during this year. The visit to a hydro power station with Archimedes Screw turbine would not be possible without him. The information of that visit were used to make the MSc realistic and adapt the study to an existing need, in the technical direction of finding solutions for real problems. Also, as an ex-industry person he had a more practical opinion in various areas, providing a less academic and more realizable perspective on the project.

I would like to give special thanks to Mr. David Mann, Managing Director of the MannPower Consulting Ltd who dedicated that morning on November to welcome us in one of the hydro power station he had set up, answer all of our questions and share thoughts and ideas. The information provided through our discussion and after were valuable for the next steps, not only for the validation, but also to exactly determine the topic of the project and realize how these turbines work and which are their advantages and disadvantages.

I would also like to thank Mr. Mathias Mueller, Area Sales Manager of the Andritz Atro GmbH. It was a great pleasure to meet him during the B.H.A. annual conference in Glasgow. The information he provided were very important, especially in the directions of fish friendliness and noise level of Archimedes Screw turbines, which are two areas with huge importance and sometimes are neglected by hydro turbine researchers.

I have to give a great thanks to Professor William David Lubitz from University of Guelph for the time he had spent to fully answer my questions in detail and providing all the information I asked him, and of course the literature he had sent me which was valuable during my research. I owe him at least an acknowledgement and I have to congratulate him and his team for the great work they conduct in Canada.

I need to give a great acknowledgment to Dirk Nuernbergk and to Alois Lashofer for the publications they have sent me and of course their incredible research on Archimedes Screw turbines. A large part of the present study steps on their numerical and experimental work which except for knowledge provided also motivation and inspiration for deeper study and understanding.

I would like to thank Mr. Mattia Schiesaro, Project Manager of the ZECO srl for the literature he had sent me. It was great to meet him and, besides his tight time schedule, we had a very interesting discussion drinking that nice espresso during the B.H.A. annual conference in Glasgow.

Many thanks go to Sean Petley, Charikleia Oikonomou, Nilla Karlsen and Tom Brindley, PhD students in Lancaster University Fluid Machinery Group (LUREG) for the time we spent during this year, and Dr. Audrius Židonis, ex PhD student in LUREG, for his help. Also, it was a pleasure to meet as an exchange student Georgios Mousmoulis ('the cavitator'), PhD student in the Laboratory of Hydraulic Turbomachines of National Technical University of Athens. I will never forget those entertaining conversations at the office during the mornings when you were preparing your tea.

Of course, I want to thank Associate Professor John Anagnostopoulos and Professor Dimitris Papantonis for their time and their hospitality when I visited the Laboratory of Hydraulic Turbomachines of the National Technical University of Athens.

Finally, I need to thank Dr. Jo Scott, Senior CFD Engineer, Mr. Gareth McMann, Sales Engineer, and Mr. Jordan Archer, Assistant Sales Engineer, for being so kind to accept my request to visit Gilkes's headquarters, meet the staff and have a short guided tour in the production line.

Last and most important I want to thank my family for their guidance and support. Without them I would have achieved nothing.

Presentations

1) LUREG Seminar Meeting Series #7 For Academic Year 2015 – 2016, Lancaster University, Lancaster, UK, 31st March 2016
2) Postgraduate Review Conference, Lancaster University, Lancaster, UK, 29th June 2016

Events Attended

1) BHA Annual Conference, Glasgow, UK, 10th – 11th November 2015

Volunteering Activities

1) Chill out session speaker on 'Environment and Energy', TEDx Lancaster University, Lancaster University, Lancaster, UK, 28th January 2016
2) Economic evaluation of potential small hydropower stations and turbine types comparison, Water21, 2016

Relevant Training

1) Engineering Research Methods, Short course, Lancaster University, Lancaster, UK, October 2015
2) Environmental Modelling with FLOW – 3D, Webinar, FLOW – 3D, 17th March 2016
3) Trends in Hydropower Financing, Webinar, IHA, 9th August 2016
4) Automated geometry optimization using FLOW – 3D CAST and CAESES, Webinar, FLOW – ED, 25th August 2016
5) Better Designs Faster with Integrated Fluid-Structure Interaction (FSI), Webinar, CD-Adapco, 20th September 2016
6) Risk and Reliability for Tidal Turbines, Webinar, DNV GL, 22nd September 2016

Visits

1) AST Hydro Power station, UK, 16th November 2015
2) Laboratory of Hydraulic Turbomachinery, National Technical University of Athens, Greece, 22nd April 2016
3) Gilbert Gilkes and Gordon Ltd, Kendal, UK, 8th June 2016

Author's Introduction

It is not very usual for the author of a dissertation, a student, to write an introduction for his own work. This happens because the dissertation is often considered as a document with no real importance, just an obligation as a part of a course, which will not be opened again in the future and will provide very few to anyone who would like to study it. The truth is that there really are many such documents of really bad quality, documents written in the last minute or full of useless information just to achieve the minimum requested number of words to graduate. And of course, although the mark may not be really good, many of them rarely fail.

The present document does not vindicate any special prize or any acknowledgement about the 'state of the art' work that presents, or a 'global first' in any field. It is no more than the work of a young engineer, who has still much to learn, in his field of his interest and specialization, inspired after the deal with the difficulties of such a real project during his short professional experience. The improvement of an engineering career is always a great think but it should not be forgotten that engineers, as many other professionals, have the education and the perception to actively participate in the improvement of many elements of our society. Publications are good for the CV but it would be preferable to be a result of hard work able to provide something new to any knowledge community.

Reading in one publication (Rorres 2000) about Vitruvius, a Roman engineer who managed almost 21 centuries ago to calculate a geometry, not very different from an approach of our days about the optimal geometry, and contribute with his writings to keep know this device until our days, I remembered the words of my first supervisor a few years ago: 'Although the technology related with hydraulic machinery looks mature and it has been developed for many years, there will be a need to reinvent many of its elements'. As the global energy needs are constantly increased, with no signs in the near future for stabilization or decrease, and the environmental charge is getting worse, there will always exist the need for the production of clean energy. Since the requested values are the produced energy and power, any potential improvements are not limited in the turbine's efficiency increase but should also focus on other fields like the overall efficiency increase by improving the installation process, except for the design, and reducing the maintenance needs.

Closing this short introduction with the words that according to Rorres Vitruvius used to close the description of his screw let's remind ourselves that although their access to any resource was negligible related with today, the request for progress and excellence led to discoveries and inventions that are not fully studied in our days:

'I have now described as clearly as I could, to make them better known, the principles on which wooden engines for raising water are constructed, and how they get motion so that they may be of unlimited usefulness through their revolutions.'

Ilias Kotronis
06/08/2016
Room 105, Flat 29, House 06
Lancaster University Graduate College Residences

Contents

List of Figures

List of Tables

Nomenclature

a	Water depth in rectangular open channel	[m]
C_{RR}	Cost of hydropower installation per kW	[£ / kW]
d	Diameter ratio d = D_i/D_o	[-]
D_o	External diameter of the screw	[m]
D_i	Internal diameter of the screw	[m]
E	Specific energy	[J/kg]
f	Fill level of the screw	[-]
g	Acceleration of gravity (g = 9.81)	[m/s^2]

H	Available head	[m]
L	Length of the rotor covered with blades	[m]
N	Number of blades	[-]
n	Rotational speed	[rps]
n_m	Manning's friction coefficient	[-]
n_{max}	Maximum rotational speed of the screw	[rpm]
N_{QE} ,N_S	Specific speed	[-]
n_q	Specific speed (German literature)	[U/min]
n_r	Rotational speed of the screw	[rpm]
Pav	Available power	[W]
P	Pitch	[m]
p_1	Pressure at the upper side of the bucket	[N/m^2]
p_2	Pressure level at the lower side of the bucket	[N/m^2]
Q	Available flow in the inlet open channel	[m^3/s]
Q_G	Leakage flow through gap between the screw and the trough	[m^3/s]
Q_O	Overfill flow when f > 1	[m^3/s]
R	Hydraulic radius	[m]
r	Radius ratio r = R_i/R_o	[-]
rstep	Number of subsections for integrated radius	[-]
R_i	Internal radius of the screw	[m]
R_o	External radius of the screw	[m]
S	Pitch ratio S = P/D_o	[-]
S_f	Gradient because of friction losses	[-]
S_o	Gradient of the channel's bed, meters of drop per 1000 m length	[-]
s_{sp}	Maximum gap between screw and trough	[m]
t	Sluice gate opening	[mm]
U_o	Speed of the flow in the delivery channel	[m/s]
U_s	Speed of the flow in the trough	[m/s]
V_B	Volume of water in each bucket	[m^3]
V_U	Volume of water entering in the screw in one turn	[m^3]
V_{tot}	Total volume of water in the screw	[m^3]
w	Width of rectangular open channel	[m]
z_{max}	Maximum water level of the bucket, full bucket, f = 1	[m]
z_{min}	Minimum water level of the bucket, empty bucket, f = 0	[m]
z_{wl}	Water level in the bucket in comparison with the minimum	[m]

z_1	Water level at the upper side of the bucket	[m]
z_2	Water level at the lower side of the bucket	[m]
β	Slope angle of the screw related with the horizontal level	[radians]
θstep	Number of subsections for integrated angle	[-]
ρ	Density of water (ρ = 1000)	[kg/m^3]
ω	Rotational speed of the screw	[rad/s]

Acronyms

ASP	Archimedes Screw Pump
AST	Archimedes Screw Turbine
EU	European Union
FSI	Fluid Structure Interaction
HPM	Hydrostatic Pressure Machine
HPW	Hydrostatic Pressure Wheel
NTUA	National Technical University of Athens
RST	Rotary Screw Turbine

Chapter 1. Introduction

The work described in the present thesis can be divided in three main areas:

1) Preliminary study of the low head turbine technologies for use in conventional hydropower stations. As it is mentioned later, this part is focused in mature, high efficiency technologies in relatively wide use.
2) Numerical modelling of an Archimedes Screw turbine. The necessary mathematics are described step by step for the estimation of the power and the losses created from each mechanism.
3) Improvements suggestions for the hydropower station with Archimedes Screw Turbine which was used as a case study for the MSc.

In part 1) there is an extended description of the hydro turbine technologies for low head hydropower. In this field the number of the commercial hydro turbines available for use is significant, related with the middle and high head, so the decision making process for the optimal turbine is more complicated. The very low head installations also have a great technical interest since few hydro technologies can successfully and in reasonable cost exploit this area.

In part 2) the numerical model was developed to calculate the hydraulic performance and a few structural details of the turbine. Analytical mathematical modelling was used based on available literature to calculate the known existing losses and simulate the operation.

In part 3) the model is validated based on manufacturer's performance curves of an existing hydropower station using Archimedes Screw turbine. After the validation three potential improvement options are proposed, from minimum to maximum modification, which also leads to different levels of improvement in the performance curve.

In this chapter there is a short introduction about hydropower as a renewable and clean source of energy focused in low head. The general structure of the thesis is also provided among with the aims and goals.

1.1 Hydropower

Hydropower or water power is a clean and renewable form of energy that is produced by water, usually fresh, in motion. The last decades many devices have been developed for the utilization of sea water which belong to various categories like tidal or wave energy devices but these are not considered conventional hydropower technologies and they are not covered in the present study. The tidal energy technologies have many elements in common with the very low head hydropower technologies although usually the percentage of the available capacity that can be exploited is limited. The fact that the installations are in the most cases bi-directional and their main elements (direction, speed of the stream) can be accurately predicted make them attractive for future study in areas with tidal streams. The motion of water is transformed in kinetic energy by rotating an impeller, so this kind of energy is extracted in most cases by rotating devices, the hydro turbines. The maximum hydraulic power at disposition of the turbine is given by the formula 1.1. It is very often to see this formula without the density (ρ) so in this case the result is in different units (kW). The energy at disposition for production for a reference flow is calculated by the formula 1.2 and it is the specific energy per volume of water or in other words the specific power per flow.

$$P_{av} = \rho\, g\, Q\, H \tag{1.1}$$

$$E = \rho\, g\, H \tag{1.2}$$

There is not a clear definition for hydropower based on nominal power (ESHA 2004). It is generally considered that between 100 kW and 1 MW a hydro power project is 'small' while between 5 kW and 100 kW is 'micro'. Based on the abstraction of the flow, hydro power schemes are categorized as run – of – river schemes, schemes with dam or schemes integrated with a canal or a pipe. There are more types, like the pumped – storage schemes, but usually they are not so frequent and their nominal power is usually high.

The main criterion of classification for the hydroelectric projects is the 'head', which is the vertical distance between the turbine's shaft or the water surface at the lower reservoir (depends on the type of the turbine) and the water surface at the upper reservoir. That head is in other words the total available pressure difference that the flow can exploit. In appendix A, the minimum head in combination with the minimum available flow for the most well-known low head hydro turbines are presented to give an idea about the combined flow with the low head and low power mentioned. As 'high head' projects are classified the projects with head 100 m or more. As 'medium head' the projects with head 30 – 100 m and 'low head' 2 – 30 m. Also the projects with head between 0.8 – 2 m are categorized as 'very low head' projects (Wiemann et al 2007). This area is the most problematic for conventional hydropower (not hydrokinetic and similar devices etc) since the cost per kW is high, the nominal power is low and the performance of the available turbine technologies able to cover this area is sometimes questionable. However, the large number of small hydro sites with low or very low head and the fact that they can be economically accessible even by investors with limited resources, because their nominal power is in many case very low, make them attractive for further study and development. Just to give an idea about the number of potential sites in this area, according to a study (Wiemann et al 2007) the structures which could be used as a weir for hydropower plants in an area of 1,000 km^2 in Germany were 1,304 locations with head difference between 0.2 and 1 m. This is equivalent to around 1.3 locations per km^2. Although it is not possible to claim that this number is representative for Europe, it is a strong evidence that many projects can still be developed, if the technology for very low head is improved and become more economical attractive, even if more than 70 % of the output of the theoretical available hydropower capabilities considered fully developed (Wiemann et al 2007).

In large scale hydropower (10 – 100's MW) it is estimated that only 33 % of the world's technically and economically feasible hydropower has been exploited, although this percentages may vary significantly from country to country (Senior 2009). In the same article is also mentioned that there is target for further development of the small hydro turbines for head below 5 m and that no economical feasible option exists for sites with head between 1 – 3 m and also that in some studies head differences below 1 m are ignored completely. As regards the hydropower technologies for that low head, there are many designs and patents but few of them are commercial in terms of combining acceptable performance and reasonable cost. From the known turbines in use for low head, it is not clear what is the absolute low limit for their use but it may have interest to be studied their ability to exploit very low head, possibly with small modification to their design.

1.2 Aims and Objectives

The aim of the present research was the development of a methodology for estimating the performance of an Archimedes Screw turbine for a hydropower scheme, based on a number of assumption. Every hydro power scheme is unique so the turbine at least, from the necessary components, should be optimally designed for it. As a free surface device, the Archimedes Screw turbines performance is sensitive to a large number of variables, many of them in correlation, so a good design is depended less on the rotor related with the most turbine types.

In this direction a numerical model was developed, based on the existing literature but also introducing new elements, trying to simulate as accurate as possible the performance. The main objective was the maximization of the produced energy of a potential scheme, taking into consideration the flow and head-duration curves. The first step in this direction was the determination of the optimal geometry, especially the main values related with the rotor (length, slope, external diameter, diameter/radius ratio, pitch ratio) to exploit the available capacity but also to reduce secondary losses, which are much more significant in many cases and very often they are not taken seriously into consideration, especially in a water to wire estimation. For the best result, the study focused on the following three directions:

- Optimal choice of the geometric characteristics
- Minimization of the losses
- Efficiency of the secondary devices (generator, transmission etc)

For the validation of the model were used manufacturer's data from an existing hydro power station. Finally an optimization algorithm was developed able to compare the performance of different designs and suggest the ones with the best efficiency according to chosen criteria.

1.3 Structure of the Thesis

The thesis is structured as described in the following outline.

Chapter 2 – Literature Review. Introduces the existing knowledge in the field of study which was collected after an extensive research in public domain during this research. The work of a number of scientists is presented in the areas of low and very low head hydro power technologies and relative areas, focusing on Archimedes Screw turbines and in the aspects that are taken into consideration during their design.

Chapter 3 – Numerical Modelling. The developed numerical model is described in detail with the mathematical background and the steps of the algorithm. A few computational issues are discussed like the effect of the discretization in computational time and accuracy of the main variables.

Chapter 4 – Validation and Results. The manufacturer's data from the project which was used as a case study are compared with the results of the simulation using the developed model under the same circumstances and the results are presented and discussed.

Chapter 5 – Discussion for potential changes and improvements of the project. After running different cases using the basic model and the optimization tool, thee potential improvement approaches are proposed for the existing project: a) for minimum interference, b) for significant interference but maintaining the same rotor and c) for major interference with new rotor.

Chapter 6 – Conclusions

Chapter 2. Literature Review

The available low and very low head hydro power commercial technologies are presented in the beginning of this chapter with their main design challenges and historical data. A few secondary information are also provided related with the design of the delivery open channel, legislation, fish protection and noise issues. The available relative to the topic AST publications are reviewed with their results in the second part.

2.1 Low and Very Low Head Hydropower

As it was mentioned in the first chapter, the low head area starts from 30 m and below, even if in many reports and references place this limit at 20 m. Many turbine types can work below this limit. There is of course the question what is the absolute lower limit in terms of head and flow combination. It is not clear which is the criterion about if a turbine can work at a specific point, in case that there are no other obstacles (e.g. cavitation). The specific speed in combination with the efficiency are supposed to be the main tools in this direction but again, the absolute minimum acceptable efficiency or specific speed are not clearly determined. For the Kaplan turbine for example, the most well-known turbine for use in low head, and the various versions of it there is a significant range of specific speed limits that are available in literature (Figure 2.1) and of course they have differences related with what the manufactures suggest, especially in the lower areas of the low head region. Each manufacturer has developed his own know-how among with improvements based on his experience through the years. In addition, it depends on the performance envelope of the other turbine types that the company provides, aiming to cover as larger area as possible of the head – flow chart. In case for example that a manufacturer does not produce Kaplan turbines but he produces Francis turbines, he will try to minimize the lower limit of the head that his Francis turbines can achieve aiming to cover this area of the low head for his potential clients. In case now that another manufacturer produces Kaplan and Francis turbines, the lower limit of the head for the Francis turbines will be around the upper limit of the Kaplan turbines that he can offer. Comparing these two manufacturers it is expected that the Francis turbines of the first can achieve reduced lower head related with the second.

The specific speed is calculated from the formula 2.1 based on by IEC 60193. For S.I units the formula 2.2 has to be used. The maximum rotational speed, except for the specific speed limits, has to do with rotor dynamic restrictions and other technical limitation related with the stiffness of the construction. For the majority of the manufacturers if it is allowed by the specific speed limits the maximum rotational speed of the turbine can be up to 1500 rpm which is the limit for the maximum rotational speed of a commercial synchronous generator in use with a hydro turbine. It is easy to understand that in case that the designer wishes a direct drive connection between the turbine and the generator, the speed of the turbine should be the same with the generator. A low speed generator is more expensive and heavier than a high speed generator for a reference power, so the high speed is preferred if it can be used.

$$N_{QE} = n\frac{Q^{0.5}}{E^{0.75}} \tag{2.1}$$

$$n_q = 60\ N_{QE} \tag{2.2}$$

$$N_S = 995\ N_{QE} \tag{2.3}$$

As it was mentioned and it is presented in Figure 2.1, there are significant variations in the specific speed of a reference turbine type in the available publications (ESHA 2004, Warnick 1984). Unfortunately the available data which were found cover only the turbine types which are more common in our days and they have generally high nominal power. No studies related with the specific speed determination of small hydro turbines in low head like Water Wheel turbines and Archimedes screw turbines were found during the present study. Also, the minimum estimated flows for the minimum heads are presented from the mentioned specific speed studies with the minimum estimated heads and flows that a significant number of manufacturers provide or are available in public domain (Appendix A).

These information were collected by their websites, where they were available. The obvious and very interesting significant result from the previous information is that there is not clear determination of the performance range for each turbine, especially as the head is getting lower. Of course, the available data only deal with the nominal performance. The minimum and maximum flow and head for the referenced minimum flow and head are not known and each company has different limits based on it's know – how. The previous published data are from the most 'famous' hydro turbine types which usually have higher nominal power and they usually cannot cover the area of very low head. Except for them there is significant number of devices for power production in low and very low head. Their nominal power is generally reduced and their performance sometimes questionable. However they provide many potential options for very low head applications.

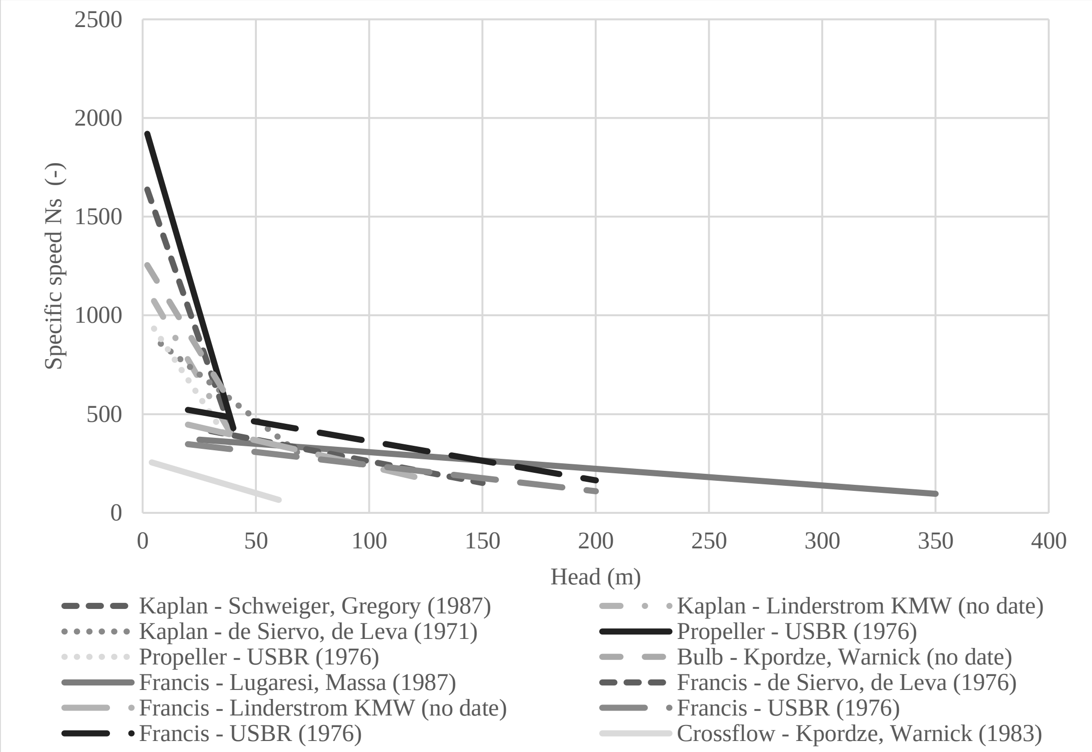


Figure 2. 1 Specific speeds for various researchers for different hydro turbine types (ESHA 2004, Warnick 1984).

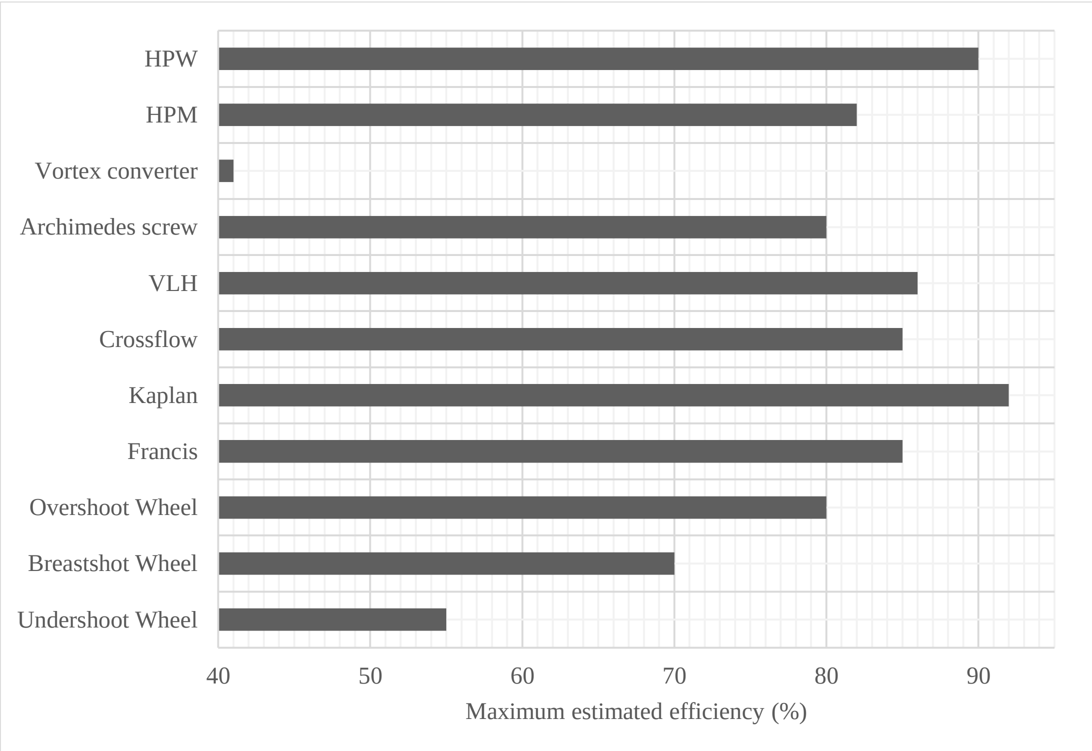


Figure 2. 2 Estimated nominal efficiency for conventional and not conventional types of turbines for use in low and very low head (Wiemann et al 2008, Bozhinova et al 2012, ESHA 2014).

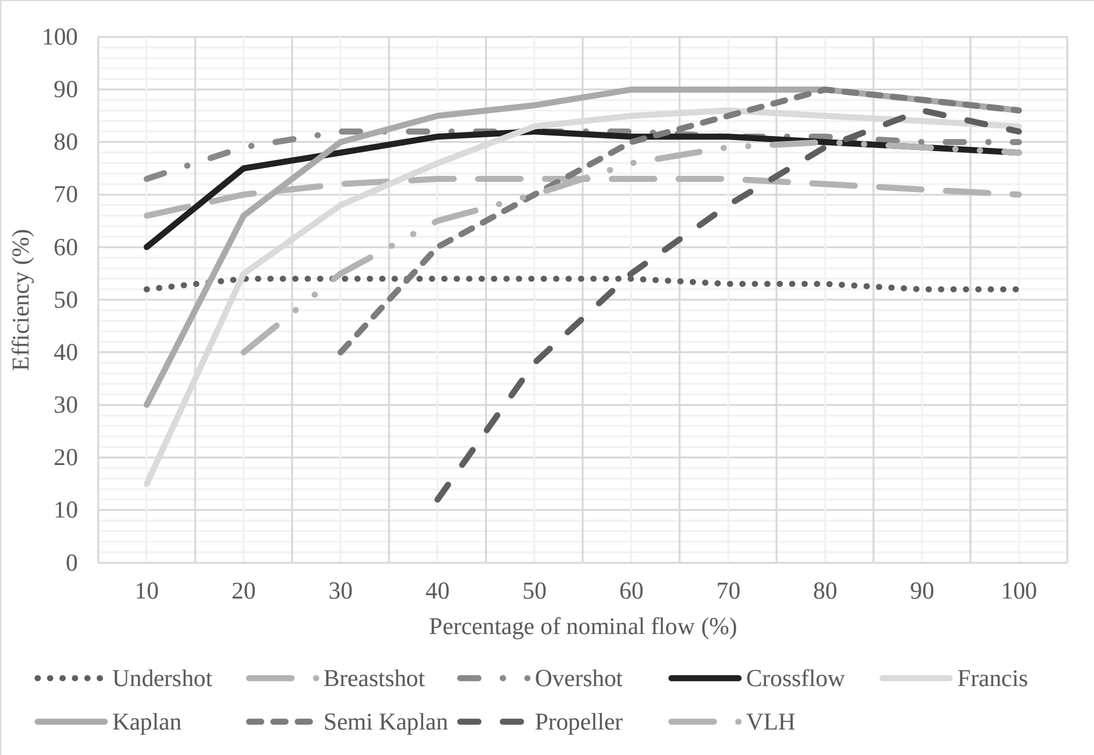


Figure 2. 3 Typical efficiency curves (Bozhinova et al 2012, ESHA 2014).

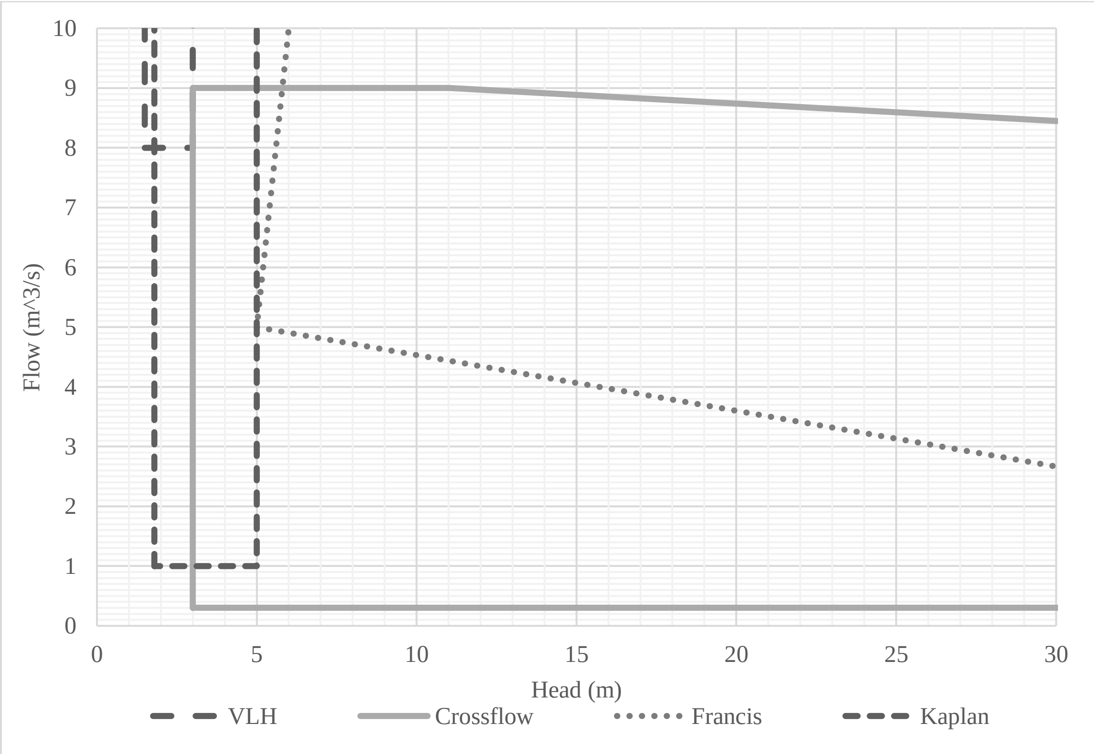


Figure 2. 4 Aspect of the performance envelopes for different hydro turbine types in low head hydropower (Wiemann et al 2008, Bozhinova et al 2012, ESHA 2014).

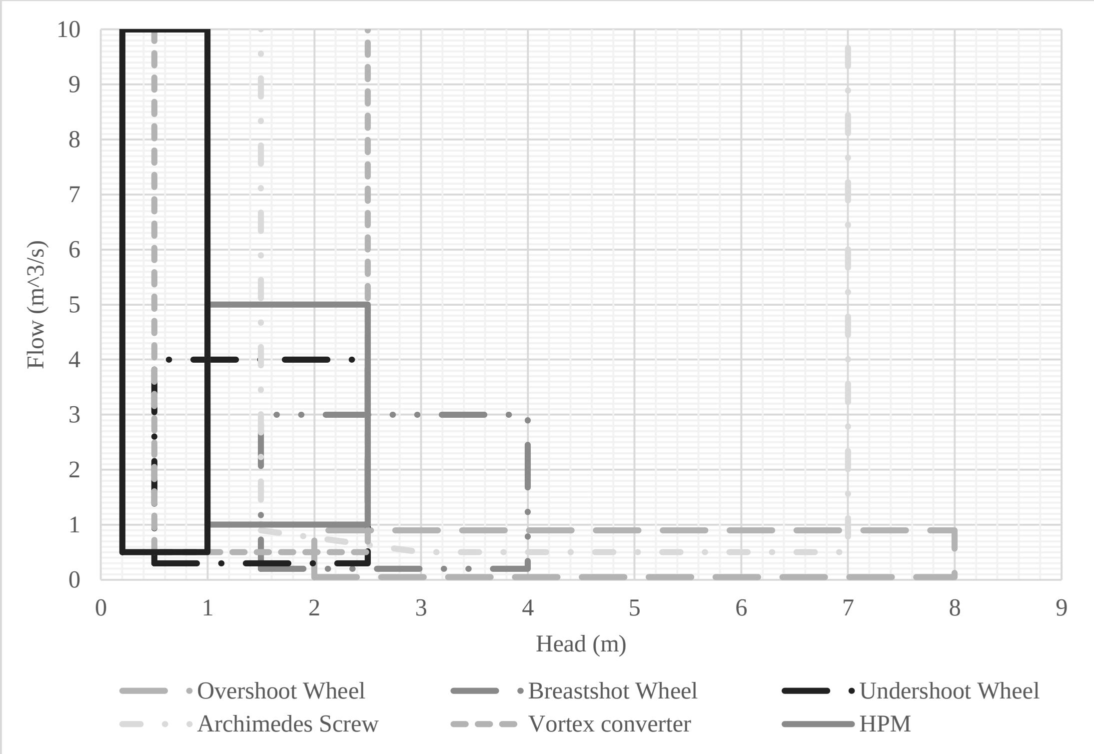


Figure 2. 5 Aspect of the performance envelopes for different hydro turbine types in very low head hydropower (Wiemann et al 2008, Bozhinova et al 2012, ESHA 2014).

The Kaplan, Francis and Crossflow turbines are the most well - known from the presented in the Figures 2.1-2.4 for the low head. They are well developed after many years of research and experience with thousands turbines of each type installed globally and they can achieve high efficiency and very high flow. The Crossflow is an impulse turbine like Pelton and Turgo. Although it is theoretically possible down to a point, the Francis and Crossflow turbines usually are not used in sites with head below 10 m.

It has been categorized based on if the rotor blades or guide vanes are variable or fixed for a reference specific speed (Figure 2.1). So as Kaplan (or double regulated Kaplan) turbine is usually considered the turbine with variable both rotor blades and guide vanes. The semi – Kaplan has variable rotor blades and constant guide vanes and the Propeller has variable guide vanes and constant rotor blades. Except for the rotor, a large part of its construction is the draft tube which is useful for energy recovery and avoidance of air entrapment (it is also used with Francis turbines). As it can be seen in Figures 2.1 and A.2, it is not clear what the lowest head that Kaplan turbines can reach. The VLH is a compact single device, recent design, using a Kaplan turbine including the generator and other equipment.

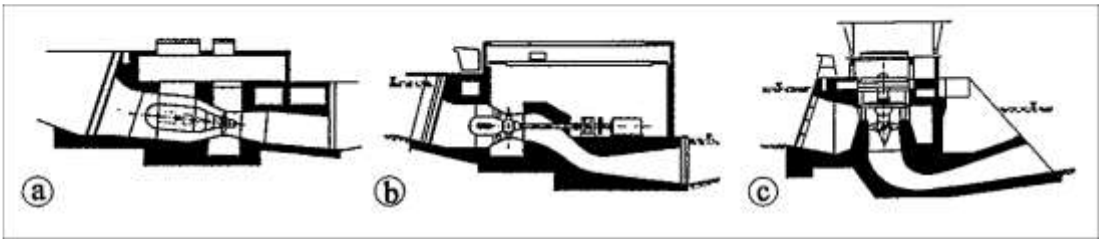


Figure 2. 6 Various Kaplan turbines in different arrangements a) Bulb turbine, b) Horizontal S – type turbine, c) Vertical (Senior 2009).

Figure 2. 7 Kaplan turbine (www.zeco.it) and VLH turbine (www.vlh-turbine.com).

Figure 2. 8 Francis turbine (www.gilkes.com) and Cross flow turbine (www.ossberger.de).

The Water Wheel family (Overshot, Breastshot, Undershot) are really old devices for power production in water mills and similar installations. In our days their modern versions are still in use for small hydro power projects, sometimes in existing mills to maintain the traditional view of the construction. In the same direction there have been efforts for advanced designs of similar with Water Wheel turbines devices (Senior 2009) although their philosophy of operation is different. The Vortex Converter has a few similar characteristics with a Francis turbine (radial inlet, axial outlet). Archimedes screw turbines, although they have been used as pumps since ancient times only a few decades ago started to be used as turbines for electricity production. Of course many more devices for electricity production from water in motion may be available in public domain. However either they are in early stages of development and there are not many information available or their performance is not so advanced as to be considered satisfactory and competitive with the devices described.

In addition there is a large number of converters for use in ‘ultra’ low head, close to zero (Wiemann et al 2007) and relatively low power but since they are actually hydrokinetic/ stream devices they are not widely used in conventional hydropower and they have not been included in the present work. It is certain that there are differences between the performance envelope and the efficiency curves above related with the same information that will provide each company for a turbine that produces.

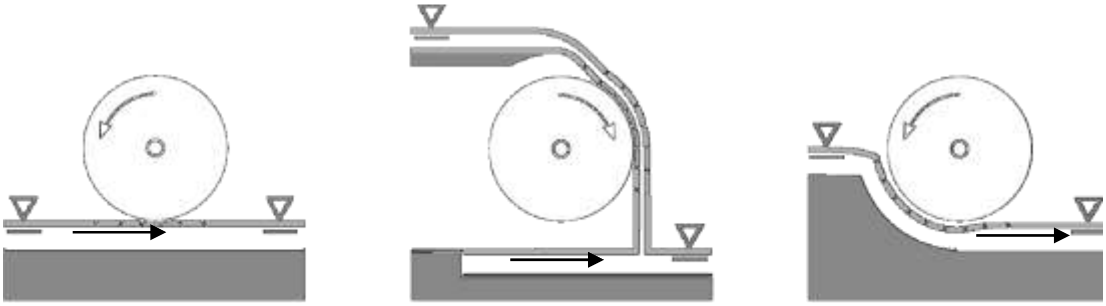


Figure 2. 9 Various types of Water Wheel, Undershot, Overshot and Breastshot (Senior 2009).

Figure 2. 10 Vortex Converter (Senior 2009).

2.2 Archimedes Screw Turbines (AST)

2.2.1 History

The Archimedes screw and the Water Wheel are the oldest hydraulic machines in use. Archimedes of Syracuse is considered to be the inventor of Archimedes screw the 3rd century B.C, although that according to a few researchers it was in use in Assyria at least since the 7th century B.C. Also, according to Strabo the Hanging Gardens of Babylon were watered by screws. The Roman engineer and architect Vitruvius gave detailed and informative description of Archimedes screw in one of his books the 1st century B.C. (Rorres 2000) and his description had a great role to keep this device known until our age. It has to be noted here that Vitruvius suggested an optimal geometry for an 8 - bladed screw which geometrical characteristics are relatively close with the calculated as optimal 8 - blades screw in the same study. Until recently the Archimedes screw was used only as pump (ASP). Only in the beginning of the 19th century started to be used for power production and it was patented in 1991 by Karl – August Radlik (Lashofer et al 2012). A few years ago the first tests of the performance as hydro turbine for electricity generation for use in low head sites were published (Brada 1999). The design of the machine makes it attractive for use under strict environmental restrictions related with fisheries.

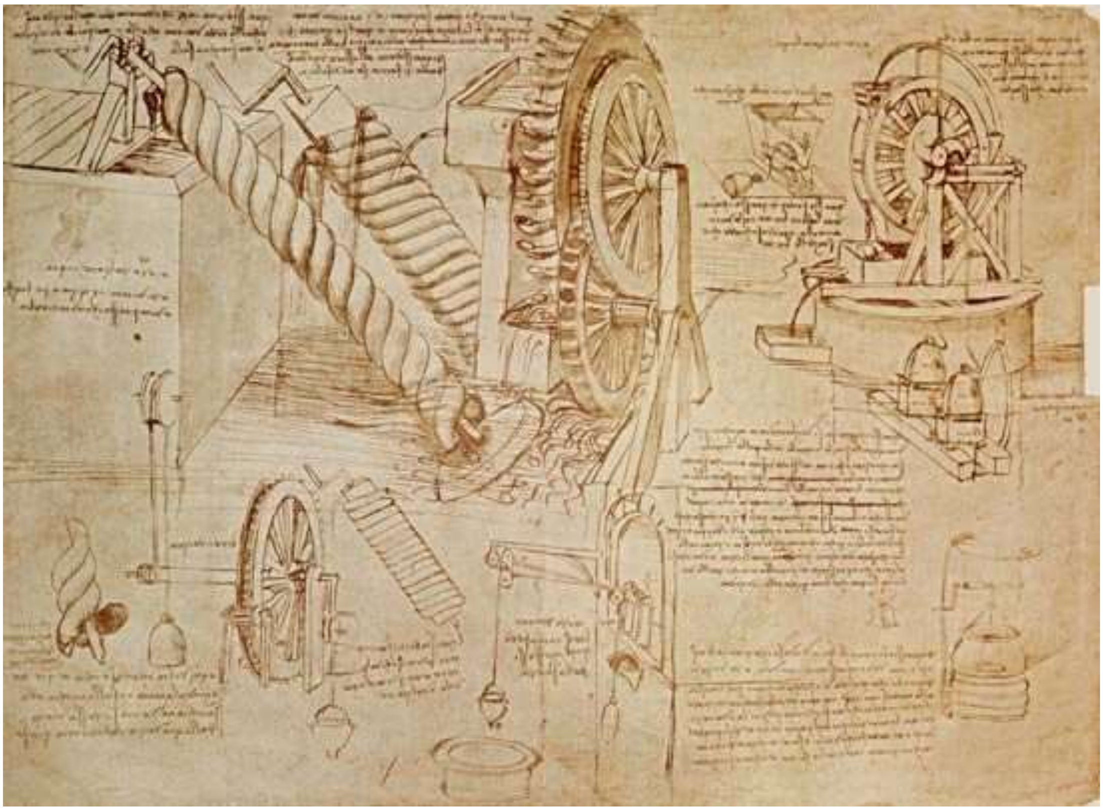


Figure 2. 11 Screw and Water Pump by Leonardo Da Vinci (www.museumofthecity.org).

The reason is that because of the low speed and large size the fish can swim through the device, if a number of limitation is respected (Fishtek 2011). The performance envelope of the AST is considered between 1 – 10 m head and 0.25 – 15 m^3/s flow (Spaans Babcock 2009). Of course these numbers are referred to the nominal points. The minimum and maximum achievable flow and head for a turbine designed for specific characteristics is depended by many parameters which will be studied in the following chapters. The upper limits are mainly defined because of the dimensional limits.

2.2.2 Size Limitations

The dimensions are limited because of the equipment capabilities during the manufacture process and because transportation restrictions which will increase the final cost. The following table (2.1) presents a few basic general rules for abnormal road transport in EU. These limits may vary from country to country but they can be used as general guideline. Based on the limits mentioned, the maximum diameter which can be transported without limitations cannot exceed about 2.5 m (total height – trailer height) and the maximum length 24 m. It is obvious that larger machines can be transported, and indeed are provided by a number of manufacturers up to 5 m diameters, but as dimensions are increased special permits are required and a necessary design of the route is necessary to avoid narrow or low passages (tunnels, bridges etc). As a result, the cost of transportation is increased and since many hydro power sites are in distant places, far from motor ways, the transportations of large size turbines can become a real challenge and result to significant installation cost, which is also supposed to be one of the main advantages of hydropower schemes with AST. Based on literature (Rorres 2000, Nuernbergk et al 2013) it can be estimated that a three blade AST in 22° slope angle can achieve maximum flow about 2.7 m^3/s and maximum head about 8 m, considering that the length of the rotor not covered with blades is 2 m, for the size restrictions mentioned.

Table 2. 1. Framework for abnormal road transport permits (European Best Practice Guidelines for Abnormal Transports).

	No permit needed	Long term permit	Corridor
Maximum width	3 m	3.5 m	4.5 m
Maximum overall length	24 m	30 m	40 m
Maximum overall height	4 m	4.2 m	4.4 m

Figure 2. 12 Examples of AST road transportation (Landustrie, Spaans Babcock).

2.2.3 Speed

For any kind of performance calculations the rotational speed is obviously necessary. The formula for the calculation (2.4) of the maximum rotational speed with main parameter the external diameter of the screw was suggested by Muysken in 1932 (Nuernbergk et al 2013). Muysken suggested this formula after experience and experimentation. The calculated speed from this formula for a reference diameter is also called sometimes 'Muysken limit' because above this limit the produced friction losses and centrifugal forces are excessive and the efficiency starts to be reduced. It is possible to increase the speed above this limit, especially in variable speed installation for short periods, to exploit higher flow with reduced efficiency however. According to (Lashofer et al 2011) most European screw manufacturers recommend and use this rotational speed.

$$n_{max} = \frac{50}{{D_o}^{2/3}} \tag{2.4}$$

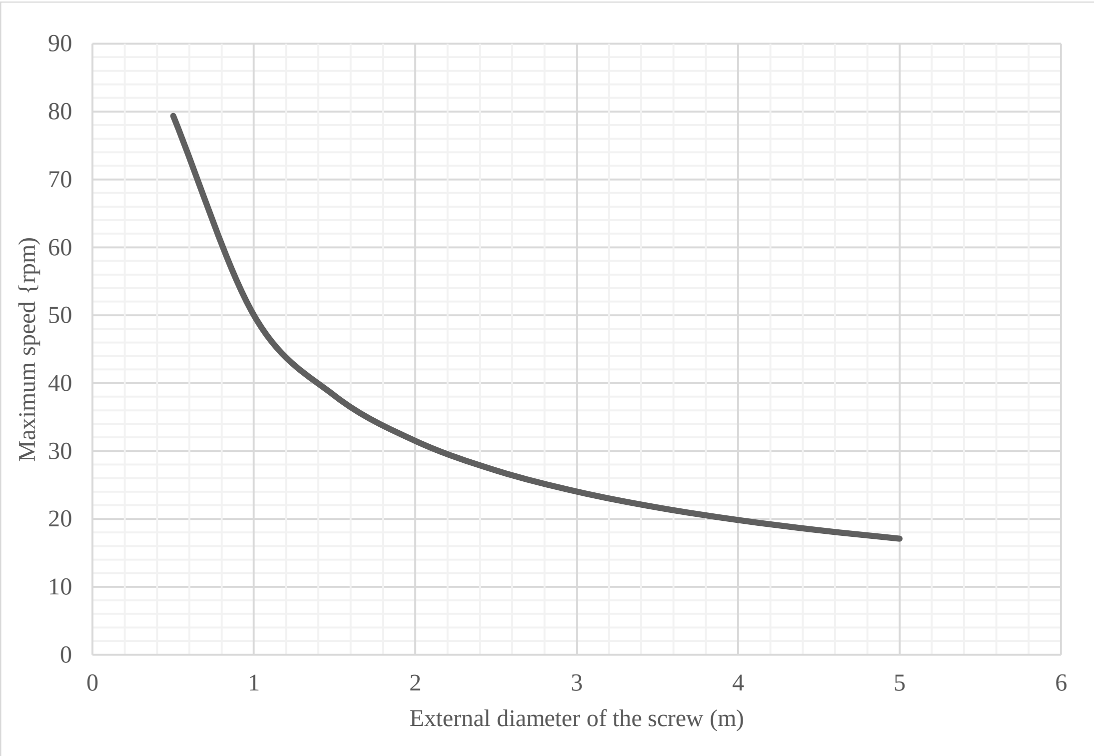


Figure 2. 13 Maximum speed as a function of external diameter (Nuernbergk et al 2013).

The lower limits of the performance envelope are not so clear as well. For minimum diameter 0.5 m, based on the same literature as before, the maximum achievable flow can be about 0.1 m^3/s. This is the maximum flow for the reference diameter, the minimum partial flow for the design mentioned can be significantly lower, below 20 % of the nominal flow in good conditions. Diameters below 0.5 m are possible to be manufactured but in this case it needs further study if the cost is competitive related with another type of turbine for the same application.

2.2.4 Gap

Another important parameter is that the designer must ensure the stiffness of the rotor, as it has to do for larger diameters as well. As the diameter is decreased, the deflection is increased for a reference length. This fact creates the need, if the deflection is significant, to increase the distance between the rotor and the trough aiming to avoid contact, which will increase the leakage losses and reduce the efficiency. The maximum gap between the screw and the trough calculated by the formula 2.5 (Nagel 1968) and this is the gap which is possible to be measured, close to the bearings of the shaft where the deflection starts. The minimum gap at the maximum deflection point will be of course smaller. However there is no published research which suggests the minimum gap. The minimization of the gap would provide a significant profit because it would lead to the minimization of the leakage losses. According to Nagel (1968), a screw manufacturer cannot guarantee the efficiency without specify the upper limit of the gap width, which makes sense because in this case he will not be able to know the leakage losses. The minimum gap can be estimated if a few geometrical details are available.

$$s_{sp} = 0.0045\sqrt{D_o} \tag{2.5}$$

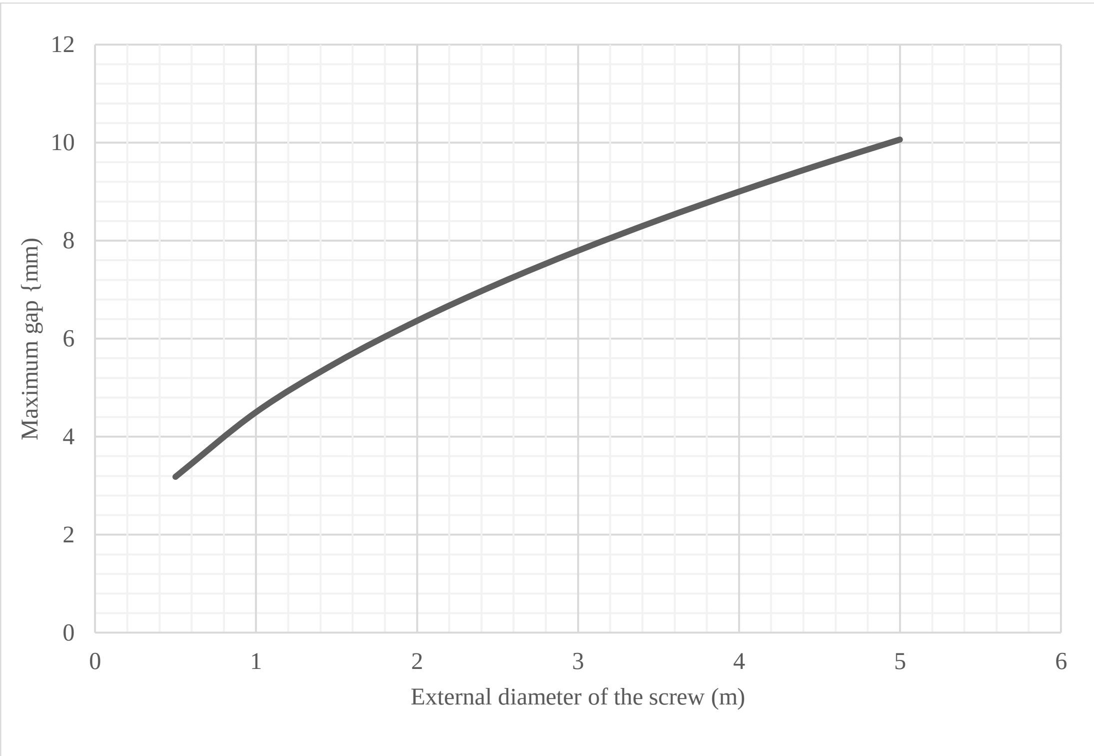


Figure 2. 14 Maximum gap as a function of external diameter (Nagel 1968).

For the minimum head there is not any obvious limitation related with the manufacture process, if there is an available length covered with blades. The actual limitation is that to exploit the total difference between the two water levels a part of the rotor should be submerged. This submerged part will lead to losses as a result of the drag because of hydrodynamic resistance, as will be described in next chapter. If the drag is high and the power is not relatively high, the efficiency will be reduced, down to not acceptable. So a really low head may be possible to be exploited but it cannot be combined with low flow, if a satisfying efficiency should be expected.

2.2.5 Materials

Except for the transportation or stiffness limits because of the dimensions of the AST, another potential parameter is the availability of raw materials. According to Kantert (Kantert 2011), the screw pumps are made of steel St.37. There are screws made of plexiglass and perhaps other materials for use in test rigs but from the research covered in the present thesis, in commercial applications where the size is high and the durability necessary, only steel is used. The dimensions of the pipe which is the core of the rotor are defined by the standard DIN 2448, which is included in the more recent standard EN 10220:2002. This standard defines the diameter of the pipes and the thickness but not the length, so the length that each manufacturer uses seems to be determined by the supplier. As a result, there are variations in the combinations of diameter and length because of transportation, design and market limitations. Each manufacturer based on his know – how has different range of products with respect to the limitations mentioned and aiming to cover the need of its clients based on his experience.

2.2.6 Number of Blades and Fish Protection

The number of blades is another area of interest. It is possible that during old times the ASP used to have larger number of blades related with today, the 8-blades study from Vitruvius (Rorres 2000) is an indication to this direction, maybe because it was believed that a larger number of blades can increase the flow. In our days the most manufacturers use 3, 4 or 5 blades in their designs (Fishtek 2011) and the maximum allowed number of blades is increased as the diameter is increased, as it is presented in figure 2.15 after the collection of data from 201 different screw systems. The number of blades is a result of the optimization that each manufacturer has done based on his experience. In the figure mentioned is also visible that in some areas of diameters more than one number of blades is used. Also, according to Tarrant (Tarrant 2011), the critical peripheral velocity for fish contact injury is 4 $\mathrm{m/s}$. According to another report however (Fishtek 2009), for various fish masses (0.21 - 6.5 kg) the impact force between 2 - 2.5 $\mathrm{Kg/cm^2}$ was found to cause indents/bruising to the flank of the fish and this was taken as the damage threshold. So, considering that the tip velocity of a screw for the mentioned maximum velocities varies from around 2 – 4.5 $\mathrm{m/s}$, to secure that there will be no injury even to large fish the blade edges should always covered by compressible bumper (Figure 2.17).

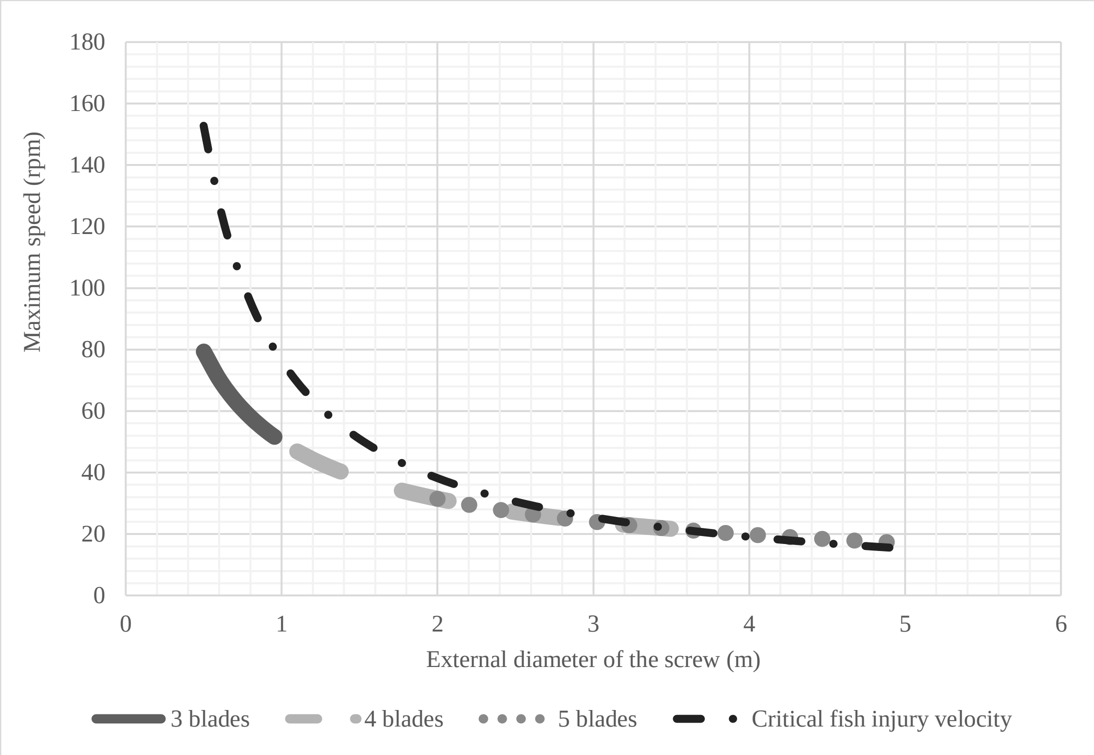


Figure 2. 15 Number of blades as a function of speed and external diameter (Fishtek 2011).

It can be noted here that a) the number of blades is increased with the diameter/flow so it has an impact on efficiency and the minimum/maximum partial achievable flow and that b) perhaps in lower diameters (and lower flows) large number of blades cannot be used because of the high speed which would become dangerous for the fish migration and would lead to the screw pumps or turbines to lose one of their significant advantages which is the fish friendliness. This is mentioned in Environmental Agency (2011), where is also determined the minimum diameter for a reference number of blades (Table 2.2). The thickness of the blades is important as well, a blade with higher thickness will lead to an impact with lower pressure and make any injury less possible but also will increase weight and the moment of inertia. To reduce impact pressure the leading edge is also covered by rubber bumper.

Table 2. 2. Minimum diameter and maximum speed for fixed and variable speed screws based on the number of blades (Environmental Agency 2011).

	Fixed speed		Variable speed	
Number of blades	**Minimum Diameter (m)**	**Maximum speed (rpm)**	**Minimum Diameter (m)**	**Maximum speed (rpm)**
5	3	24	2.3	29
4	2.2	30	1.6	36
3	1.4	40	1.1	48

Figure 2. 16 Leading edge covered with rubber bumper (Fishtek 2011).

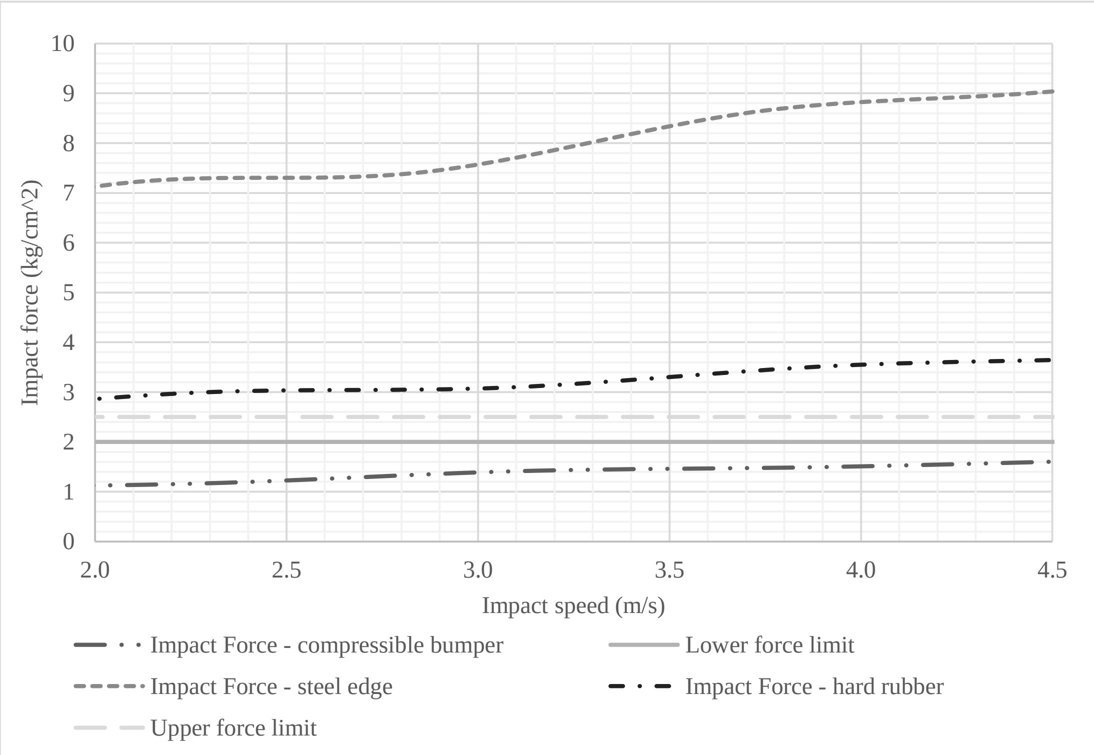


Figure 2. 17 Impact force for various speeds and materials (Fishtek 2009).

According to Tarrant (Tarrant 2011), who compares two proposed hydropower schemes at Avocliff Weir, there is a number of potential disadvantages of the AST related with other turbine types for use in the same performance range (e.g. Kaplan). These potential disadvantages are mainly related with the fact that this is a free surface turbine, so the philosophy of operation has some differences related with impulse and reaction machines.

According to the publication referenced, the AST may be fish friendly for fish length between 8 and 63 cm because of its design and allow fish migration through it but the gap between the screw and the trough, which is between 5 and 10 mm, can lead to major damage and mortality especially to small sizes.

As regards fish friendliness again, in large diameters the tip speed exceeds 4 m/s, which is considered the critical fish injury velocity and as a result the shear stress damage and mortality are increased. Two not so well known damages for the fish are barotrauma and embolism which happen because of fast and extended decompression (more than 30 %) at the inlet of the turbine. Different species are more or less susceptible to barotrauma. Many fish species have a fish pneumatic duct which helps to rapidly take in or vent gas, while others have no such duct and they must adjust their body's gas content by diffusion into the blood. However this diffusion may take hours so in case of very fast transition from higher to lower pressure a barotrauma can be caused. In low flow conditions, when the screw shutdown the forebay and the turbine will empty after a short period and the trapped fish will die. It is also possible that the affects the fish migration and spawning. Based on the Good Practice Guide from Environmental Agency, it is recommended intake velocities less than 0.25 m/s. In any case, although it is considered a very fish friendly turbine, there are many issues for further consideration related with fish protection and disturbance.

Except for the fish friendliness problems, Tarrant makes references for technical issues as well. Comparing a Hydrodynamic screw turbine with a Kaplan turbine, as proposed turbines for the scheme mentioned, significant differences are appeared in efficiency but mainly in the exploited net head and flow that each setup can deal with. The author believes that the performance curves provided by the manufacturer of the Hydrodynamic screw are not clear and easily understood and they present only the shaft's efficiency, not the water to wire, at fixed head without taking into consideration drivetrain losses, generator's, outlet drag because of the submerged end or forebay head losses. For a given geometry in this study, since the speed is fixed, the only way to regulate the inlet flow and water level is by using a sluice gate. This will result to reduced net head, related with the available, which will result to reduced overall efficiency since the net head is lower than the design net head for the available flow. However, according to the author for the energy calculation was used the available and not the operational head (the water level after the sluice gate) and as a result the calculations were not realistic.

At the end of his report, Tarrant makes a reference to the Settle hydro power station. The project seems to have reached approximately only the 85% of its rated max power. Using data from the site and manufacturer's data and applying the adjustments explained at the report related with the efficiency and the performance range, an estimated realistic output is 84,096 kWh, which is around 100,000 kWh below the installer's estimated output (183,981 kWh) used for the financial decision to proceed with the scheme. If the revised estimated annual energy output is correct, the schemes costs cannot recover within twenty years, without also including maintenance costs and other secondary costs. Based on the same calculations, the use of a double regulated Kaplan turbine, correctly installed, at the Settle site would have had an estimated annual output of 212,000 kWh.

2.2.7 Noise

In addition to the performance problems related with the flow, doubts about the noise levels are presented, which is produced by the changed direction of the water leaving the screw and back filling of the buckets, depends on if the flow is high or low. After the author's research to three hydropower stations using Hydrodynamic screw, the noise levels were so high that it should not have been possible to gain planning permission in a residential situation. Also, the high noise level is considered as an indication of inefficiency. A number of studies have been conducted in the area of noise level produced by the AST, including measurements (Skingle et al 2012) but also measurements and simulation (Dewald 2009) to estimate the noise levels in the area around the hydro power station. In the UK, assessments about noise levels related with renewable sources of energy can follow the methodology, guidelines and limits outlined by BS4142:1997, BS8233:1999 and BS7445-1:1993. Daytime operation is normally considered between 7:00 and 23:00 while night time operation between 23:00 and 7:00. The noise of an AST in consist of more than one components, which can be in general summarized as: a) mechanical noise from the generator close to the inlet, b) blade – entry and water flow noise from the inlet, c) water flow noise from the outlet and d) noise from water running through the fish passage. The noise level of the generator (A) can be reduced using a cooling system or by covering it with a noise insulation casing. The noise level at the inlet (B) and outlet (C) can be generally not high if the water levels do not vary significantly related with the nominal and the difference in the water levels before and after the transition points are limited. It has to be reminded here that the high noise level is considered an indication for reduced efficiency.

For the noise level identification a number of measurements are collected in a large range of frequencies, from sensors planed in particular point related with the origin of noise. The first assessment mentioned (Skingle et al 2012) results than 'Complaint is unlikely'. The second assessment mentioned refers to DIN EN ISO 3746 and DIN 45645 Part 1 for the type of measurement and to DIN ISO 9613 – 2 for the procedure. The collected data from three sensors are introduced to a simulation software to create a sound pressure level layout for the area around the hydro power plant.

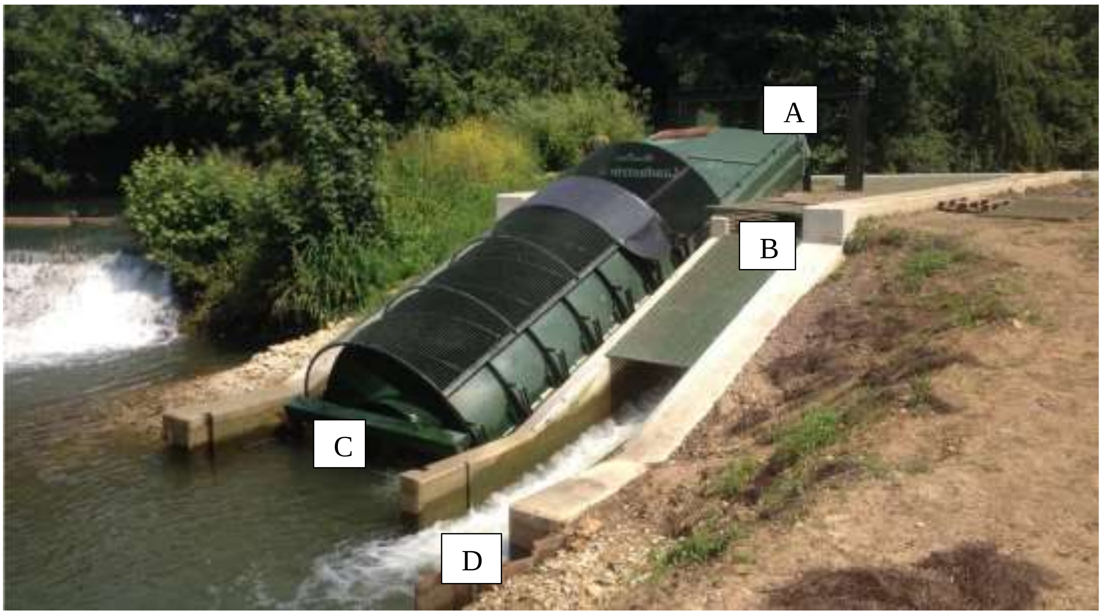


Figure 2. 18 Main sources of noise around and an AST (Skingle et al 2012).

2.2.8 Geometry and Performance

Perhaps the most significant parameters of the geometry for the performance of the turbine, after the external diameter, are also the pitch ratio and the diameter ratio. According to Rorres (Rorres 2000) the optimal radius ratio is suggested equal to 0.54. However this study does not take into consideration any leakage losses and the main optimization criterion is the maximization of the flow (maximum amount of water per turn). It has reported (Lashofer et al 2012) that almost all installed AST have pitch ration close to 1 and radius ratio close to 0.5. The mentioned report studied a sample of 74 AST at 71 sites in Europe and the founded range was 0.86 – 1.25 for the pitch ratio and 0.3 – 0.52 for the radius ratio. In the same reference was reported that the most commonly built screw is three bladed with radius ratio 0.5 at 22˚ tilt angle.

Figure 2. 19 AST blades outlet formation possibly for noise level reduction (left picture: jnbentley-news.co.uk, right picture: news.bbc.co.uk).

The present research did not manage to find out how this tilt angle has been defined as a very good slope angle for the AST. Based on numerical calculations (Müller et al 2009, Lubitz, et al 2014) which however do not fully simulate the losses mechanisms, the efficiency is increased as the tilt angle is decreased for bucket fill level equal to the optimal (f = 1,$Q_O = 0$) but when the fill level is increased (f > 1,$Q_O > 0$) this phenomenon seems to be reversed and the efficiency is reduced as the tilt angle is decreased (Lubitz et al 2014). As, it is generally expected, the experimental data provide addition information. Lyons (Lyons et al 2014) observes no significant efficiency increase for installation angle higher than 24.9˚. Lashofer (Lashofer et al 2012) presents experimental data that show some variation in the efficiency for various slope angles and show efficiency drop as the flow is increased, for the particular geometry in every slope angle tested. This seems a bit strange although the geometry of the screw and the nominal flow are not known to be able to determine the rest characteristics in detail. In the same document are also presented results from rotary screw turbine tests (RST), a different approach of a screw turbine fully covered by trough/shell which rotates with the same speed as the rotor. Although the RST design is oriented to reduce leakage and overfill losses, it has lower efficiency in the range of flows, angles and speeds which are tested, perhaps because of increased friction losses. It has been suggested (Müller et al 2011) that shallower angles lead to increased fill level of the buckets and higher efficiencies, although they result to longer turbines with higher manufacture and transportation costs. A preliminary numerical formulation has been described by a few researchers (Rorres 2000, Nuernbergk et al 2013). The total discharge that the screw can deal with is consisted of the following five components:

- Q_W : The flow that generates the torque.
- Q_G : The leakage flow between the screw and the trough.
- Q_O : The leakage flow when the screw is overfilled (f > 1).
- Q_F : The friction leakage flow formed by water that adheres to the flights.
- Q_P : The leakage flow if there is no guiding plate on the one side of the trough.

The friction leakage Q_F can be neglected since it is comparatively small. The leakage flow Q_P can be eliminated with the use of steel plate. The flow balance equation is:

$$Q = Q_W + Q_G + Q_O \tag{2.6}$$

The total inflow Q is the total available flow at the inlet and this can be the flow from an open delivery channel. Using the Manning's formula (ESHA 2004), the total inflow can be correlated with the geometrical characteristics of the inflow open channel.

$$Q = \frac{A}{n_m} R^{2/3} S_f^{1/2} \tag{2.7}$$

$$A = a\,w \tag{2.8}$$

$$R = \frac{a\,w}{w + 2\,a} \tag{2.9}$$

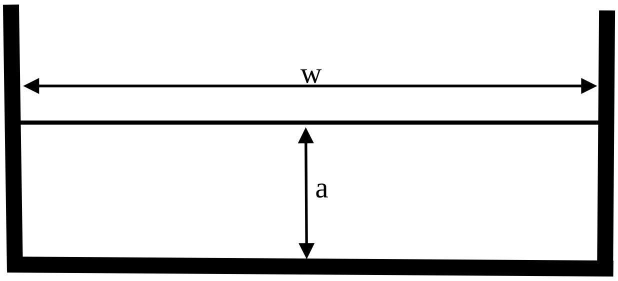


Figure 2. 20 Rectangular open channel cross section.

The flow Q_W was given by Weisbach in 1855 (Nuernbergk, Rorres 2013). Also to determine the volume per turn of the screw V_U, the volume of water that flows into or out of the screw in one turn which is the volume in length equal with one pitch, the following dimensionless parameters were introduced by Rorres (Rorres 2000).

$$Q_W = \frac{n_r}{60} V_U \tag{2.10}$$

$$\rho = \frac{D_i}{D_o} \text{ (radius ratio)} \tag{2.11}$$

$$\lambda = \frac{P \tan\beta}{\pi\, D_o} \text{ (pitch ratio)} \tag{2.12}$$

$$v_u = \frac{V_U}{\pi\, P\, {R_o}^2} \text{ (volume ratio)} \tag{2.13}$$

$$\lambda\, v_u = \frac{V_U \tan\beta}{2\,\pi^2\, {R_o}^3} \text{ (volume per turn ratio)} \tag{2.14}$$

The total number of buckets and the volume of water in each bucket across the screw are calculated by the following formulas.

$$N_B = \frac{L\,N}{P} \tag{2.15}$$

$$V_B = \frac{V_U}{N} \tag{2.16}$$

Using these simple formulas it is possible to estimate the volume of water in the screw per turn and the flow. This method cannot calculate the leakage losses as a function of the volume ratio so it is not accurate if these losses are high, except for they are given so flow can be calculated more accurate from the formula 2.6. However it can be used in the first stage of calculation to examine if the geometry (diameters, pitch etc) can handle the requested flow. Rorres (2000), using numerical simulations suggested the optimal characteristics for screws with various number of blades (1 - 25), aiming to maximize the volume per turn ratio, without however calculating leakage or overfill losses. It is not clear if the optimization criterion for the maximum flow leads to the optimal efficiency for a reference flow. In table 2.3 are presented the suggested values for screws with 3, 4 and 5 blades which are the most common in use today.

Table 2. 3 Calculated optimal characteristics for maximization of the volume per turn ratio for 3, 4 and 5 blades screws (Rorres 2000).

Number of blades	Radius ratio	Pitch ratio	Volume ratio	Volume per turn ratio
5	0.5352	0.263	0.2647	0.0696
4	0.5353	0.2456	0.2667	0.0655
3	0.5357	0.2217	0.2697	0.0598

It has not been totally clear which is the maximum efficiency that can be achieved by an AST. According to Andritz Atro’s website a hydrodynamic screw turbine can achieve up to 92 % efficiency. Lashofer (Lashofer et al 2012) after experimental study claims efficiency above 84 % with peak values of 94 %. Lyons (Lyons et al 2014), Rohmer (Rohmer et al 2016) and Dellinger (Dellinger et al 2016) present measurements or numerical estimations between 80 – 85 %. These numbers are referred to the nominal efficiency and it is not always clear if they are referred to the net or gross efficiency of the turbine, although that in case of laboratory conditions the net efficiency sounds more possible. The efficiency in partial load is an area in which more study could be conducted. Furthermore, is has not been so what is the minimum achievable flow and what is the correspondence of the device in this area. An efficiency curve not very high but relatively constant would be an advantage, if it could be achieved. An independent in detail assessment for Mannpower consulting Ltd which studies a hydro power station using Andritz Atro’s turbine (Bard 2007) results that for constant speed and flow from 21 - 60 %, the efficiency of the turbine varied from 49 % - 74 %. According to same assessment, the efficiency of the frequency converter, which is necessary for variable speed operation, was calculated from 86 - 96 %. Saroinsong (Saroinsong et al 2015) found optimal efficiency 89 % for an AST with outer diameter 0.055 m at 50 rpm with 25° slope angle under a head flow equal to the radius of the rotor.

Not many information about the specific speed of the AST were managed to be collected during the present study. Only one publication provides information about this area (Lashofer et al 2013). The data from the paper mentioned include 5312 values from laboratory measurements and 36 values from field measurements and they represent a performance range from < 1 up to 7 m head and from 2 - 45 U/min specific speed. The highest efficiency values are at low specific speeds between 3- 15 U/min. The reduced efficiency values below 15 U/min were from turbines with high tilt angle, high flow and insufficient speed, where neither in laboratory nor in field measurements were observed high efficiencies. For the laboratory measurements were used seven screw types at eight inclinations (18˚ – 32˚ in 2˚ steps). The flow rates varied from 20 l/s to 220 l/s with 20 l/s steps and the speed from 20 U/min to 80 U/min with steps of 5 U/min. The reference design used during the measurements had four blades with pitch ratio equal to 1 and radius ratio equal to 0.5. According to the authors, the AST is the turbine with the highest growth potential during the following decade.

2.2.9 Cost

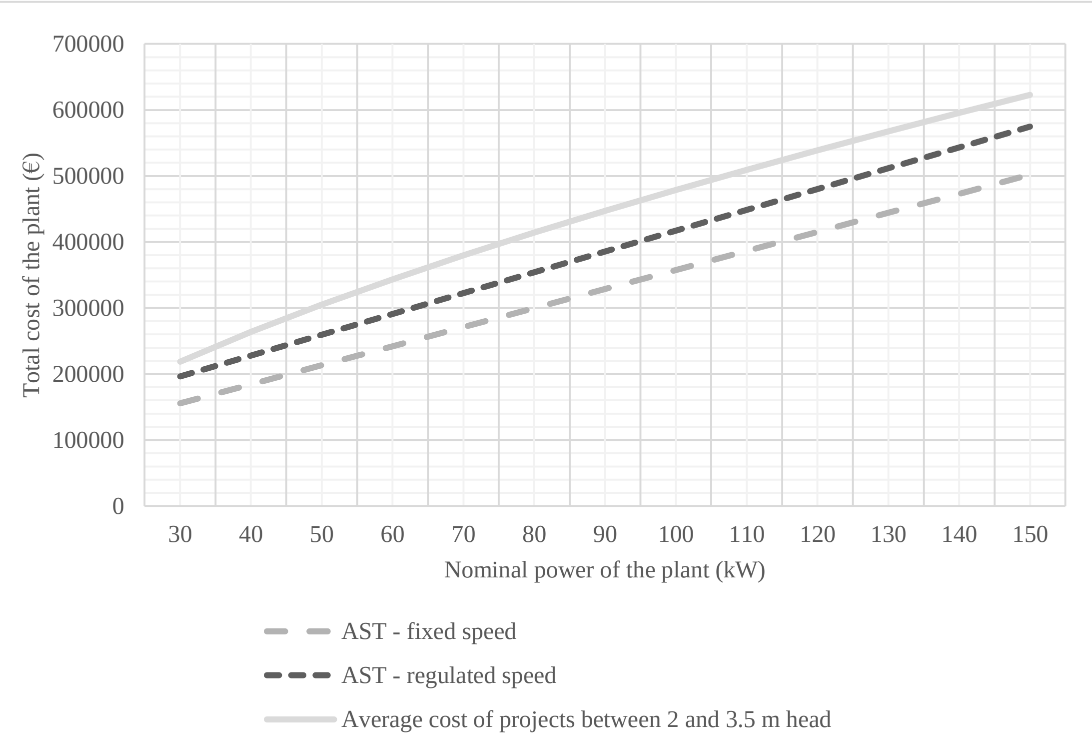


Figure 2. 21 Total project cost for hydro power plants using data from other turbine types in the area of 2 – 30 m head (Aggidis et al 2010) and fixed speed and variable speed AST (Lashofer et al 2012).

It is believed that AST are in general uneconomic (Tarrant 2011). Lashofer (Lashofer et al 2012) has published two formulas for the cost of power plant using data from 74 fixed and variable speed AST around Europe from 71 sites, in the range of 1 – 140 kW and head from 1 – 6 m, although the largest part (81 %) is up to 3.5 m head, and flows from 0.1 – 6 m^3/s, in diameters from 1 – 3.6 m.

Another very interesting information from this study is that the construction period for the projects of the sample is from 1 – 40 months and the average service time 1 hour per week. There are no reports about noise complaints from neighbours at 48% of the studied installations and the enclosure seems to be an effective measure against steady noise reduction and icing problems which were intense in temperatures below -10˚ C.

$$C_{RR} = 25000\ [Pe/H^{0.35}]^{0.65} \tag{2.17}$$

The estimated cost as a function of nominal power looks attractive based, related with curves from other source (Aggidis et al 2010), in which plants from 2 – 30 m head were studied. The data were published in 2008 (including inflation during these years) but they were collected in 1989 from Salford University Civil Engineering Ltd and they are from 50 sites in NW England with installed capacity in the range of 25 – 990 kW. However, the fact that the most of these projects may use higher efficiency turbines (Kaplan) should have as a result their income to be higher for similar hours of operation per year. Except for the used formula there are more for the estimation of the hydro power project (overall) and hydro turbine cost but the one's which were found during the present research are from even older studies. Since the Formula 2.17 covers installations with above 2 m head and the majority of the AST measurements are for head below 3.5 m the average cost of installations between 2 and 3.5 m head is presented in Figure 2.21 as comparison with the AST projects cost curves. The power in the Formula 2.17 is in kW and the result is in GBP (£).

Chapter 3. Numerical Modelling

The main goal of the present research is the understanding of AST's operation issues, advantages, disadvantages and design procedure. The aim of this approach is to make the reader able to make a preliminary design of an AST, on a realistic basis, able to work properly as one of the main subsystems of a small hydro power plant and exploit as much as possible of the available flow and head with the optimal efficiency. To do that, a numerical model, generally simple and as accurate as possible, was developed based on available literature able to take into consideration all the variables that have any known effect on the performance. The losses mechanisms were modelled based on available literature from a number of researchers. As programming environment was used the Matlab (2013 b) because of its simplicity and user friendliness and because it provides a large number of tools to directly visualize results, without need to transfer them to any other software like Excel, which leads to time saving.

3.1 Power Production and Losses Modelling

The present chapter was developed based on existing literature from a large number of studies about power production, performance and losses mechanisms in AST published during the last years (Lubitz, et al 2014, Nuernbergk et al 2013, Lyons et al 2014, Kozin et al 2014, Rohmer et al 2016, Dellinger et al 2016, Lubitz 2014, Kozyn et al 2015). The bearing losses were calculated based on a calculation method which is available from a well - known bearing manufacturer (SKF). As a summary, the calculated power sets are the followings aiming to determine the efficiency and power output both of the turbine and the total scheme:

- Total available power.
- Forebay losses
- Delivery open channel losses.
- Screening losses.
- Turbine inlet transition losses.
- Total power that can be produced in case of no losses.
- Leakage losses form the gap between rotor and trough.
- Overfill losses in case of high fill level (f > 1).
- Friction losses created by the trough and the rotor.
- Bearing resistance losses.
- Drag losses because of the submerged part of the rotor at the outlet.
- Turbine outlet transition losses.
- Transmission losses.
- Losses in frequency converter (for variable speed).

- Electrical losses (generator, transformer).

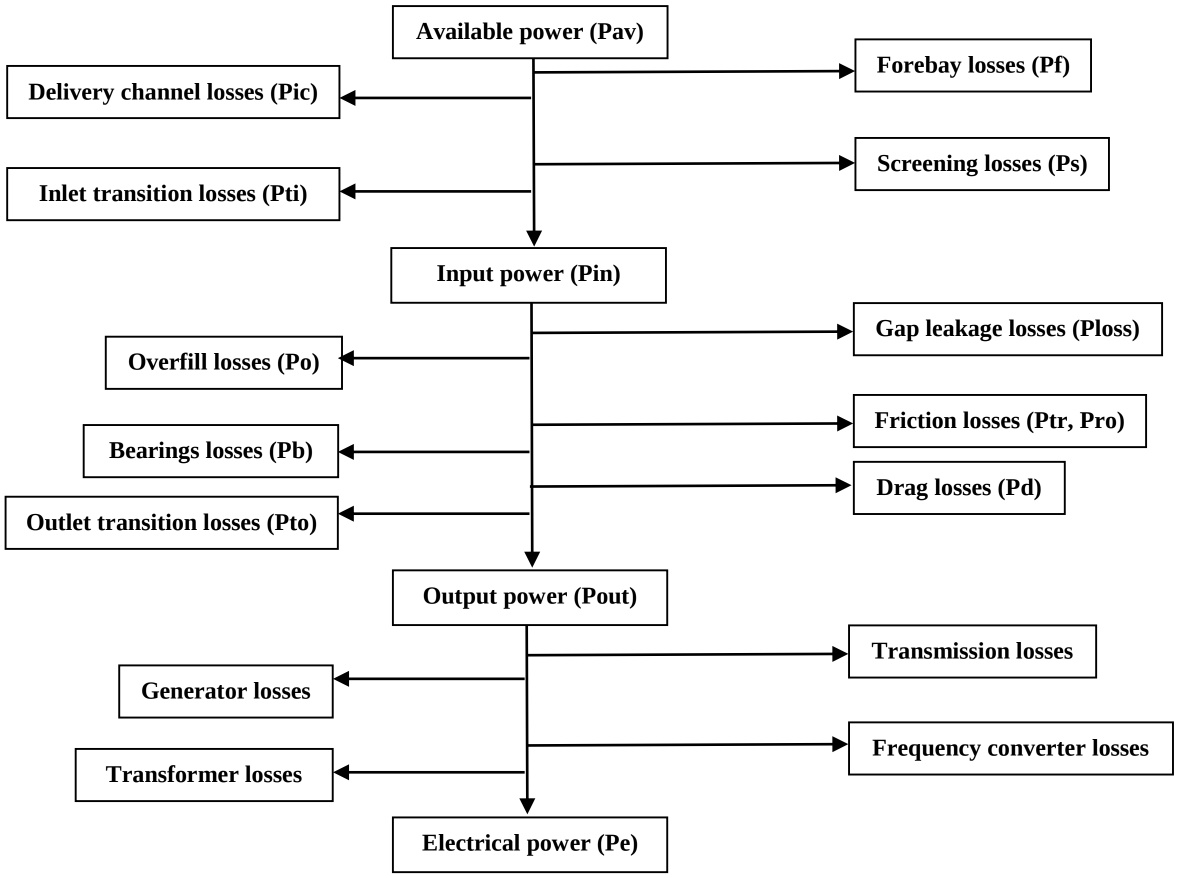


Figure 3. 1 Diagram of the power production and loss processes in a hydropower scheme using AST.

As a general outline, the algorithm needs as input all the geometrical characteristics of the screw (inner, outer radius, pitch, length, number of blades, speed, gap, slope angle, roughness), the dimensions of the upstream and downstream open channel which is assumed rectangular (width, length) and the water levels upstream and downstream, in combination with the available flow for each water level. When the optimizing tool for the geometry is being used, the changing variables are chosen and their values, single values or range, are introduced.

The algorithm starts with the definition of the fill level, from the minimum to the maximum value that has been defined with the step that has been also defined. From the fill level at each point the equivalent flow is estimated which is mapped to the available flow which is also correlated with a value for the upstream and downstream water level from the introduction of the algorithm. If the water level at the reservoir before the sluice gate is known for every upstream water level, the forebay losses are calculated. For known dimensions (length, width), surface quality of channel walls, water level and flow, the friction slope and the losses are calculated for the delivery channel. Calculating the approach speed from the flow, for known the basic geometry of the screen, the screening losses are calculated. Knowing the water level at the inlet and outlet of the turbine from the fill level and the water level upstream and downstream, the transition losses are calculated. The zwl estimated by the fill level for known geometry leads to the calculation of the total power and the overfill losses. The losses at the secondary devices (transmission, generator, frequency converter) can be estimated if the efficiency curve and the nominal power of each device is known.

$$\text{Turbine efficiency} = \frac{\mathrm{P_{out}}}{\mathrm{P_{in}}} \tag{3.1}$$

$$\text{Overall efficiency of the scheme} = \frac{\mathrm{P_{e}}}{\mathrm{P_{av}}} \tag{3.2}$$

3.2 Forebay losses

Since there are no available data from the site, in the present research no forebay losses are considered. In fact, the use of sluice gate from the reservoir to the delivery channel for the water level regulation has as a result head losses. Since the AST schemes belong to the low head range in small and micro hydropower in terms of nominal power, even a small water level difference before and after the sluice gate will lead to significant power loss in the overall, water to wire, estimation. Because of this, the use of sluice gate for water level regulation should be avoided where it is possible. The installation of such a gate is necessary for isolation in case of emergency and maintenance periods. The use of it for flow regulation can be avoided if the turbine has variable speed operation and the available flow is within the permissible limits that the diameter of the turbine can handle. If the flow does not vary significantly, the variable speed can be avoided as well. If an open channel will be built in which the turbine will be installed, parallel to a run of river, the geometry of the channel (width, height) can be defined to achieve the requested flow for the available water levels. Both the variable speed and the channel design approaches can be combined. Of course, if the water level variation upstream is very high the diameter of the rotor should be high to handle it, or else a sluice gate has to be used. In case of relatively limited variation of the reservoir water level in a future application would make interesting the comparison between the following two scenarios: a) head losses because of regulation with sluice gate and transition at the inlet of the turbine and b) no use of sluice gate for regulation and head losses only because of the transition at the inlet of the turbine. The forebay losses are considered to be created only because of water level difference and calculated using the formula 3.4. The flow passing through the opening of the sluice gate, which is the available flow for the turbine in the delivery channel, is calculated based on the guide of ESHA (ESHA 2004) from the formula 3.3. The parameter δ is equal with 90 from the mentioned reference. The use of sluice gate will lead to some friction losses because of the acceleration of the flow through the opening.

$$Q = \sqrt{2\, g\, h_{\text{upper reservoir}}}\ w\, t\, \left[0.98\left[\frac{4 + 5\, e^{-0.76\,\delta}}{9}\right] - e^{\left[\frac{t}{2\, h_{\text{upper reservoir}}}\left(1-\frac{\delta^2}{w}\right)\right]}\right] \quad (3.3)$$

$$P_f = g\, Q\, (h_{\text{upper reservoir}} - h_{\text{after the sluice gate}}) \quad (3.4)$$

3.3 Delivery channel friction losses

Before the entrance of the AST the flow is in a delivery open channel from the reservoir to the turbine's entrance. The cross section area of the channel in such applications is usually rectangular, at least during the last meters before the inlet. For the rectangular channel and for known water level, width and friction coefficient on the walls, the flow is calculated using formulas 2.7 – 2.9. In the Manning's formula for the calculation of the average speed has to be used the friction slope (S_f). Usually however, and in many publications, is used the slope of the channel's bed (S_o). This happens possibly because these two slopes do not have a significant difference and also because the friction slope is not easy to be estimated in many cases, while the bed slope is a known geometrical size.

The friction slope can be estimated if the geometry of the cross section, the flow and the Manning's coefficient are known (Apsley 2016). The specific energy difference per length (dh/dx) can be estimated (3.6) and assuming that the friction slope is constant in the length of the channel, the total specific energy change can be calculated for known length, which is equivalent with the head loss.

$$S_f = \frac{n^2 Q^2}{R^{4/3} A^2} \quad (3.5)$$

$$\frac{dh}{dx} = S_f \quad (3.6)$$

$$h_{ic} = \frac{dh}{dx}\, L_{\text{channel}} \quad (3.7)$$

$$P_{ic} = g\, Q\, h_{ic} \quad (3.8)$$

There is an equivalency between the Manning's friction coefficient n and the absolute roughness of the wall k (Charbeneau 2011) using Strickler's formula (3.9). This equivalency is very useful for calculations where the roughness (μm, ft or any other units) of a surface is known. The same publication provides a diagram about the Manning's friction coefficient (nm) in case of grassed channel as a function of the flow, the hydraulic radius of the cross section (R) and the average length and density of the grass. Almost always the open channels have at least a small quantity of grass. In low speeds of the flow (< 1 m/s) the grass has a greater development and to avoid energy losses because of friction it must be cleaned from time to time where it is possible. The diagram in the publication mentioned has a great interest because it shows that the friction coefficient n can be much higher in case of a greased channel (0.022 – 0.4) related with a clean of grass concrete open channel (0.01 – 0.02).

$$nm = 0.034\ k^{1/6}\ \ (\text{k in ft}) \tag{3.9}$$
$$nm = 0.034\ (3.28\ 10^{-6}\ k)^{1/6}\ \ (\text{k in μm})$$

3.4 Screening losses

In the most hydro power projects a screen is necessary, not only for fish protection and secure migration but also as trash rack to avoid the entrance in the turbine of debris and other materials that possible are in the flow. A number of formulas have been developed by various researchers, based on the literature covered (ESHA 2004, Raynal et al 2014, Johiah et al 2016, IS 11388), for the calculation of the head loss which is usually considered small in hydropower applications. In the present research is used a formula developed by Measburger in 2002 (Raynal et al 2014). This formula takes into consideration the bar thickness (t), the width between the bars (b), the approach velocity (U_o), the angle of inclination of the rack related with the channel bed (φ) and a correction factor (K_t) which depends on the shape of the bars. The correction factor is suggested 2.42 for rectangular bars and 1.79 for round bars. For rectangular bars, as the ones in the present research, is considered equal with 2. IS 113888 suggests the multiplication of h_t with a factor between 1.75 and 2 to include bracing and frame effects.

$$h_s = K_t\ \frac{{U_o}^2}{2g}\ \sin\varphi\ (\frac{t}{b})^{1.5} \tag{3.10}$$

$$P_s = g\ Q\ h_s \tag{3.11}$$

3.5 Total power

The power calculated in this part is the total available power, minus the power that cannot be exploited because of geometrical parameters, taking into consideration no other losses. In other words it is the theoretical maximum power that the screw can produce. The geometrical parameters mentioned are the gap between the trough and the impeller, which cannot be avoided and has as a result the gap leakage losses, and the length of the rotor which in case that it is not enough will not be able exploit the total available head and will lead to head losses because of the water level difference between the exit channel and the last bucket of the screw.

Based on Lubitz et al 2014 the calculation was developed as described in the following lines.

$$z_{min} = -R_o \cos\beta - \frac{P}{2} \sin\beta \tag{3.12}$$

$$z_{max} = R_i \cos\beta - P \sin\beta \tag{3.13}$$

$$z_{wl} = z_{min} + f\,(z_{max} - z_{min}) \tag{3.14}$$

$$f_{max} = \frac{\cos\beta\, D_o}{0.5\,(\cos\beta\,(D_o + D_i) - \sin\beta\, P)} \tag{3.15}$$

In the same document, the maximum fill level presented is 1.25. Nuernbergk reports that according to Brada a water screw can handle a flow up to 120 % of the flow in case of optimal filling (f = 1) without significant efficiency losses. It is not clear what a significant loss means, in terms of efficiency percentage drop. However it sounds feasible that the turbine's flow can be increased considerably, if an efficiency drop is acceptable, to make possible the exploitation of higher flow. If f is known or it is considered, as in the present study, it is possible to calculate the z_{wl}.

$$z_1 = r \cos\beta \cos\theta - \frac{P\,\theta}{2\,\pi} \sin\beta \tag{3.16}$$

$$z_2 = r \cos\beta \cos\theta - \left(\frac{P\,\theta}{2\,\pi} - \frac{P}{N}\right) \sin\beta \tag{3.17}$$

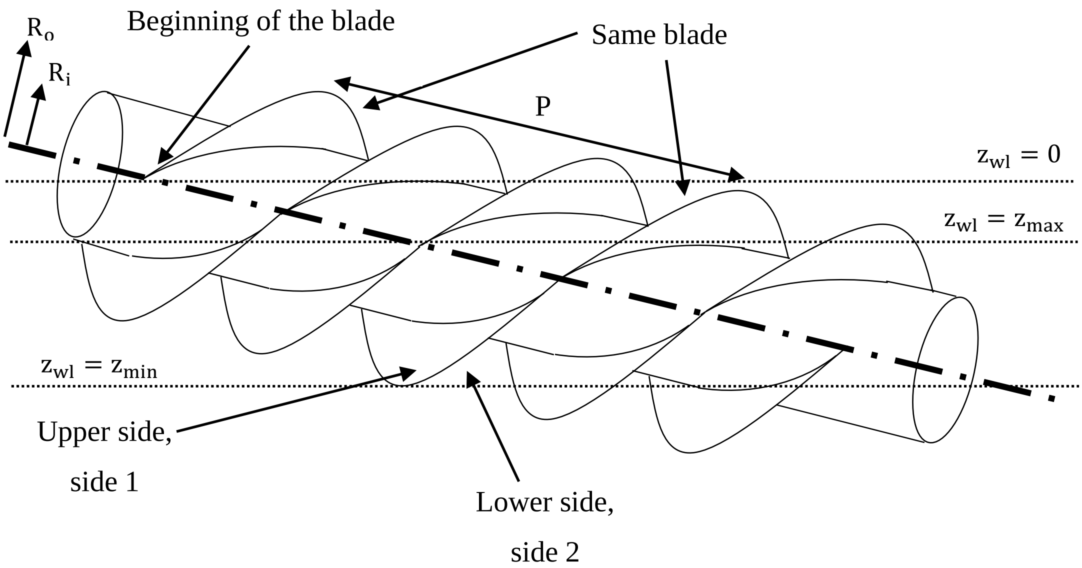


Figure 3. 2 Details that are used for the calculation of the total power (Lubitz et al 2014).

The theoretical maximum fill level, above which no more power is produced is calculated by formula 3.15. The relative height of each point on the blade is calculated by the formulas 3.16 and 3.17, integrating variable r from R_i to R_o and variable θ from 0 to 2π.

The steps (rstep, θstep) were chosen so the dr to be up to one millimeter and the dθ to be up to one degree. The formula 3.19 considers that the volume and the geometry of the water in every bucket is identical. This assumption is not absolutely realistic because in the first and last bucket there are significant variations because of transition. However, the fact that the number of buckets is calculated as real numbernand the introduction of transition losses in next chapter create a good approch to the estimation of the real torque and volume. In formula 3.23 are calculated two pressures, 1 and 2 for both sides of the blade. As input variable for the part of the algorithm which was described above, the fill level f is given, aiming to calculate the flow for various fill levels which will also result to various flows.

The volume of water in the bucket, which is useful for the estimation of the flow and the volume of water is calculated by the formulas 3.18 to 3.21.

$$\mathrm{dV} \begin{cases} 0 & z_2 > z_{wl},\ z_1 > z_{wl} \\ \left(\frac{z_{wl}-z_1}{z_2-z_1}\right) \frac{P}{N}\, r\, dr\, d\theta & z_2 \geq z_{wl}, z_1 \\ \frac{P}{N}\, r\, dr\, d\theta & z_2 < z_{wl},\ z_1 < z_{wl} \end{cases} \tag{3.18}$$

$$V = \int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} dV \tag{3.19}$$

$$dr = \frac{R_o - R_i}{rstep} \tag{3.20}$$

$$d\theta = \frac{2\,\pi}{\theta step} \tag{3.21}$$

The torque of the bucket is calculated by the formulas 3.22 to 3.24.

$$p \begin{cases} \rho\, g\, (z_{wl} - z) & z \\ 0 & z \geq z_{wl} \end{cases} \tag{3.22}$$

$$dT = (p_1 - p_2)\frac{P}{2\pi} r\, dr\, d\theta \tag{3.23}$$

$$T = \int_{r=Ri}^{r=Ro}\int_{\theta=0}^{\theta=2\pi} dT \tag{3.24}$$

The total torque which is produced in the screw is calculated by sum of all buckets by formula 3.25.

$$T_{tot} = T\frac{L\,N}{P} \tag{3.25}$$

$$P_{tot} = T_{tot}\,\omega \tag{3.26}$$

3.6 Gap leakage losses

The gap leakage losses are the losses which are created because of the gap between the screw and the trough. This gap cannot be avoided because if there was no distance the rotating screw and the non-rotating trough would be in contact. This would create many significant problems in operation like high losses because of friction, high levels of noise and deformation both for the trough and the screw. The leakage flow can be estimated by various approaches using formulas 3.27 – 3.30. Formulas 3.28 and 3.30 calculate only the leakage in case of full bucket, so they cannot be used for leakage losses calculation in variable fill level because of variable flow and head. However, they can be used to provide with just a simple calculation a very initial estimation to the designer about the leakage loss as a function of the diameter in case of full bucket (optimal fill, f = 1).

Formulas 3.27 and 3.29 can be used for the estimation of losses in a range of flow and head variation since they can take into consideration the length of the arc of the diameter that is covered with water and can leak because of the gap mentioned. The calculation of the arcs at each fill level is possible after the estimation of the intersection points between the water surface and the inner and outer diameter of the blade (Rorres 2000). It has to be reminded here that the leakage is depended on the parameters C or C_R (0.65 - 1), which may vary for different geometries so the formula which will be used has to be calibrated for the geometry of the studied screw. For the gap leakage (Q_G) many expressions have been suggested which are calculated by the following expressions (Lubitz 2014, Lyons et al 2014).

The expressions 3.28 and 3.30 cannot calculate the leakage losses in variable fill level and they assume full bucket (Lyons et al 2014) but they can give an initial estimation on a very easy way about the leakage losses in nominal flow for a reference diameter of the screw.

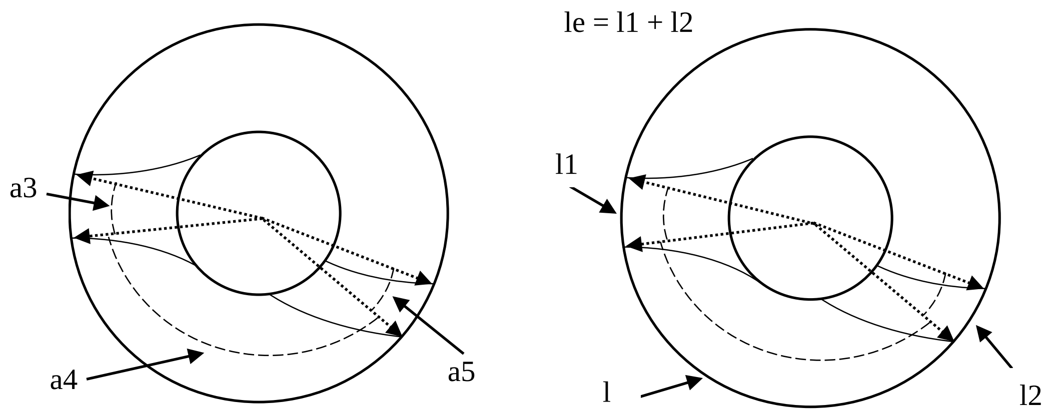


Figure 3. 3 Angles (left) and arcs (right) for the calculation of leakage losses from the formula 3.27 and 3.29 respectively.

$$Q_G = C_R\, s_{sp}\, \frac{D_o}{2} \left(1 + \frac{s_{sp}}{D_o}\right) \sqrt{1 + \left(\frac{P}{\pi\, D_o}\right)^2} \left(\frac{2}{3}\, a_3 + a_4 + \frac{2}{3}\, a_5\right) \sqrt{\frac{2\, g\, P}{N}\, \sin\beta} \tag{3.27}$$

(Muysken 1932)

$$Q_G = 2.5\, s_{sp}\, {D_o}^{1.5} \text{ (Nagel 1968)} \tag{3.28}$$

$$Q_G = C\, s_{sp} \left(l_w + \frac{l_e}{1.5}\right) \sqrt{\frac{2\, g\, P}{N}\, \sin\beta} \text{ (Lubitz 2014)} \tag{3.29}$$

$$Q_G = 5\, s_{sp}\, {D_o}^{1.5} \text{ (Lyons 2014)} \tag{3.30}$$

Formulas 3.27 and 3.29, if C = C_R and $s_{sp} \ll D_o$, they represent the same model with different variable combinations (Lubitz 2014). In the same reference, the parameter C is defined equal to 0.9 based on measurements of static leakage flow from a small screw. This value may not be valid for different screws. The parameter C_R lies in a range 0.65 – 1 (Nuernbergk et al 2013) and it is set to 1 in that paper for maximum gap leakage. The parameter l_w is equal to the angle a_4 multiplied by R_o and is equivalent to a length of arc which has water both on the upper and air on the lower surface. The parameter l_e is equal to the angle $a_4 + a_5$ multiplied by R_o and is equivalent to a similar length of arc but with water instead of air on the lower surface. The existence of water in this case leads to reduced pressure difference between the two sides of the gap. This may be the reason for the multiplier (1/1.5) of l_e in formula 3.29 which creates a reduced contribution to leakage flow for this parameter.

$$P_{loss} = g\, Q_G\, H \tag{3.31}$$

All the formulas for the calculation of gap losses have signignificant deviations from the real leakage (Kozyn et al 2015) and they can estimate the leakage on an acceptable level of accuracy only in relatively low speeds. However it is not clear, since no experimental data were found, what is the accuracy level of a reference geometry in acceptable variable speed range in combination with flow and water level variations.

3.7 Overfill losses

For more accurate estimation of the performance, leakage models were developed based on experimental data. The overflow leakage (Q_O) according to Aigner (Nuernbergk et al 2013) is given by the formula 3.32.

$$Q_o = \frac{4}{15}\,\mu\,\sqrt{2\,g}\,(\frac{1}{\tan\beta} + \tan\beta)\,(z_{wl} - z_{max})^{5/2} \tag{3.32}$$

The formula 3.32 is an expression for a triangular spillway. The coefficient μ, which depends on the shape of the weir and the flow direction, is set equal to 0.537 in the mentioned paper for the maximum overspill leakage. According to Brada (Nuernbergk et al 2013), a screw can handle up to 120 % of the optimal flow without significant efficiency losses because the overfill losses are not high up to this point. The overfilled flow does not participate to torque generation.

The overfill losses are calculated when the flow is increased above the flow for optimum fill (f = 1). It is possible that the overfill creates more losses, except for the leakage, because of the 'disturbance' of the flow in the next bucket, but this opinion needs further research in a future study. The coefficient μ is suggested equal to 0.537 as a spillway in the most publications (Lubitz et al 2014, Dellinger et al 2016, Nuernbergk et al 2013). According to Rohmer however (Rohmer et al 2016) this number does not take into consideration the kinetic energy and the surface velocity distribution and he suggests μ = 1.0633 for better accuracy.

$$P_o = g\,Q_o\,H \tag{3.33}$$

3.8 Friction losses

The friction losses is a result of the contact between the flow and the walls. In this problem there are two types of friction losses: a) because of the contact between the fluid and the trough (transport) and b) because of the contact between the fluid and the rotor (rotation). There are also friction losses across the delivery channel from the reservoir to the inlet of the turbine which will be studied in following chapter. The friction losses calculation methodology here is based on a number of available publications (Kozyn et al 2016, Nuernbergk et al 2013, Dellinger et al 2016).

$$L \begin{cases} \dfrac{H}{\sin\beta} & L \le L_{max} \\ L_{max} & L > L_{max} \end{cases} \tag{3.34}$$

$$A_{avg} = \frac{V_{tot}}{L} = \frac{V\,N}{P} \tag{3.35}$$

$$y_a \begin{cases} \sqrt{{R_o}^2 - y^2} - \sqrt{{R_i}^2 - y^2} & |y| \le R_i < R_o \\ \sqrt{{R_o}^2 - y^2} & y > R_i, |y| \le R_o \\ 0 & |y| > R_o \end{cases} \tag{3.36}$$

$$A_{avg} = \int_{-R_o}^{h_{avg}-R_o} y_a\,dy \tag{3.37}$$

For the estimation of the friction losses, and the transition losses later, the estimation of the average crossover surface of water (A_{avg}) and the average water level in the trough (h_{avg}) are necessary and also the volume of water in the bucket (V) which was calculated earlier (formula 3.19) and the active length of the screw L. L_{max} is the length of the screw covered with blades.

The average water level (h_{avg}) is estimated using numerical iteration for the formula 3.37. When the right part is equal to the left, $h_{avg} = y_a$. The losses because of transport are calculated by formula 3.47. The Darcy - Weisbach friction coefficient f is calculated using one form of Colebrook equation developed by Zigrang and Sylvester in 1982, which has very small error compared with the implicit Colebrook equation (Brkić 2011).

$$f = \left[\frac{1}{4\log_{10}(\frac{k}{3.7\,4\,R} - \frac{5.02}{Re}\log_{10}\left(\frac{k}{3.7\,4\,R} - \frac{5.02}{Re}\log_{10}\left(\frac{k}{3.7\,4\,R} + \frac{13}{Re}\right)\right))}\right]^2 \tag{3.38}$$

$$h_f = f\,\frac{L}{4\,R}\frac{{U_s}^2}{2g} \tag{3.39}$$

The hydraulic radius R is calculated as in 2.9 as the ratio of flow cross sectional area to the wetted perimeter and Re is the Reynolds number. Roughness (k) and hydraulic radius (R) must have the same units, here both in m. Typically, the Darcy – Weisbach friction factor f is determined by the Moody diagram. The methodology which used here has the advantage that correlates the factor f with the roughness and the Reynolds number, based on available literature, which makes possible the direct calculation of the f from the properties of the flow and the wall surface.

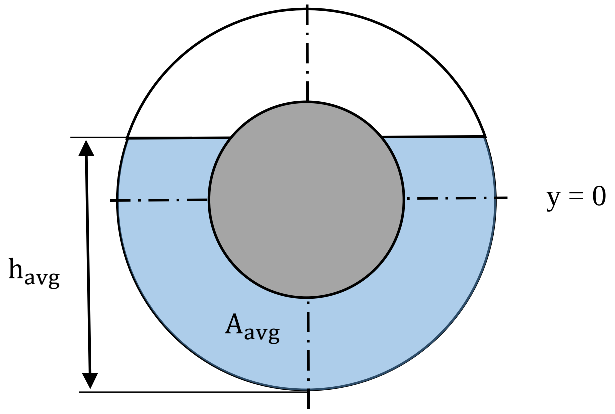


Figure 3. 4 Cross sectional view (idealized) of the flow in an AST.

$$\tau_t = f_t \frac{\rho\, {U_s}^2}{8} \tag{3.40}$$

$$\tau_s = f_s \frac{\rho\, {U_s}^2}{8} \tag{3.41}$$

$$dA_t \begin{cases} 0 & z_2 > z_{wl}, z_1 > z_{wl} \\ \left(\frac{z_{wl}-z_1}{z_2-z_1}\right) \frac{P}{N} R_o\, d\theta & z_2 \geq z_{wl}, z_1 \\ \frac{P}{N} R_o\, d\theta & z_2 < z_{wl}, z_1 < z_{wl} \end{cases} \tag{3.42}$$

$$A_t = \int_{\theta=0}^{\theta=2\pi} dA_t \tag{3.43}$$

$$dA_s \begin{cases} 0 & z_2 > z_{wl}, z_1 > z_{wl} \\ \left(\frac{z_{wl}-z_1}{z_2-z_1}\right) \frac{P}{N} R_i\, d\theta & z_2 \geq z_{wl}, z_1 \end{cases} \tag{3.44}$$

$$\frac{P}{N}\, R_i\, d\theta \qquad\qquad z_2 < z_{wl}, z_1 < z_{wl}$$

$$A_s = \int_{\theta=0}^{\theta=2\pi} dA_s \tag{3.45}$$

$$W_{tr} = \rho\, \frac{N\, P}{L}\, V\, h_{tr} = N\, P\, (\tau_t\, A_t + \tau_s\, A_s) \tag{3.46}$$

$$P_{tr} = g\, Q\, h_{tr} \tag{3.47}$$

Nuernbergk had suggested that the flow in an AST is similar to an open channel flow, in which the Manning's coefficient can be used to approximate the shear friction (Kozyn et al 2016), and in this case the shear stress on the trough (τ_t) and the shear stress on the main shaft, not including the blades, (τ_s) can be determined. After calculating the surfaces of the trough and shaft which are equivalent for each bucket (3.43, 3.45), the forces because of the shear stresses can be estimated (F_t, F_s) and the total wall energy loss because of friction. Nuernbergk also suggested the expression 3.46 for the wall energy loss because of friction.

So far were calculated the losses from friction with the shaft and trough because of transport. In other words, these losses would exist again if the screw was not rotating. There are also friction losses because of the rotation created from the contact between the water, blades and the main shaft. The calculation of losses because of rotation is a bit more complicated because of the geometry of the blades which has to be modelled. Again, 1 is referred to the upper (downstream) and 2 to the lower (upstream) surface of the blade which is covered by water. Formulas 3.55 and 3.56 estimate the weighted average wetted radius for plane surfaces 1 and 2.

$$U_r = R_i\, \omega \tag{3.48}$$

$$\tau_r = f_r\, \frac{\rho\, {U_r}^2}{8} \tag{3.49}$$

$$P_{rs} = \tau_r\, U_r\, A_s\, \frac{N\, P}{L} \tag{3.50}$$

$$dA_1 \begin{cases} 0 & z_1 > z_{wl} \\ \dfrac{\sqrt{4\,\pi^2 r^2 + P^2}}{2\,\pi\, r}\, r\, dr\, d\theta & z_1 \leq z_{wl} \end{cases} \tag{3.51}$$

$$dA_2 \begin{cases} 0 & z_2 > z_{wl} \end{cases} \tag{3.52}$$

$$\frac{\sqrt{4\,\pi^2 r^2 + P^2}}{2\,\pi\,r}\, r\,dr\,d\theta \qquad\qquad z_2 \le z_{wl}$$

$$A_1 = \int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} dA_1 \tag{3.53}$$

$$A_2 = \int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} dA_2 \tag{3.54}$$

$$r_{1av} = \left(\frac{\int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} r^3 dA_1}{A_1} \right)^{1/3} \tag{3.55}$$

$$r_{2av} = \left(\frac{\int_{r=Ri}^{r=Ro} \int_{\theta=0}^{\theta=2\pi} r^3 dA_2}{A_2} \right)^{1/3} \tag{3.56}$$

$$P_{ro} = \frac{N\,P}{8\,L}\, f_b\, ({r_{1av}}^3\, A_1 + {r_{2av}}^3\, A_2) \left(\frac{\pi\,n}{30}\right)^3 \tag{3.57}$$

The Manning's coefficient n is the same for the blades and the main shaft because they belong to same part (rotor) and they are both made from the same material (steel). According to EJP (EJP 2008) the coefficient n for welded steel is between 0.1 and 0.14 and for the present study it is assumed equal to 0.12. The trough, although it was concrete in older installations, is covered by a sheet metal made of steel in our days. The coefficient n is considered 0.12 for the trough as well.

3.9 Bearings losses

The bearings of the rotor create resistance because of the friction between their parts (rings, balls etc) which however is not high. For the estimation of the resistance moment created by the bearings are used two simplified formulas from the website of SKF. The first formula calculates the frictional moment of the bearing (3.58), assuming that the bearing operates in normal conditions, it has good lubrication and the bearing load $Pe \approx 0.1\,C$, where C is the dynamic load rating of the bearing in kN. The second formula (3.59) estimates the power loss as a result of bearing frictional moment in kW. For the present study it has been considered that both the bearings of the screw are sealed spherical roller bearings (SKF 2000, SKF 2007). This arrangement although is simple, can easily accommodate misalignments which may appear from potential manufacture difficulties because of the large size of the turbine. Also, it has been considered that the upper bearing can receive both axial and radial load while the lower only radial, because since the lower bearings works under more difficult conditions (underwater) related with the upper, it should not be loaded the same. The radial (F_r) and axial (F_a) forces that the bearings with receive are calculated from the structural model of the rotor. The equivalent load of each bearing (Pe) is calculated according to SKF (SKF 2000) guide for spherical roller bearings (3.60). The parameter μ is the coefficient of friction for the bearing and in the studied method is considered constant and equal to 0.0018 for this bearing type. The parameters used in formula 3.60 (2.5, 1, 3.7, 0.67, 0.27) are for the particular bearings and their dimensions (SKF 2000), and they should change in case of using different bearing or different dimension of it.

$$M = 0.5\ \mu\ Pe\ d \tag{3.58}$$

$$P_b = 0.105\ M\ n_r \tag{3.59}$$

$$Pe \begin{cases} 2.5\ F_a + F_r & \dfrac{F_a}{F_r} \leq 0.27 \\ 3.7\ F_a + 0.67\ F_r & \dfrac{F_a}{F_r} > 0.27 \end{cases} \tag{3.60}$$

3.10 Drag losses

The drag losses are produced because of the resistance which is created on the submerged part of the screw. Kozyn (Kozyn et al 2015) presents an experimental effort to quantify these losses based on an initial numerical estimation using dimensionless flow and outlet water level parameters. Nuernbergk has suggested a formula (3.61) for the optimal downstream level (Dellinger et al 2016).

$$H_{2_{opt}} = (R_i + R_o)\sqrt{1 - \left(\frac{\tan\beta\ P}{2\ \pi\ R_i}\right)^2}\cos\beta - \frac{P}{N}\sin\beta \tag{3.61}$$

3.11 Transition losses

Transition losses are, on a very simplified way, the losses which are appeared when the flow goes in and out the part of the open channel where the trough and the screw are. For the transition from the inlet to the trough, Nuernbergk (Nuernbergk et al 2012) suggested the use of Bernoulli energy equation in one dimension with the Borda – Carnot equation for head loss, which is also used by Kozyn (Kozyn et al 2015). The only exit head loss estimation, from the trough to the outlet, which was found during this research has been done by Kozyn and uses the Borda – Carnot coefficient. These losses generally considered minor in pipe systems with significant length because they are very small related with the friction losses. The velocity in the following formulas is squared, so its contribution is very high. This method for transition losses and the Borda – Carnot coefficient can be used with good accuracy for both flow in open channels but also for flows in pipes. However, it should be reminded that this method was developed for relatively smooth transition between open channel and it is not clear how accurate can include the contribution of the trough's slope, which is not small, and the rotation of the screw which may create a more complicated condition related with the modelled one at the inlet and outlet.

For the present research a head loss approach at the inlet and outlet is used from available studied literature (Chanson 2004, Tullis et al 2008, Tullis et al 2008, Massey et al 2006, Oertel 2004) using Borda – Carnot formula, considering the inlet and outlet are abrupt abrupt enlargement and contraction areas. Oertel (Oertel 2004) introduces the Weisbach's contraction coefficient a, which is used to estimate the participation of the narrowing after the wide area in the head loss from one open channel to another in case of contraction. It has been found experimentally that Weisbach's coefficient is obtained for $A_{i+1}/A_i < 0.7$. The inlet and outlet of the screw are assumed that can be either enlargement or contraction since the criterion for this is the ratio of the areas on both sides of the cross-section surface.

$$a = 0.63 + 0.37\left(\frac{A_{avg}}{A_1}\right)^3 \tag{3.62}$$

$$h_{ti} \begin{cases} \frac{{U_o}^2}{2\,g}\left(1 - \frac{A_1}{A_{avg}}\right)^2 & A_1 \leq A_{avg} \\ \frac{{U_o}^2}{2\,g}\left(\frac{A_1}{a\,A_{avg}} - 1\right)^2 & A_1 > A_{avg} \end{cases} \tag{3.63}$$

$$a = 0.63 + 0.37\left(\frac{A_2}{A_{avg}}\right)^3 \tag{3.64}$$

$$h_{to} \begin{cases} \frac{{U_s}^2}{2\,g}\left(1 - \frac{A_{avg}}{A_2}\right)^2 & A_{avg} \leq A_2 \\ \frac{{U_s}^2}{2\,g}\left(\frac{A_{avg}}{a\,A_2} - 1\right)^2 & A_{avg} > A_2 \end{cases} \tag{3.65}$$

$$P_{ti} = g\,Q\,h_{ti} \tag{3.66}$$

$$P_{to} = g\,Q\,h_{to} \tag{3.67}$$

3.12 Frequency converter losses

In case of wide fluctuation in the site's flow characteristics, mainly in head, the use of variable speed constant frequency system, or frequency converter or inverter as it is also known among the technical staff, is possible to be used. The frequency converter is used to connect the generator via a DC link to the grid and can 'synchronize' to the grid even before the generator starts rotating (ESHA 2004). The cost of a system is high and it should be used, if it is necessary, only where there no other options. For example, if the turbine installed should be a propeller, based on the performance range, from the Kaplan family which are the main alternative turbines in low head, it is much preferable, in terms of cost, simplicity and efficiency, to be installed a double regulated Kaplan turbine without a frequency converter than a Propeller turbine (without regulated blades and guide vanes) with a frequency converter. It is not actually possible to improve the energy production compared to a double regulated Kaplan turbine (ESHA 2004). This happens mainly because of the relatively high losses of the frequency converter compared with the losses of the hydraulic system that is being used for the regulation of the blades, especially as getting away from the design point. Furthermore, if the variation of the flow characteristics is very extensive, the use of frequency converter without regulated blades and vanes may not be able to exactly adapt the operation point.

The frequency converter is used to allow variable speed operation. The advantage of an AST with variable speed is significant in reduced flow, especially in case of large flow variations. For the research covered in the present study only a few references were found for use of electronic equipment for continuous regulation of speed in screw turbines (Lashofer et al 2012, Bard 2007). According to Lashofer, hydro power stations suffer an overall efficiency loss of $\geq$ 3 % in case of frequency converter use, although it is a preliminary conclusion and further measurements are necessary. Bard presents measurements of the variable control system (inverter) efficiency as a function of the project's overall output power (after the inverter and the transformer) and the generator's output power (before the inverter). The maximum overall power of the studied project is 65.24 kW, while the maximum power of the inverter is 55 kW. The accuracy in power readings is estimated better than $\pm$ 2.5 %. The measurements do not cover the field of high output power, above 70 %, and it seems that there is large drop of the frequency converter efficiency for output power below 50 % of the maximum. Bard makes a smart suggestion in his report, to switch over the variable control system to constant speed at high flows, avoiding the efficiency drop because of the variable speed system. The trend line in the figure 3.5 is an estimation and does not come from the report mentioned. In case that there is no frequency converter (fixed speed AST), the power entering the transformer is the power exiting the generator.

$$P_i = P_{gen} \, n_i \tag{3.68}$$

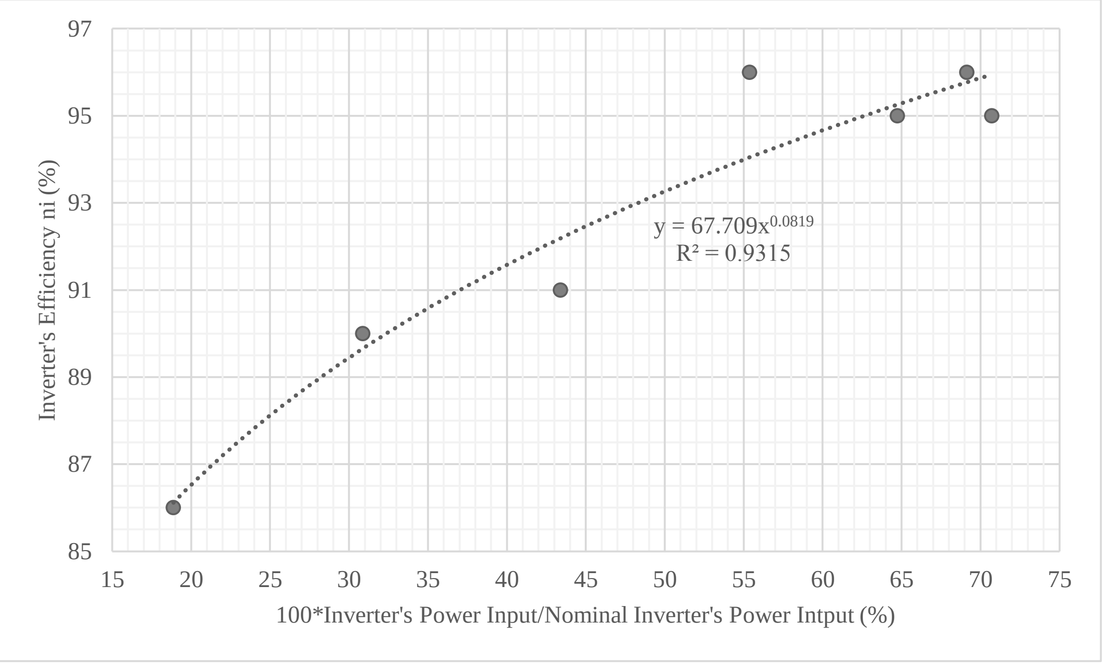


Figure 3. 5 Efficiency of the frequency converter as a function of the input power of the inverter (Bard 2007).

Figure 3. 6 Variable speed constant frequency systems or frequency converters (www.boschrexroth.com).

3.13 Transmission and electrical losses

The very low speed of a hydrodynamic screws (either as a turbine or as a pump) makes a transmission system necessary. Theoretically, if the requested speed of the screw was equal to a generator's speed with a specific number of poles, a direct drive connection would be possible. In practice however it is not possible to avoid transmission with screw hydraulic turbomachines because a) only in very specific diameters (formula 2.4) the speed can result to be the same, or close enough to it, as the requested from the generator to be connected directly (e.g for external diameter 1 m the speed should be 50 rpm, the same with a synchronous generator with 30 pairs of poles) and b) even if it was possible the generator for such low speed would be huge and very expensive because of the large number of poles which are necessary to achieve that low speed. An AC generator is also called alternator, a common term in renewable energy applications. Sometimes, at least in documents related with hydropower applications, as generator is meant the overall systems of the turbine with the electrical generator and the secondary electromechanical equipment.

Since the speed is low in this type of turbines there is always need for a drivetrain system. The existing experience from the maintenance and troubleshooting of the wind turbines (Sheng, et al 2013) has shown that the drivetrain is one of the most sensitive parts of the equipment and it is responsible for a large number of failures and significant downtime periods. The most well-known transmission systems for industrial applications are the gearbox or a bevel or helical gear or parallel shaft connection, the belt drive and the chain drive but in hydro power applications, for the transmission between the turbine and the generator usually is used either gearbox or belt drive, if direct drive is not an option. In many instances, if the technical or other limitations impose it, it can be used a combination of the options mentioned like gearbox and belt drive or gearbox and bevel gear. This of course would lead to increased complexity and efficiency drop.

From unofficial discussions with European manufacturers during this research, for the studied applications (low speed screw turbines) the estimated nominal efficiency of a gearbox is around 97 % and the estimated nominal efficiency of a belt drive system is around 94 %, both in case of 1500 rpm generator. According to the same discussions, the efficiency of the transmission system is defined mainly, at least for the power range in the area of 'micro' and 'small' hydropower, from the speed ratio. As the speed ratio is increased the number of stages is increased, either concerning gears or pulleys, and the efficiency of each stage rises in the number of stages. In addition the belt drive system has similar cost. So from the information collected so far, although the belt drive is a simple option and easy to maintain, the gearbox seems to be the optimal option since it is more efficient (at least in nominal point), more compact and in about the same cost. Based on the unofficial discussions mentioned, the belt drive can be used effectively even with significant power but with relatively high speed machines (e.g. Kaplan). In case of low speed machines, the belt drive is suggested only in low power. For the present research the efficiency curves for the gearbox and the generator (asynchronous) were taken from the website of Permanent Magnet Generator Applications (P.M.G.A.), where there is a performance comparison of a hydro power plant with a double regulated Kaplan turbine using an asynchronous generator with a gearbox in one case and a permanent magnet generator in the other. No research on permanent magnet generators was made during the present study but it is definitely suggested for the future since it seems that a permanent magnet generator has high efficiency and significant advantages, according with the comparison described in the website mentioned. For an overall water to wire power calculation a transformer is assumed just before the connection with the grid, which is necessary for the final transformation of the power to electricity, with efficiency 99 %. A number of pictures are provided in the next pages giving examples about the view of the generator, the transmission and the transformer which are used in hydropower stations.

$$P_{gea} = P_r \, n_{gea} \tag{3.69}$$

$$P_{gen} = P_{gea} \, n_{gen} \tag{3.70}$$

$$P_e = P_i \, n_t \tag{3.71}$$

Figure 3. 7 Efficiency of the gearbox (ngea) and the asynchronous generator (ngen) as a function of the load (www.pmga.eu).

Figure 3. 8 Figure 3.7 Horizontal (www.hitzinger.us) and vertical (www.marellimotori.com) hydro power generators.

Figure 3. 9 Parallel shaft gearbox (www.powerpal.co.uk) and belt drive system (www. canyonhydro.com) in small hydro turbines.

Figure 3. 10 Archimedes screw turbines using belt drive + gearbox (www.emersonindustrial.com) and parallel shaft gearbox with no belt drive (www.renewablesfirst.co.uk).

Figure 3. 11 Large transformers installed in hydropower plants (www.seatrasformatori.it).

Chapter 4. Validation and Results

The validation of the results is top priority of any kind of simulation or modelling. Without validation it is not possible to be proved that the results are in correct direction. For the validation of the present results was arranged a visit to an existing hydro power station with AST. The meeting with the developer and the discussion of potential research directions and any existing gaps according to his experience was more that useful for the formation of the topic. To validate the developed model, manufacturer's data were used, achieving finally satisfactory approach with the numerical results for this level of study.

4.1 The Case Study

After the first weeks of the course and the finalization of the topic, it was suggested that any contact with experts from the industry in the field of AST hydropower development would be very useful. For this reason, a hydropower station in operation with an AST was used as a case study.

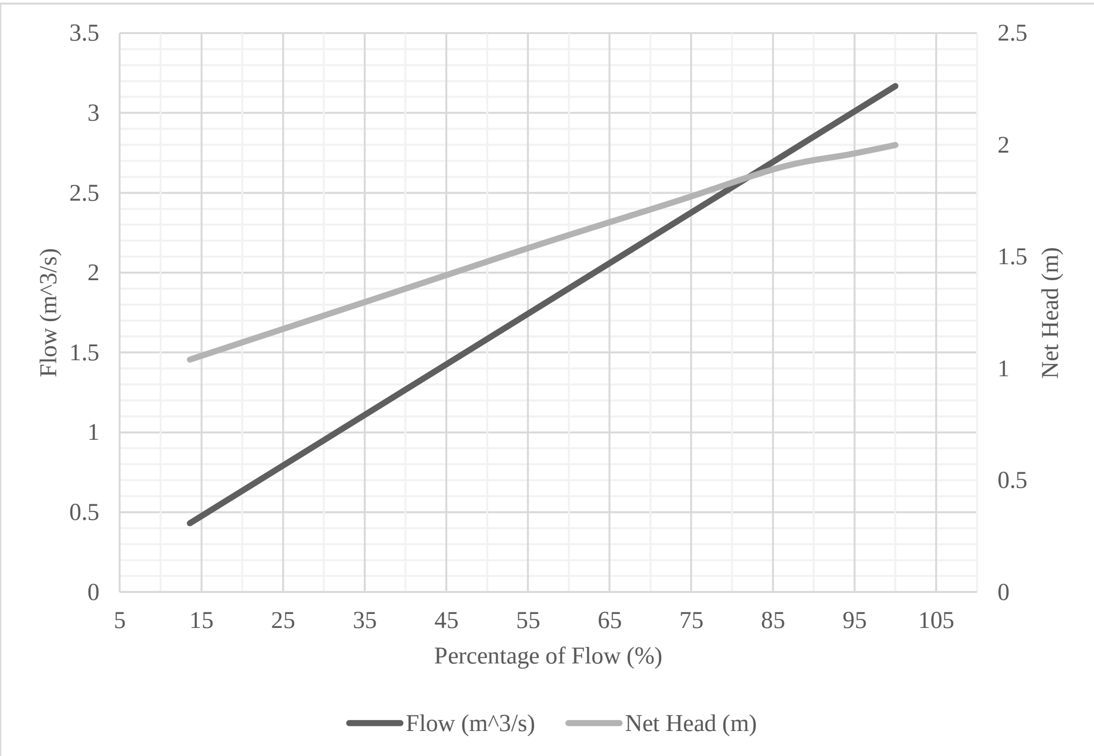


Figure 4. 1 Manufacturer's curves for flow and net head as a function of the percentage flow for the studied hydro power station.

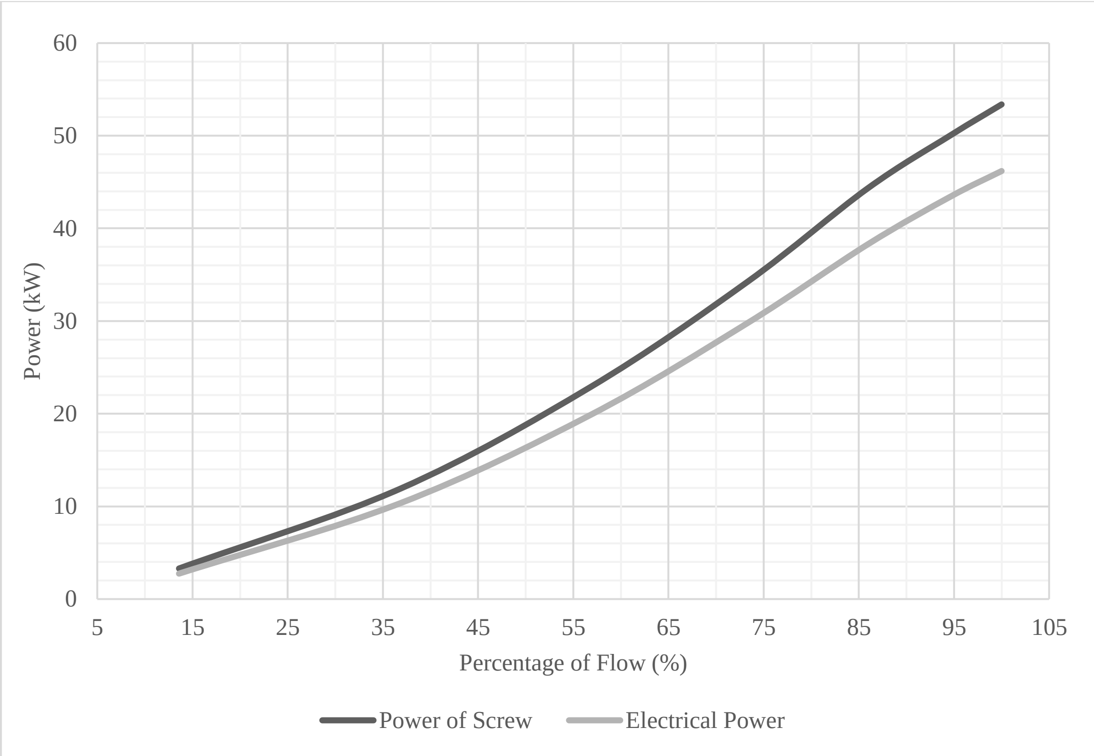


Figure 4. 2 Manufacturer's curves for the power produced by the turbine and the power produced by the station as a function of the percentage flow for the studied hydro power station.

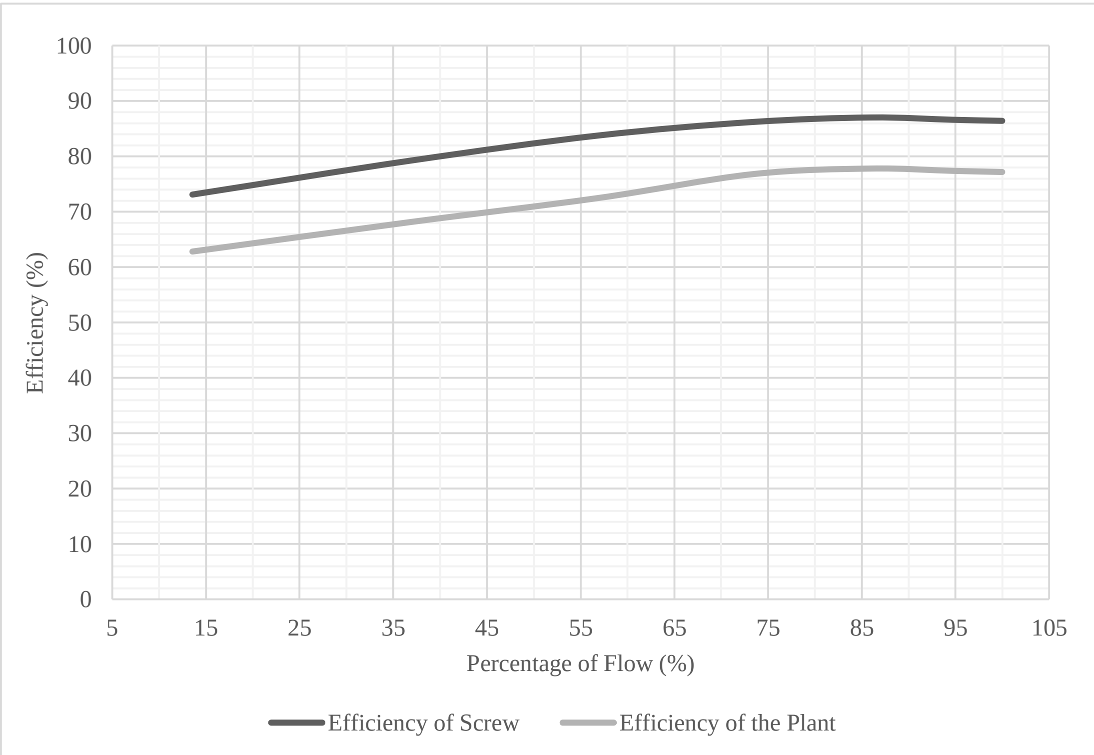


Figure 4. 3 Manufacturer's curves for the efficiency of the turbine and the efficiency of the overall project as a function of the percentage flow for the studied hydro power station.

4.2 Convergence and Sensitivity

The mathematical model described in previous chapter has numerical integrations in some parts for calculations using geometric steps. The main steps of interest for the accuracy of the results are the r_{step} and θ_{step} which are the number of divisions between the radius and the angle of regression respectively. The numbers of radial and angular steps were increased starting from very small number of steps and increased until the difference in results become negligible. A relatively small number of steps would provide an advantage by reducing the simulation time. The convergence was visualized for the power output of the turbine (P_{out}) and the volume of the bucket (V) in optimal fill level (f = 1). The radial and axial steps numbers were chosen to achieve difference lower than 0.5 %, and the radial and axial step length not exceed 1˚ and 1 mm respectively. The simulation time in Figure 4.5 illustrates the time that the model needs to create results explained in previous chapter, including the hydraulic calculations like the turbine's power output and the volume of the bucket which are represented in following graphs.

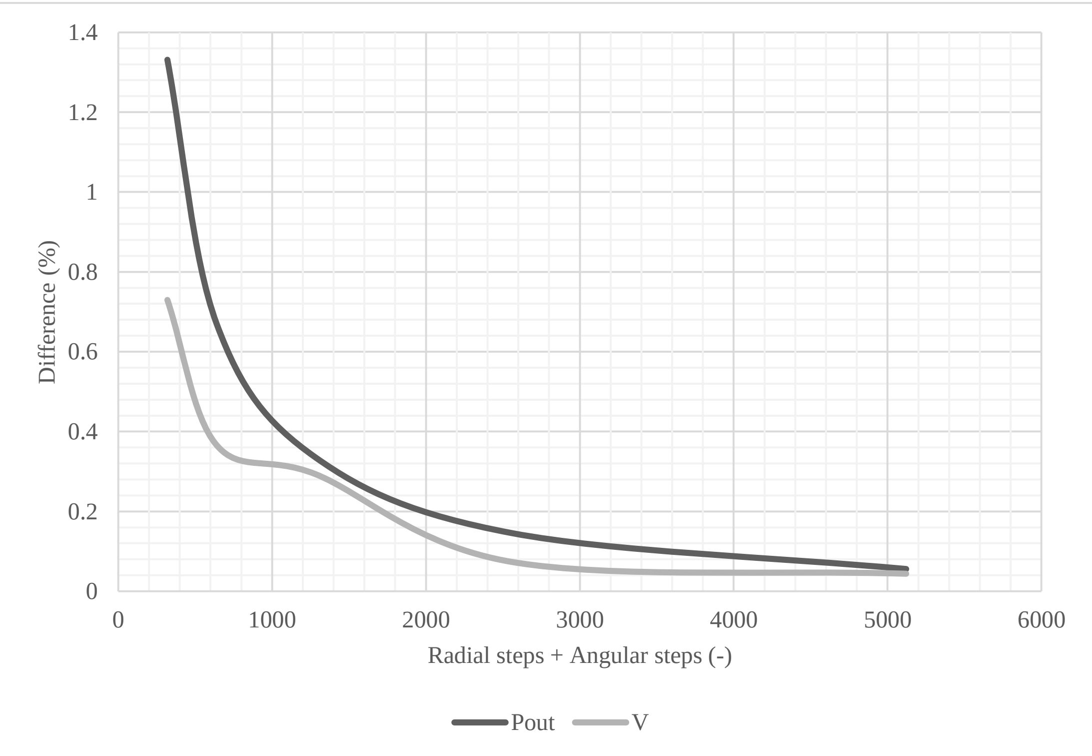


Figure 4. 4 Percentage differences in water volume in the bucket (V) and turbine power output (Pout) as radial and angular steps numbers are increased.

Except for the radial and axial steps there have been introduced a few more step number variables. However their contribution in terms of simulation time and any variable difference is small.

To ensure the proper operation of the mathematical model, the sensitivity of the main output parameters to the input parameters was examined by changing one input parameter every time and plot the percentage difference of the output variable as a function of the percentage difference of the input variable. As for the convergence, the screw is studied here in optimal fill level (f = 1) and has no overfill losses. The plotted parameters are the most basic geometric parameters which are chosen during the design procedure and, as it was described in the literature review, they are all related each other, directly or indirectly. The external diameter (Do) was not introduced to this study because the flow has a great dependence on it, so if it changes the exploited flow changes significantly and the requested performance range cannot always be covered. The length (L) and the slope angle (β) were not introduced as well because it would lead to different head coverage and the performance range studied here could not be covered.

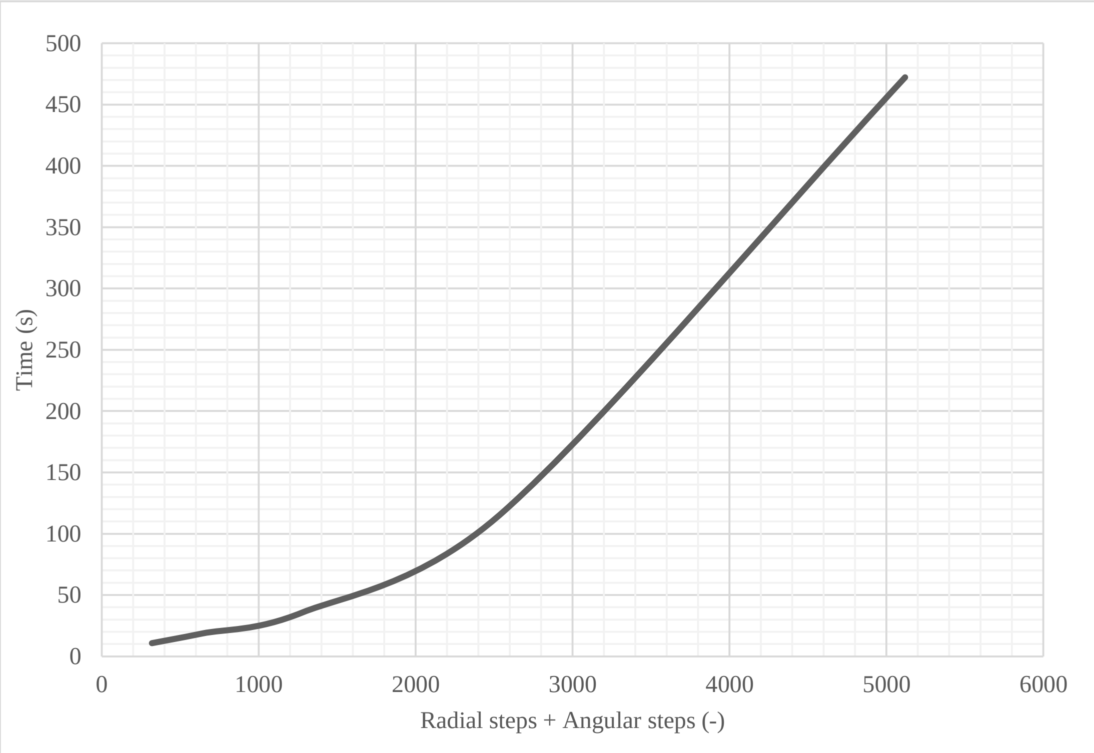


Figure 4. 5 Simulation duration as radial and angular steps numbers are increased.

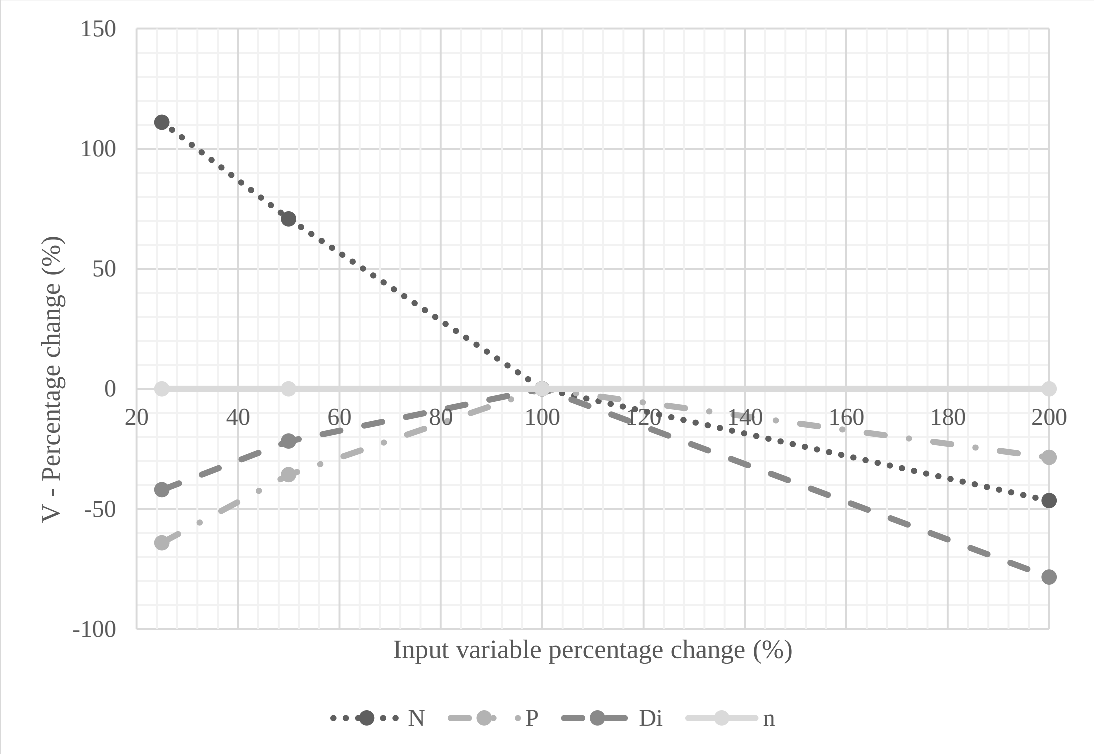


Figure 4. 6 Effect of number of blades (N), pitch (P), internal diameter (Di) and speed (n) in the bucket water volume.

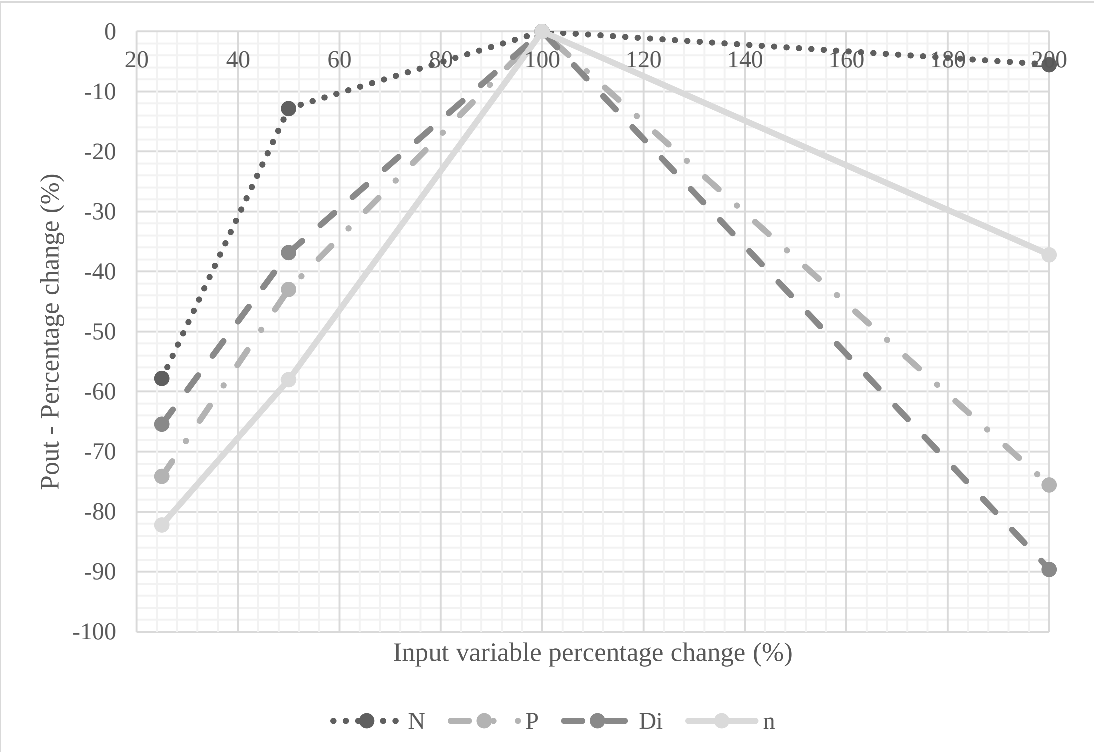


Figure 4. 7 Effect of number of blades (N), pitch (P), internal diameter (Di) and speed (n) in the power output of the turbine.

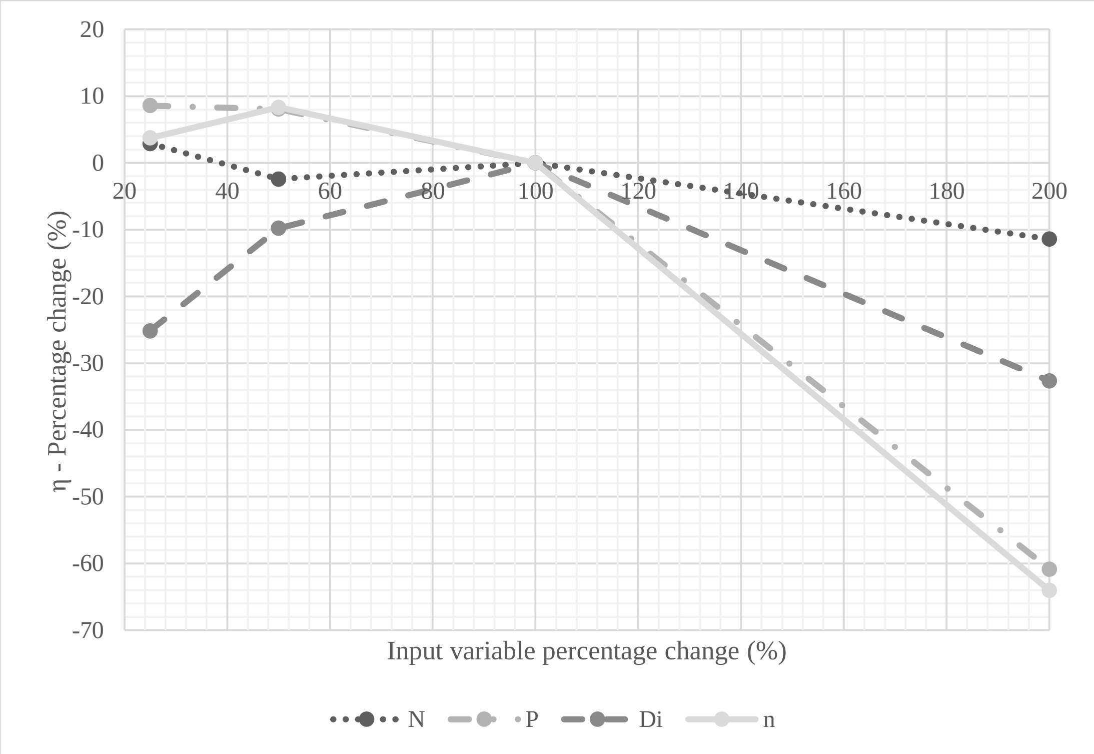


Figure 4. 8 Effect of number of blades (N), pitch (P), internal diameter (Di) and speed (n) in the efficiency of the turbine.

The nominal value (100 %) of each parameter is the value of the particular parameter for the turbine that was used as a case study. The nominal number of blades for example, that the case study device has, is 4. From the figure 4.7 it seems than the chosen variables values are very close to the optimal ones as regards the power output of the turbine and a small reduction to the pitch and the internal diameter is possible to maintain the power and increase slightly the efficiency. The reduction of the speed and the pitch looks to have a positive effect to efficiency, perhaps because of friction reduction, but such change would lead to power reduction because of flow reduction. The above charts illustrate only one change of one variable every time so they cannot estimate the effect of any combined changes of the input variables to the output variables.

It has to be clarified that, as it has already mentioned above, the plotted results are for optimal fill level. It does not mean that the flow in each case is equal each other or equal to the nominal. For example, at 120 % of the pitch the flow is not necessary the same with the 120 % of the speed. This can become understandable up to a point (because it shows the volume and not the volume per time) from figure 4.6. Since the turbine studied is manufactured by a well-known company, its geometry is definitely already optimized. Any further optimization may be possible but it demands scanning of each variable in combination with all the others in the area around the design point for each parameters using very small step.

4.3 Validation

For the validation of the results produced by the numerical model were used the manufacturer's curves presented on chapter 4.1. The initial plan was to collect real data from site while it was in operation, however this never carried out. After the regulation of the model, based on the results presented in chapter 4.2, the geometric parameters were introduced. The pitch was estimated. From the same drawing a few more interesting details were extracted related with the construction of the canal, the inlet and the outlet, the sluice gate and the trough. The gap between the impeller and the trough is assumed based on formula 2.4, because there were no information about it. As water level at the inlet (H1) is considered the level values which were available with the curves. It is not known if there is significant difference between it and the level before the reservoir because of the sluice gate. It is assumed that the delivery channel losses are only because of the water friction with channel wall from the sluice gate until the inlet of the turbine. Also, a minor loss has been calculated because of the screen a close to the entrance of the trough.

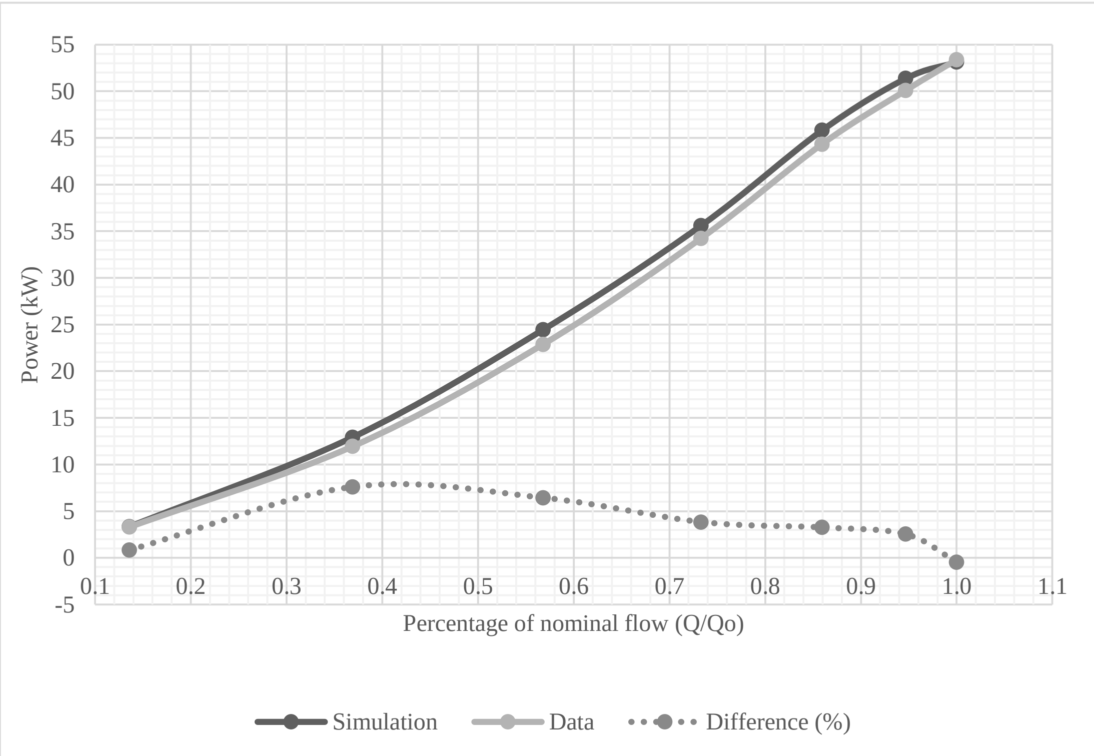


Figure 4. 9 Calculated data using the numerical results in comparison with the data provided by the manufacturer of the turbine for the power output of the turbine.

The estimated power output of the turbine in comparison with the power output provided by the manufacturer of the turbine is presented in Figure 4.9. The curves have a percentage difference between -0.46 % (minimum) and 7.59 % (maximum) with average percentage difference 3.43 %.

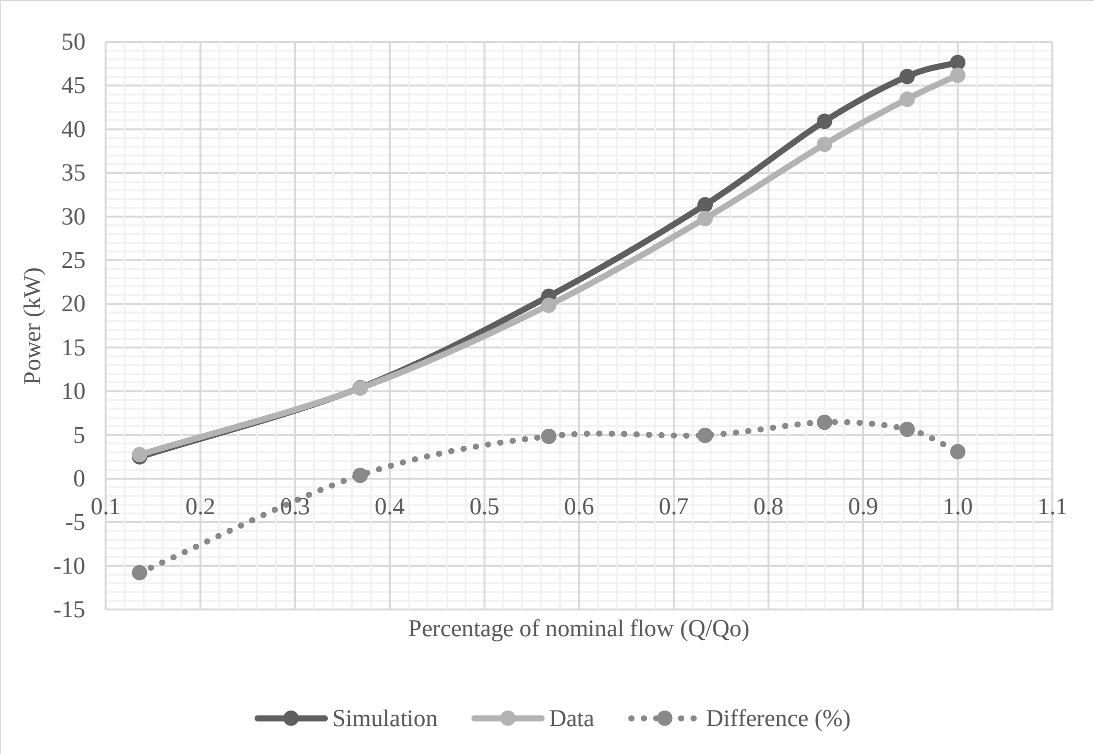


Figure 4. 10 Calculated data using the numerical results in comparison with the data provided by the manufacturer of the turbine for electrical power output of the plant.

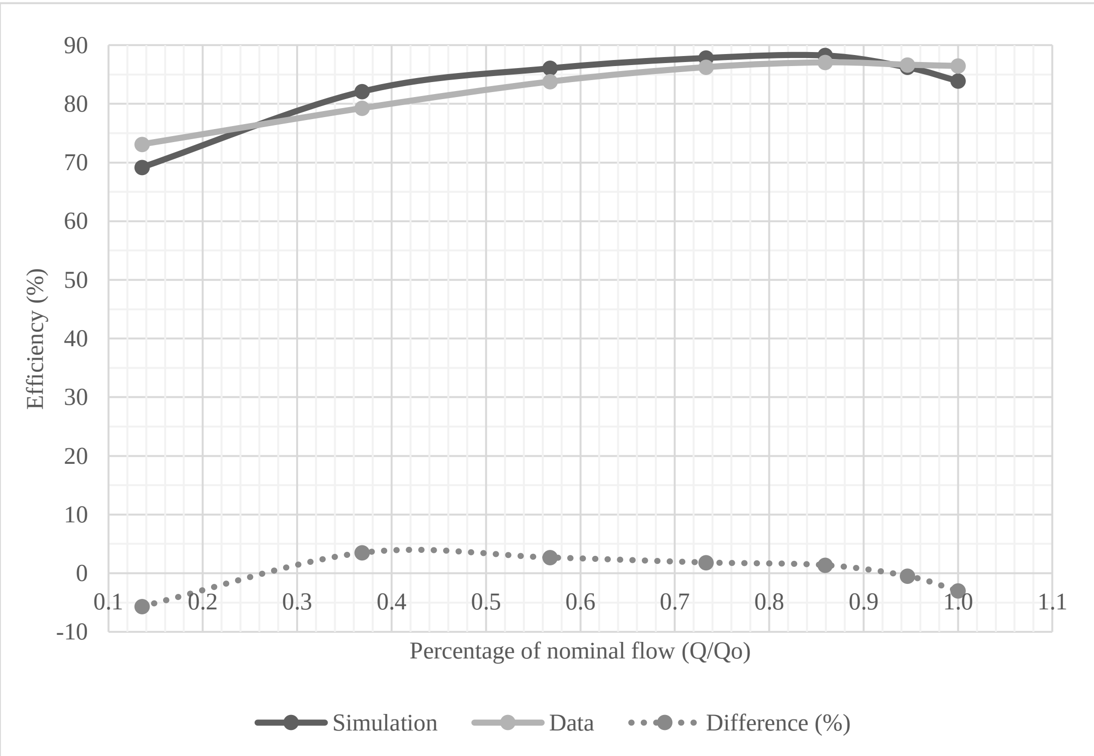


Figure 4. 11 Calculated data using the numerical results in comparison with the data provided by the manufacturer of the turbine for the efficiency of the turbine.

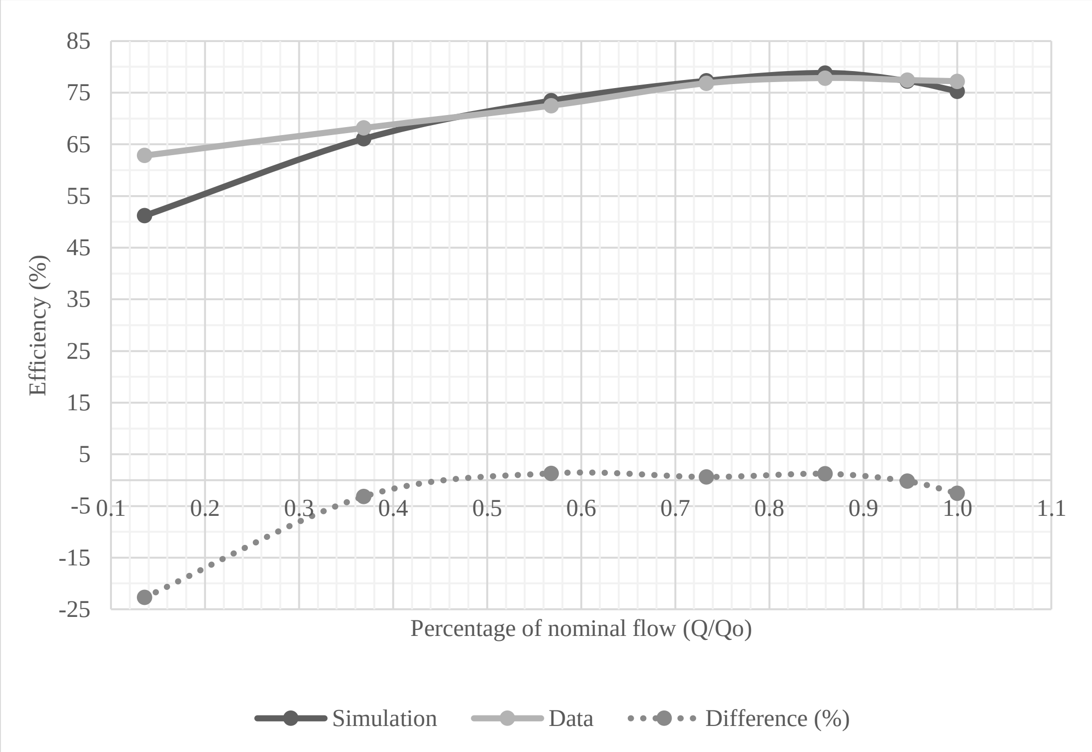


Figure 4. 12 Calculated data using the numerical results in comparison with the data provided by the manufacturer of the turbine for the electrical efficiency of the plant.

On the same way, the estimated electrical power output in comparison with the power output provided by the manufacturer of the turbine is presented in Figure 4.10. The curves have a percentage difference between -10.79 % (minimum) and 6.43 % (maximum) with average percentage difference 2.07 %. The estimated efficiency of the turbine is in Figure 4.16. The percentage difference of the curves is between -5.65 % (minimum) and 3.47 % (maximum) with average percentage difference 0.02 %.

For the end (Figure 4.12), the electrical efficiency curves. The percentage difference in this case is between -22.71 % (minimum) and 1.32 % (maximum) and the average is -3.63 %. In this this case the convergence is not so good, mainly in lower flow, but it is justified from the difference between the efficiency curves introduced in the model for the transmission and the generator (Figure 3.8) and the data for the transmission and the generator that the same manufacturer for another project supplied (Figure 4.13) according to Tarrant (Tarrant 2011) and it is assumed that they were used for the curves provided by the manufacturer (4.9 - 4.12).

The convergence of the calculated and available data is considered satisfactory for this depth of study. The deviation in mainly in electrical efficiency and power, especially in low flow fraction was expected here since it is not known if the efficiency curves used for the generator and the gearbox fit with the real ones.

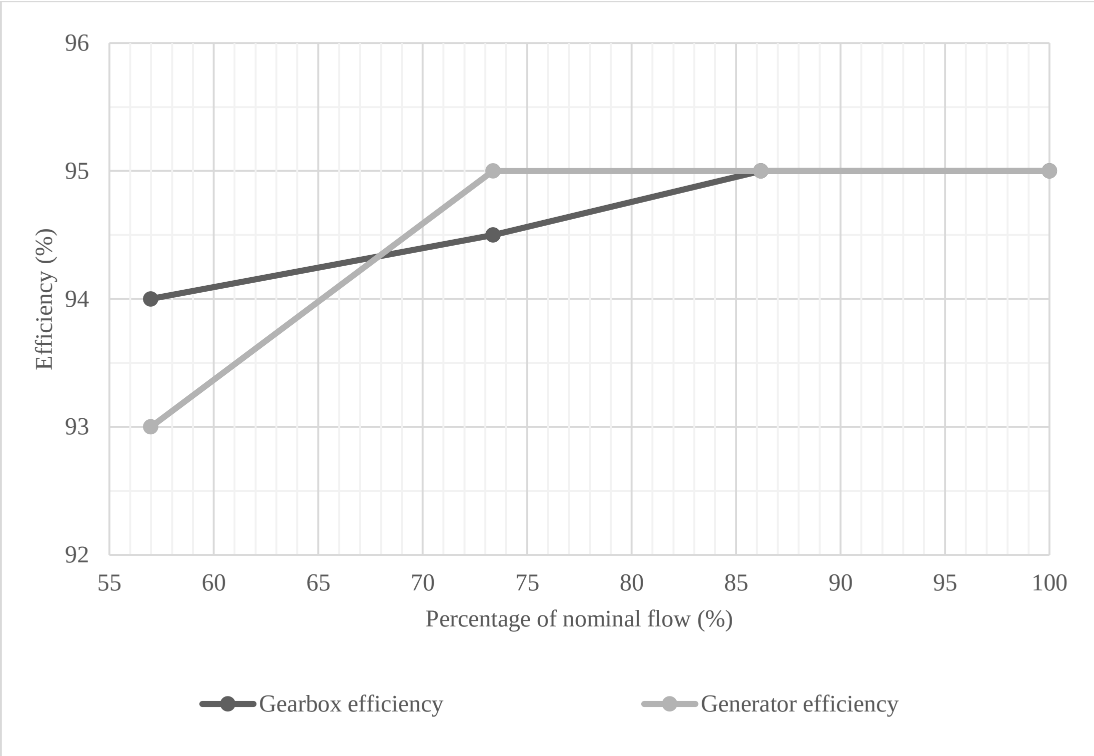


Figure 4. 13 Curves for the efficiency of the gearbox and the generator, as a function of the flow, supplied by the manufacturer of the turbine for another project (Tarrant 2011).

Except for this, the deviation occurred is reasonable because of a significant number of assumption like the roughness of the surfaces, the Manning's coefficient for the walls of the channel, the bearings diameter, the pitch and many more. This conclusion confirms, based on the assumption and the results presented above, that the validation for the mathematical model, which was explained in the previous chapter, is positive and it can be used for the initial estimation of an Archimedes screw turbine performance and other useful information related with the performance of the project, in a potential site. The percentage difference which was used as a convergence criterion above (Figure 4.8 - 4.12, 'difference') is estimated based on the formula 4.1, for the same flow fraction in each point.

$$\mathrm{PD} = 100\,\frac{y_{\mathrm{simulated}} - y_{\mathrm{data}}}{y_{\mathrm{simulated}}} \tag{4.1}$$

In figures 4.15 and 4.16 the losses are presented for the reference geometry as a percentage of the total available capacity. In Appendix B also, each loss is presented separately to make the reader understand what the effect of each loss mechanism in efficiency and how this effect changes with the suggested modifications.

4.4 Results

As it has already mentioned, according to Tarrant (Tarrant 2011) the curves provided by the particular manufacturer do not take into consideration the forebay losses and the effect of drag, as it happened in Figures 4.9 – 4.12 for the calculated curves as well to be able to validate the model. The forebay losses cannot be estimated here since there are no details neither for the water level at the reservoir nor for the sluice gate position every time to be able to calculate any head losses from the reservoir until the inlet of the turbine. The effect of drag and the losses because of it can be estimated and are introduced in the results presented in the following figures.

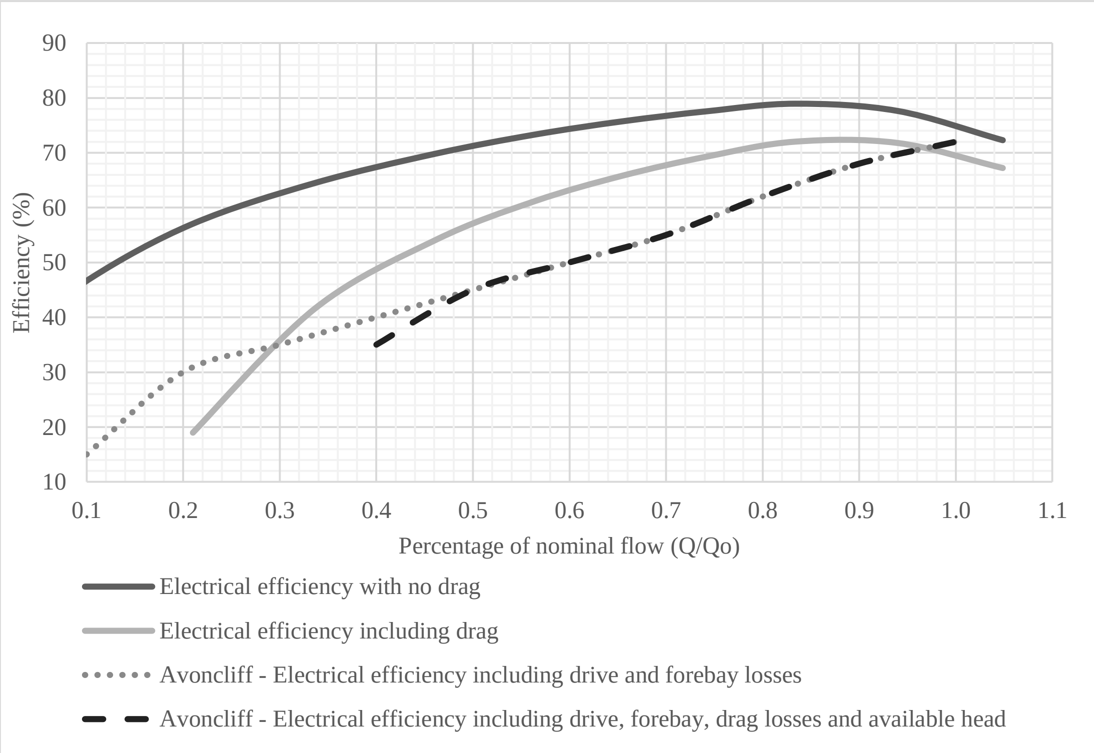


Figure 4. 14 Calculated electrical efficiency with and without the effect of outlet drag. As a comparison the estimated curves from Avoncliff (Tarrant 2011) have also plotted.

Also, there are two other minor losses introduced in the following results which come from the belts connecting the gearbox and the generator and the trash rack before the inlet of the trough in the studied hydro power station, trying to make any estimated data as more realistic as possible.

In the figure 4.14 is obvious the difference between the electrical efficiency including the losses because of the drag created by the tail-water and the efficiency not including them. The difference is significant and gives an idea about the participation of the submerged part of the screw in created losses. The curves about Avoncliff were taken by literature (Tarrant 2011) and they have created a) based on the information for the project which are available in the report, b) the curves provided by the manufacturer of the AST which will be potentially installed there and c) the adaption of the curves to the specific site, based on empirical estimations from other projects. It is considered then that the real electrical efficiency (without forebay losses) curve for the studied hydro power station is the one presented in figure 4.14.

This efficiency curve is a result of a number of losses applied on the produced capacity because of the equipment. In many turbine types all the attention is focused on the improvement of the turbine but the truth is that usually there are other, more significant origins of inefficiencies. The Archimedes screw turbines, have a few extra disadvantages in this field mainly because of the size, the speed and the fact that they are free surface turbines. The large size (for relatively large flow), except for the transport and the construction difficulties, leads to large surface on which there is friction with the fluid. This loss is not so high but it is not negligible. Since the friction has a linear correlation with the squared speed the friction per surface is possibly higher in smaller diameters where the speed is higher. The low speed, which is responsible from one point of view for the large size, makes necessary the use of a transmission system. This increases the cost, the complexity, the maintenance difficulty and mainly it introduces one more efficiency to a construction for which the efficiency is not already its main advantage. In addition, the Kaplan turbine which is the main competitor in the flow and head range studied here, except for the very high efficiency, because of the high speed it does not need transmission in many cases, having one more advantage in terms of losses. The free surface design philosophy is perhaps the greatest advantage and the greatest disadvantage the same time. The greatest advantage since because of it the project is simplified, there is no need for penstock, draft tube, inlet formation, main inlet valve, and all the attention all of them require, but instead of penstock there is the open channel and instead of the main valve there is sluice gate. The greatest disadvantage because the most of the losses are results of this free surface approach. The inlet open channel although it is usually short and has smaller surface in contact with the flow related with a penstock it has higher friction factor because of its surface related with the penstock.

Also, if the inlet water level varies significantly, the forebay losses and the transition inlet losses cannot be avoided, even if they can be restricted with correct design. The water level at the outlet open channel is responsible for the transition outlet losses and the drag on the submerged part of the impeller which of course exists in the reaction turbines as well but perhaps not in so high percentage of the available power. The drag losses may be reduced if the submerged part or the speed is reduced but it may lead to head losses for reference geometry and slope angle.

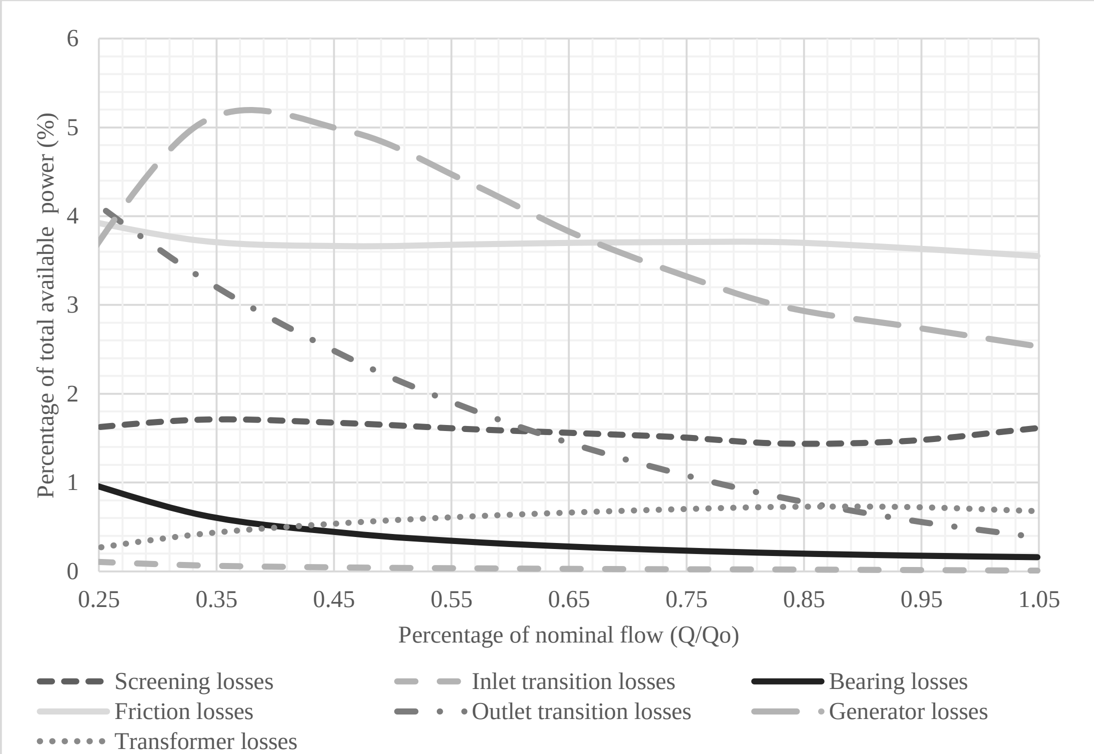


Figure 4. 15 Estimated minor losses (maximum 5 %) as a percentage of the total available capacity.

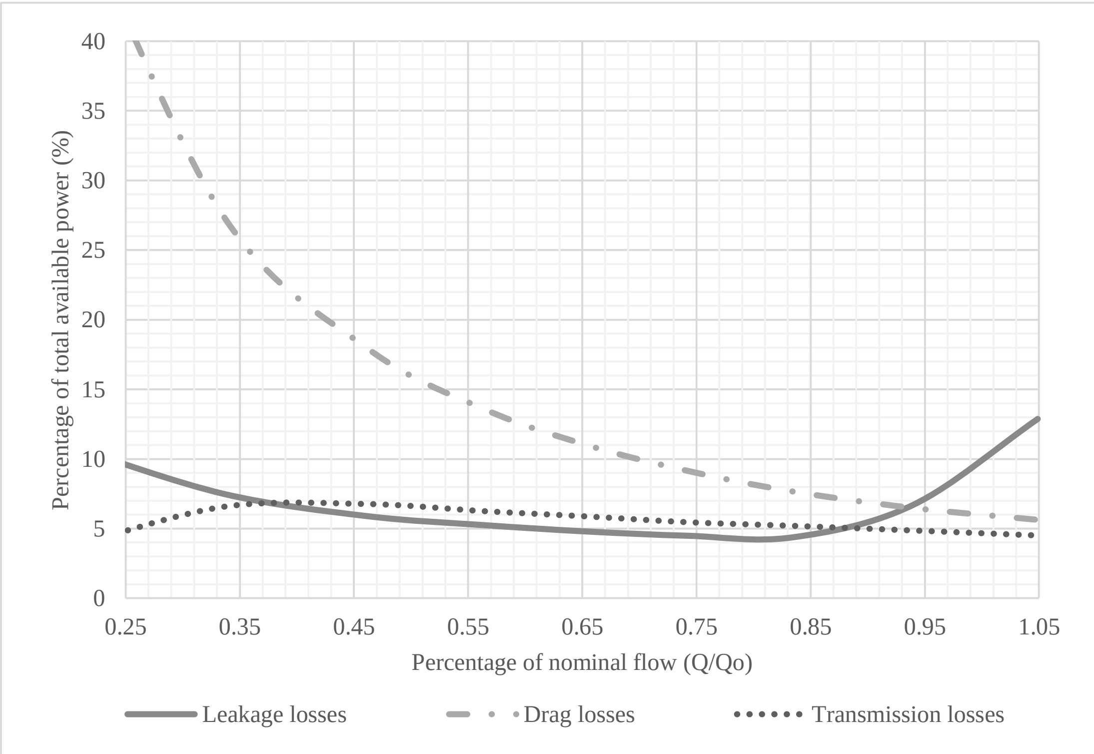


Figure 4. 16 Estimated significant losses as a percentage of the total available capacity.

The Figures 4.15 – 4.16 represent each mechanism of losses as a percentage of the flow fraction for the studied site. In the first diagram, the friction losses may be able to be reduced a bit by using antifriction coating. The inlet transition losses seem low because the inlet water level used is the one provided by the manufacturer with the performance curves and it is not certain if the horizontal axis of the graph is addressed to the equivalent performance based on laboratory or site measurements of the manufacturer. If the inlet water level varies significantly related with the equivalent fill level, the inlet transition losses can be high. In the second diagram the outlet transition losses depend on the outlet water level. The leakage losses can possibly be reduced if the size of the gap can become smaller. The drag losses, which represent a very high percentage can be reduced by reducing or by minimizing the submerged part of the rotor.

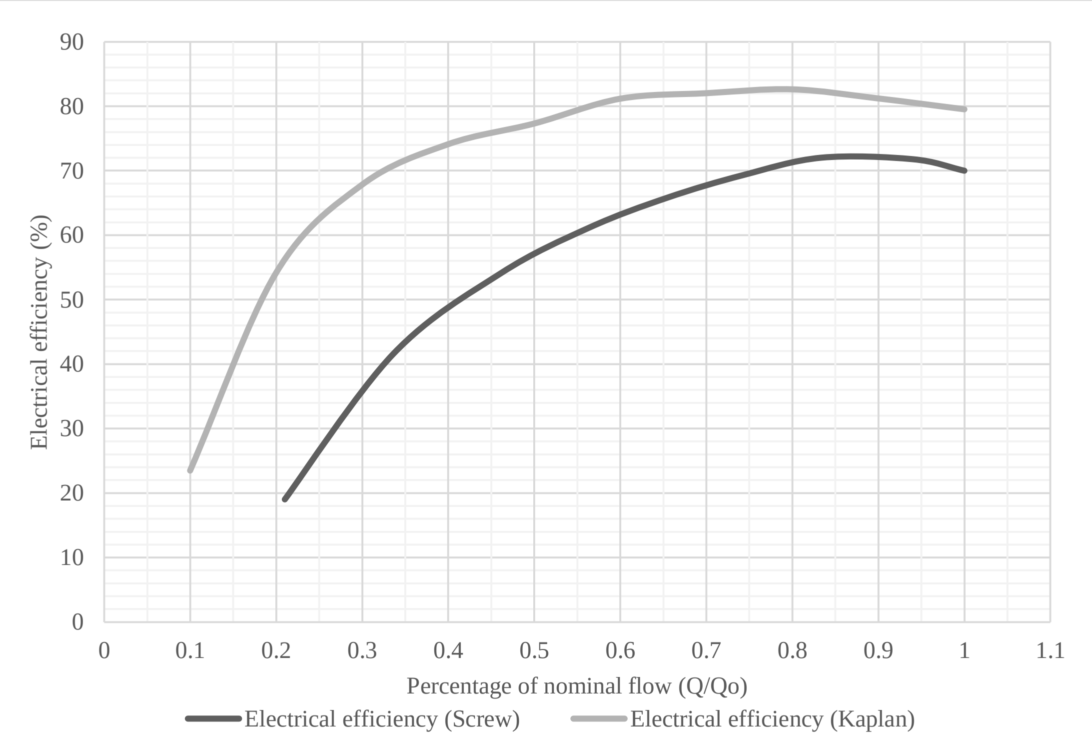


Figure 4. 17 Electrical efficiency of the studied Archimedes screw turbine and a double regulated Kaplan turbine for the hydro power project.

Using the shaft efficiency curve from the figure 2.3 for a double regulated Kaplan turbine, which is the main alternative of an AST in low head, it is possible to be estimated the electrical efficiency in case of use in the hydro power station studied instead of an Archimedes screw turbine (figure 4.17). For the transmission and generator losses were used the same information with the previous calculations. For the same nominal characteristics ($Q_{nominal} = 3.2\ m^3/s$, $H_{nominal} = 2$ m), the estimated rotational speed for the Kaplan turbine can be up to 380 rpm. This significant higher speed related with the screw turbine may lead to a gearbox with less stages and as a result a bit higher efficiency related with one which is necessary with an AST. In case of a project with similar head but lower flow ($< 1.2\ m^3/s$,) the Kaplan could be used without transmission at all since it's speed would be high enough (> 600 rpm) for a direct drive connection with a generator and as result increased efficiency because there would be no transmission losses.

Chapter 5. Discussion for Potential Improvements of the Project

The final stage of the study is the suggestions of ways and methods which could improve the performance of the project, for the available information and data, based on the developed model. The main criterion of improvement is the ability to perform higher efficiency in nominal and partial flow. Three directions of improvement are suggested for low, middle and extended modification of the existing plant, which have equivalent effect on the improvement of the performance.

5.1 Fields of Possible Improvement

As it was written earlier, research areas aiming to investigate potential improvements in the performance of the plant is something that was discussed during the meeting in the studied hydropower station. From the study so far, especially in the previous chapter, is clear that there are many parameters that generate losses and reduce the energy production. Based on figure 4.14, there is a strong evidence that the water to wire efficiency of the power plant with the existing setup has been overestimated. In fact the water to wire efficiency may be even lower than the estimated if the forebay losses are significant because of the head loss from a high water level difference before and after the sluice gate at the delivery channel entrance. This overestimation may be a result of many parameters. As Tarrant (Tarrant 2011) says for one of the graphs presented in his report, the curves provided by the particular manufacturer '*…are not clear nor easily understood and could be so confusing that customers will ignore or misinterpret them and rely on the performance comparison graph which only shows the screw shaft efficiency at fixed rated head…*'. The figure 4.11, assuming that the numerical model achieves an acceptable representation, look to represent only the efficiency of the shaft, the ratio of the power just after the screw divided by the available power just before the screw, without taking into consideration the drag from the submerged part of the turbine. From the figure 4.15 seems that the drag is the most important mechanism that creates losses. Of course, for the same turbine in the same flow the drag may be different if the water level after the outlet of the turbine changes. Except for the drag, there are many parameters of inefficiency, as it was described in previous chapter, mainly because of the free surface approach of the Archimedes screw turbines, which have to be taken into account for a proper approximation of the performance because from the figures 4.15 – 4.16 it is obvious that they are not negligible. Many of these parameters exist in impulse and reaction turbines as well. However, because of the fact that these turbines have usually much higher power than an average hydrodynamic screw turbine and higher shaft efficiency, these losses can be considered as negligible. Of course in case of an impulse or reaction turbine there are other losses, like the head losses because of the friction on the penstock wall, which have to be included in any calculation. In any case, they do not have significant effect from the inlet and outlet water level variations as a free surface turbine. So to be competitive a free surface turbine is better to be installed in sites where there are not significant water level and flow variations.

Based on the figures 4.15 – 4.16, theoretically the studied project can be improved if neutralized or at least minimized the reasons that create losses, one by one. Three stages of improvement for the scheme are going to be presented for different levels of modification, with equivalent effect in performance and modifications of the performed work. In appendix B the losses created by each mechanism are presented in separated diagrams to show their effect in the performance range and how they change in the following stages of modification.

5.2 Stage 1st

The first stage aims to the minimum modification of the scheme, in terms of money and effort and as a result the performance improvement is small. This stage is proposed considering that any modifications should not last long, so any income lost because of the downtime period will be limited. Also the cost should be limited aiming to make the application of the proposed changes attractive, even as an experiment. The final parameter considered for this stage is that the external geometry of the structure, mainly the turbine, of the scheme should not be modified at all and from any point of view. The particular changes included in stage 1 are oriented to already installed turbines as low cost and limited modification upgrade actions.

The present stage aims to losses reductions in the following areas from the ones presented in Figures 4.15 – 4.16: screening, inlet, friction and transmission losses. Judging from the Figures mentioned, the efficiency gain should not expected significant (except for the transmission perhaps), but in return it can be fast, simple and low cost. Three changes are suggested here.

The first is the removal of the belt drive. The belt drive makes a bit more compact the transmission system, because the generator is placed above the gearbox and the whole assembly becomes shorter, but it introduces at least 2 - 3 % of losses, considering that is has only one stage, (in case of more stages for larger speed ratio it is increased, 4 – 6 % for two stages) between the gearbox and the generator which can be easily avoided. In addition this move makes the construction simpler, because there is one less part which needs inspection and maintenance, and reduces the cost of the maintenance, since the belt has to be replaced every a few thousand hours. As it can be seen in figure 3.10 there are AST installations both with and without belt drive, depends on the approach of the developer. The first option may looks more compact but in fact it is more sensitive and less efficient.

The second is the use of anti-friction coating for the rotor and the metallic surface of the trough. During the research there was a contact request from a number (6) of companies, from the UK and other countries, which produce anti friction coatings for use in water turbines, pumps, pipes, valves, ship propellers, external part of the ship hull tec. Four of them answered and two of them provided details about the roughness of the surface after the use of their product to estimate any difference related with ordinary coatings. In practice, the roughness of a well painted metallic surface is around 15 μm. According to the data provided by the manufacturers of the anti-friction coatings, the surface of the turbine after the use of their products can achieve roughness around 0.08 – 0.09 μm. After further contact with one of the manufacturers came out that the expected lifetime is more than 10 years before the surface of the turbine needs repainting. Any damages (e.g. cavitation) can be repaired by local application of a small quantity of coating and it can be done very easy on site. The mentioned product is a two coat system. As it can be seen in Figure B5 the estimated gain between the current situation and stage 1 because of the coating only is about 0.1 %. Because mainly of the low speed, the friction losses are not high. The limited improvement in combination with the not negligible cost make the use of it in AST questionable. However, it seems that it may provide a benefit with small pumps and turbines where there is high friction because of high speed combined with limited size.

The third is the modification of the delivery inlet channel. By modification is meant both the cleaning of the walls from algae and the reduction of the walls friction by smoothing their surface. The slope of the bed is assumed that is can be about 1 m of drop every 1000 m of length (S_o = 0.001) for so short length, which is around 15 m measured in Google Earth. These changes can reduce the Manning's friction coefficient from around 0.03 (estimated) to 0.01 (minimum for concrete wall). This will increase a bit the speed of the flow calculated by the Manning's formula (for the reference width and depth) and as a result the screening losses will be slightly increased (Figure B2) but the delivery channel losses will be reduced (Figure B1). The Manning's coefficient for the wall of the channel was considered so far equal to 0.03. This was estimated based on the size and density algae the day of the visit on the site (no test, only optical observation) in combination with the speed and hydraulic radius of the cross section (Charbeneau, 2011). An open channel covered by stones, like the delivery channel in the studied scheme, may have even higher friction coefficient (0.035) but as it was covered by algae around the wetted perimeter it was considered that the wall can be assumed as 'weedy' and not 'stony'. From other reference (EJP 2008) it seems possible that the Manning's coefficient can be reduced down to 0.01 for a clean and very nice finished smooth open channel made of cement walls. Of course, in case that this modification is fulfilled, the stones on the wall have to be covered with a layer of cement and neat finish it to be able to minimize the friction coefficient in practice. This will reduce friction losses in the channel because of the contact between water and wall. The generally low speeds in the channel because of the slope, friction and hydraulic radius is possible to have as a result intense development of algae on the walls which has to be cleaned relatively regularly to maintain the friction factor of the wall as low as possible. The need for friction reduction is of course larger in case of a channel with large length or large slope, especially in combination with power plants with not high power.

5.3 Stage 2nd

The second stage is suggested considering as main difference a significant cost increase of the modifications related with the first stage and more extensive works. In fact, the second stage includes all the actions suggested for stage one plus extra actions for further improvement of the performance. This stage aims to all the minor and major modifications possible to limit the losses mechanisms possible from the ones presented in figures 4.15 – 4.16, but without change or redesign any of the main components of the plant (turbine, generator etc).

The present stage aims to losses reduction in the following areas from the ones presented, except for the gain already explained for stage one: friction, outlet transition and drag losses. Judging from the figure mentioned, the efficiency gain is expected to be significant mainly because of the reduction in the drag losses. Two changes are suggested here, supposing that the changes from stage one will be introduced as well.

The first is the introduction of a constant frequency variable speed system or frequency converter. This system will allow the regulation of the speed mainly for different flow fraction, but it is also possible to differentiate the speed aiming to handle losses because of other reasons like water level variations, by taking the advantage of the software and the installed sensors. On this way the turbine will be possible to exploit down to very low flow fraction because, since the speed will be reduced, the effect of the drag will be reduced. In addition, because the speed will be lower related the fixed speed operation for the largest part of the performance envelope, the friction and the outlet transition losses will be reduced since they are correlated with the squared speed. Unfortunately, the frequency converter has two important disadvantages. The first is that it is very expensive, at least for middle and high power. The second is that it has itself a percentage of electrical losses (Bard 2007) and as a result there will be a percentage of losses which is significant in low power and cannot be avoided. This percentage of losses may not be negligible but it is much lower related with the occasion that such a system is not used, at least for the flow and water level conditions of the studied scheme.

The second is the change of the slope angle of the turbine. This demands some extended reconstruction of the slide under the trough. The placement of the turbine in lower slope angle will have an effect on the fill level of the buckets because for reference flow the fill level will be reduced and as a result the gap leakage losses will be reduced, at least in low and middle flow fraction, but the most important, will reduce the resistance of the submerged part, since the size of the submerged part will be reduced, and the losses because of the drag. There will be also an impact to the outlet transition losses but this also depends on the water level of the outlet channel. In the case studied here the water level of the outlet channel is considered constant, which is not very often in real problems. The suggested slope angle in this stage is 21˚ because further reduction would lead to reduction of the exploited head, even if it would reduce more the drag losses. The formula 3.32 suggests an optimal water level or better an optimal level of immersion for the outlet part of the rotor which is responsible for the drag. This level of immersion is related with the main geometrical characteristics (internal and external diameter, slope angle, pitch and number of blades) but is also depended on the relative place of the rotor with the inlet and outlet channel and length of the rotor which is correlated with the head, except for the slope. As a result, the suggested level of immersion by the formula cannot always be applied, for reference water levels, geometry, slope and length, but it can be used as guideline. In another approach, the result of the formula can be taken as reference and based on this can be calculated the length and the slope. This is not usually practical because the slope cannot take any value. In case of an 'unusual' slope angle new problems will be arisen because trying to minimize the drag other losses will be increased. Moreover, the outlet water level in the most case varies from season to season. Also in the studied case, as in many cases, the flow exiting the turbine goes to a very short channel and actually it outflows directly to the lower reservoir-lake. As a result the speed at this short outlet channel is very low and the outlet water level is determined only by the lower reservoir water level.

5.4 Stage 3rd

The third stage suggested is actually a reconstruction of the power plant or an approach that suggests what could change if the project was made from the beginning. In this case the parameters of time and cost are considered that will be very high, exactly because it is like setting up a new hydro power station from the beginning. The proposals included in this stage aims to achieve the maximum performance which is possible to be achieved for the particular flow and water level conditions using the particular type of hydro turbine (AST) with the necessary secondary equipment and with respect to environmental and other limitations for the site (screening etc) to make the performance maximization able to be realized from any point of view. The actions suggested in the two previous stages are included here too.

The major difference here related with the two previous proposals is the change of the turbine (rotor and trough) which is the most expensive single component, and the largest considering transport and assembly, and definitely the most significant one for the optimal operation of the power plant. The new/proposed rotor will have the same external diameter related with the present one and the same nominal speed, but will have different internal diameter, pitch, length, slope angle and number of blades which is allowed to be up to five for this diameter and this speed based on table 2.2. This changes will give to the proposed turbine an advantage as regards the drag and the outlet transition losses as it can be seen in Figures B6 and B7. The estimated pitch ratio (S) and radius ratio (d) for the existing screw have been estimated from the available information as 1 and 0.45 respectively for the current turbine. A number of different simulation has been done for different geometry setups (d = 0.45 – 0.55, S = 0.8 – 1.2) aiming to compare the performance for different geometric options. The results are illustrated in figure 5.1.

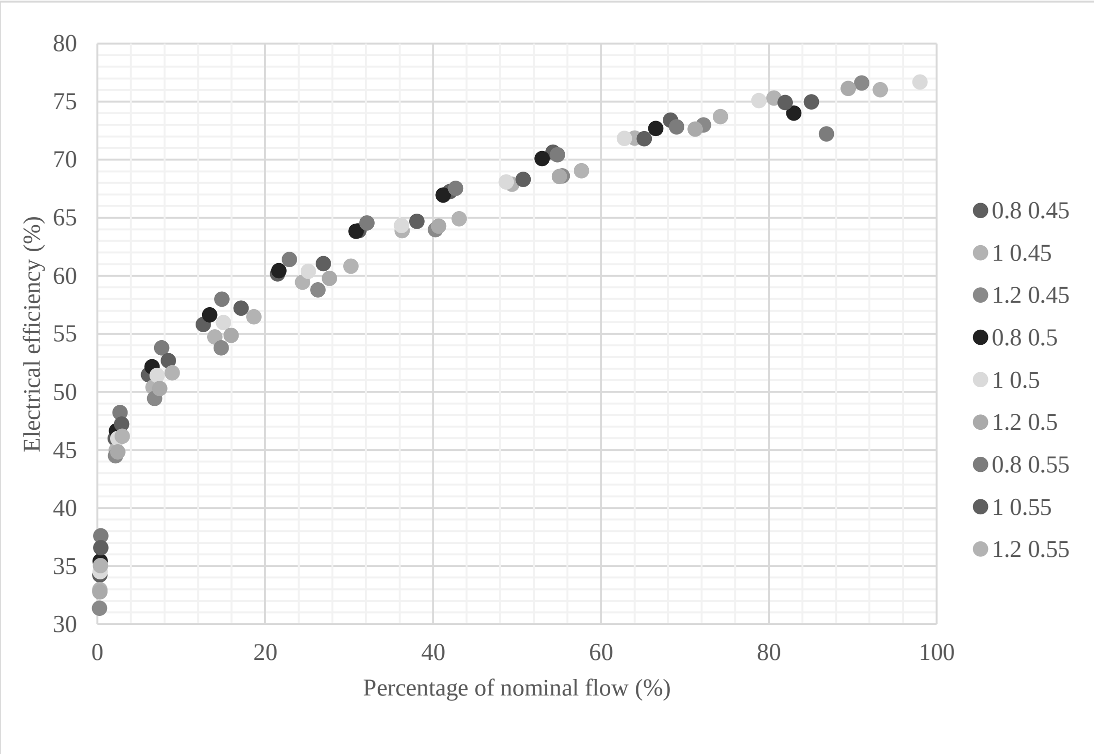


Figure 5. 1 Electrical efficiency estimation for different pitch ratio and radius ratio in various flow fractions for the studied scheme.

Based on the results of these simulations, the use of pitch ratio equal to 0.8 and radius ratio 0.55 will result to efficiency increase in the range up to around 70 % of the nominal flow.

Above the 70 % of the nominal flow there is an efficiency drop for the suggested design and the existing setup seems to have the optimal performance of all setups simulated in the range 70 – 100 %. From the data collected so far it seems that the scheme has a large flow variation and needs to work in partial load for a period of time.

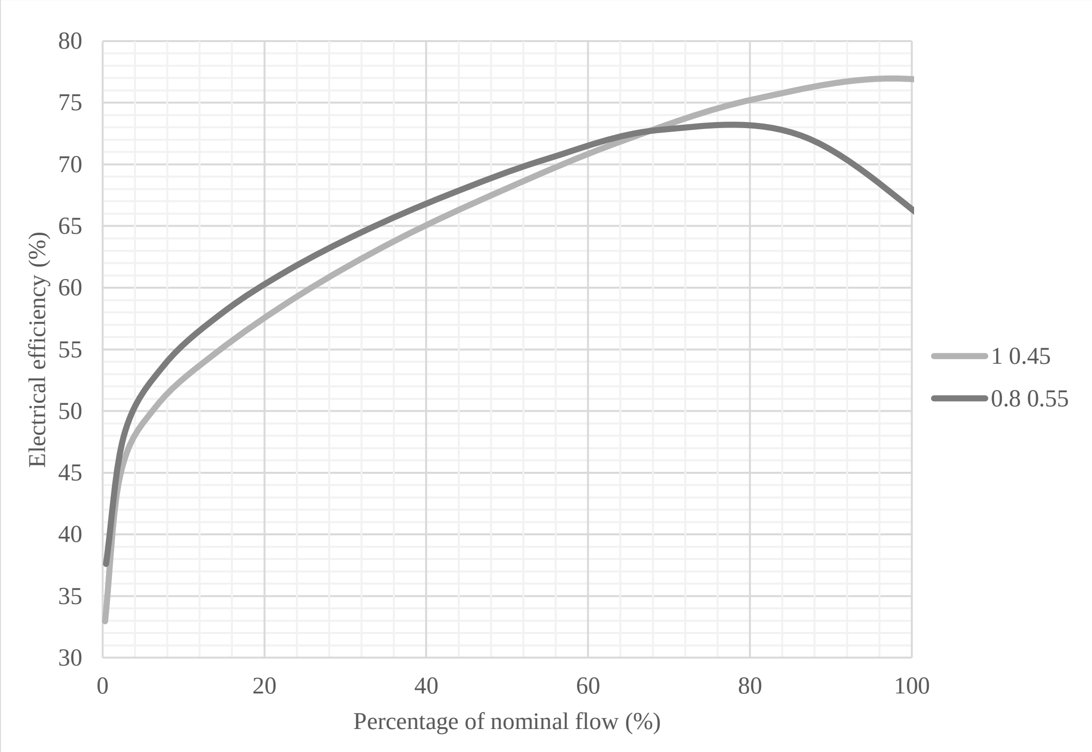


Figure 5. 2 Electrical efficiency estimation for the two different setups of pitch ratio and radius ratio for the studied scheme.

One of the claimed advantages of the AST is the relatively stable performance curve with significant efficiency down to very low flow. Looking it from a different perspective, the nominal efficiency is not supposed to be one of the advantages because it is not considered high, which is easy to understand comparing it with the efficiency curve of a double regulated Kaplan turbine (figure 6.1). So it is suggested the reinforcement of the efficiency in partial load for the studied project aiming to the best exploitation of the largest amount of the flow by using a new design (d = 0.55, S = 0.8). As it was mentioned above, the new design is suggested to have five blades which is the maximum permitted for the external diameter and the speed of the mentioned turbine and it is generally the highest that is used by the majority of the manufacturers for the AST. Also, for the new design is suggested the increase of the rotor length to 6.6 m and the reduction of the slope angle to 18˚. Although that the described changes may increase a few loss mechanisms like inlet transition and friction, they will achieve a significant reduction to major losses like the drag increasing the overall performance of the turbine without losing any of the available head.

The second change, which is very important as well and is related with the first one, is the reduction of the gap between the screw and the trough aiming to minimize the gap losses. After the study of the deflection, taking into consideration the mass and the speed, the clearance which secures the avoidance of contact can be reduced by reducing the distance between these two parts. This will have as a result the reduction of the leakage losses as it can be seen from the formulas 3.18 – 3.21, which is clear leakage and this part of the flow is lost for the turbine and cannot be produce power at all from it. The minimization of the gap, if the deflection which is related with the geometry allows it, is a simple and easy way to increase the efficiency in case of the design from the beginning of such a turbine. However the contact between the rotor and the trough must be avoided in any case because this would create high resistance moment and extra leakage losses from possible deformation of the surface of the trough and perhaps imbalance of the rotor. Using the formula 2.4 it is estimated that the maximum gap between the rotor and the trough can be up to around 7.3 mm. To avoid contact for the studied case is estimated that even a much smaller distance is acceptable to avoid contact and it creates no other issues based on the research conducted so far. It is not known if such a small distance can be achieved in practice during the assembly of the turbine and the trough for a real site. However it seems that it is possible a large reduction of the gap which will have as a result the reduction of leakage losses which is a large percentage of the overall losses in reduced flow. As it can be seen in Figure B4, such a change will have a major positive impact in efficiency in low flow.

Chapter 6. Conclusions

The aims of the MSc, which were defined by the author and his supervisor in the beginning of the year, were achieved in the majority. The theoretical background and the numerical model, which were the principle elements to be in the position to estimate the performance and improve it, were developed covering the initial thoughts of the author for a tool able to make an overall approach, water to wire, and not only for the turbine.

The most important conclusion for the author, is that because they are free surface devices they have a number of significant loss mechanisms, in terms of performance, which are not easy to understand if they do not carefully quantified, and it seems that they do not exist or they are not so significant in other low head turbines. As it can be seen in the figures 4.15 and 4.16, the inlet and outlet transition losses, the leakage and the drag losses are more or less significant, for the particular turbine and they are a result of free surface design. The transition losses can be reduced if the water levels upstream and downstream do not vary significantly from the water level inside the turbine, which is not easy to be regulated. The lower gap can lead to reduced leakage losses. In any case, further experimental work is absolutely necessary to prove the mentioned considerations and the estimations for the losses from the numerical model.

Based on the results of the model but also studying the available literature, even a very well designed and installed AST has significantly lower efficiency related with a Kaplan turbine, as it can also be seen in figure 6.1. The overall efficiency makes the distance between them greater since the AST always needs a transmission system, because of the very low speed. The Kaplan is a relatively high speed turbine, in many cases it does not need transmission, and as a result the connection with the generator can be direct, avoiding any losses because of the gearbox or any other similar systems (belt drive, chain drive etc). Following the same thought, to exploit large variations in the available conditions, a variable speed converter is used in many cases, since the turbine has no moving parts, which introduces even more losses. The Kaplan turbine can achieve high performance in extended range using its moving blades and guide vanes and in this case the losses introduced from the control system of them are literally negligible.

There is an evidence (figure 2.22) that the installation of a plant using AST has lower cost in comparison with the same plant using other type of turbine. This makes sense, up to a point, because AST are generally simple devices without moving parts and complex geometries and they do not need significant preparation of the installation area, like draft tube, water passage etc. However the size of them definitely increases the transport cost. Also, it is not known what is the lifecycle cost including maintenance etc. In any case, the reduction of their installation cost may be able to balance the low efficiency in many cases.

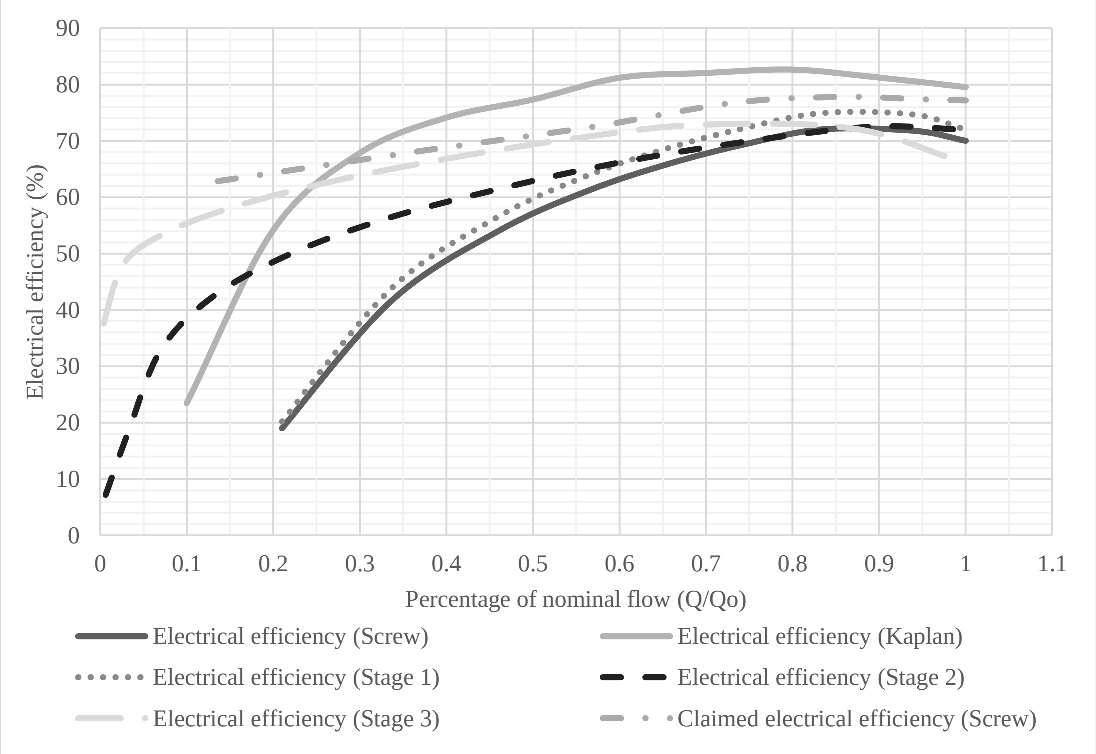


Figure 6. 1 Electrical efficiency curves for the studied hydro power station in the current condition (claimed and estimated) and after the suggested changes, including also the estimated curve for a double regulated Kaplan turbine.

References

1. AGGIDIS, G.A. LUCHINSKAYA, E. ROTHSCHILD, R. HOWARD, D.C. (2010). The cost of small – scale hydro power production: Impact on the development of existing potential. *Renewable Energy*, 35 (12), 2632 – 2638.

2. AMEC ENVIRONMENT AGENCY (2014). *Avocliff Weir - Comparison of Hydropower Schemes, Final Consultant's Report.* [pdf] UK: Amec Environment & Infrastructure UK Limited. Available at: <https://www.gov.uk/government/uploads/system/uploads /attachment_data/ file/299380/Avoncliff_AMEC_Comparison_of_hydropower_schemes_ Feb14_Appendix_4.pdf> [Accessed 19 July 2016].

3. ANDRITZ ATRO (2014) *Screw pumps*. [pdf] Available at: < https://www.andritz.com/oi-andritz-atro-gmbh > [Accessed 19 July 2016].

4. APSLEY, D. (2016) *Gradually – Varied Flow*. [pdf] Available at: <http://personalpages. manchester.ac.uk/staff/david.d.apsley/lectures/hydraulics3/gvf.pdf> [Accessed 19 July 2016].

5. BARD, N. (2007) *River Dart Country Park Archimedes Screw System Performance Assessment*, UK, Nick Bart Hydro Services.

6. BELZONA, *1341*, [pdf] Available at: <http://www.belzona.com/en/products/1000/1341. aspx> [Accessed 19 July 2016].

7. BOZHINOVA, S. HECHT, V. KISLIAKOV, D. MULLER, G. SCHNEIDER, S. (2012) Hydropower converters with head differences below 2.5 m. *Proceedings of the Institution of Civil Engineers*. [pdf] Available at: <http://eprints.soton.ac.uk /354873/1/__soton.ac.uk_ ude_PersonalFiles_Users_gm2_mydocuments_Publications%20GM_Bozhinova%20et%2 0al%202013.pdf > [Accessed 19 July 2016].

8. BRADA, K. (1999) Hydraulic screw generates electricity from micro hydro power stations (translated from German). *Maschinenmarkt Wurzburg, Mitteilung*, 105 (14), 52 - 56.

9. BRKIĆ, D. (2011) Review of explicit approximations to the Colebrook relation for flow friction. *Journal of Petroleum Science and Engineering*, 77, 34 – 38.

10. Bulgaria – Servia IPA Cross – Border Programme 2007CB16IPO006 2011 - 2 - 229, Reclaiming Rivers for Implementation of Vital and Environment – Friendly Renewable Energy Source - Rivers (2014) *Comparative study of small hydropower stations*. [pdf] Available at: <http://riversproject.eu/pdf/Comparative%20study.pdf> [Accessed 19 July 2016].

11. CHANSON, H. (2004) The Hydraulics of Open Channel: An Introduction, 2nd Edition, Oxford, UK, Elsevier.

12. CHARBENEAU, R.J. (2011) *Open Channel Hydraulics*. [pdf] Available at: < http://www. ce.utexas.edu/prof/maidment/CE365KSpr14/Visual/OpenChannels.pdf > [Accessed 19 July 2016].

13. Comparing the Darcy Weisbach equation with the Manning Equation. [pdf] Available at: < http://www.members.optusnet.com.au/msephton/wip/rel_darcy_manning. pdf> [Accessed 19 July 2016].

14. CORROCOAT, *Fluiglide*. [pdf] Available at: <http://www.corrocoat.com/wp-content/uploads/2015/10/Fluiglide.pdf> [Accessed 19 July 2016].

15. DELLINGER, G. TERFOUS, A. GARAMBOIS, P.A. GHENAIM, A. (2016) Experimental investigation and performance analysis of Archimedes screw generator. *Journal of Hydraulic Research*, 54 (2), 197 – 209.

16. DELLINGER, G. (2015) *Etude expérimentale et optimisation des performances hydrauliques des vis d' Archimède utilisées dans les micro centrales hydroélectriques*. Ph.D. University of Strasbourg.

17. DEWALD, J. (2009) *Determination & Evaluation of Noise Emissions & Noise Pollution in the Environment of the Hydrodynamic Screw Located in Dittigheim*, Birkenau, Gemany, Technak.

18. DEWALD, J. (2009*) Determination & Evaluation of Noise Emissions & Noise Pollution in the Environment of the Hydrodynamic Screw Located in Lauda*, Birkenau, Gemany, Technak.

19. EJP (2008) *Manning's Equation & Table of N Values, Engineering & Technical Data R – 15*. [pdf] Available at: < http://www.ejprescott.com/media/reference/ManningEqua&Tableo fNR-15.pdf > [Accessed 19 July 2016].

20. ENVIRONMENTAL AGENCY (2011) *Interim guidance on the permitting of Archimedes 3, 4 & 5 blade screws, Version 1, Appendix 1*. [pdf] Available at: <http://www.archonenergy.com/Documents/Fish%20Studies/UK% 20studies/GPG%203,4%20&%205%20 Blade%20Arch %20screw%20%20guidance%20%20final%2027%20Sep%202011.pdf>[Accessed 19 July 2016].

21. ESHA (2004) *Guide on how to develop a small hydropower plant*. [pdf] Available at: <http://www.esha.be/fileadmin/esha_files/documents/publications/GUIDES/GUIDE_SHP/GUIDE_SHP_EN.pdf> [Accessed 19 July 2016].

22. ESHA (2014) *Small and Micro Hydropower Restoration Handbook. The All – Inclusive Replicable Model. The RESTOR Hydro Project.* [pdf] Available at: < http://www.restor-hydro.eu/documents/2014/08/small-and-micro-hydropower-restoration-handbook-the-all-inclusive-replicable-model.pdf> [Accessed 19 July 2016].

23. European Best Practice Guidelines for Abnormal Road Transports, European Commission, Directorate – General for Energy and Transport. [pdf] Available at: <http://ec.europa.eu/transport/road_safety/vehicles/doc/abnormal_transport_guidelines_en.pdf> [Accessed 19 July 2016].

24. EN 10220: 2003. *Seamless and welded steel tubes – Dimensions and masses per unit length*. Brussels: European Committee for Standardization

25. FISHTEK CONSULTING (2009) *The Archimedes screw turbine assessment of the three leading edge profiles. 8 mm steel edge. Hard rubber. Compressible bumper.* [pdf] Available at: <http://www. archonenergy.com/Documents/Fish%20Studies /Fishtek/FISHTEK_ Archimedes%20Screw_Leading%20Edge%20Risk%20Assessment.pdf> [Accessed 19 July 2016].

26. IEC 60193: 1999. *Hydraulic turbines, storage pumps and pump – turbines: model acceptance tests*. Geneva: International Electrotechnical Commission.

27. IS 11388: 2012. *Recommendations for design of trash rack for intakes (Second revision).* New Delhi: Bureau of Indian Standards.

28. JOSIAH, N.R. TISSERA, H.P.S. PATHIRANA K.P.P. (2016) An experimental investigation of head loss through trash racks in conveyance systems. *Engineer, 49 (1), 1 – 8.*

29. KANTERT, P.J (2008) *Manual for Archimedean Screw Pump: Adviser and Decision Guidance for Planners, Owners and Decision – Makers*, F. Hirthammer Verlag.

30. KIBEL, P. COE, T. (2011) *Archimedean screw risk assessment: strike and delay probabilities.* [pdf] UK: Fishtek Consulting.Available at: < http://www.fishtek.co.uk /downloads/Fishtek-screw-risk-assessment-example-report.pdf> [Accessed 19 July 2016].

31. KOZYN, A. LUBITZ, W.D. (2015) Experimental validation of gap leakage flow models in Archimedes screw generators. *Renewable Energy in the Service of Mankind Vol I*, 365 - 375.

32. KOZYN, A. (2016) *Power loss model for Archimedes screw turbines*. MSc. University of Guelph.

33. LANDUSTRIE (2009) *Landy screw pumps & hydropower screws*. [pdf] Available at: <http://www.ecsengineeringservices.com/wp-content/uploads/2014/01/130107-folder Screw-Pumps-hydropwer-screws-UK-ECS-lr.pdf> [Accessed 19 July 2016].

34. LASHOFER, A. CASSIDY, T. HAWLE, W. PELIKAN, B. (2011) *Hydropower as restoration opportunity for obstructed river reaches?,* HYDRO 2011, Practical Solutions for a Sustainable Future, 2011, Prague, Czech Republic.

35. LASHOFER, A., HAWLE, W., KALTENBERGER, F., PELIKAN, B. (2013). *Die Wasserkraftschnecke – Praxis* , Österreichische Wasser- und Abfallwirtschaft, 65(9), 339–347.

36. LASHOFER, A. HAWLE, W. KAMPEL, I. KALTENBERGER, F. PELIKAN, B. (2012) *State of technology and design guidelines for the Archimedes screw turbine.* HYDRO 2012, Innovative Approach to Global Challenges, 2012, Bilbao, Spain.

37. LUBITZ, W.D. (2014) *Gap flow in Archimedes screws.* CSME International Congress. June 1 – 4, 2014, Toronto, Ontario, Canada.

38. LUBITZ, W.D. LYONS, M.W.K. SIMMONS, S. (2014) Performance Model of Archimedes Screw Hydro Turbines with Variable Fill Level. *Journal of Hydraulic Engineering*, 140 (10).

39. LYONS, M.W.K. (2014) Lab testing and modelling of Archimedes screw turbine. MSc. University of Guelph.

40. MASSEY, B. WARD – SMITH, J. (2006) Mechanics of Fluids, 8th edition, Abingdon, Taylor & Francis Group.

41. MAS HYDRO (2014) *Screw pumps product catalogue*. [pdf] Available at: <http://www.mas-hydro.cz/galerie/tinymce/Screw%20pump.pdf> [Accessed 19 July 2016].

42. MÜLLER, G. SENIOR, J. (2009) Simplified theory of Archimedean screws. *Journal of Hydraulic Research*, 47 (5), 666 - 669.

43. NAGEL, G. (1968) *Archimedean screw pump handbook*. Schwabisch, Gmund, RITZ – Pumpenfabrik OHG.

44. NUERNBERGK, D. LASHOFER, A. HAWLE, W. PELIKAN, B. (2013) Betriebsarten von Wasserkraftschnecken. *Wasserwirtschaft* 103 (7 - 8), 35 - 40.

45. NUERNBERGK, M.D. RORRES C. (2013) Analytical Model for Water Inflow of an Archimedes Screw Used in Hydropower Generation. *Journal of Hydraulic Engineering*, 139 (2), 213 - 220.

46. OERTEL, H. (2004) *Prandtl's Essentials of Fluid Mechanics*. New York, Springer.

47. Permanent Magnet Generator Applications. Available at: <http://www.pmga.eu/> [Accessed 19 July 2016].

48. RAYNAL, S. COURRET, D. CHATELLIER, L. LARINIER, M. DAVID, L. (2014) An experimental study on fish - friendly trashracks – Part 1. Inclined trashracks. *Journal of Hydraulic Research*, 51, 56 – 66.

49. ROHMER, J. KNITTEL, D. STURTZER, G. FLIELLER, D. RENAUD, J. (2016) *Modelling and experimental results of an Archimedes screw turbine*. *Renewable Energy*, 94, 136 - 146.

50. RORRES C. (2000) The turn of the Screw: Optimal Design of an Archimedes Screw. *Journal of Hydraulic Engineering*, 126 (1), 72 - 80.

51. SAROINSONG, T. SOENOKO, R. WAHYUDI, S. SASONGKO, M. (2015) The effect of head inflow and turbine axis angle towards the three row bladed screw turbine efficiency. *International Journal of Applied Engineering Research*, 10 (7), 16977 - 16984.

52. SENIOR, J.A. (2009) *Hydrostatic Pressure Converters for the Exploitation of Very Low Head Hydropower Potential*. Ph.D. University of Southampton.

53. SHENG, S. KELLER, J. GLINSKY, C. (2013) Gearbox Reliability Collaborative Update. *Sandia Reliable Workshop*, National Renewable Energy Laboratory, Albuquerque, U.S.A., 13 – 14 November 2013.

54. SKF (2007) *Spherical roller bearings.* [pdf] Available at: <http:// www.skf.com/binary/21148465/6100_EN.pdf> [Accessed 19 July 2016].

55. SKF (2000) *The ECO line squeezes costs in screw pumps*. [pdf] Available at: <https://www.ahrinternational.com/PDF_catalogues/SKF/058936_skf_focus_spherical_roller_bearings.pdf> [Accessed 19 July 2016].

56. SKINGLE, S. PAGE, M. (2012) *Hydropower Archimedean Screw Noise Impact Assessment*, Suffolk, Sound solution consultants. [pdf] Available at: <http://www.osneylockhydro.co.uk/wp-content/uploads/2013/03/Noise-assessment-report.pdf> [Accessed 19 July 2016].

57. SPAANS BABCOCK (2009) *Screw Generator.* [pdf] Available at: <http://www.edaenv.ca/_mndata/edaltd/uploaded_files/screwgenerator.pdf> [Accessed 19 July 2016].

58. TARRANT, M. (2011) *Comparative merits of the two proposed hydropower schemes at Avoncliff Weir.*

59. TULLIS, B.P. ROBINSON, S.C. (2008) Quantifying culvert exit loss. *Journal of Irrigation and Drainage Engineering*, 134 (2), 263 - 266.

60. TULLIS, B.P. ANDERSON, D.S. ROBINSON, S.C. (2008) Entrance loss coefficients and inlet control head - discharge relationships for buried - invert culverts. *Journal of Irrigation and Drainage Engineering*, 134 (6), 831 - 839.

61. WARNICK, C.C. (1984) *Hydropower Engineering*, Englewood Cliffs, New Jersey, Prentice – Hall.

62. WILLIAMSON S.J, STARK B.H, BOOKER, J.D (2011). *Low Head Pico Hydro Turbine Selection using a Multi – Criteria Analysis, World Renewable Energy Congress, 8 – 13 May 2011, Linkoping, Sweden.*

63. WIEMANN, P., MÜLLER, G., SENIOR, J. (2007). *Review of current developments in low head, small hydropower*. 32 nd IAHR Conference, 1 – 6 July 2007, Venice, Italy.

64. WIEMANN, P., MÜLLER, G., SENIOR, J. (2008). *Risk management and resolution strategies for established and low technologies in the low head small hydropower market.* [pdf] Available at: <http://www.hylow.eu/knowledge/all-download-documents/3%20P%20Wiemann.pdf.> [Accessed 19 July 2016].

Appendices

Appendix A. Minimum head for low head turbines

Minimum head and estimated flow at this head based on various published studies and information from manufacturer's websites are presented here. In many cases the manufacturers offer options for lower flow but the charts aim to show the lowest point of the performance envelope as regards the head.

Figure A. 1 Crossflow turbines.

Figure A. 2 Kaplan turbines and similar versions.

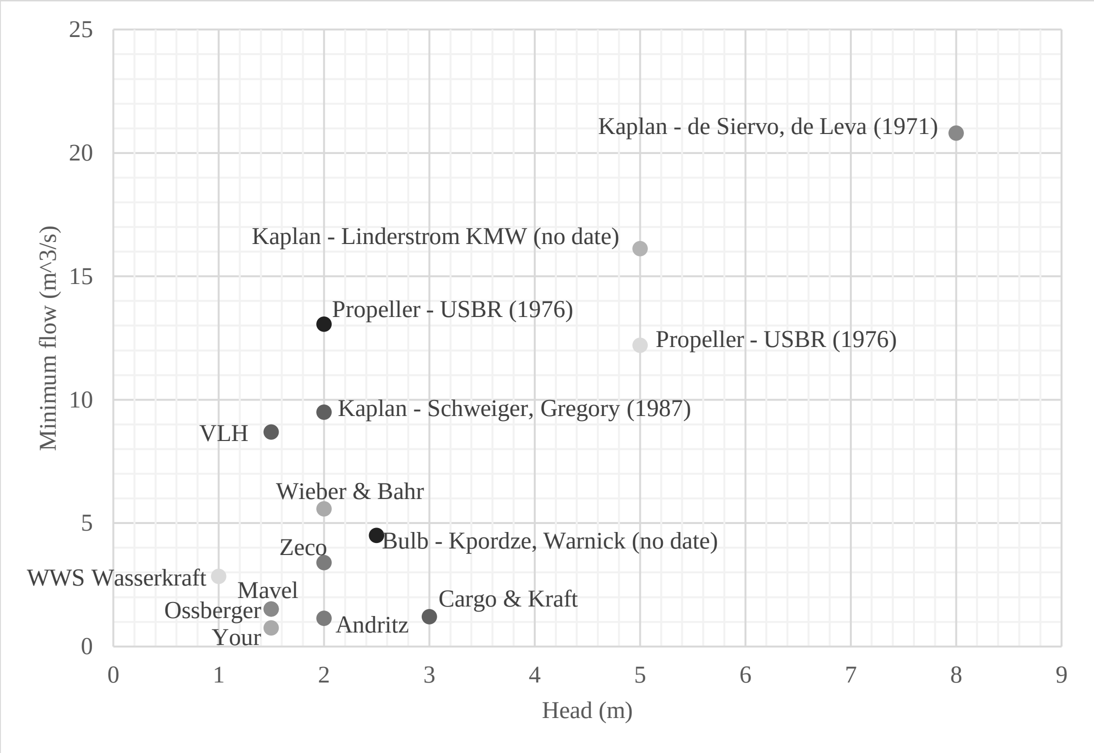


Figure A. 3 Francis turbines.

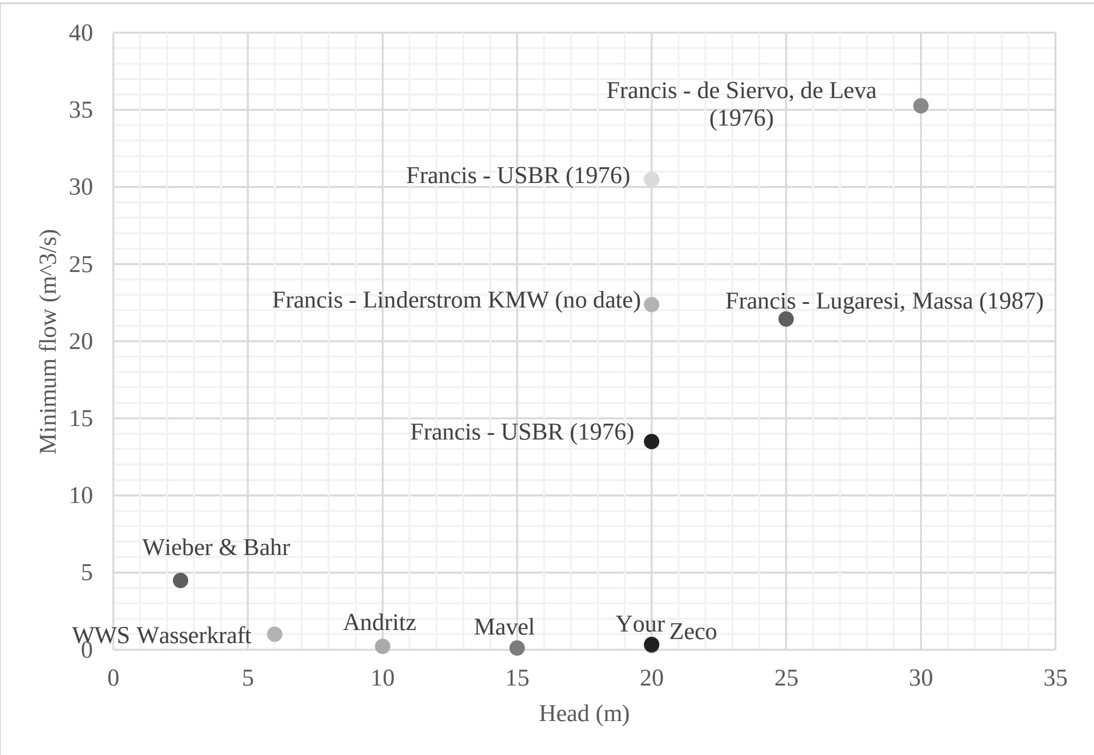


Figure A. 4 Water Wheels turbines.

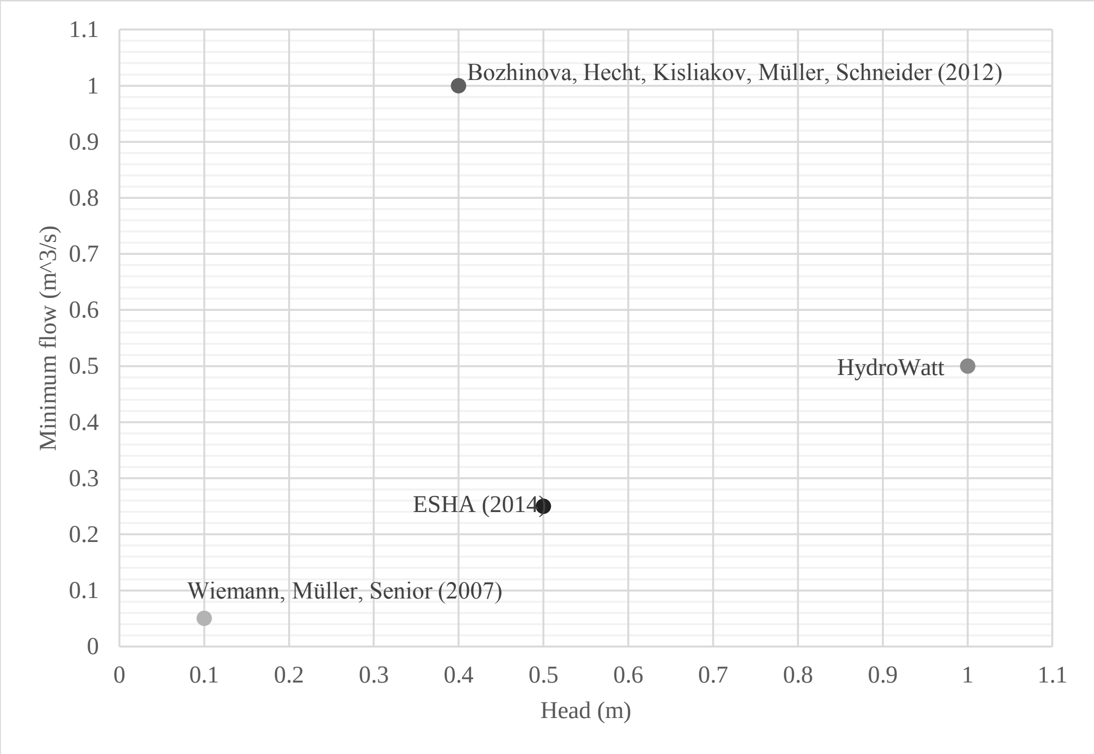
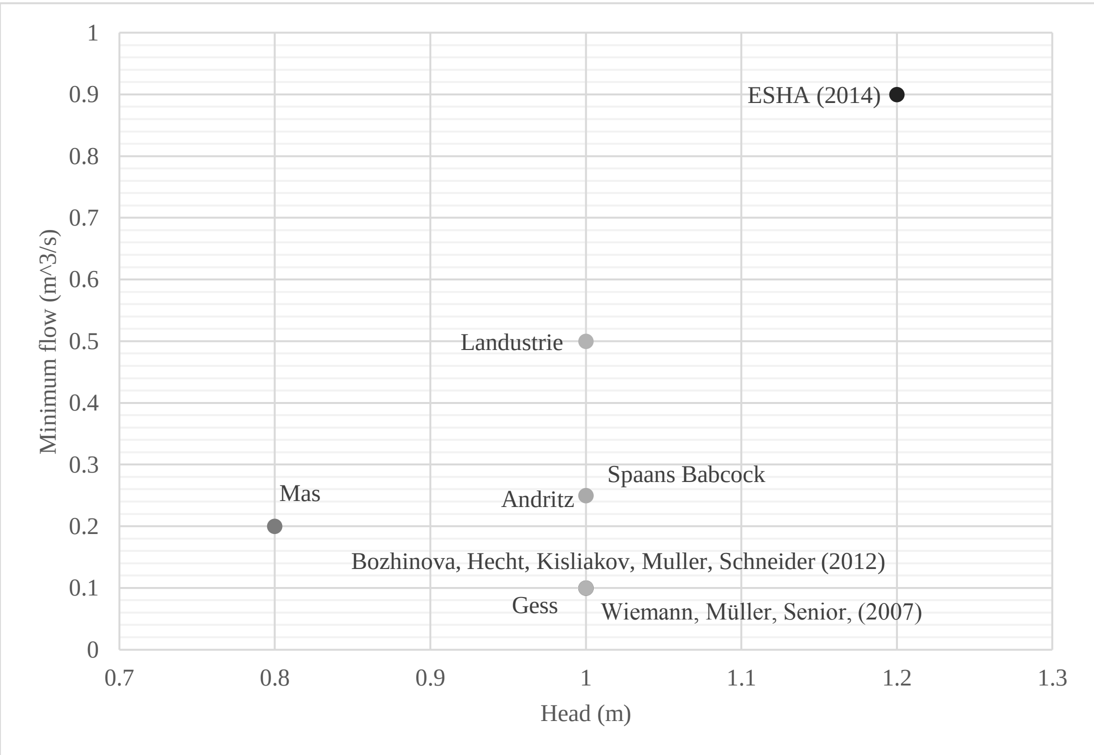


Figure A. 5 Archimedes Screw turbines.



Appendix B. Losses categorized by different loss mechanisms

Charts for each category of the losses are presented here for the three stages described in chapter five and the existing design which is in operation now. The vertical axis represents each loss as a percentage of the total available power for the setup in each stage and the horizontal axis the flow fraction. The charts make clear what is the effect of the changes suggested in each stage in the efficiency of the scheme aiming to provide guidance in case of a decision making process for any potential upgrade of the project studied during this research.

Figure B. 1 Delivery channel losses.

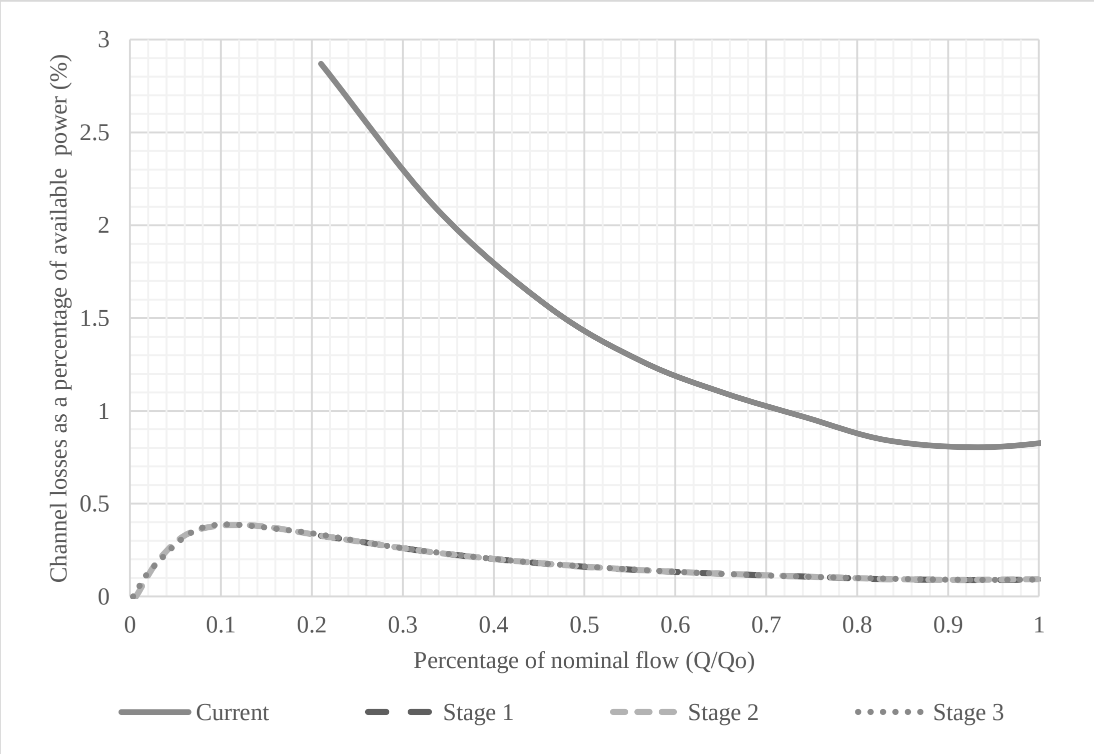


Figure B. 2 Screening losses.

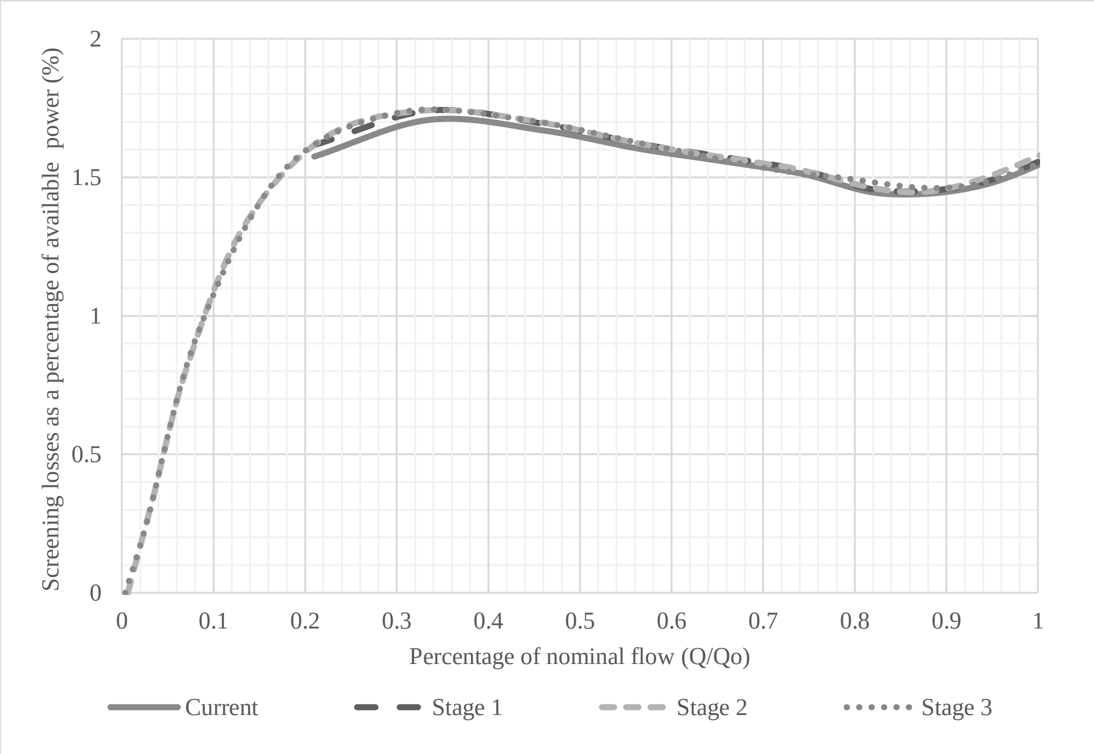


Figure B. 3 Inlet transition losses.

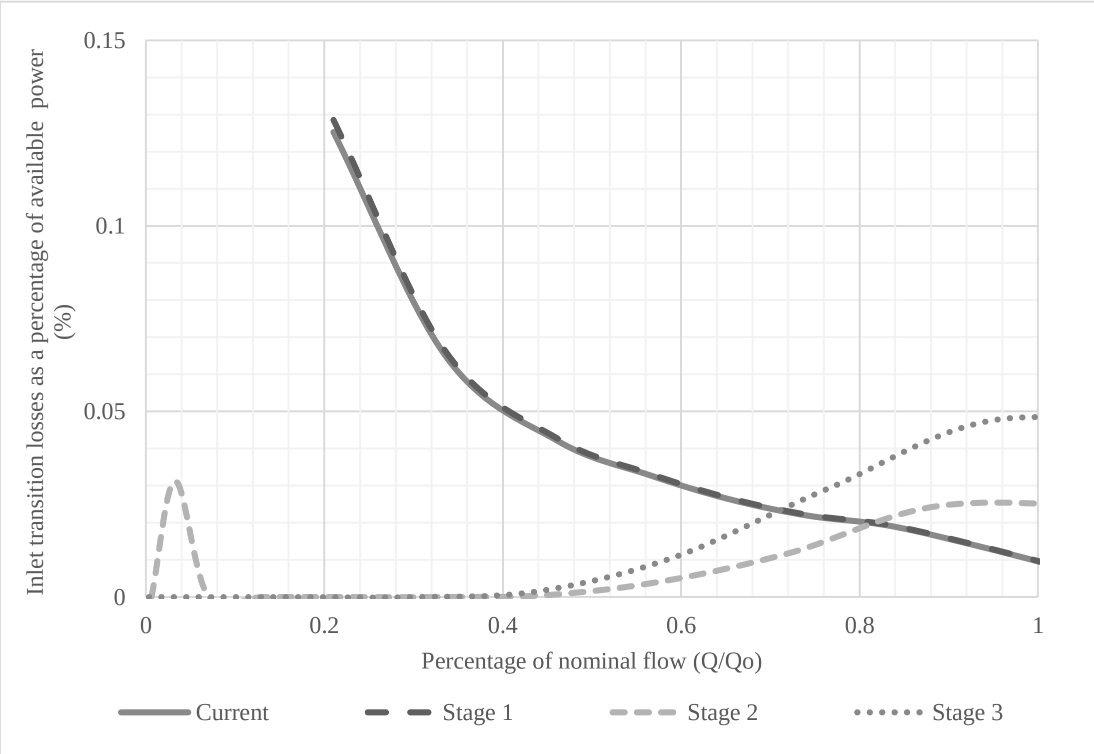


Figure B. 4 Leakage losses.

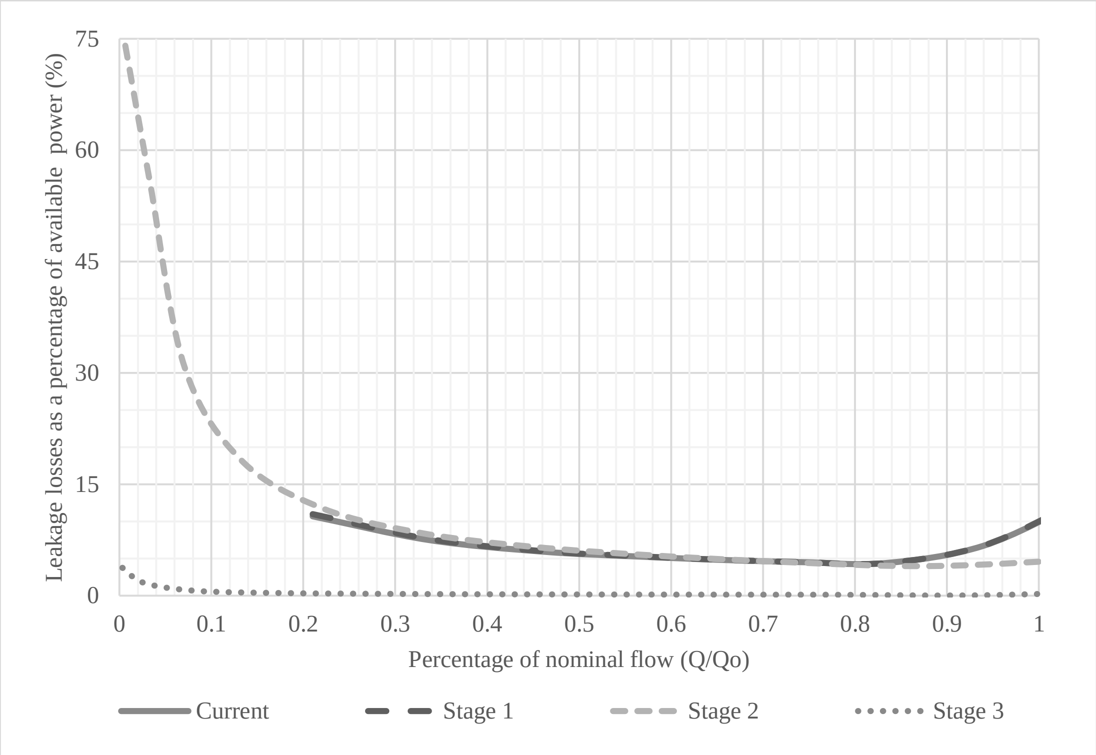


Figure B. 5 Friction losses.

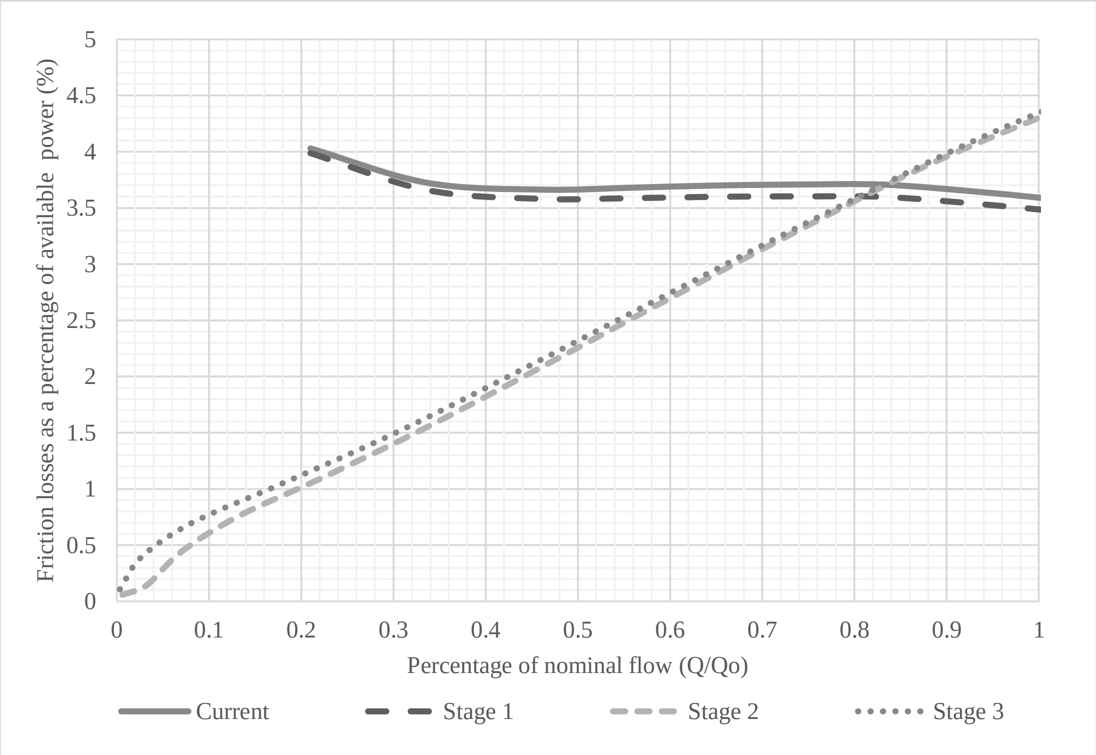


Figure B. 6 Outlet transition losses.

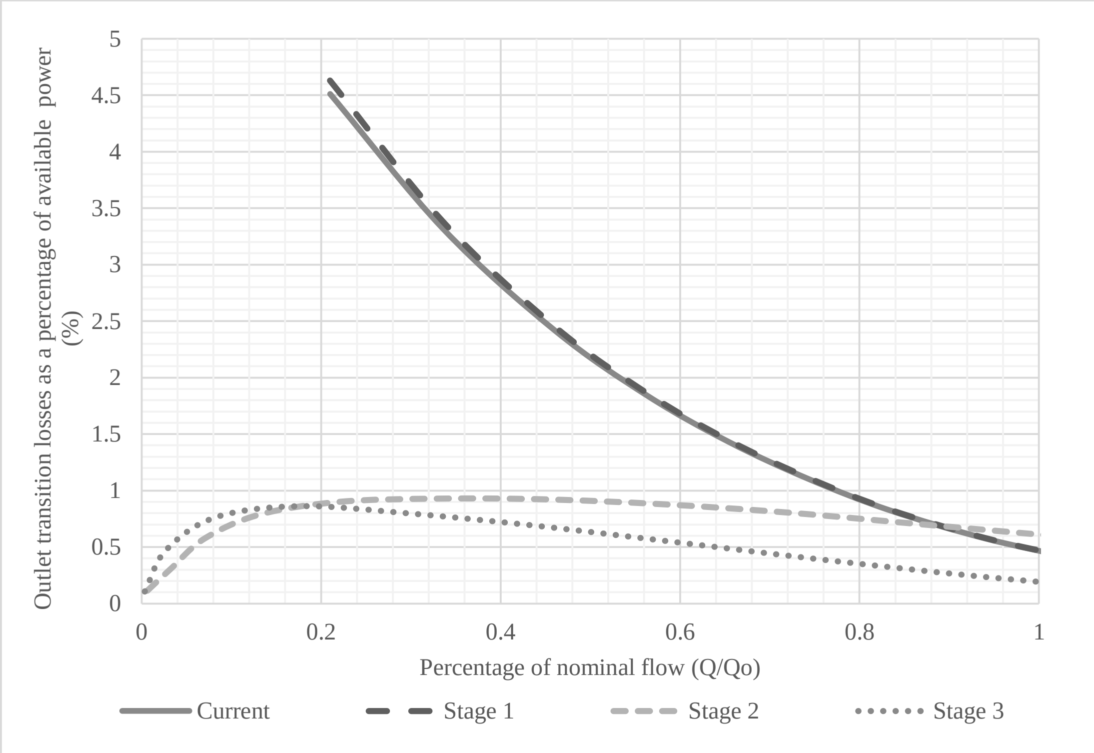


Figure B. 7 Drag losses.

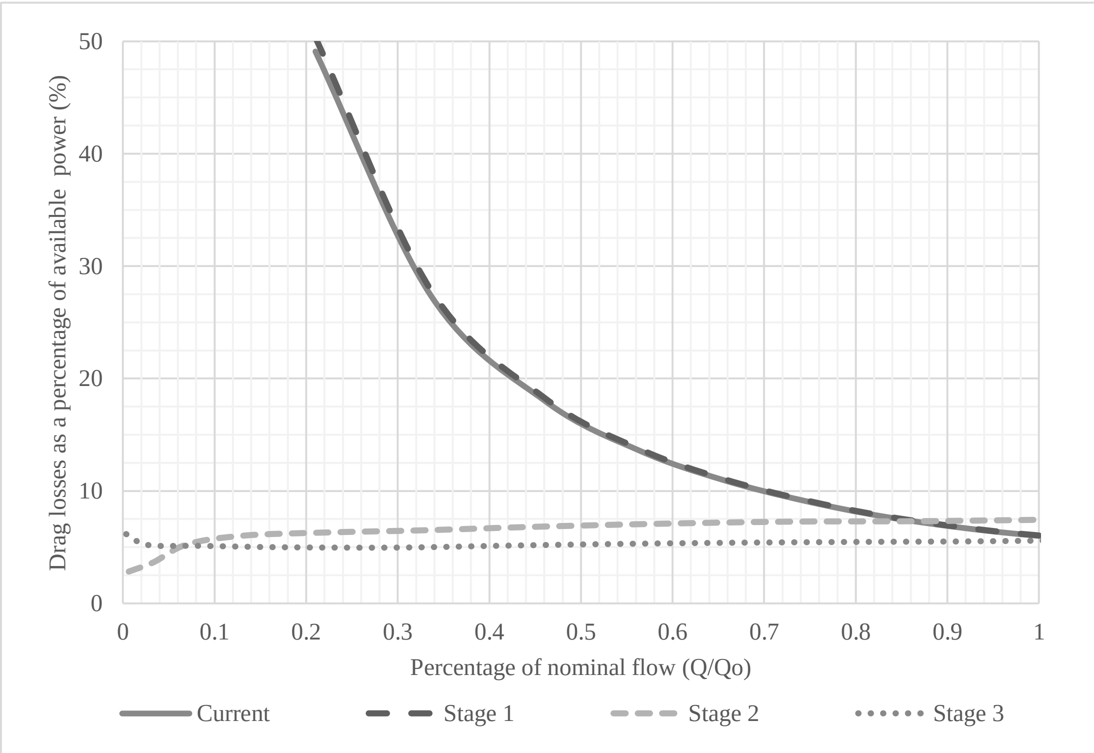


Figure B. 8 Bearings losses.

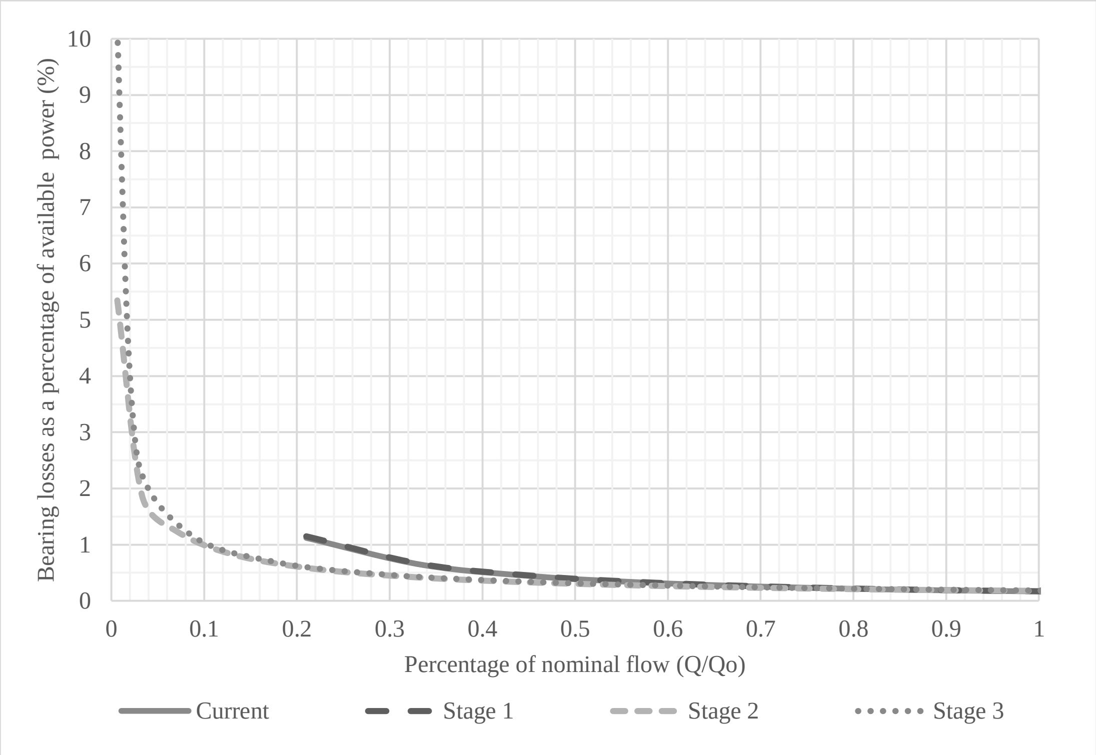


Figure B. 9 Transmission losses.

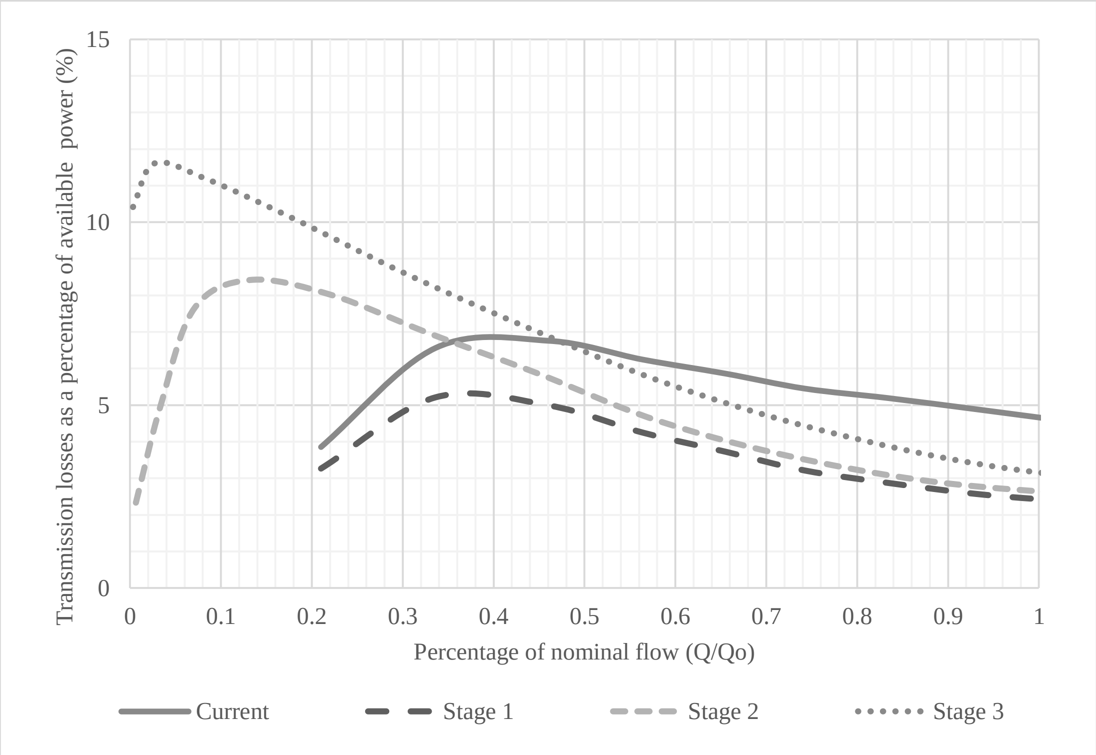


Figure B. 10 Generator losses.

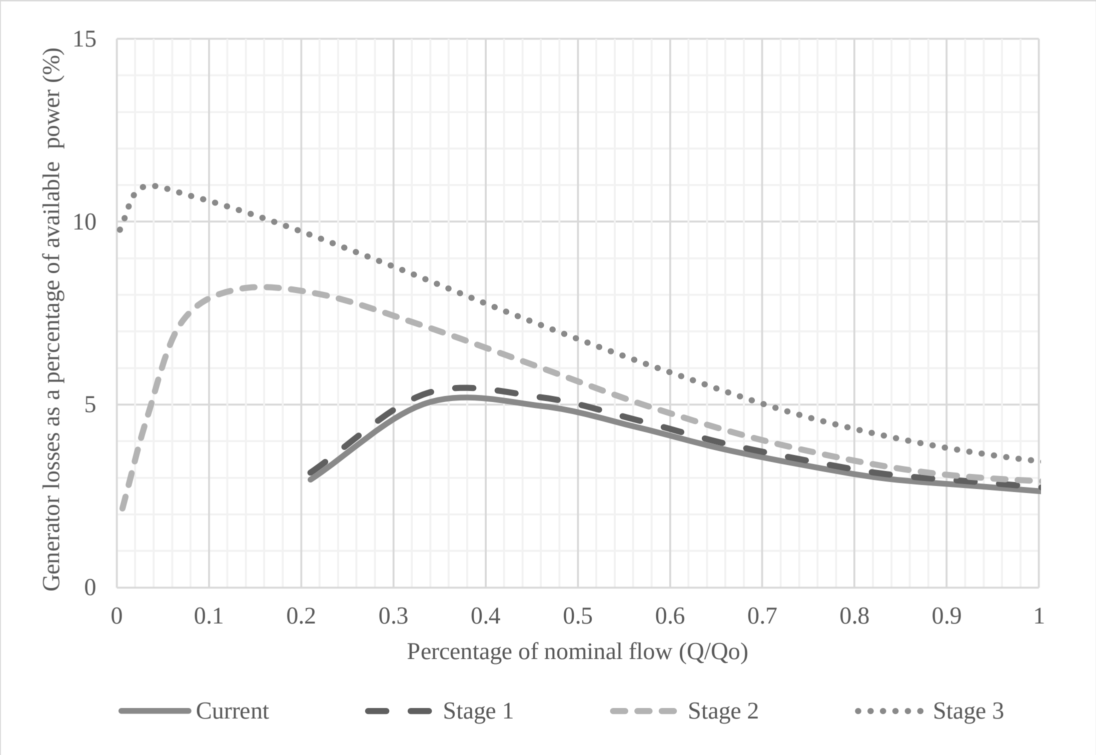


Figure B. 11 Frequency converter losses.

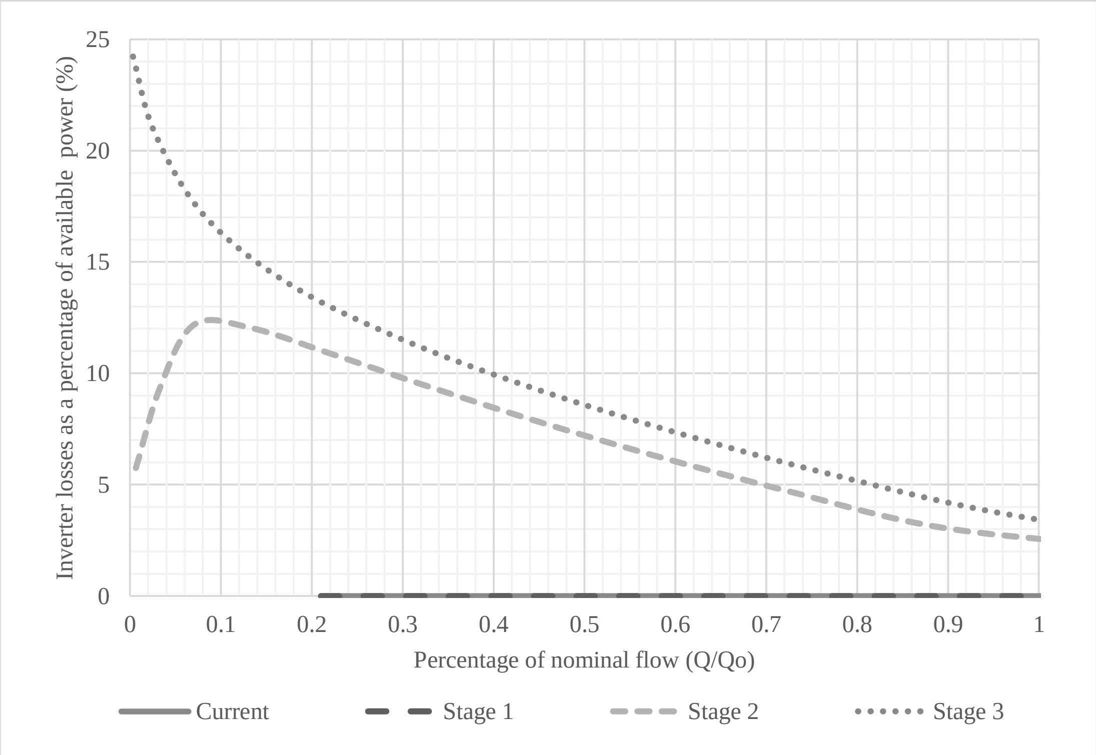
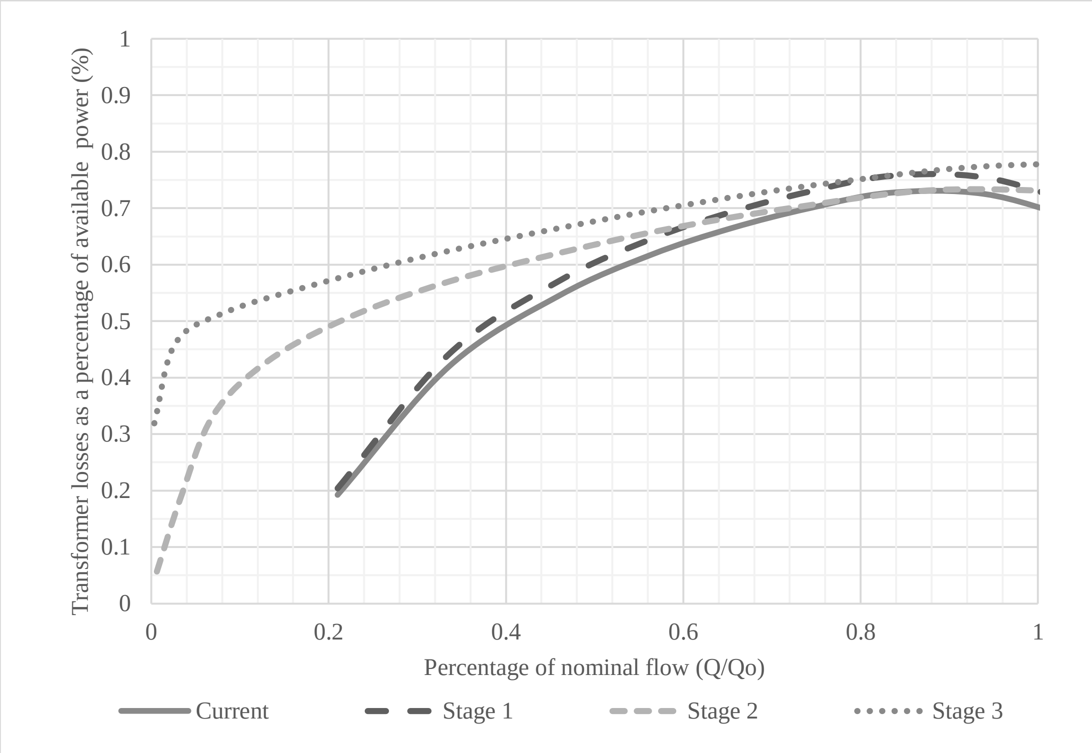


Figure B. 12 Transformer losses.



Appendix C. Discussion - Recommendations

The presented study attempted to develop a methodology, based on available literature, for the estimation of the performance of a conventional hydropower station using an Archimedes screw turbine for electricity generation. The novelty of the present research is that attempts an overall, as total project, water to wire estimation of the various losses mechanisms and is not focused only in the turbine losses as it happens many times, aiming to suggest improvements of the performance in an existing scheme. The results of the simulation seem that it may be possible the reduction of the losses by different design techniques which depend on which loss needs to be limited.

Based on the results of the study presented in figure 2.22, for head between 2 and 3.5 m and for capacity between 30 and 150 kW, the average cost of a hydro power scheme using a Kaplan turbine is 5228 €/kW, while the same cost for hydropower scheme using variable speed Archimedes screw turbine is 4579 €/kW (12.4 % reduced) and for a scheme using fixed speed Archimedes screw turbine is 3851 €/kW (26.3 % reduced).

The nominal power of the projects with Archimedes screw is limited, up to 500 kW, and in combination with the free surface design, two challenges are appeared: a) because of the low total power there is a significant contribution in losses production from sources that considered negligible in high power (e.g. screening) and b) there are loss mechanisms because of the free surface approach which do not exist, or at least not in the same form, in impulse and reaction turbines (transition, drag, forebay losses). These challenges make difficult the understanding of how the Archimedes screw works and which are really the advantages and disadvantages from its use.

It is not far from the reality to say that it is an attractive option for low and very low head schemes, especially as the number of available sites is rapidly reduced, the nominal power is reduced as well, and the competition pushes the investors to look for even cheaper options for the equipment of the projects every time. As it was mentioned in the literature review, there are opinions according to which the Archimedes screw turbines is the type of the turbine with the highest potential in the following years. However it must also be admitted that except for the advantages of the turbine itself, there is another possible reason that explains up to a point why this turbines becomes more and more famous: the marketing. Archimedes screw turbines are used widely in hydropower development for less than twenty years. Their supporters, except for the element of mystery that they use to promote them as a so called 'ancient' device existing for more than twenty-five centuries adapting mature technology, they present them as an alternative for Kaplan turbines, simpler, cheaper but with similar performance. But is this true? The Kaplan turbine is available in many versions (vertical, horizontal, single regulated, double regulated etc) covering almost all the available commercial range in low head. It has been developed since around a century ago and it is possible that its development limits are close to be reached, if they are not already reached, since it incorporates very high efficiency in very wide flow fraction (Figure 2.3).

The Kaplan turbine may be more complex because of a number parts that does not exist in case of an Archimedes screw turbine (penstock, draft tube, variable blades etc) but on the other hand its size is much more compact for a reference flow and power, and possibly overcomes the advantage of simplicity that the screw may has. As the flow is increased, the diameter of the Kaplan rotor tends to be increased and can often reach 2-3 m for large flow in small hydropower applications, which is again much smaller related with the 5 m that an Archimedes screw rotor can reach. It has to be noted that not all the manufacturers provided 5 m diameter turbines because, as it was explained in literature review, it is believed that the transport cost in this is very high and the transportation to middle and long distances lead to high increase of the cost and as a result the low cost, which is promoted as one of the main advantages, is lost.

In figure 6.1 are summarized the available and the estimated in case of improvement electrical efficiency curves for the Archimedes screw turbine which is installed in the studied hydropower station, with the estimated electrical efficiency of a double regulated Kaplan turbine which could had been installed instead of the Archimedes screw. It seems that even in the most advanced design scenario the electrical efficiency of the plant using Archimedes screw turbine is around 10 % less efficient (at some points more) than the same scheme with a Kaplan turbine in the range between 60 – 100 % of the nominal flow. Below 60 % the difference is reduced and the setup with the Kaplan turbine may have lower efficiency in the range up to 25 % of the nominal flow. It is said as 'may' because usually the numerical models have lack of accuracy in low flows because many geometric values cannot be calculated properly (e.g. the hydraulic radius) and because in this area complex phenomena are appeared in the flow which reduce the efficiency and cannot be modelled, at least with a simple analytical model like the one developed in the present study. The differences in efficiency mentioned are for the optimal design. The current design, according to the results of the simulation, has much lower performance related with the Kaplan and the electrical efficiency is reduced between 10 % (100 % of the nominal flow) and 35 % (20 % of the nominal flow). These numbers are very far from the claimed electrical efficiency which is also presented in the same Figure, although even the claimed electrical efficiency of the studied scheme using Archimedes screw turbine is reduced from 0 to around 8 % in the range from 25 to 100 % of the nominal flow.

The presented curves for the Archimedes screw may actually be very lower than the presented ones because of the forebay losses which are the result mainly because of the head loss before and after the sluice gate of the delivery channel. There are no information for these losses or for the difference between the water level in the delivery channel and the upper reservoir which would make their estimation possible. In any case, because the capacity and the exploited head of the project is low, the forebay losses can be a large percentage of the available power even with a limited water level difference before and after the sluice gate. In addition, it has to be reminded that during the calculation was considered fixed water level at the lower reservoir which does not happens often in real power plants. In case that this water level changes for various flows the drag and the outlet transition losses reduce efficiency, except for the head reduction which will lead to a power reduction. It has to be mentioned as well that a Kaplan turbine is not affected by water level variation in terms of losses increase, except for the head reduction which lead in the same way to power reduction.

So taking into consideration the large difference in the electrical efficiency, even for the optimal suggested design, it is estimated that the 12.4 % difference in the installation cost is not meant to cover the difference of the reduced energy production because of the reduced efficiency for a period of a few decades that the turbine will be operational. The project using an Archimedes screw turbine is really cheaper but its performance is so much reduced that cannot compete the setup with a double regulated Kaplan turbine which would overcome the small benefit of the reduced installation cost in a few years. Unfortunately no flow data recordings were available that could be used for the estimation of the produced annual energy and any investment evaluation values for the project (IRR, NPV etc).

It has to be noted however that this advantage in performance which was studied so far has to do only with the double regulated or full Kaplan, which is the turbine from the Kaplan family with regulated both guide vanes and rotor blades. It was briefly explained in the literature review that there are two more turbines in the Kaplan family with different characteristics in terms of blade and vane regulation. They are generally known as semi-Kaplan and Propeller, although from country to country their names sometimes vary in practice. The semi-Kaplan is the same with the full Kaplan with only difference that its guide vanes are fixed and the rotor blades are variable. The Propeller is again the same with full Kaplan but this time the guide vanes are variable and the rotor blades are fixed. So in fact both of them are somehow semi-Kaplan turbines or single regulated but the regulated stage is different every time. These turbines have reduced cost related with the double regulated but not much reduced because actually their only difference is that they do not have the one out of two regulation mechanism that changes the angle of the blades. Their performance however is significantly reduced related with the double regulated in reduced flow. As it is illustrated in Figure C.1, where the electrical efficiency of the advanced design Archimedes screw turbine calculated below and the electrical efficiencies of these two Kaplan types are presented, their performance is much lower related with the curve of a full Kaplan from figure 6.1. The fact that they do not have fully adjustable system makes them unable to adapt themselves in flow conditions far from the design point. From the Figure it seems that the Archimedes screw turbine has advantage in performance below the 80 % of the nominal flow related with the Propeller and below 60 % related with the semi Kaplan. The difference in efficiency is increased in favor of the Archimedes screw as the flow is reduced. That leads to the estimation that its overall performance is better in schemes where there is significant variation in the flow conditions while the Kaplan family turbines presented in Figure C.1 will be better in case that the flow varies from 20 % (Propeller) to 40 % (Semi Kaplan) related with the nominal flow. The low difference in cost in combination with the large difference in performance related with the double regulated Kaplan turbine are the main reasons why usually they are not preferred and as ‘Kaplan’ is meant in practice the double regulated Kaplan since the other two types described here are used rarely in low head applications. It has to be reminded here that the efficiency curves of all the Kaplan turbines were taken from the available literature so they are ‘a general case’. The curves provided by the manufacturers may differ because the ones from the literature represent a realistic but almost ideal performance, from the ones presented here, especially in reduced flow. On the other hand, for the estimation of the electrical efficiency assumed a transmission system, which may be always necessary for the Archimedes screw turbines because of their very low speed, but very often it is not necessary for the Kaplan turbines (in case of the studied scheme it would be necessary) since, because of their relatively high speed, they can be designed for direct drive connection with the generator. The direct drive would make not necessary the use of transmission and as a result the losses because of it, which are not negligible especially in low flow, would be possible to be avoided achieving higher electrical efficiency.

In our days the Kaplan turbine is usually the first type of turbine that has a developer in his mind when he starts studying a low head project, which is reasonable based on its advantages described so far. In the first chapter were presented the available technologies for low and very low head schemes exploitation with acceptable performance. The Archimedes screw is one of the two hydraulic machines known from ancient times but until recently it was used exclusively as a pump. Because of this, and as results from the study that reveals that there are still many potential areas for research, it is difficult for someone to claim that the Archimedes screw is an absolutely mature device for use as a turbine for electricity production. On the other hand, the family of the Water Wheels, which is the second hydraulic machine known for thousands of years, was always used for power production and in our days for electricity production. The Water Wheels are available in different versions and depends on the application and the design, they are known with various names. The most well-known are the Undershot, Breastshot and Overshot water wheels. As it is presented in figure 2.9, they are used in low and very low head applications and the name occurs from the head and the position of the point that the flow meets the rotor related with the axis of rotation. The Water Wheels share the most of the advantages of the Archimedes screw turbines like the simplicity in calculation and design, no moving parts and the relatively low cost. In the same way, they share the main disadvantages like the large size for low power which makes them as a result unable to exploit high power, the low speed (6 – 20 rpm) that makes the transmission necessary and the reduced efficiency in some cases, at least related with the Kaplan turbine. Perhaps the only disadvantages of the Water Wheels related with the Archimedes screw turbines are that they cannot be used as pumps, which is sometimes useful in waste water applications, and that the fish cannot swim through them so they are not so fish friendly and ladders and other constructions are necessary to ensure the fish migration. Of course, ladders and constructions to help fish migration are used with Archimedes screw turbines in many cases as well because as it was explained in the literature review, they are not as fish friendly as it is believed since they can create serious damages to the fish entering the buckets because of the pressure difference and serious injuries from entrance in the gap between the trough and the rotor (small fish) or impact with the blades in high tip speed (large fish).

In figure C.2 are presented the electrical efficiency curves for the Water Wheel turbines, based on the information presented in figure 2.3, in comparison again with the electrical efficiency curve of the advanced design Archimedes screw from the previous chapter. As it was mentioned above for the curves of the Kaplan turbine, the efficiency curves for the Water Wheel turbines are ‘a general case’ and the curves provided by the manufacturers may differ significantly, based on the flow characteristics and the know-how of each manufacturer. From the curves plotted in figure C.2 it seems that the Archimedes screw turbine has a large advantage related with the Undershot Water Wheel turbine since its electrical efficiency is between 10 and 15 % higher. It has also higher electrical efficiency related with the Breastshot Water Wheel turbine from 1 to about 8 %. However the Overshot Water Wheel turbine looks to has better performance and its electrical efficiency is increased between 1 and 8 % related with the Archimedes screw turbine, approaching even more the performance curve of the double regulated Kaplan turbine presented in figure 6.1.

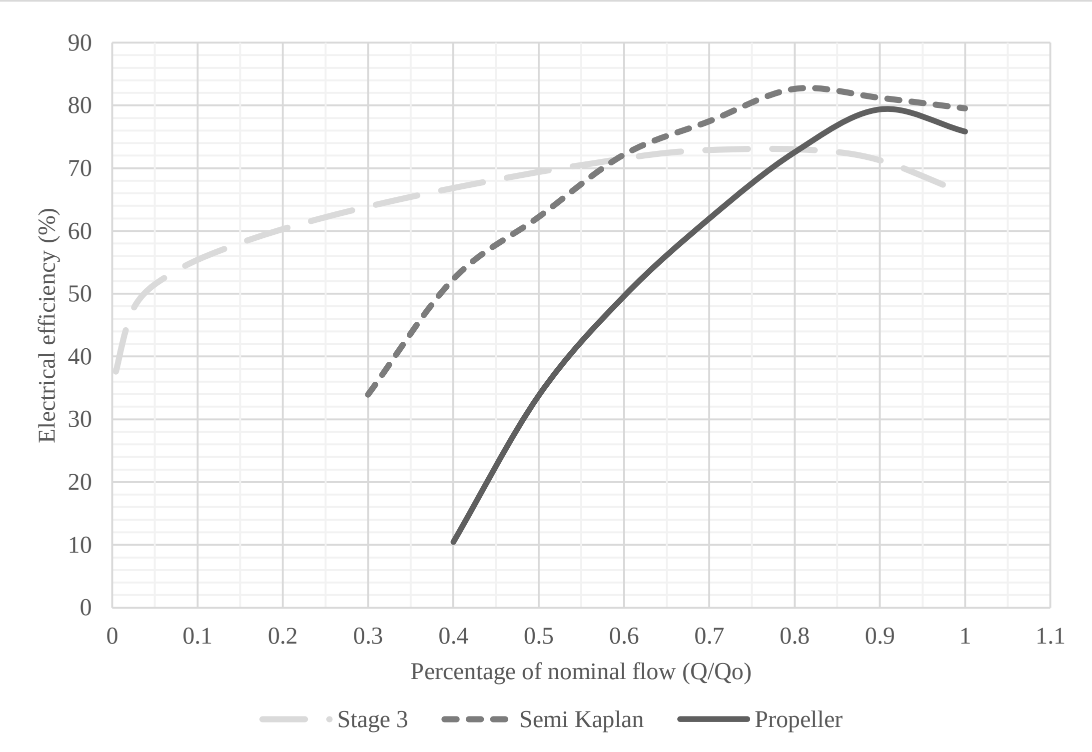


Figure C. 1 Electrical efficiency curves for the suggested advanced design Archimedes screw turbine, a typical semi-Kaplan hydro turbine and a typical Propeller hydro turbine.

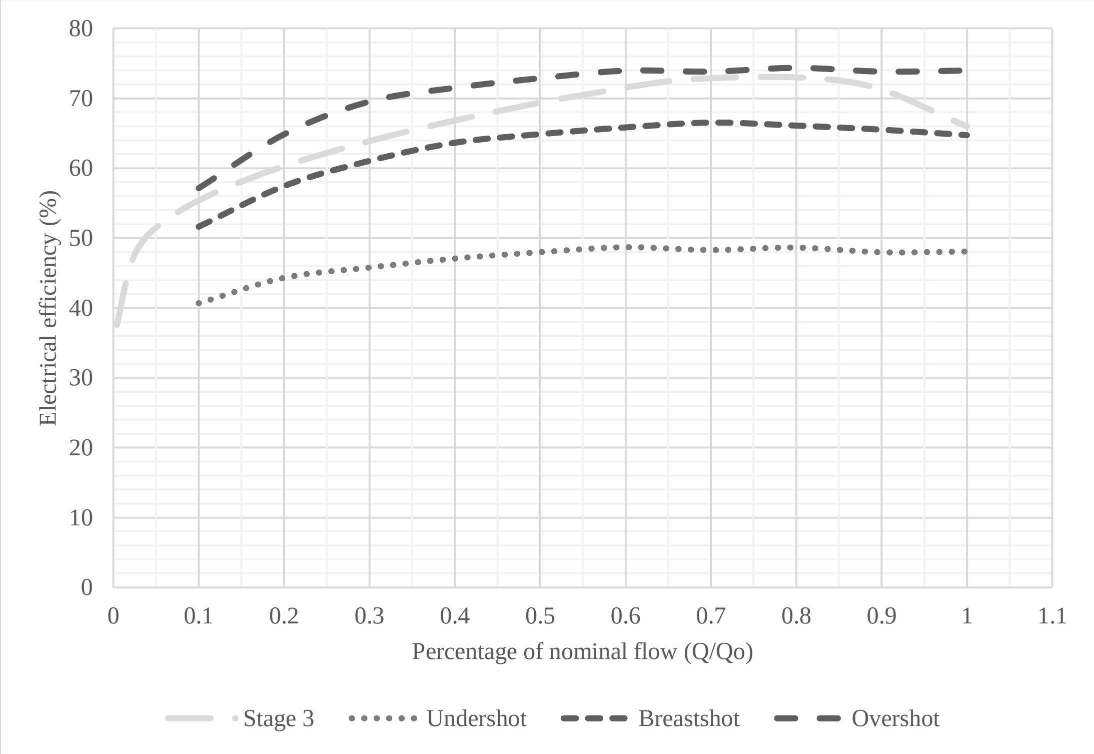


Figure C. 2 Electrical efficiency curves for the suggested advanced design Archimedes screw turbine and the Water Wheel hydro turbines.

Unfortunately, it is doubtful if an Overshot type Water Wheel turbine can reach below the 2 m as it is necessary in the studied scheme to cover the available head (around 1 m minimum) because as it is presented in figure 2.3 it can be used for nominal head down to 2 m and the maximum flow it can achieve is less than 1 m^3/s which is not enough to cover the range needed (around 3.2 m^3/s maximum). Of course, the available information are based on the available literature as it was described in the first chapter and it is possible that the limits in head flow are not so determined. In other words, it is possible that as a general rule the Overshot Water Wheel can acceptably work between 2 and 8 m head and between 0.1 and 0.8 m^3/s flow but it is also related with the sources that the available literature has used (companies, books etc) and it is difficult to be proved that the particular turbine cannot have acceptable efficiency in performance envelopes that slightly vary from the range mentioned. The same applies to all the types of hydro turbines, the literature can be used as a general rule but manufacturer can offer slightly or not different ranges and characteristics based on his knowledge and his experience to cover the market needs. As regards the Kaplan turbine for example, if someone makes a search on the websites of the main manufacturers (some of them are mentioned in figure A2) will realize that for the same turbine in practice the minimum head can be from 1 to 8 m.

As a general conclusion, although many information are not available (e.g flow-duration curve, forebay losses, tail water level variation) to be able to have an absolutely clear view, there is an indication that for the studied hydro power station, which was used as a case study in the present research on Archimedes screw turbines, the installation of such a turbine for the exploitation of the available capacity was not the optimal choice.

The accuracy of the model and the reality of this claim cannot be certified without any experimental data but it seems that the curve represents somehow the hydraulic efficiency of the installed turbine in different flow conditions and perhaps in fixed head. In case of a reaction turbine the efficiency estimated using this approach would be accurate enough as hydraulic efficiency. In case of an Archimedes screw turbines however its free surface design is highly affected from the water level variations of the inlet and outlet channel and as a result they have to be taken seriously into consideration to achieve an acceptable estimation of the performance. The installation of a double regulated Kaplan turbine is almost certain that would lead to higher annual energy production and as a result higher income for the owner. Also, because it would be fully covered by the power house, the noise levels would may be lower, which would be important if the scheme was installed close to residences.

Based on the collected information available in public domain so far, the results of the simulations and a number of unofficial discussions and opinions exchange with people from industry and academia related with hydropower, the author believes that a double regulated Kaplan turbine, properly installed, will always have better electrical efficiency from an Archimedes screw turbine, based on the technology existing so far, at least in the largest part of the available flow and head that they will be compared. The electrical efficiency of the Archimedes screw turbine will be even more reduced in case of significant inlet and outlet channel water level variations which will increase the drag and the transition losses. According to the available literature (Aggidis et al 2010) the cost of the Kaplan turbine per kW is increased for flows below 5 m^3/s. In this area the Archimedes screw installations may be able to become more competitive, in terms of investment evaluation of a hydropower scheme, taking into consideration that they cannot compete installations using Kaplan turbines in performance. To do that the manufacturers have to be careful when they form their pricing policy because it does not make sense two types of hydro turbines with significant difference in performance to have comparable installation cost.

Closing the wring process of the thesis are going to be quoted a few thoughts that came out during this year and directions for anyone who will decide sooner or later to study a close field in a future research procedure.

The Archimedes screw turbine, as it has already mentioned, is a very recently introduced commercial hydro turbine. The term commercial is used because from time to time various devices and patents are appeared for electricity production in hydropower sites, however either because of limited performance or for other reasons they are rarely utilized widely. It is important to remember that the traditional hydro turbine types that usually used (Pelton, Francis, Kaplan) are in use since many decades ago, perhaps more than a century, in their current or similar form and in our days their development has reached top level in more or less all the critical areas (performance, serviceability, cost etc). Also, the main manufacturers, not only in Europe, have produced thousands of machines for each type creating a huge record that has helped to enrich their experience and optimize their turbines. Archimedes screw have been used as pumps since many centuries ago. According to Nagel (Nagel 1968) before the 2nd W.W. there were more than 300 screw pumps installed only in Germany. This shows that there was a large sector including companies, technical staff, workshops, universities etc with activity around Archimedes screw pumps development. The screw pumps are in wide use today in small sizes for industrial viscous fluids like lubricant but also in large sizes as Archimedes screws for irrigations and similar purposes.

Archimedes screw turbines started to be used for electricity production almost two decades ago. It is possible that there was a number of efforts before but the first official patent for 'Hydrodynamic screw for energy conversion' was given just in 1997 to Karl August Radlik. The same period the first experimental work started to be published by Brada showing for the first time in history that this ancient device can be used for electricity production with a very promising for further development performance. So the whole scientific work in this type of turbines has been in progress since less than two decades ago with a relatively small number of people have conducted any research on this field.

The truth is that in our days a small number of companies, related with the number of companies that produce the more traditionally used turbine types, offer Archimedes screw turbines and the most well-known of them are in Europe. They all have their own technology but from the information available in public domain they generally provide turbines up to 10 m head and up to 10 – 15 m^3/s flow. Also, depends on their capabilities the maximum dimensions in length and diameter they can construct varies, possibly because of the market needs and the high transportation cost that the large diameter machines demand for middle and high road distances.

Archimedes screw turbines are often suggested as a low cost but acceptable performance alternative option for Kaplan turbines in low head applications. However, according to a number of people in hydropower community, their performance is not as high as the manufacturers disseminate and in the same time their installation cost is not much lower than a Kaplan, without of course taking into consideration the maintenance cost, especially for the transmission system which is necessary with this type of turbines and in case of works that need rotor disassembly. The size of the rotor for the studied site (7 t, 2.6 m diameter, 7.8 m long) makes understandable the disassembly difficulty related with a reaction turbine of the same nominal power. The free surface design creates loss mechanisms that do not exist in other turbines, or at least they do not have the same form and they are very limited related with the available power, mainly the transition, leakage and drag losses. The drag losses can be minimized, at least in low flow, using a frequency converter with the disadvantage of the introduction of new electrical losses and the significant cost increase since it is a very expensive device. The free surface has also another effect in the performance because of the water level variations. In the usual types of turbines a water level variation of the upstream (decrease) or the downstream (increase) would lead only to a reduction of the available power and as a result a reduction to the produced power, if this variation was not so high to change the performance curve. In Archimedes screw turbines, even if the water level varies on that way to maintain the available net head constant, it is possible that the turbine will encounter very high loses in case of significant reduction of the inlet water level or significant increase of the outlet water level for reference flow. On the same way, it is possible that it will not be able to exploit inlet water level over a point or outlet water level under a point and as a result there will be head losses, since the design of the turbine is such that the exploited net head is correlated, among other parameters, with the length of the screw and the slope angle.

But Archimedes screw turbines do not have only disadvantages. If the flow conditions are proper for its installation, and this is a big if, it can achieve a good enough efficiency in a large part of the flow range. In case of interest in the future for further study a number of potential areas of interest can be suggested based on the short experience that was developed during this year based on this study.

In the technical field first of all, it would have major interest if a test rig could set up and an extended experimental process take place. The results of such tests could provide valuable information for the behaviour of the device in a large number of scenarios, especially the scenarios that represent the not ideal installation conditions (extended water level variations, inlet water level reduction combined with flow increase, high speed etc). The problem in this case is that it is not easy to measure and quantify the effect of each loss mechanism separately with accuracy. It is suggested that in case of a test rig construction the dimensions of the tested rotor should be small enough to be easily exchangeable and able to be manufactured in a 3D printer. This will make possible the test of various geometries without the effort of changing a large geometry rotor every time. It is also suggested that the main non geometrical variables of the flow (upper water level, lower water level, rotational speed, flow, slope angle) should be easily regulated in the rig and independently each other. Of course, since the size will be very small the limitations because of similarity and specific speed have to be taken into consideration.

A CFD simulation effort is not suggested as the first priority of a future study, except for there is a long period and much computational resources (parallel computing) available. A normal size Archimedes screw model for a computational simulation will lead to a very large number of elements because of the physical dimensions including, water, air (two phase) and except for the trough and the rotor, a part of the inlet and outlet channel. Also, because of the low speed the physical time will be very large because a transient simulation with many rotation of the screw will be necessary for an acceptable result. The preparation of the mesh will be time consuming, considering the duration of an MSc course but it is possible. The main problem is the simulation since the available licenses for the parallel computing available in the department (128 in August 2016) is not enough for such a time consuming simulation in less than a few weeks, considering also that there are more students that use them for their research. It would be possible the simulation of a very small turbine which would also have relatively high speed and as a result the simulating time would be much smaller, but it is doubtful if the simulation of a very small machine could provide results that can be used for large scale machines.

Based on the speculations of the author during the effort to understand how these devices work, an interesting field of study is considered the effect of drag losses and how it is related with the speed, diameter and water level of the tailwater. It would also be interesting the study of the effect of the variable speed to the inlet transition losses. Finally, it would be very useful if the forebay losses could be avoided by removing the sluice gate and making possible the exploitation of the total available inlet water level either using variable speed and/or variable slope or in a more challenging approach by using a large variable geometry inlet nozzle.

Except for the performance, the manufacturability is another interesting field because of the dimensions of the device. A field of study in this area would be the possibility to manufacture large screws using composite materials, and even better if on this way a modular manufacture of the rotor in many parts with assembly on site would be possible. This could ideally lead to large reduction of the transport cost, which is a problem for long road transportations, since the maximum dimensions of a single part would be reduced.

For the end, an economical study that could investigate and quantify the cost of manufacture and transport as a function of the size would have major interest since it is possible that at least the manufacturing cost is not so high and it could result that the real cost of the turbine can be significantly lower than a Kaplan turbine. It is possible that the performance cannot increased much. However if the installation cost can be reduced enough, Archimedes screw turbines can be a very competitive option for low head hydropower applications in the following years.