ADDIS ABABA UNIVERSITY

Addis Ababa Institute of Technology

School of Civil and Environmental Engineering

Investigation of Hydroelectric Generation Options in Addis Ababa city municipal Wastewater Treatment Plants

A thesis Submitted to Addis Ababa Institute of Technology, School of Graduate Studies, Addis Ababa University In partial Fulfillment of the Requirement for the Degree of Master of Science in Civil and Environmental engineering

Major in Hydraulics Engineering

By

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Declaration

I, the undersigned, declare that the thesis is my original work, & has not been presented for a degree at any university and that all sources of materials used for the thesis have been duly acknowledged.

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This is to certify that the thesis prepared by Akelile Tsigeberhan, entitled: Investigation of Hydroelectric Generation Options in Addis Ababa city municipal Wastewater Treatment Plants and submitted in partial fulfillment of the requirements for the degree of Master of Science (Hydraulic Engineering) compiles with the regulations of the University and meets the accepted standards with respect to originality and quality.

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Abstract

Energy recovery through local hydropower generation is a way of coping with high electricity expenditures in wastewater systems (WWS). There are some hydropower units operating in wastewater treatment plants, showing that there is an interest for this type of small-hydro, but there is still a lack of awareness of its potential. This study assess the hydroelectric potential in the existing wastewater treatment plants in Addis Ababa city, by introducing energy recovery turbines (pump as a turbine) on the downstream side of wastewater treatment plants.

In wastewater treatment plants especially the electro-mechanical equipment consumes a considerable amount of energy in the treatment process. Specially, the newly constructed WWTP in Addis Ababa are totally dependent on electricity to function. Hence, by introducing a wide range of “pump as turbine” configurations that can offer cost-effective means of generating energy decentrally at heads from 2.5 to 300 meters and rates of flow ranging from 10 to 5000 liters per second, it is possible to recover the wasted potential.

In this study, industrial centrifugal pump running as turbine were analyzed using Ansys cfx computational fluid dynamics (CFD) software. The pump was simulated in direct and reverse modes and complete characteristic curves of the pump in direct and reverse modes were obtained.

The model was built up, starting from the real geometry, with a commercial tridimensional code, which is particularly suitable to simulate pumps. The numerical model was simulated. First, the model results were compared with the data declared by the pump manufacturer and the simulation model showed good accuracy. Then after using the validated model, the reverse mode was analyzed. The boundary conditions were changed in order to running the pump as a turbine. The results that were obtained from the simulation of pump in turbine mode is compared with the result obtained from Chapallaz design method. The results showed good accuracy. During the analysis the selected site has power production capacity between 4.1-147 kW depending on the site conditions.

Finally a relationship between flow, head and power was developed and sites with flow rate of above 40,000 m^3/day and head above 25m will be economically feasible sites for hydropower generation in wastewater treatment plants.

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Abbreviations

Water supply systems (WSS)

Wastewater systems (WWS)

Wastewater treatment plants (WWTP)

Energy recovery turbine (ERT)

Greenhouse gases (GHG)

Municipal waste water (MWW)

Pump as turbine (PAT)

Million gallon per day (MGD)

Ultra-Low Head turbine (ULH)

Micro Hydro Project (MHP)

Pico Hydro Project (PHP)

Revolution per minute (rpm)

Giga watt hour (GWh)

Best efficiency point (BEP)

Computational fluid dynamics (CFD)

High density polyethylene (HDPE)

Unplasticized Polyvinyl Chloride (UPVC)

Energy generating equipment (EGE)

Civil works (CW)

Downstream treated effluent micro-hydropower plant (DTE-HP)

Ethiopian Energy and Power Cooperation (EEPCO)

Total Watt Hour (TWH)

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1. Introduction

Society’s energy consumption worldwide has increased by up to 600% over the last century. This increase has been a direct result of population growth since the industrial revolution, in which energy has been provided mainly by fossil fuels. Nevertheless, today and in the near future, renewable energies are expected to be more widely implemented to help maintain sustainable growth and quality of life. Special attention must be paid to those new strategies that are related to energy recovery. These new techniques have raised interesting environmental and economic advantages. (Modesto et al. 2017)

Within urban infrastructure both water supply systems (WSS) and wastewater systems (WWS) are net consumers of electricity, in particular for pumping and treatment (also monitoring and operation). Their respective business cases are however often quite different, water being normally sold, whereas wastewater collection and treatment is less commonly paid for and therefore is often treated as a liability by the public administration. The treatment of wastewater in highly energy consuming. (Cecile et al. 2017)

The reduction of electricity consumption in WSS and WWS has been matter of growing interest in the last decades (Focus on Energy 2016) Generally, in recent years, an effort has been made to increase the efficiency of the main electricity-consumers, mainly by changing the treatment technology and only exceptionally by adapting local processes for electricity production (e.g. heat to electricity, biomass to electricity). Recent developments have shown that it is possible to attenuate the energy consumption by generating electricity within wastewater treatment plants (WWTPs), which can be consumed on site or exported to the grid. (Pakenas 1995). Local energy production (also called energy recovery) can be achieved with methane production from sludge digesters, sludge incineration and through hydropower, whilst keeping the performance of the main WWTP system unaffected.

Addressing the problems of energy sustainability in the water and wastewater industry thus requires a thorough review, and research into technologies that are cost effective and sustainable for each location. The industry thus need to harness renewable and non-polluting resources that are at its door step. (Theophilus 2010)

Hydropower generation in WWTPs presents several advantages. In general it is environmentally friendly, since it is a renewable energy source which produces no greenhouse gas emissions and offers synergies with existing infrastructure that reduce the civil works. Furthermore it does not require a water diversion dam or flooding of land by a reservoir (Saket 2008). More specifically hydropower generation can be adjusted to the load curve of the wastewater inflows and the WWTP energy consumption profile. High and low flows correspond to high and low energy usage in the treatment works and thus the devices will produce power to match the peak demand. Power can be produced locally at the same time that power is needed to supply the treatment plant ("local power to local consumer"), regardless of the cost or sell prices of electricity and without the need for transportation and step up/down voltage transformation.

Therefore, the development of renewable energy has a promising future, if the potential exploitation is considered. This promising development has positive aspects (e.g., lower environmental impact and generation of stable electrical supply) compared to other renewable energies (e.g., intermittent generation, such as solar or wind). In addition, this type of energy generation can be very important in the development of multipurpose water systems, where generation is another possible water use.

From an economical point of view, small hydropower schemes in WWS may benefit from policies favorable to renewable energies. There is a generalized acceptance of the importance of promoting renewable energy sources to ensure environmental protection and sustainable development (Romero 2012)

However, not all wastewater treatment works can take full advantage of this exciting technology because the feasibility of hydropower for any site depends principally on the available net head and the flow regime of the treatment works to generate enough energy to be worth the investment. To optimize a hydropower scheme, either the head or the flow must be significant. Treatment facilities located within hilly terrain, mountains and valleys are likely to produce a significant amount of power at a relatively low cost and with a shorter payback time. This thus gives an advantage to some treatment works due to their geographical location. Kirk, (1999, citied in Gaius-obaseki 2010, pp 207)

In Addis Ababa, there are a number of wastewater treatment plants and all of them release the treated water to a river or a drainage system. Because the potential is not well thought out and given a proper attention from the government, we are losing a huge amount of renewable energy that could be easily harnessed with little resources and effort from these facilities.

The implementation of hydroelectric power in wastewater plants is primarily in the developmental stages. Few plants worldwide have implemented this technology to date, but the potential exists for some facilities to do so. There are 14 municipally owned wastewater plants located In Addis Ababa city. Namely (Bole Arabsa 1A, 2A and 2B, Tulu Dimitu 1, 2, and 3, Oromia, Mekanisa, Kilinto, Bole Bulubula, karakore, degent, chefy and kality). The WWTPs average treatment capacities differ greatly from 75,000 m3/day for Kality site to 400 m3/day for Degent site. The water flowing through these treatment plants provides a potential source of energy that may be reclaimed and transformed into electricity, depending on the site conditions. Usually after treatment, the effluent will be disposed to a nearby river or stream near the site. But a nowadays by introducing and using Energy recovery hydro turbines and pumps as turbines, we can generate power from the effluent flows. Also, the site conditions and flow amount and characteristics should also be considered when assessing this kind of projects. The electricity generated from the system may then be used to contribute to the demand of the treatment plant either through net metering or by selling the electricity back to the supplier.

1.1 Statement of the Problem

Almost all wastewater treatment plants that are found in Addis Ababa city discharged there effluent to nearby drainage system or river without considering or exploiting the potential hydropower that can be generated from these infrastructures.

Because the potential is not well thought out and given proper attention from the government, the country is losing a potential source of renewable energy that could be harnessed with little resources and effort from these facilities. Therefore, this study try to assess the potential hydropower energy that could be generated and reclaimed from these wastewater treatment plants.

1.2 Objectives

General objective:

- To assess the hydroelectric power generation options from the existing wastewater treatment plants in Addis Ababa city.

Specific objectives:

- To identify potential wastewater treatment plant sites for power generation.
- To develop an election procedure for potential wastewater treatment plant site in order for power generation
- To estimate the potential power that can be generated from the potential sites.

1.3 Research Question

- Is it possible to generate hydroelectric power from wastewater treatment plant?
- What are the potential location for hydropower generation in wastewater treatment plants?
- What are the criteria's used in order to select a wastewater treatment sites for hydropower generation?
- How much power can be generated from the influent and effluent of wastewater treatment plants?

1.4 Scope of the study

- To undertake hydropower potential assessment from existing wastewater treatment plants that are found in Addis Ababa city.
- To develop an evaluation method to assess the viability of hydropower energy recovery at any particular WWTP.

1.5 Limitation of the study

The following limitations apply to this study:

- For flow calculation, the average daily treatment capacities of the WWTP were taken. Currently all wastewater treatment sites has not reached their full treatment capacities.
- For head calculations, Due to small elevation difference between the sewer and the treatment plants, only the downstream power generation option were considered.
- The simulated results were not verified with actual experimental investigations.

1.6 Significance of the Study

These study try to show, the hydroelectric power potential of the water passing through wastewater treatment plants. By installing energy recovery pump as turbines, it’s possible to reclaim the wasted hydropower potential of the inflow and outflow of water. In the near future, part of the growth of hydropower production must come from the retrieval of potential energy embedded in water distribution networks. Considering the potential reduction of natural resources due to extensive agriculture or unsustainable water use on a global level, any investment in energy water recovery is crucial. Therefore, the whole water cycle must be included in the process of energy recovery, including both drinking and treatment systems. This coupled water-energy nexus will allow the consideration of these systems as a new sustainable and efficient source of energy.

1.7 Organization of the thesis

The content of the thesis is outlined briefly in this section as follows

Chapter 1: Deals with general introduction, statement of the problem, research objectives and research questions of the thesis and states the whole over of the thesis

Chapter 2: Deals about literature review and gives detail information about potential locations for low head hydropower, hydropower generation from wastewater treatment plants, Africa and Ethiopia experience on power generation from wastewater and WWTP, hydropower generation location in wwtp, energy self-sufficient wwtps, micro hydraulic turbines, pump-as-turbine (pat) and criteria for selection of pump running in turbine mode

Chapter 3: deals about the general descriptions of the study area, and the research methodology that is the research methods, the research methods used, and the data collection, analysis and interpretations.

Chapter 4: Elaborates about the result and discussions of the thesis in light of the specific research question raised and research objectives

Finally, **Chapter 5** deals about the conclusion and recommendation resulted from the study work and suggest topics for further research.

2. Literature Review

2.1 Ethiopian Energy Policy

According to EREDPC (2002), Ethiopia's Energy consumption is predominantly based on biomass energy sources. Ethiopia is committed to shaping its economic future. The government has adopted a strategy for sustainable economic development, which places agriculture as its driving force. This strategy is referred as Agricultural Development Led Industrialization (ADLI) Strategy. It envisages the structural transformation of the Ethiopian economy through exportled growth, which feeds into an interdependent agricultural and industrial development.

Though Ethiopia is endowed with vast energy resources (Hydro, solar, wind, geothermal, biomass, natural gas, etc.) it has not been able to develop, transform and utilize these resources for optimal economic development.

2.1.1 Rationale for the policy

Energy is critical for economic development. Its importance stems from the fact that energy is basic input in all productive activities, including the household sector.

- To develop and utilize the country's energy resources on the basis of Ethiopia's overall development strategy;
- To assist other economic sectors to meet their development objectives by putting in place a clearly defined energy policy;
- To save scare foreign exchange resources and to ensure that energy is efficiently utilized;
- To ensure reliable and secure energy supplies to cushion the economy from external and internal disruptions of supply as well as price fluctuations;
- To ensure that development of energy resources is benign to the environment;
- To formulate comprehensive energy prices and to ensure economic profitability;
- To ascertain what energy technologies and equipment are appropriate for and compatible with the country's economic development needs; and
- To raise the efficiency of the energy sector and develop the necessary institutional and man power capabilities to undertake energy development programs.

2.1.2 Priority of the Policy

The Government of Ethiopia’s energy sector policy priorities are:

- To place high priority on hydropower resource development, as hydrological resources are Ethiopia’s most abundant and sustainable energy forms;
- To take appropriate policy measures to achieve a gradual transition from traditional energy fuels to modern fuels;
- To pay due and close attention to ecological and environmental issues during the development of energy projects;
- To set issues and publicize standards and codes which will ensure that energy is used efficiently;
- To develop human resources and establish competent energy institutions; and
- To provide the private sector with necessary support and incentive to participate in the development of the country’s energy resources.

2.2 Low Head Hydropower

“Low head” hydropower schemes can’t be exactly defined worldwide. In general, describes plants which head is less than 15m. A low- head hydro scheme requires a large passage/opening to accommodate a high volume of flow, making low head turbines inevitably large in size & expensive consequently creating a number of engineering challenges. In addition, low head schemes suffer from a lot of flow fluctuation due to the variation in head water & tail water levels. Singal et al, (2008, citied in Gaius-obaseki 2010, pp. 55)

Low head sites tend to be more common in runoff river hydro schemes. But can also be available in wastewater treatment outfalls. For low head sites, the conventional procedure is to select and install reaction turbines such as a Kaplan turbine, either as a vertical axis unit, a right-angle drive or a bulb turbine or an open flume Francis turbine with an adjustable vane, Although these turbines are technically efficient, they are prohibitively expensive for smaller schemes. (Furukawa et al, 2009)

Although the past couple of centuries have seen large scale development in high head turbines, this is not the case for low head turbines. The two main reasons for this are that conventional turbines are expensive and hence the cost of installing a low head turbine is often comparable with a large scale hydro scheme Kirk, (1999, citied in Gaius-obaseki 2010, pp 207) and also low head turbines are thought to have negative effects on the ecology of the river. However, in recent years, there has been a shift in research towards developing low head hydropower. This is because the majority of the available hydropower resources are low head. (Gaius-obaseki 2010)

One of the identified factors militating against the uptake of micro hydropower application within the industry is, a lack of reliable technology options to harness the low heads that exist within treatment plants. The majority of wastewater treatment plants were built on a flat landscape to take advantage of the gravitational flow of sewage into the treatment works. The low head difference therefore makes high head hydropower application technically unfeasible. The option is thus a compact ultralow head turbine, which can be located before or after treatment plant. Such a turbine should be able to utilize the existing civil structures and be environmentally friendly, cost-effective, durable, a high load factor installation and efficient as a conventional turbine. (Gaius-obaseki 2010)

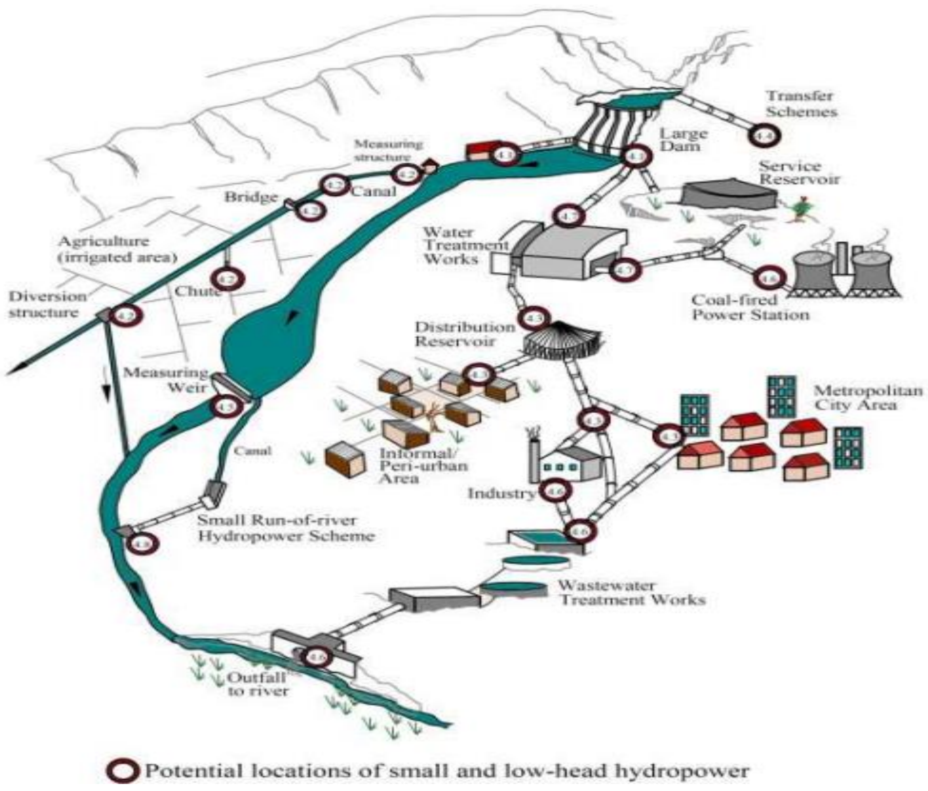
2.3 Potential Locations for Low Head Hydropower

The physical characteristics of the various types of hydraulic structures, together with a net available head and water flow data, will primarily dictate the choice of suitable turbine/generator unit(s) for installation. The figure below illustrates various possible locations that could have low-head hydropower potential. (I. Loots et al. 2015) studied a review of available low head hydropower technologies. Study was carried out in South African region. They identified the sites where technologies can be implemented which are grouped as follow:

- Irrigation systems (canals and conduits) and
- urban areas (industrial and urban discharge, storm water systems and water distribution systems).

There were some important parameters which they consider when planning a low head hydropower plant like reliability of flow, nearby utilization of electricity, environmental factors and water quality. They also mentioned that because of hydropower development, primary function of the infrastructure and its function should not be compromised.

Figure 1: Typical low head hydropower locations



Source Adapted from Loots et al. (c.2015, p. 20)

These new technologies can be applied to existing water systems, with the purpose of producing energy (Ramos et al., 2013), irrigation canals and wastewater treatment plants are one of the possible location for power generation..

2.3.1 Hydropower Generation on Irrigation canals and networks

Micro hydropower can use different hydraulic heads or diversion schemes in small dams in rivers and ravines, in open irrigation channels and in drainage systems. Similarly, in open channel irrigation systems, energy recovery can also be implemented by installed turbines in small dams or irrigation. An example of these installations is the analysis made by (Butera and Balestra, 2015), who determined the potential generation by hydropower plants for the Piedmont region (Italy). This region has an installed capacity of 46 MW, of which 45% is pico hydropower, 49% is micro and 6% is small, with an average hydraulic potential of 1.5–2 kW/ha. (Tarragó, 2015) developed a preliminary study in the Alqueva's irrigation system, where twenty-two hydrostatic pressure machines were studied in different locations with hydraulic heads below three meters. Using this assumption, the theoretical energy recovery reached 406.64 MWh/year in 67,932 ha of this region.

Preliminary values of recoverable energy have been obtained using average circulating flows of the irrigation system (Tarragó, 2015). These show the importance to analyze these networks in terms of recovery energy. By determining variability of flows and pressure in any point or line on the irrigation network has advantage (determining instant values of flows and pressure) allows performing the analysis of energy recovery in any point on the network.

2.3.2 Hydropower potential from Irrigation networks in Ethiopia

According to (Awulachew et .al 2015) Ethiopia has vast cultivable land (30 to 70 million hectares(Mha), but only about a third of that is currently cultivated (approximately 15 million hectares(Mha), with current irrigation schemes covering about 640,000 ha across the country and total irrigable land potential in Ethiopia is 5.3 million hectares assuming use of existing technologies, including 1.6 million hectares through rainwater harvesting and ground water. This means that there are potential opportunities to vastly increase the amount of irrigated land.

The Ministry of Water and Energy has identified 560 irrigation potential sites on the major river basins. The total irrigable land in Ethiopia is estimated to be around 3.7 million hectares (without considering the groundwater potential and gently sloping areas). The area under irrigation development to-date is estimated to range between 160,000-200,000 hectares for the entire country (Awulachew et. al 2015)

Table 1: Review of Multipurpose projects in Ethiopia

No.	Projects	Function
1	Fincha Neshe (FAN) Hydro Power Plant	Hydropower generation and irrigation
2	Geba Hydro Power Plant	Hydropower generation and irrigation
3	Genale Dawa IV Hydro Power Plant	Hydropower generation and irrigation
4	Tana Beles Hydro Power Plant	Hydropower generation and irrigation
5	Koka Hydro Power Plant	Hydropower generation and irrigation

The use of water networks built for irrigation can be used for energy development. The advantage of using existing networks is that the initial cost is lower compared to other configurations. In the case of irrigation or drinking water networks, the pressure caused by the strong slope between the reservoir and the consumers, has to be wasted in the surge tank. Instead of reducing the pressure it is often. Technically and financially possible to use a small pelton turbine which uses this pressure.

Regarding the theoretical recoverable energy in a network, it mainly depends on the orography of the irrigation area. The networks with larger gradients between the supply and the consumption points have greater possibility to recover energy, if the appropriate machine is selected. According to (Modesto et al 2016) the energy recovery can be analyzed in different parts of the network:

i) **In plot of cultivation**—in this case, the private user needs to reduce pressure down to 30 m w.c. to carry out drip irrigation. Generally, the user installs a pressure reducer to dissipate the excess energy. This element can be replaced by a pico-turbine to generate energy for self-consumption. This energy can be used in remote-control system, cleaning of filters, lighting and others similar consumptions.

ii) **In the hydrant pipe**—when the hydrant supplies to flat topography, reduction of pressure can be done. In an operating network, this reduction is carried out with a pressure reducing valve. This recovery could potentially be done if a suitable turbine could be installed.

iii) **In pipe branch**—in networks with large extension and irregular orography, some parts of the network can achieve higher pressure than necessary, forcing pressure to be reduced on a pipe branch. Currently, this reduction is possible by using a reducing valve installed on this branch. These valves can be replaced by turbines or pumps as turbines (PAT) depending on the system characteristics to increase the energy efficiency of the network.

2.4 Advantages and disadvantages of power production in irrigation canals

2.4.1 Pros for Irrigation Canals:

- **Efficient & Reliable -** When you compare the hydro energy with other renewable energies, it won’t be hard to notice that hydro is one of the few that produces a continuous supply of electrical power. Plus, one of the main advantages regarding the hydro power is the fact that it is more needed during the winter months and this is when exactly large quantities of electricity produced.
- **Cost Effective -** The reality is that the difference between the expenses of building a larger dam and a smaller dam is already huge. However, when you consider the costs of building a mini-hydro power plant in irrigation canals, this is, by far, the most affordable solution.Depending on the location as well as on site electricity requirements, a small mini-hydro power plant can cost between €80 000 and €400 000.

2.4.2 Cons for Irrigation Canals:

- **Requires Specific Site Characteristics -** To take full advantage of the mini-hydro power plant in irrigation canals, you need to make sure that you have the right site. Some of the factors that you will need to take into account include the stream size (including the drop, output, and flow rate), the distance between the mini-hydro and the location of the canals, and the balance of the system components, pipelines, transmission lines, controllers, batteries, and inverter

- **Expansion Is Not Possible -** When you can find the perfect site for having a mini-hydro power, the truth is that, in most occasions, the flow and size of the streams will not allow the possibility of expanding it in the case more electricity is needed.
- **Summer Months May Be A Concern -** In many locations, the stream size varies depending on the season. Usually, summer months tend to present less water flow which results in less energy produced. This is why it is important to know if all the energy requirements are going to be met during the entire year during the planning and research stages

2.5 Hydropower Generation From Wastewater Treatment Plants

Hydropower production in WWS exists in several countries, mainly in conveyance systems, but innovative solutions are being developed as well to install micro-hydropower units inside the distribution networks themselves. Regarding WWS, however, the number & typology of pilot installations is very limited, due to nature of the resource wastewater and the quantity of suspended materials, conveyance system (often free surface flow), available head (generally below 100 m) and discharge stochasticity. Given the widespread distribution of WWS and WWTPs & the weight of their electricity consumption on the operation costs, integration of small-size hydropower units may be viable anywhere in the world even in locations where traditionally hydropower has had little interest. (Cecile et al. 2017)

Some cases of hydropower plants installed in wastewater systems around the world are presented in Table below.

Table 2: Inventory of hydropower plants in wastewater systems (countries with available data)

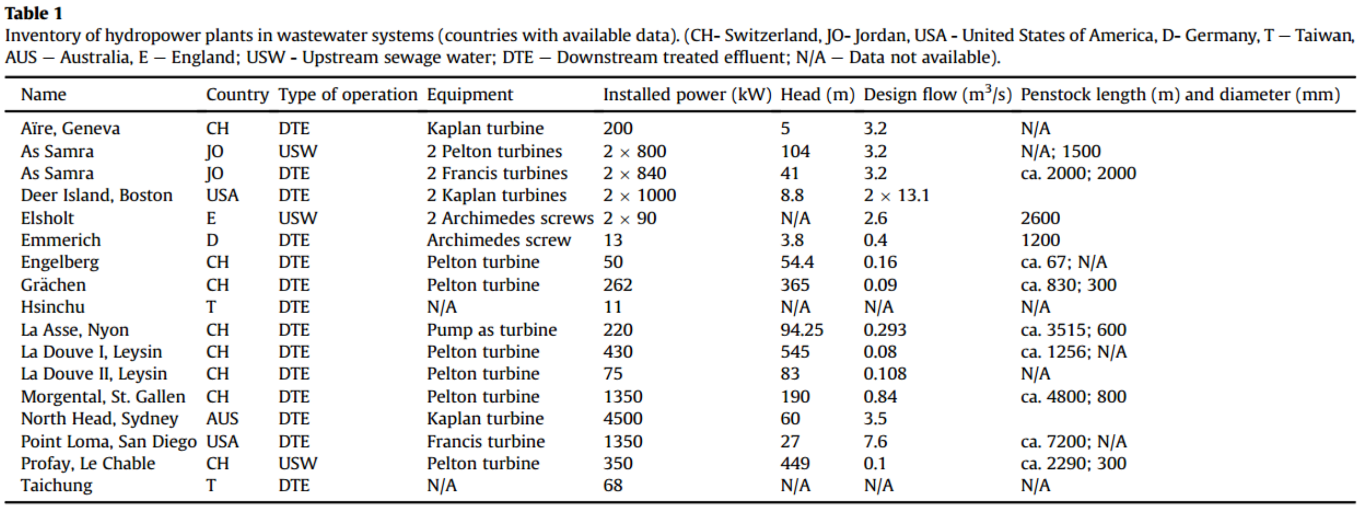


Table 1
Inventory of hydropower plants in wastewater systems (countries with available data). (CH- Switzerland, JO- Jordan, USA - United States of America, D- Germany, T – Taiwan, AUS – Australia, E – England; USW - Upstream sewage water; DTE – Downstream treated effluent; N/A – Data not available).

Name	Country	Type of operation	Equipment	Installed power (kW)	Head (m)	Design flow (m^3/s)	Penstock length (m) and diameter (mm)
Aïre, Geneva	CH	DTE	Kaplan turbine	200	5	3.2	N/A
As Samra	JO	USW	2 Pelton turbines	2 × 800	104	3.2	N/A; 1500
As Samra	JO	DTE	2 Francis turbines	2 × 840	41	3.2	ca. 2000; 2000
Deer Island, Boston	USA	DTE	2 Kaplan turbines	2 × 1000	8.8	2 × 13.1	
Elsholt	E	USW	2 Archimedes screws	2 × 90	N/A	2.6	2600
Emmerich	D	DTE	Archimedes screw	13	3.8	0.4	1200
Engelberg	CH	DTE	Pelton turbine	50	54.4	0.16	ca. 67; N/A
Grächen	CH	DTE	Pelton turbine	262	365	0.09	ca. 830; 300
Hsinchu	T	DTE	N/A	11	N/A	N/A	N/A
La Asse, Nyon	CH	DTE	Pump as turbine	220	94.25	0.293	ca. 3515; 600
La Douve I, Leysin	CH	DTE	Pelton turbine	430	545	0.08	ca. 1256; N/A
La Douve II, Leysin	CH	DTE	Pelton turbine	75	83	0.108	N/A
Morgental, St. Gallen	CH	DTE	Pelton turbine	1350	190	0.84	ca. 4800; 800
North Head, Sydney	AUS	DTE	Kaplan turbine	4500	60	3.5	
Point Loma, San Diego	USA	DTE	Francis turbine	1350	27	7.6	ca. 7200; N/A
Profay, Le Chable	CH	USW	Pelton turbine	350	449	0.1	ca. 2290; 300
Taichung	T	DTE	N/A	68	N/A	N/A	N/A

Source: Cecile et al. (2017)

Micro-hydropower system using waste water from community neither requires a large dam nor is land flooded. Only waste water from different parts of the city is collected to generate power which has minimum environmental impact. (Saket & Varshney 2007). Micro hydroelectric power generation system based on municipal waste water (MWW) is highly fluctuating in nature. But, it is possible to match the capacity factor of a turbine with the flow profile and energy consumption at a sewage treatment works. High and low flows equate to high and low energy usage in the treatment works and thus the turbine will produce power to meet the base and peak flows. This can be achieved by selecting a turbine capable of working under variable flow conditions. Most turbines can work at a certain percentage below and above their rated flow within certain efficiency; Notable among them is the impulse turbine. A variable flow turbine will ensure a constant supply of electricity all year round. Flows into a sewage works are diurnal; A turbine designed for a high flow will remain idle during dry weather periods and a turbine designed for dry weather flow will run throughout the year. If a system depends fully on hydropower for power generation, the design flow should be flow that is available 95% of the time. (Harvey 1993)

Turbines and energy recovery machines could be incorporated into wastewater treatments whether in open channel flow or in pressurized flow. However, a proper study of the hydraulics would be required to ensure that the turbine installed does not cause a network blockage, which could lead to consent failure, so protection mechanism should be implemented before generating power. A reaction turbine or, alternatively, Pump as turbine (PAT) or vortex turbines can be used to extract the power from the flow of water. (Gaius-obaseki 2010)

2.5.1 Africa experience on power generation from wastewater and WWTP

Africa is the most underdeveloped continent with regard to hydropower generation with only 6% of the estimated potential exploited. This should not be seen as a burden, but rather as an opportunity. (Jonker, 2011)

There are many untapped hydropower potential in Africa. Using new technologies, now it is possible to develop previously unfeasible sites. One of these potential sources for hydropower development is from the inflow and outflow from wastewater treatment plants. These potential has not been considered in any of the African countries. Because the potential is not well thought out and given proper attention from the government, the content is losing a potential source of renewable energy that could be harnessed with little resources and effort from these facilities.

But when we see the biogas production from WWTPs, there are few African counties that exploit these energy source. South Africa, Ghana and Kanye have commercial Biogas digesters facilities. Biogas (a methane-rich natural gas) derived from anaerobic digestion and captured at WWTW plants provides a renewable energy source which can be used for electricity, heat and biofuel production.

WWTW use a lot of electricity for pumping and aeration. International reference indicates that around 60% of the electricity consumption of the WWTW can be offset by the electricity generated at the biogas plant. In general, biogas projects will require a long-term investment of 7 – 10 years or more. The initial indication is that larger WWTWs with an inflow in excess of 15ML/day show financial viability based on the amount of electricity they could generate. As emphasized, site-specific modelling would be required to confirm any estimation, given the variability of characteristics of each plant.

With electricity price increases set to triple Johannesburg Water’s electricity bill from R100 million to over R300 million over the next ten years they identified the need to cut back on electricity usage. Northern Works treats about 43 ML of sewage/day. It is the City of Johannesburg’s largest wastewater treatment works and the site of its first biogas to energy project.

The plant produces electricity from biogas using three 376kWe (KWh equivalent = heat and power) combined heat and power (CHP) gas engines. The electricity produced is consumed on-site. Currently it produces 10% of the treatment work’s power requirement. However, once all of the digesters have been refurbished, and all of the sludge is treated anaerobically, the CHP plant should produce some 56% of the on-site power requirements

2.5.2 Ethiopia experience on power generation from wastewater and WWTP

Before a decade in Ethiopia there were few WWTP that were operational which were limited to the major cities. The treatment plants capacity differ from 24000 m^3/day in Addis Ababa to 24m^3/day in Dire Dawa. (Federal Democratic Republic of Ethiopia, 2017)

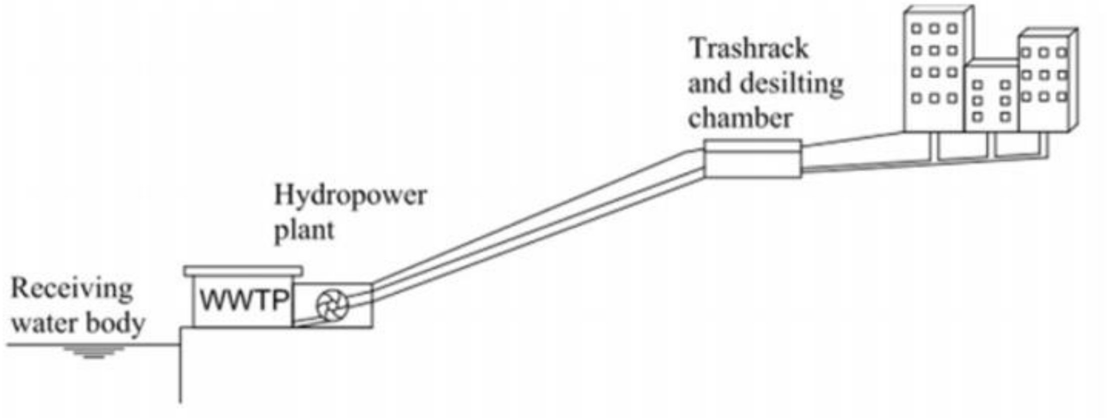
All of the WWTP found in Ethiopia discharge there treated water to nearby river or a drainage system without considering the potential for hydropower generation. Proper attention should be given for these facilities as they can support the energy demand from these facilities. Saving energy through energy efficiency improvements can cost less than generating, transmitting, and distributing energy from power plants, and provides multiple economic and environmental benefits. Energy savings can reduce operating costs for local governments, freeing up resources for additional investments in energy efficiency and other priorities. (Theophilus 2010)

Kality WWTP has a potential for biogas production. The treatment plant have a potential to produce 12000 m^3/d. The average hourly potential production capacity is 520m^3/h. Taking in to consideration 520m^3/h * 6.5KWH/nm^3 (low calorific value of biogas per normal m^3) * 0.38(electric efficiency of CHP units) electric power in the range of 1.3MW can be produced.

2.6 Hydropower Generation Location in WWTP

According to (Cecile et al. 2017) there are two possibilities to generate electricity from wastewaters. The first one is before the WWTP (Choulot et al. 2012; Griffin, 2000) and the second one is after the WWTP. Hydropower generation is particularly challenging in the untreated wastewater network, as the high solids content of raw sewage can create problems for traditional turbines, such as increased wear and tear and clogging of the rotating parts. In such case, the wastewater network of a built-up area will lead to a forebay equipped with a thin trash rack and a rack cleaner. Such micro-hydraulic turbines are frequently blocked with foreign matter such as fallen leaves, twigs, and refuse, and they occasionally lose their function. A filter installed upstream of the micro-hydraulic turbine can remove the foreign matter. However, such equipment increases the operation cost of micro hydraulic turbines. Suspended solids could create wear on the turbine blades, so the blade should be constructed from stronger material that isn’t damaged easily. Also the whole turbine should be explosion proof because of the nature of the sewage and different gases in the system. The wastewater is then led through a penstock to the WWTP, situated at a lower elevation, where it passes through the turbine before being treated through the usual process. The turbine has to be set as close as possible to the elevation of the treatment basin to maximize the head.

Figure 2 : Upstream sewage water mini-hydropower plant

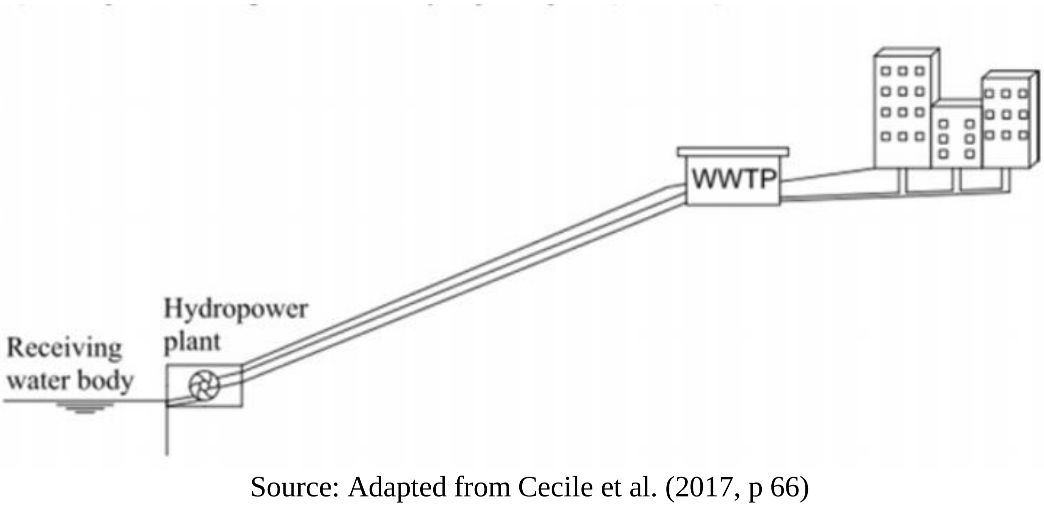


Source: Adapted from Cecile et al. (2017, p 66)

The second possibility is after the WWTP (McNabola, et al. 2014; Pakenas, 1995; Gaius-obaseki, 2010). In this case, the treated water that comes out of the WWTP is led down through a penstock to a turbine before being discharged to a water stream or a lake. To maximize the head, the turbine will then be close to this restitution. The effluent channel would be ideal location for a turbine because of the quality of the water.

For some sites, the hydropower project can lead to improving the cost efficiency of a longer penstock to reach a water stream where dilution can be more significant.

Figure 3: Downstream treated effluent mini hydropower plant



Source: Adapted from Cecile et al. (2017, p 66)

In upstream operation, the placement of the water tank will condition the head but also the amount of collected wastewater. On the other hand, downstream operations are influenced by the topographic elevation difference between the WWTP and the outlet.

To infer on the location of this point, a comparative analysis of two different criteria is proposed for each WWTP. The higher the collection point is placed, the higher the available head will be. However, the discharge rate at a higher collection point will be smaller than if the collection is done at a lower altitude, since only a smaller part of the population will be contributing which means the smaller power we generate if we select high elevation. So detail assessment is needed.If water is delivered directly from the treatment plant to the power house it has the advantage of being constant which means that a turbine installed will operate at its peak efficiency at all times, thus providing a consistent electricity supply. Such a turbine should be able to utilize the existing civil structures and be environmentally friendly, cost-effective, durable, a high load factor installation and efficient. There are a few different options for low head turbines that can be retrofitted or installed into water and wastewater facilities. At present, there are few water turbine that can be used based on research studies, manufacturer literature and case histories. Some of these turbines are similar and operate with the same concept as conventional turbines but are modified to take many economic and engineering factors into consideration. (Cecile et al. 2017)

2.6.1 Energy self-sufficient WWTPs

For decades, energy efficiency optimization has been carried out in order to work towards energy self-sufficient WWTPs. Some examples of energy self-sufficient WWTPs have been constructed successfully. In the table below 12 WWTPs that have achieved >90% energy self-sufficiency.

Table 3: Energy self-sufficiency of full-scale WWTPs.

No	*Name of WWTPs*	*Location*	*Capacity (MGD)*	*Energy self-sufficiency (%)*
1	Grevesmuhlen	Germany	4	100
2	Wolfgangsee-Ischl	Austria	5	100
3	Strass im Zillerta	Austria	6	100
4	Gloversville–Johnstown Join	USA	11	100
5	Sheboygan Regional	USA	11	100
6	Gresham	USA	13	100
7	Prague Central	Czech	42	94

		Republic		
8	Zürich Werdhölzli	Switzerland	67	100
9	East Bay Municipal Utility District	USA	70	100
10	Point Loma	USA	175	100
11	Davyhulme	England	200	96
12	Joint Water Pollution Control Plan	USA	300	97

Source: Yifan Gu et al. (2016)

Micro turbines have a presence in large wastewater treatment plants. The wastewater treatment in Point Loma, San Diego, CA uses the flow of water through the facility to create electricity to sell back to the San Diego Gas and Electric Company. The project totaled $1.2 million, including turbine cost, piping cost, electrical cost, transmission lines, engineering time and other labor associated with the project. This was partially funded by the California Energy Commission and the State of California, totaling $780,000. This treatment facility generates 1.35 megawatts of renewable energy using hydroelectric turbines (United States Environmental Protection Agency, 2013). This application is an example of a large treatment plant integrating hydroelectric turbine technologies into an already existing system. (Marissa et al. 2014)

2.5.2 Micro hydraulic turbines

Micro hydropower’s such as lower head turbines are the most prospective renewable energy sources that have received considerable attention because of its potential to generate green energy 10 kW to 500 kW but the development of this kind of energy is very low due to the civil work cost. The literature research, in the areas of the hydropower sector, signifies that the installation of the project is dependent of the head and the discharge of the site. Usually high head refers to Pelton turbine installation: moderate head and moderate discharge for Francis turbine: low head and high discharge for Propeller and Kaplan turbine. Concern, now, is shifted on the utilization of unutilized sites i.e. low head and low discharge on which this project was focused. A new technology of Ultra-Low Head turbine (ULH), which can be installed even in irrigation canal, waste water treatment plant and other small man made canals, was employed that works in the principle of static pressure difference unlike conventional turbine. (Pradeep .P et al. 2016)

Micro water current turbines are most suited for places where there is an almost a constant flow of water, throughout the year. The under developed and developing countries can use this techniques to provide electricity, this scheme can be a real solution for power scarcity problems. (Tanbhir et al. 2016)

Micro-hydraulic turbines, which can be installed in small rivers and irrigation canals for power generation, are frequently blocked by foreign matter, such as fallen leaves, twigs, and refuse. As such blockage causes a reduction in turbine performance, the development of a micro-hydraulic turbine with excellent foreign matter passage performance is desired. (Uchiyama et al. 2016)

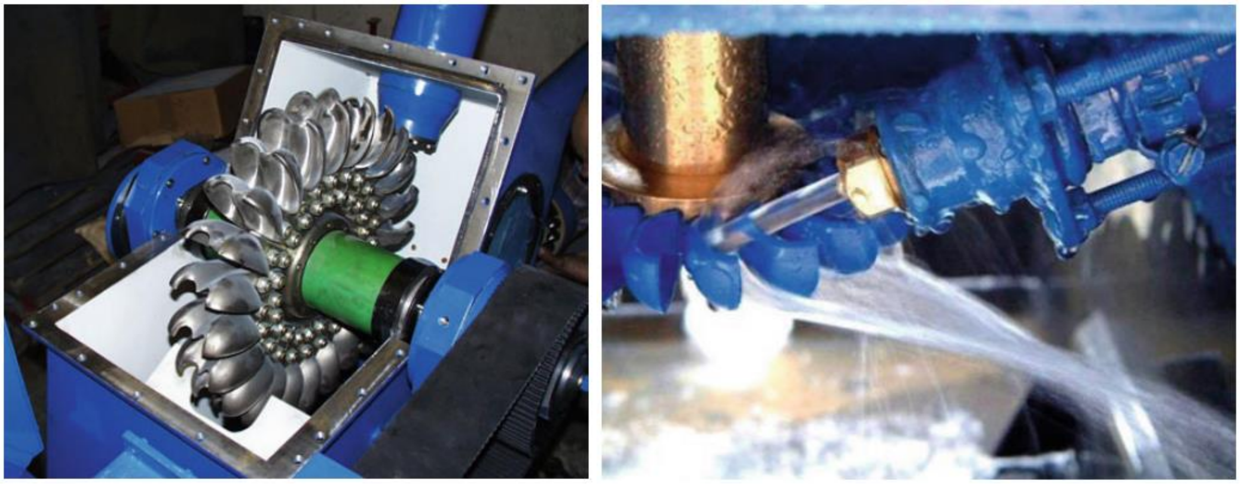
Micro hydro is perhaps the most mature of the modern small-scale decentralized energy supply technologies used in developing countries. Compact ultralow head turbine running at a medium speed, which can be located in the discharge pit, submerged or located prior to the exit flow from the treatment works. Such a turbine should be able to utilize the existing civil structures and be environmentally friendly, cost-effective, durable, a high load factor installation and efficient as a conventional turbine. There are a few different options for low head turbines that can be retrofitted or installed into water and wastewater facilities. At present, at least six types of water turbine can be used based on research studies, manufacturer literature and case histories. Some of these turbines are similar and operate with the same concept as conventional turbines but are modified to take many economic and engineering factors into consideration. (Theophilus 2010)

There are many turbine types available to convert hydraulic energy to mechanical energy. However, only a few can be applied in micro/pico hydroelectric power stations. Pelton/Turgo, cross flow and propeller turbines and pumps as turbines (PAT) are the types that have been widely used to produce energy in micro/pico hydroelectric power units. (Armando .C et al. 2018)

2.6.2.1 The Pelton/Turgo Turbine

The Pelton turbine is an impulse turbine, invented by Lester Allan Pelton (1829–1908), and is perfect for high head hydro plants. The high pressure flow enters the runner by means of a single/multi jet nozzle system. The turbine discharge can be regulated by a needle valve to adjust the output power. The runner is of a cylindrical shape consisting of several buckets, which have a special design to capture the energy from the jets of water. The jets strike the centers of the buckets and exit from the sides. The design and layout of a series of buckets is realized in order to avoid the possibility that the flow into one bucket influences the others. The recommended number of buckets is between 18 and 24. (R. Hothersal, 2004)

Figure 4: Pelton and Turgo turbine

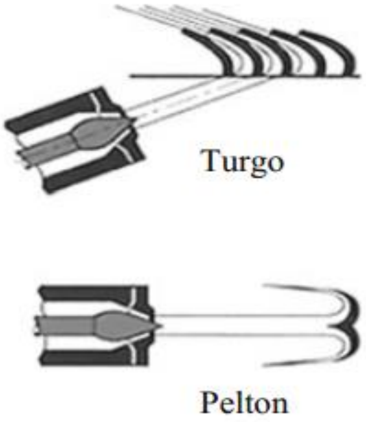


a b

Source: Adapted from Armando (2018, pp. 7-8)

A deflector is installed between the runner and the nozzle deviating the water jets to prevent them from entering the runner in emergency situations, such as the increase in the rotational speed in no-load conditions (Fig. shows a pico Pelton turbine). The Turgo turbine is very similar to the Pelton turbine. However, the jets of water enter the buckets at an angle of approximately 20°. Therefore, the Turgo turbine can be applied in power units with a lower head and higher flow capacity. The Figure b above shows a Turgo turbine and the figure below compares the inlet flow in a Turgo and a Pelton turbine. (Armando .C et al. 2018)

Figure 5: Comparing the inlet flow between a Turgo and a Pelton turbine

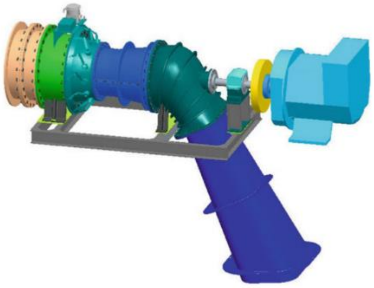


Source: Adapted from ESHA (2004, pp. 157-158)

2.5.2.2 The Propeller Turbine

The Propeller turbine is of a reaction turbine type and is useful for low heads and high flow rates. The turbine is installed in a pipe and the generator can be installed inside or outside. The draft tube can be conic or cylindrical (Fig. 1.7). In the absence of a spiral case the inlet can even be provided with a set of fixed /movable guide vanes. The manufacture of this turbine is easier than the Kaplan or Francis turbines. (Armando .C et al. 2018)

Figure 6: Propeller turbine



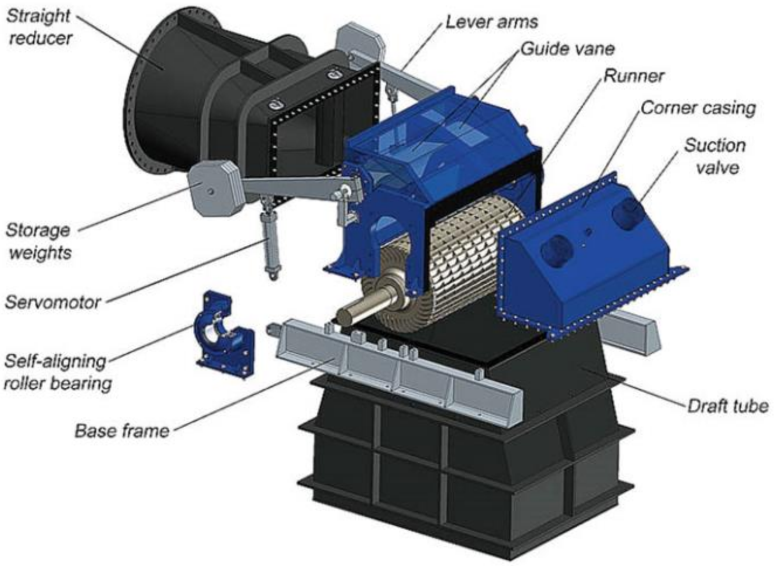
Source: Adapted from Armando (2018, pp. 9)

2.5.2.3 Cross Flow Turbine

Cross flow turbines are also called Banki, Mitchell or Ossberger turbine. A cross flow turbine comprises a drum shaped runner consisting of two parallel disc connected together near their firm by a series of curved blades. A cross flow turbine has its runner shaft horizontal to the ground in all cases.

The cross flow turbine is simpler in design and cheaper to manufacture than other types such as Pelton, Turgo, and Francis. Cross flow turbines are mostly used in remote power systems in developing countries, and have a typical efficiency in the range of 70–85%. Despite the efficiency being lower than other types, the cross flow turbine exhibits a flatter efficiency curve with varying runner angular velocity, ω, which can be an important advantage. (Ram Adhikari and David Wood 2018). The efficiency of a cross flow depends on the sophistication of its design. A feature such as vacuum enhancement is necessarily expensive as it requires the use of air seals around the runner shaft as it passes through the casing and an airtight casing. Sophisticated machines attains efficiencies as high as 85%. Part flow efficiency down to less than a quarter of full flow can be maintained at high values by the arrangement for flow partitioning illustrated in figure below. (Harvey 1993)

Figure 7: Cross flow turbine



Source: Adapted from Armando (2018, pp. 8)

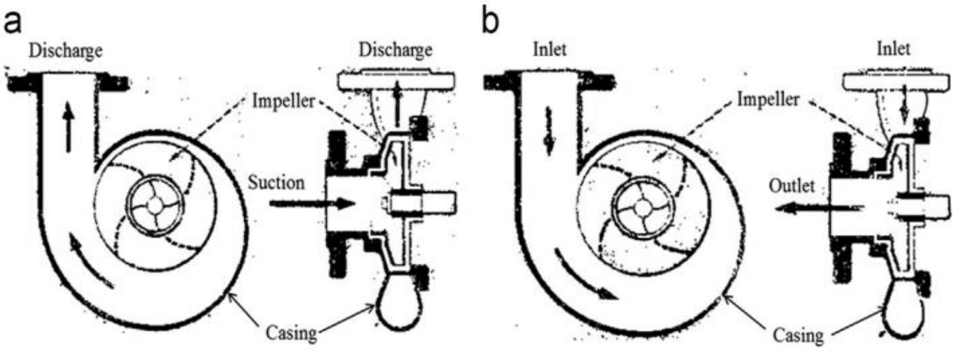
2.5.2.4 Pump-as-Turbine (PAT)

A pump used as a turbine (PAT) can deploy the hydropower potential extremely efficiently and economically with straightforward technical means. PAT are suitable for applications where pressure differences are to be reduced or where the head and flow rate of an installation can be exploited. The power generated can be used either for internal purposes or to fed into the public grid. Thanks to low investment costs, PAT solutions pay for themselves after a very short time.

Pumps are relatively simple and easy to maintain and they also have a competitive maximum efficiency when compared to conventional turbines. Perhaps the major benefit is that mass production of pumps means that they are comparatively much more cost-effective than conventional turbines. PAT systems are cost-efficient and widely available. In addition, they have simple design and easy maintenance compared to conventional turbines. (Jasmina et al. 2014)

According to (Chapallaz et al. 1992), PATs are practically applicable in the areas of power generation. Pump-turbines have been used for several decades with power ratings of several megawatts. Standard pumps are now more and more used in Micro Hydro Project/ Pico Hydro Project MHP/PHP schemes (5 to 500 kW). For pumped storage scheme, pump-turbines are specifically used to operate in both modes; pumping water into an elevated storage lake overnight at low tariff electricity and during the day, generating peak demand electricity through the same machine operating in turbine mode. The application of PAT for power generation requires the use of either a synchronous generator or induction generator (motor as generator) coupled directly to the rotating shaft of the PAT. For this reason, a nominal speed corresponding to one of the synchronous speeds of the generator (e.g. 750, 1000, 1500 or 3000) Revolution per minute (rpm) should be chosen.

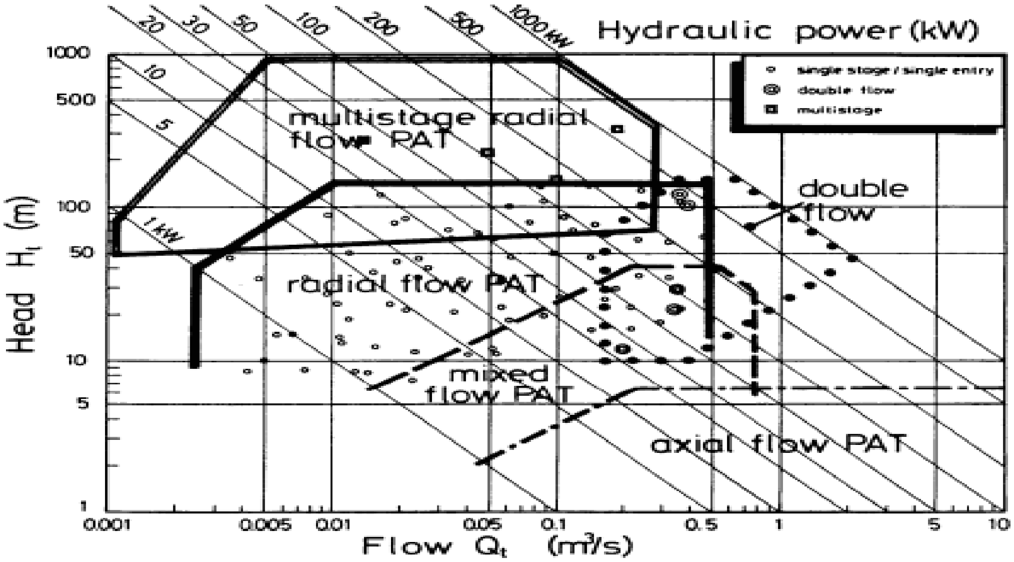
Figure 8: Centrifugal pump in (a) Pump and (b) Turbine modes



Source: Adapted from P. Vasanthakumar et al. (2016, p. 696)

As there are many different types of pumps that can be used as a turbine (Chapallaz et al. 1992) gives the rough guide in figure below to aid the choice Multistage pumps are only typically used in cases where the head is very high, and when the flow rate is high either multi-flow pumps or a system of single flow pumps in parallel is used.

Figure 9: Different pumps as turbine operation ranges



Source: Adapted from Chapallaz et al., (1992, p.15)

2.5.2.5 Field of PAT Application

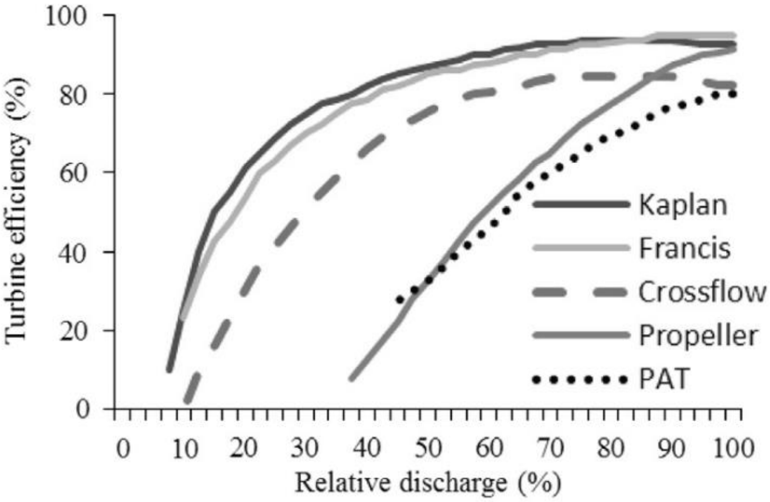
The identification of potential energy production points, the choice of the main design parameters and, finally, the selection of the best design solution are crucial aspects in micro hydro power (MHP) exploitation. (Armando .C et al. 2018)

PATs can be installed and operated in (natural streams, water supply systems, irrigation, waste water treatment and water drainage).

2.5.2.6 Disadvantages of PAT

Some of the limitations when using Pump as Turbine instead of conventional turbine in reference to (Chapallaz et al., 1992) are discussed below.

- **Lack of Hydraulic Control Device**: A control valve must be incorporated in the penstock line to start and stop the PAT. Hydraulic losses and cost of the installation will increase due to introduction of valve.
- **Peak Efficiency:** Efficiency of PAT is lower than sophisticated turbines of the medium to high output range. PATs reach efficiency not more than to locally manufactured Cross flow or Pelton turbines.
- **Lower Efficiency at Part Load:** If PATs are operated at other than the design flow (i.e. below their BEP), a relatively rapid drop of efficiency will occur



2.5.2.7 Historical Development of PAT

When (Thoma and Kittredge 1931) were trying to assess the complete characteristics of pumps, they accidentally realized that pumps could function very efficiently in the turbine mode. Later (R. Knapp 1941) published the absolute pump characteristics for a few pump designs based on experimental investigations. The further development of the technology of pumps as turbines, as well as its dissemination, has been inadvertently nurtured by the research on water hammer in pumping stations and on pump-turbine made possible by the indirect benefits derived from R&D on advanced technologies is a pattern that can be observed in many other areas. In 1950s and 1960s, the model of pump storage power plants, in the range of 50–100 MW, was evolved mostly in urbanized countries to satisfy the peak power load. In later years, chemical industries became an additional area for the application of PATs for energy recovery system. Even in water supply networks applications of this technology were found. This condition provides an effective phase of research and then beyond, standard manufactured pumps were calculated in turbine mode. In later years, many other techniques were developed by lots of researchers S. (Rawal and J. Kshirsagar 2007).

The expertise for the use of PAT for hydro power generation did not exist earlier. Though, advances in electrical equipment control technologies, rotary motion and torque have created the opportunity of the operation of pump rotating in reverse mode for power generation (J. Fernandez 2004). (A. Agostinelli and L. Shafer 2013) many pumps were tested in turbine mode over the years and accomplished that when a pump operates in a turbine mode, its mechanical operation is smooth and quiet; its peak effectiveness is same as in pump mode.

2.7 Energy Recovery in Wastewater Infrastructure

The flow of sewage effluent can also be directed through a penstock under pressure, through a MHP turbine to recover energy in wastewater treatment infrastructure (Gaius-obaseki, 2010). This can be carried out at treatment works outfalls or inflows; and the inlet to pumping station wet wells or in sewer mains where sufficient flow and pressure is available.

Investigators have reported on the feasibility of sewage-treatment outfalls for energy recovery using pumps as turbines (PATs). For example, a demonstration project built in 1993 in Switzerland used a PAT to produce up to 210 kW of electrical power from a sewage outfall (Williams, 1998). In India a demonstration energy recovery plant has been constructed on a sewage storage tank located on a University campus. Although the plant capacity was reported as just 190 W, the project was still deemed economically viable (Saket, 2008).

2.8 Criteria for Selection of Pump Running in Turbine Mode

Selection of a proper pump to be used as turbine is a big problem in installation of PAT for a particular site. PAT can become a cost-effective alternative to traditional turbines if the turbine mode curves can be determined (Teuteberg, 2010). Normally, the pump manufacturers do not provide characteristic curves of their pumps working as turbines. This makes it difficult to select an appropriate pump to run as a turbine for a specific operating condition. One of the main objectives of all PAT researchers all over the world has been to build a method that would make accurate predictions of the turbine operation of pumps.

A large number of theoretical and experimental studies have been found in the literature for the prediction of performance of PAT in which relations between best efficiency point (BEP)s in pump and turbine modes were derived based on either efficiency or specific speed in pump mode. The specific speed is one of the main parameters of turbo-machines that characterizes the type of the runner (i.e. radial, mixed or axial flow), blade shape, spiral casing and other design features. The BEP parameters in turbine mode are different from that in pump mode. Hence, the relation between these parameters is derived by the investigators in terms of head and discharge correction factors (h and q), which are defined as under:

$$h = H_t/H_p \qquad (1)$$

$$q = Q_t/Q_p \qquad (2)$$

The historical development in the performance prediction methods of PAT (up to late 90s) shows the relations for h and q derived by different researchers. The deviations between performance predicted by these methods and experimental results have been found to be around ±20% or even more (Chapallaz et al., 1992; Williams, 1995). Therefore, these methods are confined to preliminary selection of pumps to be used as turbines which is important to obtain a rough estimation of turbine mode characteristics from the pump mode characteristics (Nautiyal et al., 2011).

2.8.1 Selection of PAT

According to (S.Derakhshan and A Nourbakhsh 2008), the selection of pump for use in turbine mode has become a difficult task since pump manufacturers do not normally provide characteristic curves of their pumps working as turbines. This makes difficulty to select an appropriate pump to run as a turbine for a specific operating condition. While this draw backs holds for the use of PAT, different method have been proposed to choose a pump for a PAT system based on the required turbine mode characteristics.

The first method is (Chapallaz et al. 1992). He has presented a selection charts based on more than 80 experimentally tested pump results operating in turbine mode. In his method, first turbine mode specific speed is calculated using nominal turbine flow Q_n (from flow-duration curve) and H_{nt} net turbine head in equation 2.10.

$$n_{qt} = N_t \frac{Q_{nt}^{0.5}}{H_{nt}^{0.75}}$$

The PAT to be selected should run under these head and flow conditions near its best efficiency point. Then by using turbine mode specific speed, the pump-mode specific speed is calculating using equation 2.1

$$n_{qp} = \frac{n_{qt}}{0.89}$$

Where n_{qp} is pump-mode specific speed

The efficiency of the pump can be obtained using the graph in Figure D-1; pump efficiency as function of specific speed n_{qp} and flow Q_{np}. This figure indicates the maximum pump efficiency which can be attained by standard pumps for given head / flow conditions. For preliminary analysis, the nominal pump mode flow rate Q_{np} can be obtained using the available flow Q_{nt} divided by 1.3.

By using pump specific speed and efficiency, turbine design conditions are converted into pump design conditions using conversion factors. These conversion factors are indicated on appendix D. The conversion factors for head, CH, can be obtained using Figure D-2 and conversion factor for flow rate, CQ, can be read using Figure D-3.

Applying the conversion factors CH and CQ on the design head and flow in turbine mode to determine pump performance parameters at pump best efficiency point (BEP) pat the proposed turbine speed is the next step. The nominal pump head and flow rate at turbine speed Nt can be obtained using equation 2.12 and 2.13.

$$H_{np}(N_t) = \frac{H_{nt}}{C_H}$$

$$Q_{np}(N_t) = \frac{Q_{nt}}{C_Q}$$

Lastly, converting pump design conditions at turbine rated speed into pump rated speed using the similarity laws. The general selection charts in pump manufacturers' brochures indicate the type of pump and its rated speed which would probably accommodate the required head/flow conditions for the installation. However, the rated pump speed Np does in most cases not correspond to the proposed turbine speed Nt. Therefore, the head/flow values must be transformed into new head/flow conditions valid for nominal pump speed Np by applying the affinity laws.

The pump mode head and flow rate at pump rated speed can be calculated using equation 2.14 and 2.15 respectively.

$$H_{np}(N_p) = H_{np}(N_T)(\frac{N_t}{N_p})^2$$

$$Q_{np}(N_p) = Q_{np}(N_T)(\frac{N_t}{N_p})$$

By using the rated pump head and flow rate, the required pump can be selected for the use in turbine mode.

The second procedure for selecting proper PAT is proposed by (S.Derakhshan and A Nourbakhsh 2008). In this procedure, the first step is to calculate pump specific speed in its operating point, Nsp by using equation 2.16.

$$N_{sp} = 0.3705 * N_{st} + 5.083$$

Where Nsp (m, m^3/s) and Nst (m, kW) and are the pump and turbine specific speeds in their rated points, respectively.

The second step is to calculate the specific speed in dimensionless form using equation 2.17.

$$\alpha_P = \frac{N_{SP}}{g_{0.75}}$$

The third step is to calculate the gamma (γ) parameter by inserting equation 2.17 in 2.18.

$$\gamma = 0.3705 * \alpha_P + 0.6464$$

After finding the dimensionless parameter gamma (γ), the head ratio (the head ratio of pump to turbine mode) is obtained using equation 2.19.

$$h = [\frac{N_P}{N_T} * \gamma]^{-2}$$

Where Np - is pump mode rated speed

Nt - is turbine mode rated speed

Then the pump head at the rated (nominal) point, H_{pr}, is obtained using the head ratio in equation2.19 and becomes:

$$H_{np} = \frac{H_{nt}}{h}$$

Where H_{tr} is the available head for the PAT

The rated (nominal) flow rate of pump, Q_{np}, can be obtained using Nsp and H_{pr} while choosing rated speed of pump, Np. The proper PAT can be easily selected when the rated head H_{pr}, flow rate Q_{pr} and speed Np are known. These define the design point at which a pump should work in order to function at its best efficiency point as a turbine.

2.8.2 Theoretical studies for selection criteria

Different methods for forecasting PAT performance have been proposed in the literature; some based on pump-mode performance and others are on the geometry of the machine. Williams (1994) compared eight different PAT performance prediction methods based on turbine tests on 35 pumps in the specific speed range of 12.7 to 183.3 and studied the effects of poor turbine prediction on the operation of PAT. The difference between the predicted BEP and the actual BEP for the PAT was studied using the prediction coefficient (C).

Sharma (1998, cited in Swamy 2017, p.19) discussed the suitability of different pumps which can be used as turbines. The analysis of performance curves in pump and turbine mode plotted by Grant and Bain (1985, cited in Swamy 2017, p.19) revealed that the location of turbine BEP is at a higher flow and head than the pump BEP. The study showed that, ratio of turbine capacity and head at BEP to pump capacity and head at BEP varies with specific speed in the range of 1.1 to 2.1.

(Yang et al. 2012a) developed a theoretical method of predicting performance of PAT on the basis of former research results, through theoretical analysis and empirical correlation. The effects of variations of pump specific speed and pump maximum efficiency on h and q were studied and observed that two pumps with same specific speeds may have different h and q. In the next step, a centrifugal pump was simulated in direct and reverse modes using commercial 3D Navier-Stokes computational fluid dynamics (CFD) code available in ANSYS-CFX which has utilized a finite-element based finite-volume method for discretization of the transport equations. The comparison of proposed method with other two methods viz. Stepanoff (1957) and Sharma (1998) revealed that BEP characteristics predicted by the proposed method and CFD were more accurate than the other two methods. The slight difference between experimental and numerical results was found which may be attributed to the negligence of leakage loss through balancing holes, mechanical loss caused by mechanical seal and bearings and the surface roughness value set on the machine's surface.

$$h = 1.2 / \eta_p^{1.1} \qquad 3$$

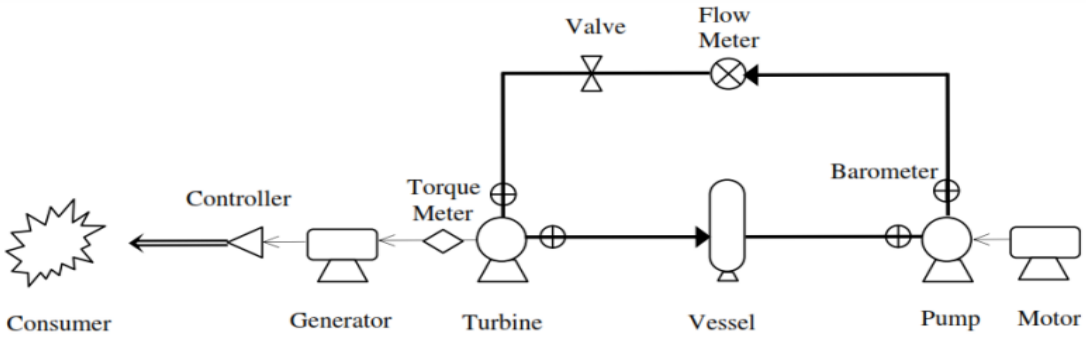
$$q = 1.2 / \eta_p^{0.55} \qquad 4$$

2.8.3 Experimental studies for selection criteria

Derakhshan and Nourbakhsh (2008b) predicted the BEP of centrifugal pump running as turbine using theoretical analysis based on "area ratio" method developed by Williams (1992). The maximum efficiency of PAT was calculated as the ratio of net power output from the turbine and the hydraulic power supplied at the inlet. The net power output was worked out by subtracting various losses in the turbine (e.g. volute power losses, leakage losses, kinetic energy losses at the outlet, hydraulic losses and mechanical losses) from the gross power. A complete mini hydropower test rig established in the laboratory of University of Tehran, as shown in Figure 10, it was used for the experimental verification of theoretical results. At BEP, the values of discharge number, head number, power number and efficiency predicted by theoretical methods were found to be 1.1%, 4.7%, 5.25% and 2.1% lower than that of corresponding experimental values. These deviations may be due to assumptions made in the evaluation of the volute and the impeller losses. The equation of maximum efficiency of PAT was derived as

$$\eta t = \frac{Pnt}{\gamma \times Qt \times Ht} = \frac{(\gamma \times Qt \times Ht) - Pvt - Plt - Pet - Pit - Pmt}{\gamma \times Qt \times Ht} \qquad 5$$

Figure 10: Mini hydropower test rig established in University of Tehran



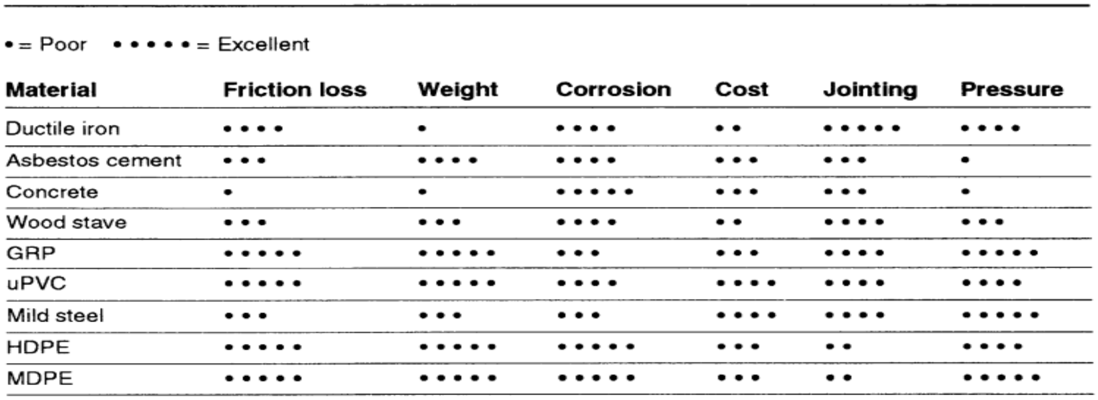
Source: Adapted from S. Maddulety Swamy (2017, p. 24)

(Agarwal 2012) reviewed the research carried out by some researchers and reported that conversion factors for PAT can be decided on the basis of theoretical and numerical studies but its performance cannot be predicted accurately. The need for further research to develop a general model for calculating the conversion factors was emphasized.

2.9 Factors to be considered in Penstock Selection and Design

The factors that need to be considered in designing and selecting the penstock material are briefly described in this section. The most important factor to be considered while designing the penstock pipe is the material to be used as a penstock. Usually mild steel and HDPE pipes are used in MHP. There are several factors to be considered when selecting material to be used for the penstock pipe. These are: Design pressure, method of joining, diameter and friction loss, weight and ease of installation, accessibility of the site, terrain, soil type, design life and maintenance, weather conditions, availability, relative cost and transportation to the site.

Table 4: Comparison of penstock materials



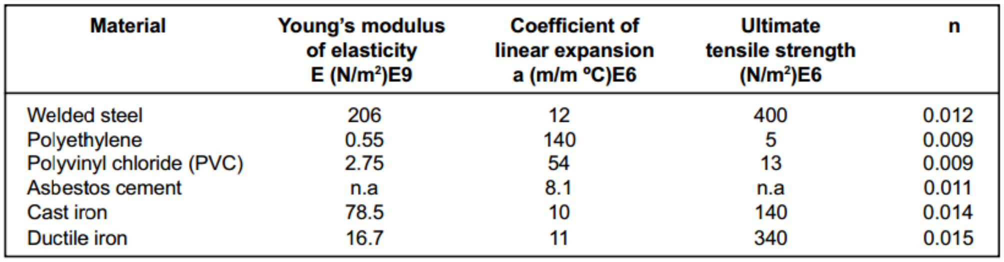
• = Poor • • • • • = Excellent

Material	Friction loss	Weight	Corrosion	Cost	Jointing	Pressure
Ductile iron	• • • •	•	• • • •	• •	• • • • •	• • • •
Asbestos cement	• • •	• • • •	• • • •	• • •	• • •	•
Concrete	•	•	• • • • •	• • •	• • •	•
Wood stave	• • •	• • •	• • • •	• •	• • • •	• • •
GRP	• • • • •	• • • • •	• • •	• • •	• • • •	• • • • •
uPVC	• • • • •	• • • • •	• • • •	• • • •	• • • •	• • • •
Mild steel	• • •	• • •	• • •	• • • •	• • • •	• • • • •
HDPE	• • • • •	• • • • •	• • • • •	• • •	• •	• • • •
MDPE	• • • • •	• • • • •	• • • • •	• • •	• •	• • • • •

Source: Harvey (1993, p.109)

The table above illustrates the possibilities of using different kinds of material based on various factors. The more the number of “stars” the more favorable is the material type under different characteristics. For example, if friction loss and cost were the major concern in selection, uPVC type of penstock would clearly be the first choice. (Harvey 1993)

Table 5: Materials used in pressure pipes



Material	Young's modulus of elasticity E (N/m^2)E9	Coefficient of linear expansion a (m/m °C)E6	Ultimate tensile strength (N/m^2)E6	n
Welded steel	206	12	400	0.012
Polyethylene	0.55	140	5	0.009
Polyvinyl chloride (PVC)	2.75	54	13	0.009
Asbestos cement	n.a	8.1	n.a	0.011
Cast iron	78.5	10	140	0.014
Ductile iron	16.7	11	340	0.015

Source: Adapted from ESHA (1998, p .140)

The diameter is selected to reduce frictional losses and therefore energy losses within the penstock, which also depends on a trade-off between penstock cost and power losses Selecting as small a diameter as possible to minimize cost and selecting as large a diameter as necessary to minimize losses. A simple criterion for diameter selection is to limit the head loss to a certain Percentage that is a loss of power (head) of 4% is usually acceptable. The wall thickness is selected to accommodate the pressures encountered during plant operation. (ESHA 1998)

2.9.1 Penstock sizing and costing

Because the penstock is often the most expensive part of the installation, it is worth putting some effort into minimizing its cost. One reason it is expensive is that the penstock must be strong enough to withstand the very high pressures which results from sudden blockage of the water flow. These high pressures are only temporary and are known as surge or water hammer pressures. They travel in waves of positive and negative pressure throughout the length of the pipe. In some cases it is the negative pressure wave which destroy the penstock by inward collapse. But it is sufficient to design the penstock for the worst positive surge, such that it has a wall thickness suitable to accommodate negative as well as positive surge. (Harvey 1993)

2.9.2 Determination of Friction Loss (h_f) in Penstock

The relation for determination of friction loss in penstock was first given by Chezy (1775). Subsequently, improved relations were given by various researchers. Some of these relations are for laminar flow having Reynold number (Re) less than 2000, some are for transition flow having Re in between 2000 and 4000 and some are for turbulent flow having Re more than 4000. Poiseuille (1841) developed the relation for laminar flow. Hazen Williams (1902) and Blasius (1913) developed the relations for turbulent flow. The flow in penstock shall be turbulent flow as penstock with very small diameter like 0.1 m and flow velocity as 2.0 m/sec shall have Re as 2×10^5 which is in the range of turbulent flow. The relation developed by Hazen Williams is not much used for penstock as this relation can be effectively used for the flow velocity up to 1.0 m/sec which is much less than the flow velocity in penstock. The relation developed by Blasius is also not applicable for penstock flow as relation given by him is applicable for flow having Re up to 1×10^5 which is less than the Re of flow in penstock. The relations developed by Darcy-Weisbach (1850), Manning (1891) and Scobey (1930) are the most commonly used relations to determine the friction loss in penstock. (Singhal & Arun 2015)

The diameter is selected as the result of a trade-off between penstock cost and power losses. A simple criterion for diameter selection is to limit the head loss to a certain percentage. Total penstock head loss should not be more than 10% of the penstock gross head. (Pushpa .C 2006)

According to the U.S. Department of the Interior (1967), losses in trash rack and entrance can be taken at 0.1, 0.2, and 0.5 ft of head, respectively, for velocities of 1.0, 1.5, and 2.5 ft/sec at the penstock entrance. Engineering monograph no. 3 further indicates that entrance losses for bell-mouthed entrances would be 0.05 to 0.1 times the velocity head at entrance and for square-mouthed entrances the loss would be 0.1 times the velocity head at entrance. The U.S. Department of the Interior (1967) gives also the following values for K: for large gate valves, K = 0.10; for needle valves, K = 0.20; and for medium-size butterfly valves with a ratio of leaf thickness to diameter of 0.2, K = 0.26. (Warnick, C. C 1984)

2.10 Power

2.10.1 Generator

Typically in micro hydro systems the torque from the output shaft of the PAT is converted into electricity by use of a generator. This provides great flexibility for the use of the power as the electricity is easy to transport and use for multiple devices at the same time.

In converting the energy from the shaft into electricity some energy is lost. As the power from the turbine may be used to drive a pump, there will again be losses when the electricity is used in the pump motor. Kaya et al. (2008) report that motor efficiencies can range between 70% and 96% and higher efficiency motors normally cost 15-25% more than standard motors. Generator performance is comparable to motor performance and thus the range of typical total efficiencies for just the electrical sub-system would be between 50% and 92%. The efficiency of the motor is also relative to the load as motors running at partial load will be less efficient. It is thus crucial to choose the correct size for the motor and therefore also the generator.

Williams (1996) reports that synchronous generators were used previously but induction motors proved to be more suited to the application. They are more robust because of the method of construction which uses cast bars instead of windings on the rotating part. Most pump units are supplied with three-phase induction motors.

2.11 Wind and solar Energy Resources Potential of Ethiopia

2.11.1 Wind Energy Resource

According to a preliminary study recently completed by Hydro China International Engineering Company (HCIE), Ethiopia's wind energy resource amounts to over 1,300GW.The country is divided in to two regions by the Great Rift Valley and has a diverse landscape coving mountains and plateaus in its highlands and deserts and steppes in its lowlands. The large differences in altitude provide a good wind resource potential. In the Ethiopian highland plateaus wind speed become maximum during the dry season. Whereas, in the low land, such as Ogaden and the eastern parts of the country, average wind velocity reaches maximum values between May and August, that is the rainy season in most of the country (GTZ, 2010).

EEPCo has a short and long term plans to mix the country's the dominant hydro power electrical energy production with other renewable energy sources, such as wind energy, solar energy and geothermal energy. Towards this, two wind park projects are under construction; these are Ashegoda Wind Park (near Mekele) with installed capacity of 120MW and the second one is Adama Wind Park which is divided in to two phases. Currently Adama phase one has been commissioned.

2.11.1.1 Advantages and disadvantages of power production by wind energy

Pros of Wind Energy:

- **Clean & Renewable Source of energy -** Wind is free. In the event that you live in a geological area that gets a lot of wind, it is ready and waiting. As a renewable asset, wind can never be drained like other regular, non-renewable assets.
- **Promotes Cost-Effective Energy Production -** The cost of wind-generated electricity has fallen from nearly 40¢ per kWh in the early 1980s to 2.5-5¢ per kWh today depending on wind speed and project size.
- **Maintenance -** Wind turbines require little maintenance, and their parts last for long periods. In fact, the average lifespan of a small, domestic turbine is 20 years

2.11.1.2 Cons of Wind Energy

- **Wind Reliability -** Wind doesn’t generally blow reliably, and turbines usually function at about 30% capacity. Serious storms or high winds may cause harm to your wind turbine, particularly when they are struck by lightning.
- **Threat to Wildlife -** The edges of wind turbines can actually be unsafe to natural life, especially birds and other flying creatures that may be in the area. There isn’t really a way to prevent this, but it’s definitely something that you want to make sure that you are aware of be possible consequences that may come up as a result of it.
- **Noise and Visual Pollution -** Wind turbines make a sound that can be between 50 and 60 decibels. People have widely varied reactions to seeing wind turbines on the landscape. Some people see graceful symbols of economic development and environmental progress or sleek icons of modern technology. Others might see industrial encroachment in natural and rural landscapes.
- **Suitable to Certain Locations -** Wind energy can only be harnessed at certain locations where speed of wind is high. Since they are mostly setup in remote areas, transmission lines have to be built to bring the power to the residential homes in the city which requires extra investment to set up the infrastructure

2.11.2 Solar Energy Resource

Ethiopia receives 4.55 to 6.5 kWh/m^2/day annual average of solar insolation throughout the country (Breyer et al. 2009) . This varies significantly during the year, ranging from a minimum of 4.55 kWh/m^2 in July to a maximum of 6.55 kWh/m^2 in February and March. Other literatures describe the yearly average radiation to be in the range from 4.25 kWh/m^2 in the areas of Itang in the Gambella regional state (western Ethiopia), to 6.25 kWh/m^2 around Adigrat in the Tigray regional state (northern Ethiopia

Ethiopia exhibits excellent prerequisites for a nearly 100% renewable energy supply. Hydro and solar resources offer the chance of renewable energy supply in an economically, ecologically and socially sustainable way. Such PV-hydro potentials have been already analyzed for Ethiopia (Breyer et al. 2009).

2.11.2.1 Pros of Solar Energy

- **Renewable -** Solar energy is a renewable energy source. This means that we cannot run out of solar energy, as opposed to non-renewable energy sources (e.g. fossil fuels, coal and nuclear)
- **Abundant-** The potential of solar energy is beyond imagination. The surface of the earth receives 120,000 terawatts of solar radiation (sunlight) – 20,000 times more power than what is needed to supply the entire world.

- **Geographically widely available -** The level of solar irradiation that falls upon the earth varies with the geography of the planet. Generally, the closer to the equator the more solar energy but what most don't realize is that solar energy can be used anywhere.
- **Silent -** There are no moving parts involved in most applications of solar power. There is no noise associated with photovoltaics. This compares favorably to certain other green-techs such as wind turbines

2.11.2.2 Cons of Solar Energy

- **High initial cost -** While a reduced electric bill is an advantage, there are initial costs for the equipment, panels and installation that could be more than $20,000. Also, if you have devices that run on DC currents, those will be more expensive to power directly.
- **Weather dependence -** The most important item with solar panels is the sun. If you live in an area prone to cloudy days or are simply having some storms and darker days for an extended period, this will negatively impact how the system runs. Your system will likely be less productive in winter months than summer months.
- **Energy Storage is Expensive -** Energy storage systems such as batteries will help smoothen out demand and load, making solar power more stable, but these technologies are also expensive.

3. Methodology

The methodology describes the key issues required to assess the hydropower potential and to identify the most promising sites. The procedures used for selecting the sites and type of devices used for hydroelectric power generation will also be discussed in detail. Lastly, the economical evaluation of the different option will be discussed.

It took out several steps, including site visits, data collection, measurements and field surveys and Analysis of the required equipment and technology (with financial analysis). The selection process comprises the following main stages.

3.1 Description of the Study Area

Addis Ababa is the capital and largest city of Ethiopia. It is the seat of the Ethiopian federal government. According to the 2008 population census, the city has a total population of 3.385 million inhabitants. Addis Ababa lies at an elevation of 2,300 meters (7,500 ft) and is a grassland biome, located at 9°1′48″N 38°44′24″E coordinates. The city is in fast development and expansions, recent housing developments are taking place on the outskirts of the city mainly in the Eastern, Western and Southern expansion areas of the City. Most of the houses under construction are located in Akaki kality, Bole, Yeka, Nifas Silk Lafto and Kolfe Keranyo Sub Cities.

Generally, the City is divided in to three main catchments namely, Kality, Eastern and Akaki. Before few years, Addis Ababa has only one major central wastewater treatment plant at kality. This treatment plant was serving only small proportion of the potential sewer connection demands as the network was limited to Kality catchment. And there was no wastewater treatment plant in the other catchments. In addition to this central system, the Eastern part of Addis is also served by the Yerer Drying Beds and Lagoons. And Akaki Catchment covers the southern part of Addis downstream to Kality Waste Water Treatment Plant and part of Bole, Lafto, Saris, Bolebulbula and Akaki. Akaki catchment is further divided in to two sub catchments namely Chefe and Akaki Sub-catchments. AAWSA as the main responsible authority has devised different mechanisms to improve the service by expanding the sewer network in Kality catchment and upgrade the existing wastewater treatment plant and constructing new wastewater treatment facilities in other catchments. Chefe wastewater project being part of akaki catchment is aimed to serve the Southern part of addis abeba and is under construction. It will take a few months to finalize and start operation. On the other hand, the City Government has continued constructing ten thousands of condominium houses in several locations. The government of the city is constructing a compacted wastewater treatment plant for the new Condominium sites. These Package Waste Water Treatment Plants are Bole Arabsa 1A, 2A, & 2B, Bole Bulbula, Kilinto, Tulu Dimtu-1, 2 & 3, Mekanisa Kotari, Oromiya, Karakori-I and Degnet.

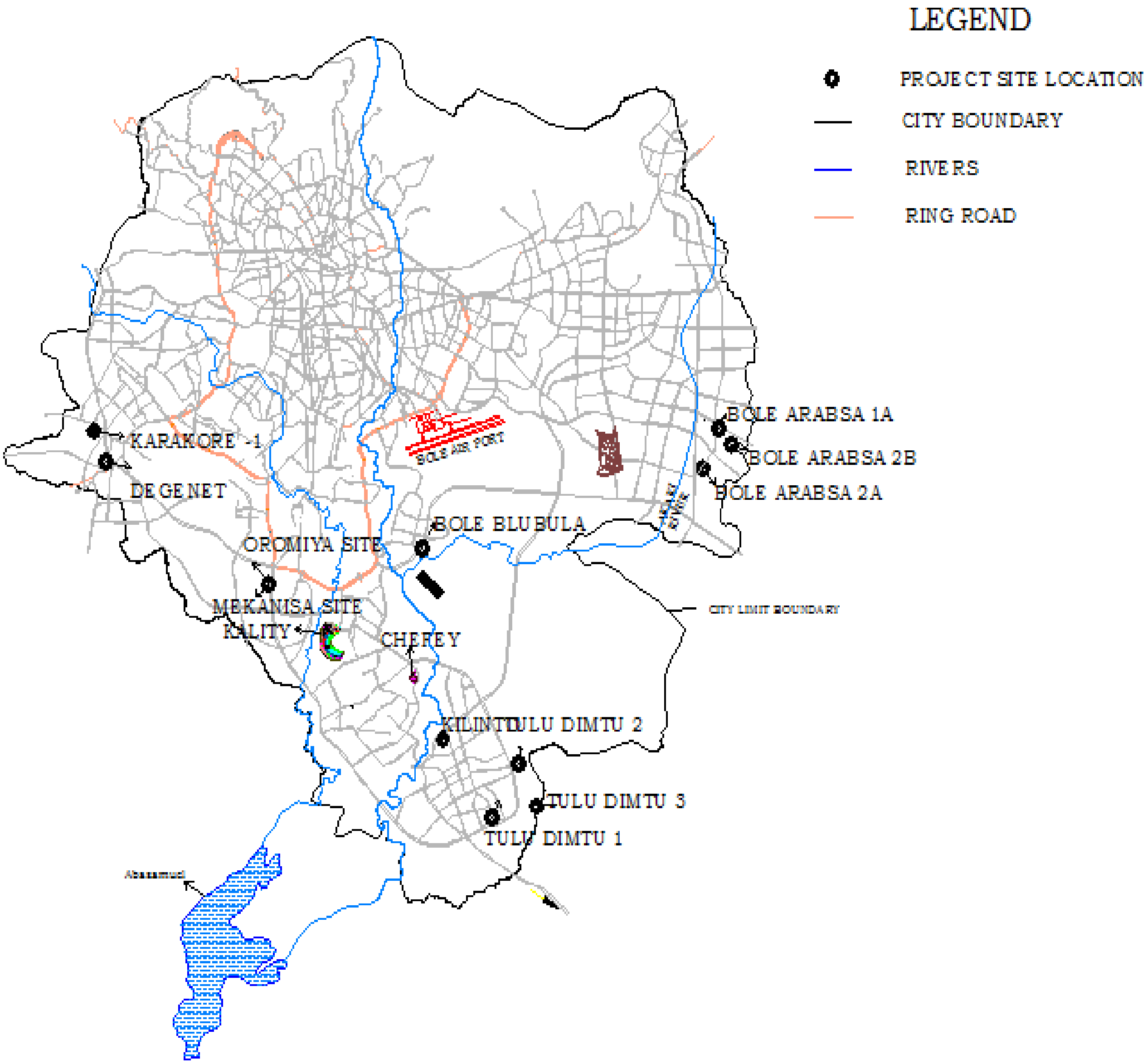


Figure 11: Addis Ababa city WWTP locations

Fig -2 provides a distribution of the head and flow for the surveyed plants in Addis Ababa city. The plants with the highest product of head and flow represent the best opportunities.

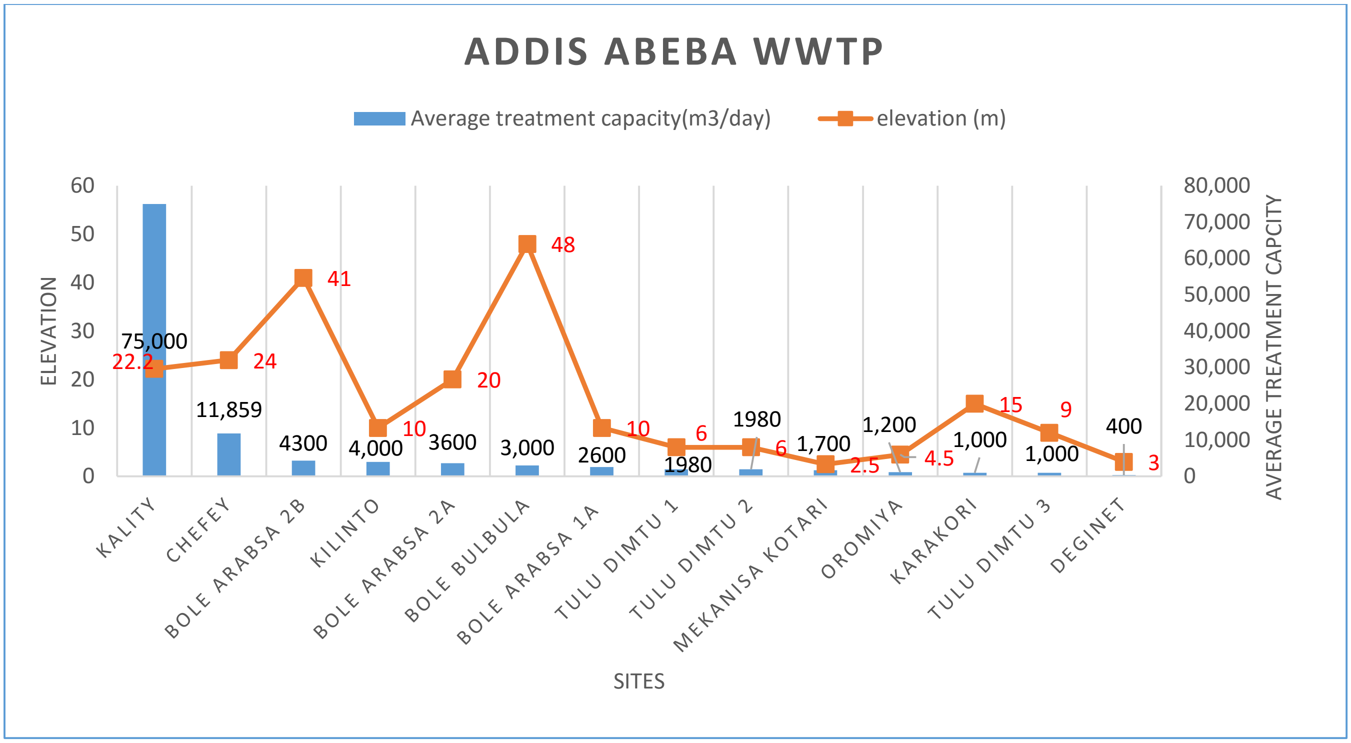


Figure 12: Addis Ababa wwtp capacity and available heads

3.2 Data collection

During Site visits appropriate data for this study was gathered and analyzed. These includes flow and topographic data. The flow data and characteristics of influent was taken from design documents and laboratory results that was provided by consultant's offices and Addis Ababa water supply and sewerage authority (AAWSA) specifically Water and Sanitation Development and Rehabilitation Project Office. The topographic data of each sites was taken and verified from the consultant involved in these.

During the study, the following aspects were analyzed: type of wwtp, average treatment capacity, average hourly flow, available gross head and potential electrical power.

3.2.1 Site visits

Generally, the City is divided in to three main catchments namely, Kality, Eastern and Akaki. Before few years Kality was the only treatment plant that was serving the central part of the city. But these catchment only serve the old city. The rest part of the city was planned to be served by chefy wwtp, this treatment plant only serve part of the akaki catchment. For the rest of the city especially for the newly constructed condominium housing; that are found on the outskirt of the city. A number of wwtp were constructed. This study included all the existing wwtp that are found in Addis Ababa city. After the site visit, all are listed in Table 1.

Table 6: Visited WWTPs Addis Ababa city

No.	Site Names	Type of WWTP
1	Kality	Conventional Treatment Process
2	Chefey	Conventional Treatment Process
3	Bole Arabsa 2B	Membrane bioreactor Process
4	Kilinto	Membrane bioreactor Process
5	Bole Arabsa 2A	Membrane bioreactor Process
6	Bole Bulbula	Membrane bioreactor Process
7	Bole Arabsa 1A	Membrane bioreactor Process
8	Tulu Dimtu 1	Membrane bioreactor Process
9	Tulu Dimtu 2	Membrane bioreactor Process
10	Mekanisa Kotari	Membrane bioreactor Process
11	Oromiya	Membrane bioreactor Process
12	Karakori	Membrane bioreactor Process
13	Tulu Dimtu 3	Membrane bioreactor Process
14	Deginet	Membrane bioreactor Process

3.2.2 Flow data and characteristics

The flow data includes (average treatment capacity, total suspended solid, biochemical oxygen demand (BOD), chemical oxygen demand (COD) and total nitrogen (TN)) was collected to analysis the flow amount and characteristics.

Table 7: Addis Ababa city WWTPs influent flow data and characteristics

Site Names	Average treatment capacity(m^3/day)	TSS (kg/day)	BOD (kg/day)	COD (kg/day)	TN (kg/day)
Kality	75,000	1020	799	1598	76.5
Chefey	11,859				
Bole Arabsa 2B	4300	1505	1290	3225	258
Kilinto	4,000	1400	1200	3000	168
Bole Arabsa 2A	3600	1260	1080	2700	216
Bole Bulbula	3,000	1050	900	1275	126
Bole Arabsa 1A	2600	910	780	1950	156
Tulu Dimtu 1	2000	700	600	1500	120
Tulu Dimtu 2	2000	700	600	1500	120
Mekanisa Kotari	1,700	595	510	1275	71
Oromiya	1,200	420	360	900	72
Karakori	1,000	350	300	750	42

Tulu Dimtu 3	1,000	350	300	750	60
Deginet	400	140	120	300	16.8

The flow characteristic of the incoming wastewater was considered for electromechanical equipment selection and calculation.

3.2.3 Installed capacity of the wwtp

The installed power capacity of the wastewater treatment plants were gathered from each sites. The power requirement of the conventional treatment plants (Kality & Chefy) were found to be smaller than those which depend heavily on electromechanical equipment's.

Table 8:- Installed capacity of each treatment plants

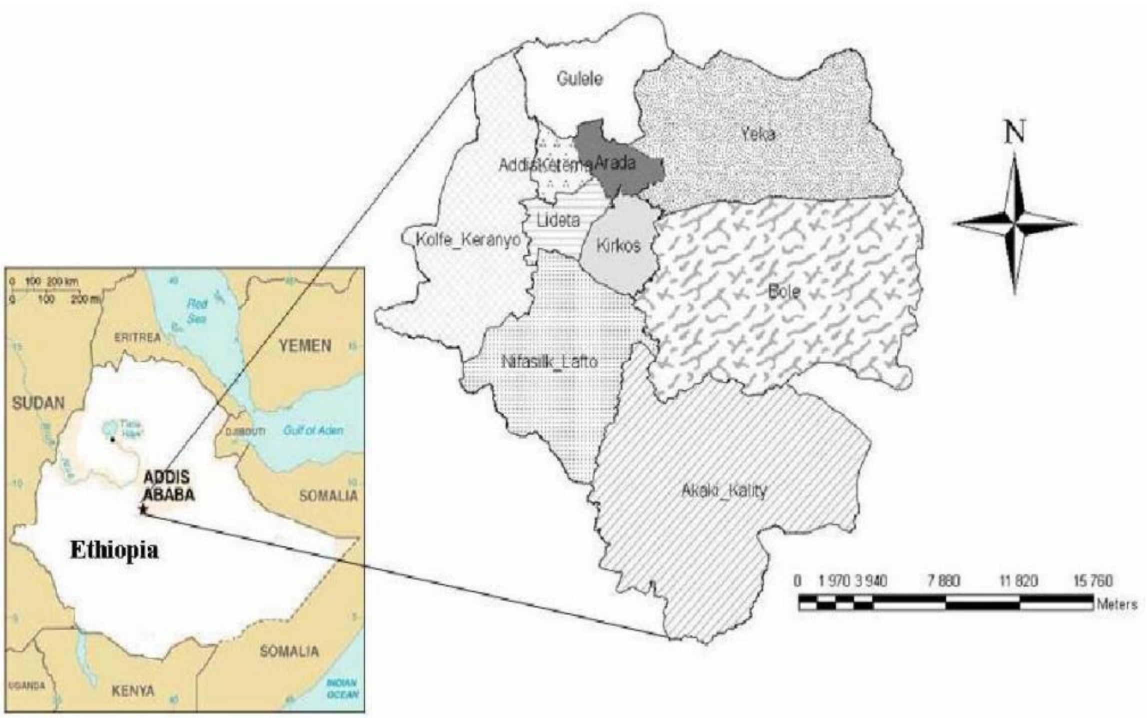
Sites	Installed capacity KW
Kality	250
Chefey	213
Bole Arabsa 2B	309
Kilinto	290
Bole Arabsa 2A	291
Bole Bulbula	240
Bole Arabsa 1A	212
Tulu Dimtu 1	164
Tulu Dimtu 2	163
Mekanisa Kotari	155
Oromiya	113
Karakori	105
Tulu Dimtu 3	102
Deginet	45

3.2.4 Topographic survey data

A contour map with one meter interval was used for each sites and the surrounding area for detailed planning analysis. Depending on the topographic features, available data and site visit potential sites were identified for further planning.

Also all ground levels for each sites were verified and confirmed by actual surveying work.

Figure 13: Map of Addis Ababa



3.3 Data Analysis

3.3.1 Potential site analysis

The identified potential sites in table 8 were screened and assessed based on the nominal discharge and available head near the site. The other ten sites listed in table 7 were eliminated during the selection process due to small head and flow availability in these sites. The available head for these sites were small between (2-10) m and the topography features of these area were flat. And also when looking at the flow range, it is small between (0.005 – 0.046) m^3/s. When considering these two factors together they greatly affect the potential power production of sites. Where η is the global plant efficiency, ρ is the water density (kg/m^3), g the acceleration due to gravity (m^2/s), H_{net} is the net head available at the turbine (m) and Q is discharge (m^3/sec). $\boldsymbol{\eta}$ was assumed to be 0.7 which is an average accepted value for combined electromechanical efficiencies.

Table 7: Sites eliminated during initial evaluation

Site Name	Average treatment capacity/outfall (m3/s)	Available Gross Head (m)	Potential power production P = η*p*Q*g*h(KW)
Kilinto	0.046	8	2.53
Bole Arabsa 2A	0.042	13	3.75
Bole Arabsa 1A	0.03	10	2.06
Tulu Dimtu 1	0.023	3.2	0.51
Tulu Dimtu 2	0.023	2.4	0.38
Mekanisa Kotari	0.02	2.5	0.34
Oromiya	0.014	4.5	0.43
Tulu Dimtu 3	0.012	2	0.16
Karakori	0.012	2	0.16

Table 8 presents the specific technical information for those visited sites, which are considered to be feasible for micro and pico hydropower generation due to their respective heads, flow and power potentials. These sites are Kality, Chefey, Bole Arabsa 2B and Bole Bulbula.

Table 8: Potential WWTP for Power Generation

Site Name	Average treatment capacity(m^3/s)	Gross Head (m)
Kality	0.868	23
Chefey	0.137	30
Bole Arabsa 2B	0.050	41
Bole Bulbula	0.035	48

3.3.2 Flow analysis

When constructing a hydraulic power plant, the daily mean water flow rate is measured in advance at the construction site. In the above mentioned treatment plants, they are not working with their full capacity due to small flow from the network. According to the design of each specific sites, by now the treatment plants should have been working with their full capacity but they are not. Also many of the wwtp plants are found in outskirts of Addis Ababa and people have not fully started living in this condominium houses. It will take at least 2 years for the treatment plants to achieve design flow. Because of this reason I was forced to take the designed Average hourly flow for the flow calculations.

For flow calculations the following designed data are taken

Table 9: Average hourly flow of potential sites

Sites	Average hourly flow(m^3/hr)
Kality	3125
Chefey	494.1
Bole Arabsa 2B	179.2
Bole Bulbula	125

3.3.3 Site selection and topographic Analysis

The gross head elevation and horizontal distance between the WWTP and the selected locations for each sites has been listed below

Table 10: Gross head elevation and Horizontal Distance of identified locations downstream of WWTP

Site Name	Elevation of the site(m.a.s.l)	Potential site D/S(Elevation) (m.a.s.l)	D/S Potential site Distance from WWTP (m)
Kality	2150	2130	304
Chefey	2098	2068	510
Bole Arabsa 2B	2282	2242	140
Bole Bulbula	2181	2137	430

3.3.4 Determination of penstock material and diameter

The basic rule in lay out the system is keep the penstock as short as possible. It is always important to calculate the most economical diameter of penstock. The diameter is selected as the result of a trade-off between penstock cost and power losses. A simple criterion for diameter selection is to limit the head loss to a certain percentage. Total penstock head loss should not be more than 10% of the penstock gross head. The recommended initial trial internal diameter (D) can be calculated as:

$$D = 41 * Q^{0.38}$$

Where:

Q - Is design flow in l/s

3.3.5 Head losses in the system

After selecting the material and the diameter of the penstock pipe it is necessary to calculate the head loss in the pipe length. The head loss in penstock comprises of friction loss (hf) due to resistance offered by penstock inner surface and other losses.

Total head loss = major head loss (h_f) + minor head loss (H_{minor}).

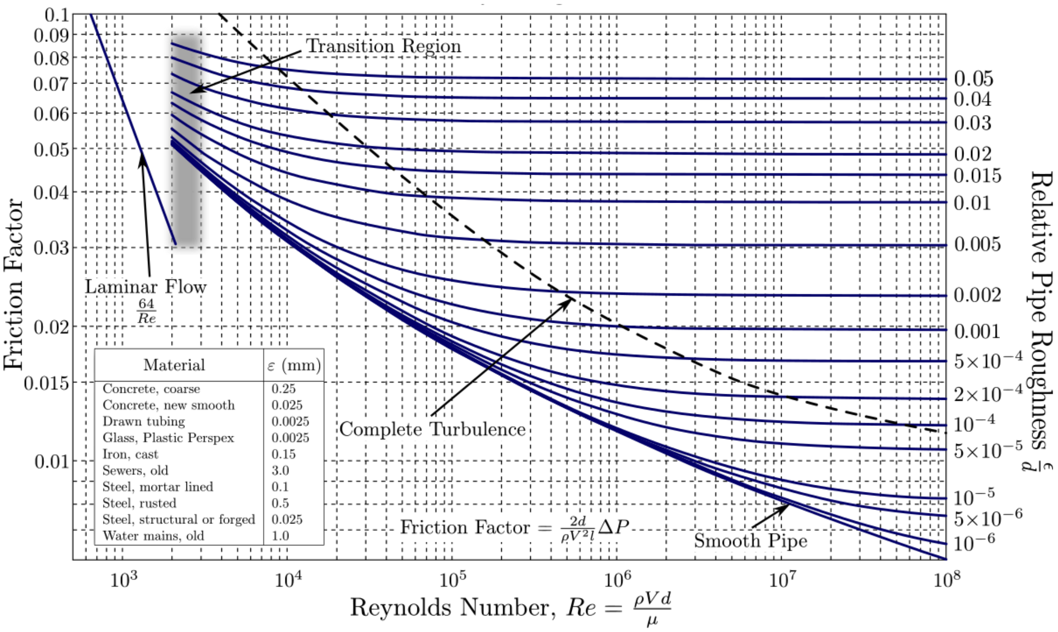
Where Major Head loss (h_f) is given by Darcy-Weisbach formulas:

$$H_{friction} = f * \frac{L}{D} * \frac{V^2}{2*g}$$

Where

- λ- friction factor according to the Moody diagram
- L- length of pipe section with constant diameter in (m)
- d-diameter of the pipe in (m)
- v-average velocity in (m/s)
- g- gravitational force

Figure 14: Moody Diagram



Material	ε (mm)
Concrete, coarse	0.25
Concrete, new smooth	0.025
Drawn tubing	0.0025
Glass, Plastic Perspex	0.0025
Iron, cast	0.15
Sewers, old	3.0
Steel, mortar lined	0.1
Steel, rusted	0.5
Steel, structural or forged	0.025
Water mains, old	1.0

Where Minor loss are given by

$$H_{minor} = \frac{V^2}{2g}\left(K_{entrance} + K_{bend} + K_{contraction} + K_{valve}\right)$$

Where:

- v-average velocity in (m/s)
- g- gravitational force
- K- (local) loss coefficient

3.4 Turbine selection mechanism

The specifications of the electro-mechanical equipment for a hydropower site are mainly defined by the nominal discharge and net head (available gross head minus losses). In case of long (pressure) pipes, the turbine characteristics have an influence on the risk of water hammer (transient pressure rise).

The equipment proposed for installation has to be integrated in the existing infrastructures taking the main purpose of the infrastructure carefully into consideration. To avoid interferences, flexible technology is required to allow a maximized utilization of the available energy potential.

Besides head and flow, the following criteria have to be considered when it comes to turbine selection:

- Possibility to integrate the equipment in the existing infrastructure,
- Turbine efficiency,
- Maintenance and
- Proposed system costs.

3.4.1 Turbine Cost Prediction

The equations given by Ogayar and Vidal (2009) predict the cost per kW for various common turbines over a range of power output under 2 MW.

- **Pelton Turbines**

$$COST = 17.693P^{-0.3644725}H^{-0.281735}$$

- **Francis Turbines**

$$COST = 25.698P^{-0.560135}H^{-0.127243}$$

- **Kaplan turbines**

$$COST = 19.498P^{-0.58338}H^{-0.113901}$$

- **Semi-Kaplan turbines**

$$COST = 33.239P^{-0.58338}H^{-0.113901}$$

3.5 Pump as Turbine

It is require that the pump to be selected should operate in turbine mode under given site head/flow rate conditions to delivers shaft power output for power generation. And the PAT nominal speed must be chosen in such a way that the available machine should operates near its best efficiency point under the given site conditions. 1500 RPM is often used.

The basic classification of pumps may be given according to their flow pattern. For rotodynamic machines, the pump industry uses the concept of specific speed to describe the type of pump in more general terms.

$$n_q = n\frac{\sqrt{Q}}{H^{3/4}}$$

Where n= proposed pump speed (RPM)

Q= pumping discharge (m^3/s)

H= pump head increase per stage (m)

As a general rule, axial flow pumps are usually selected for pumping large volume of water against relatively low heads (1 to 15m) whereas mixed flow pumps are used for intermediate lifts (6 to 30m) over a wide range of flow rates.

Pumps operated as turbine requires a net head which is between 30 and 150% higher than in pump mode to operate in their best efficiency points. In other words, for a given site (head \flow conditions) a smaller pump must be selected in turbine mode than for the same conditions in pump mode.

3.5.1 Selection Procedure of PAT

3.5.1.1 Design head and flow

- Q_{nt} nominal turbine flow (from average flow of wwtp)
- H_{nt} nominal or net turbine head (from the head loss calculation)

3.5.1.2 Specific speed in turbine mode

Calculate the specific speed of the installation in turbine mode

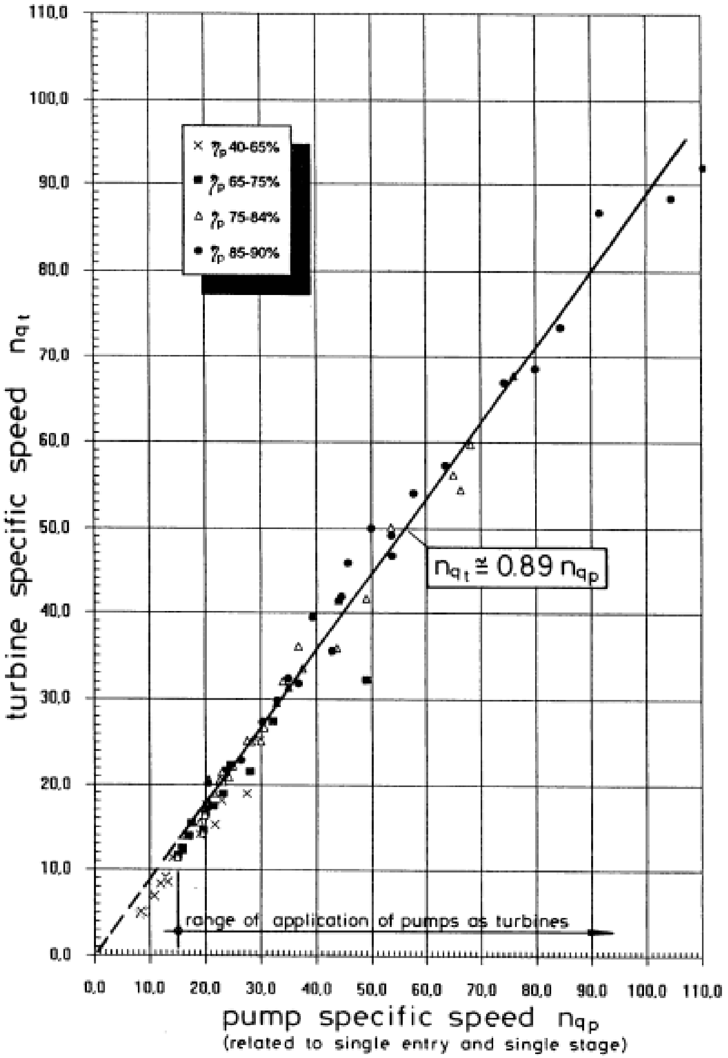
$$n_{qti} = n_{qti}\frac{\sqrt{Q_{ti}}}{{H_{ti}}^{3/4}}$$

Where n_t is the proposed PAT speed in rpm ($=min^{-1}$)

3.5.1.3 Pump-mode specific speed

In order to be able to use the diagram --- (conversion factor), the turbine mode specific speed calculation with formula --- above must be related to a pump-mode specific speed. It has been found from test results that the relation between turbine and pump mode specific speed takes a fairly constant value: turbine specific speed corresponds to approximately 0.89 times pump specific speed.

Figure 15: turbine-mode specific speed versus pump-mode specific speed



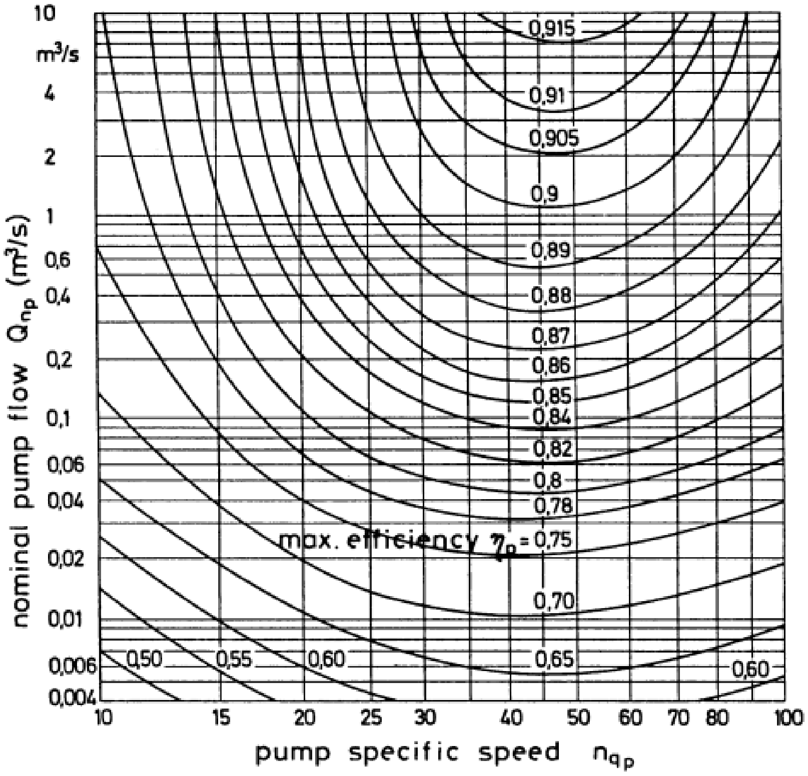
$$n_{qp} = \frac{n_{qt}}{0.89}$$

Note that pumps of specific speed $n_{qp} < 15$ should not be used as turbines. Efficiency of such impellers are low (long and narrow passages incurring friction losses) and the performance of such pumps as turbine cannot be predicted accurately.

3.5.1.4 Pump efficiency

The conversion of turbine design conditions into pump design conditions depends on the efficiency of the machine. The figure below indicates the maximum pump efficiency which can be attained by standard pumps for given head/ flow conditions. Using the available flow Qt divided by 1.3 (= average conversion factor) provides sufficiently accurate values for pre-selection.

Figure 16: Maximum pump efficiency in function of specific speed n_{qp} and flow Q_{np}



3.5.1.5 Converting turbine design conditions into pump design conditions

Using the figure below with pump specific speed and efficiency, read the conversion factors for head C_H and flow C_Q.

$$\text{Conversion factor for head} \qquad C_H = \frac{H_{nt}}{H_{np}}$$

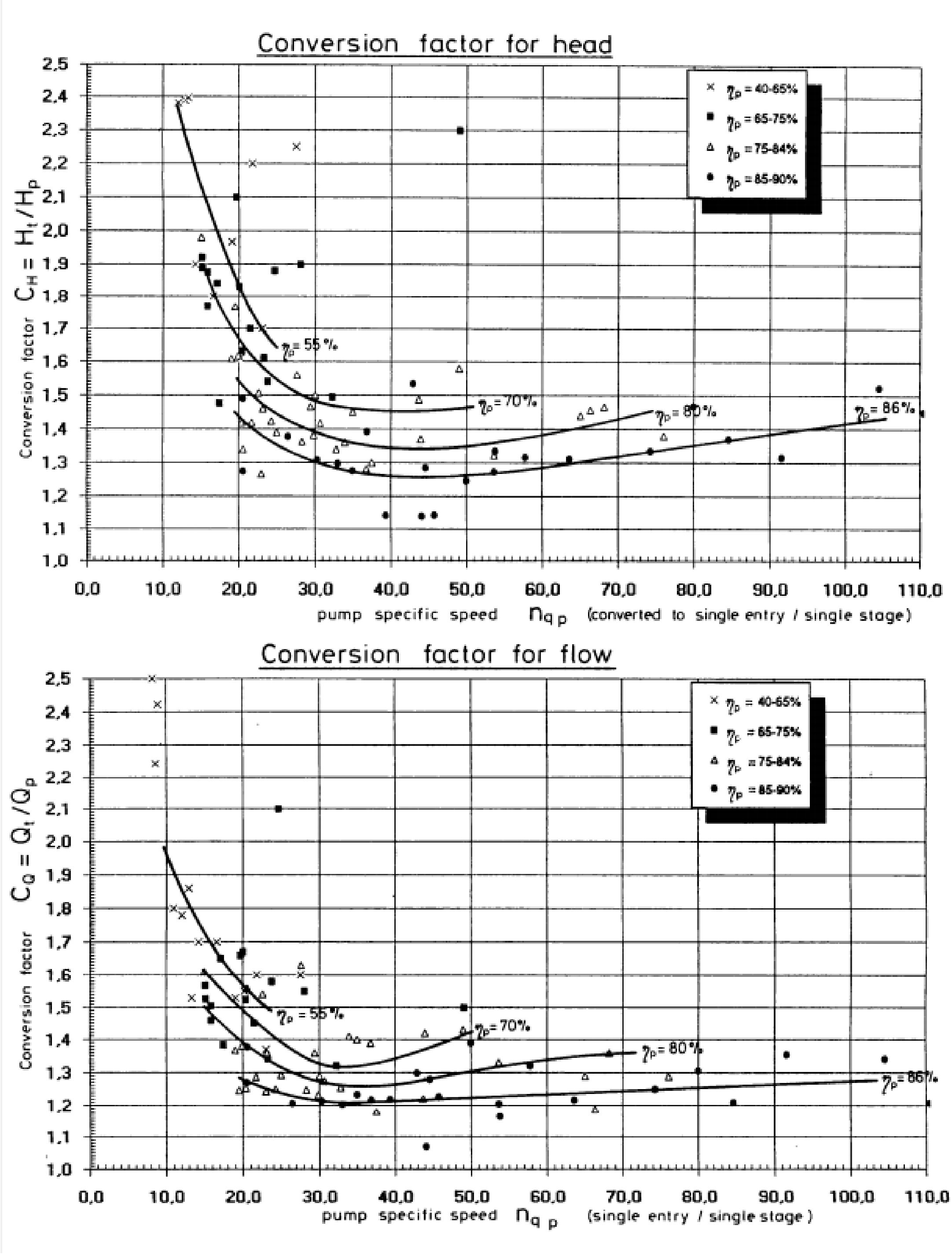
$$\text{Conversion factor for flow} \qquad C_Q = \frac{Q_{nt}}{Q_{np}}$$

Applying the conversion factors C_H and C_Q on the design head and flow in turbine mode, we obtain pump performance parameters at bep p (pump best efficiency point) at the proposed turbine speed.

$$\mathit{Nominal\ pump\ head\ at\ turbine\ speed\ } N_t \qquad H_{np}(N_t) = \frac{H_{nt}}{C_H}$$

$$\mathit{Nominal\ pump\ flow\ at\ turbine\ speed\ } N_t \qquad Q_{np}(N_t) = \frac{Q_{nt}}{C_Q}$$

Figure 17: Turbine mode performance (conversion factors related to pump best efficiency)



3.5.1.6 Converting pump design conditions at turbine rated speed into pump rated speed

The general selection charts in pump manufactures brochures indicates the type of pump and its rated speed which would probably accommodate the required head/flow condition for the installation. However, the rated pump speed n_p does in most cases not correspond to the proposed turbine speed n_t. Therefore, the head/flow values Hnp / Qnp according to the above must be transformed into new head/flow conditions valid for nominal pump speed n_p.

$$H_{np}(N_p) = H_{np}(N_T)(\frac{N_t}{N_p})^2$$

$$Q_{np}(N_p) = Q_{np}(N_T)(\frac{N_t}{N_p})$$

3.5.1.7 Converting the best efficiency point of the selected pump into turbine mode

i. Calculate the specific speed of the pump

$$n_{qpi} = n_p \frac{\sqrt{Q_{pi}}}{{H_{pi}}^{3/4}}$$

ii. Read the conversion factor for head and flow from Fig: Turbine mode performance above using the maximum efficiency of the selected pump as indicated by the manufacturer.

- Conversion factor for head: $C_{H\,max} = 1.1 * C_H$ $C_{H\,min} = 0.9 * C_H$
- Conversion factor for flow: $C_{Q\,max} = 1.075 * C_Q$ $C_{Q\,min} = 0.925 * C_Q$

iii. Calculate the turbine efficiency point (bep t) for maximum and minimum conversion factors

Nominal turbine head at pump speed n_p $H_{nt\,(n_p)} = C_H H_{np}$

Nominal turbine flow at pump speed n_p $Q_{nt\,(n_p)} = C_Q Q_{np}$

iv. convert these turbine bep conditions into the nominal speed n_t using the affinity laws as under point 6 above

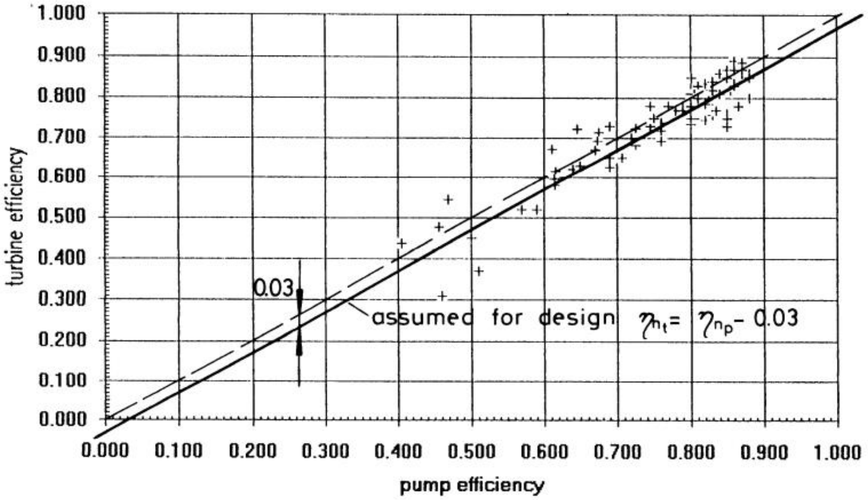
$$Nominal\ turbine\ head\ at\ turbine\ speed\ nt \quad H_{np}\ (N_p) = H_{np}(N_T)(\frac{N_p}{N_t})^2$$

$$Nominal\ turbine\ flow\ at\ turbine\ speed\ nt \quad Q_{np}(N_p) = Q_{np}(N_T)(\frac{N_p}{N_t})$$

3.5.1.8 Power output in turbine mode at the best efficiency point

Test results have shown that maximum efficiency in turbine mode is on average slightly lower than in pump mode. In the following, we propose a turbine maximum efficiency 3% below pump maximum efficiency.

Figure 18: Turbine best efficiency versus best efficiency & assumed law for design computations.



$$P_{nt} = \rho g Q_{nt} H_{nt} (\eta_{p\,max} - 0.03)$$

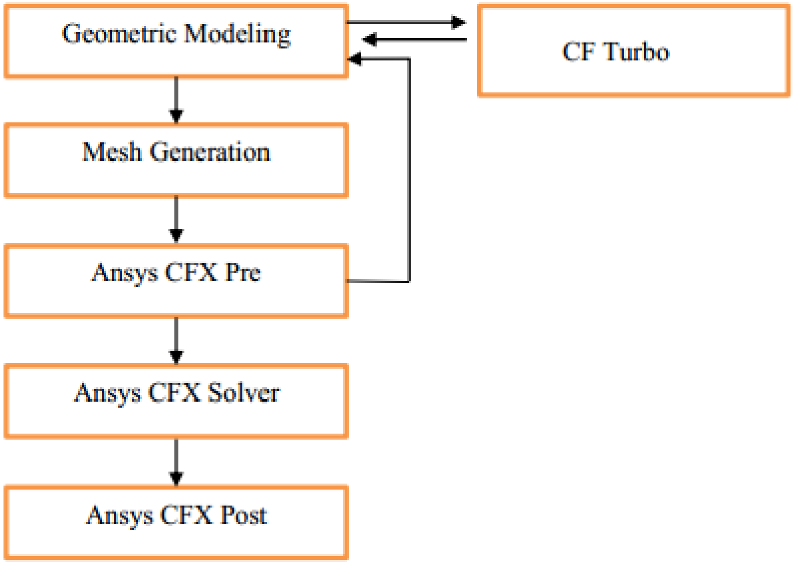
3.6 Principal Steps of CFD Simulation of PAT

ANSYS CFX is a commercial 3D Navier-stokes CFD code that utilizes a finite-element-based finite-volume method to discretize the transport equations. It is a completely embedded solver, thus it creates no time step restriction and is measured easy to recognize. It is also a united solver meaning that the momentum and continuity equations are solved. This approach reduces the number of steps required to obtain convergence and no pressure improvement term is required to retain mass alteration, leading to a healthier and correct solver.

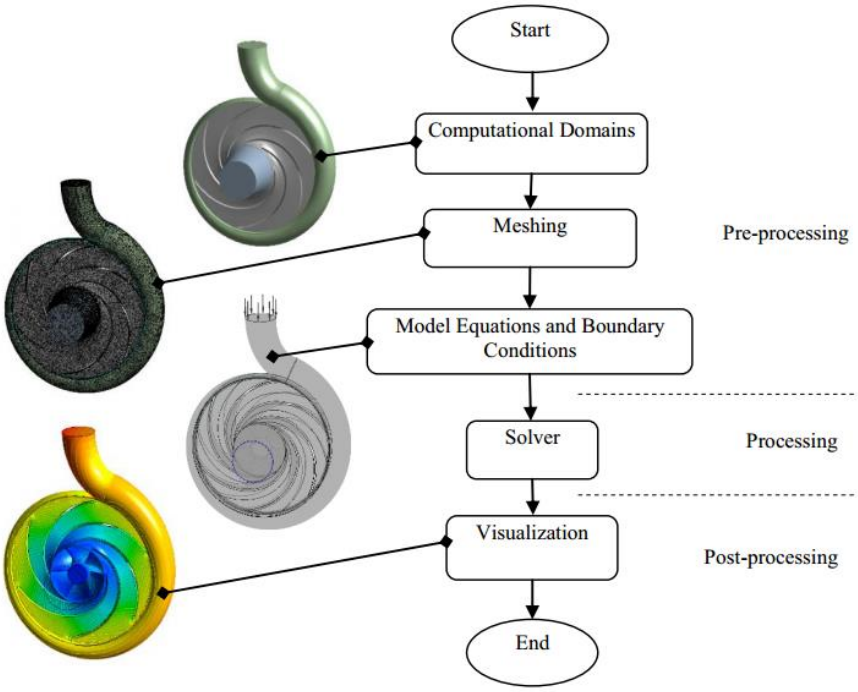
Computational Fluid Dynamics (CFD) is a useful technique that's used to perform fluid flow analysis once a model and finite volume grid on the model is generated. The general steps undertaken for problem analysis using CFD include defining boundary conditions, fluid properties, executing the solution and displaying the result.

The steps undertaken to perform a CFD analysis of Pump as Turbine (PAT) are summarized in the figure below.

Figure 19: Flow chart for simulation process



A



B

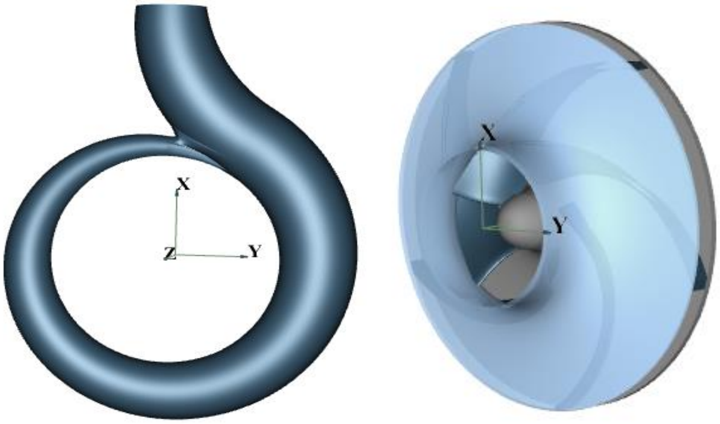
3.6.1 Geometric Modeling

Modeling of the geometry of a centrifugal pump running in turbine mode involves defining the Impeller and volute components. Geometric modeling of draft tube is also considered since it is one of the major component in PAT system.

3.6.2 Impeller and Volute

The volute and the impeller are generated using CF turbo. CF turbo is software package used to model and design turbo machineries like pumps, ventilators, compressors, turbines interactively. The software is easy to use and does enable quick generation and variation of impeller and volute geometries. Several models can be displayed, compared and modified simultaneously.

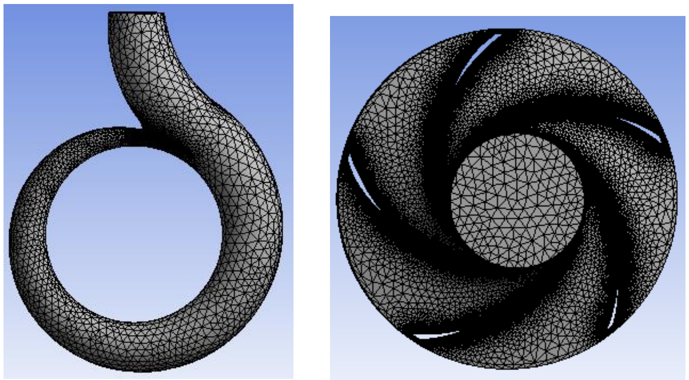
Figure 20: CF turbo impeller and volute design



3.6.3 Mesh Generation

Meshing is the process of discretizing of a region under consideration in to a set of control volume. Mesh for different components of the PAT system like volute casing and impeller fluid volumes are generated in Ansys ICEM. After modeling of the geometry of impeller and volute is finished, the two components were exported to design modeler in Ansys workbench using “.stp” format. As it can be seen from the workflow diagram in Figure, since the volute casing and impeller are imported together, they are meshed in a single meshing cell.

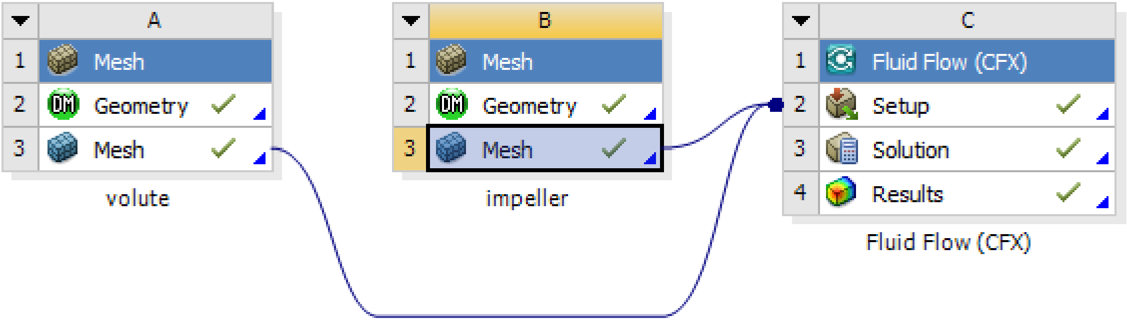
Figure 21: Ansys cfx volute and impeller meshing



3.6.4 Pre-processing in ANSYS CFX

The pre-processing is made using ANSYS CFX .After the volute and impeller components are meshed, mesh cell displays a right mark to show meshing has completed. Then the mesh files are automatically transferred to the setup cell of the project scheme. After the mesh file is transferred to the setup cell, preprocessing starts in Ansys CFX. The transferred mesh file contains information about the boundary name.

Figure 22: Ansys cfx project schematics



Analysis Type

In each "Flow Analysis" in ANSYS Pre there is a tab called "Analysis Type". This is where whether steady-state or transient simulation is defined, and Thus for PAT simulation case, the steady state option was chosen.

Domains Definition

For the simulation of the PAT alone, two fluid domains were created; one for the stationary volute and the other for the rotating impeller by clicking the “domain” icon on the toolbar. After clicking the domain icon, the domain details view appears in the workspace. On basic setting tab in the workspace, the reference pressure, turbulence model, fluid type and domain motion are defined. The reference pressure in both domains was set to 1 atm. The k-ε model’s selected for turbulence model with scalable wall function. The fluid type for both domains is selected from the CFX data base.

The domain motion for the volute domain is set to stationary while for the impeller domain, the domain motion is set to rotating and angular velocity is specified. The impeller domain is made to rotate at angular velocity 157.08 rad/s (1500 RPM). In this case the angular velocity is positive because the domain was modeled to rotate in the positive rotation direction about the z-axis.

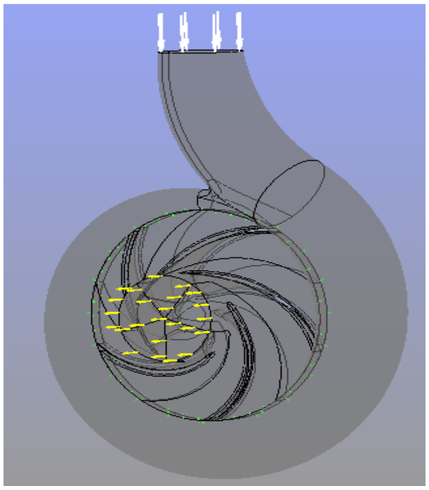
Setting Boundary conditions

The inlet and outlet boundary condition were set by imposing a constant 0 Pa total pressure on the casing inlet surface and variable mass flow rate on the impeller outlet surface respectively. 5% medium turbulence intensity for the inlet conditions is considered. For the simulation where draft tube is involved, the boundary condition imposed on impeller outlet is used on draft tube outlet surface. Figure shows the PAT model after all boundary conditions imposed on the fluid domains.

Table 12: Summary of boundary conditions for PAT model

Domain Name	**Domain motion**	**Surfaces**	**Boundary Condition**
Volute	Stationary	Volute casing wall	Smooth no slip wall
		Volute casing Inlet	Inlet
Impeller	Rotating	Impeller Hub	Smooth no slip wall
		Impeller Shroud	
		Impeller Blade	
		Impeller Outlet	Outlet

Figure 23: PAT model with boundary conditions



Setting Interface

There exists one interface between two domains when simulating a PAT model. The interface is found between the stationary volute and rotating impeller domain. When draft tube is included in the simulation, a second interface exists between stationary draft tube and rotating impeller domain. Interface between domains is created using the “interface” icon on the toolbar.

Between volute and impeller domain, the interface is created first by defining the interface type between these domains. The interface type is set to fluid-fluid interface. The next step is to define the interface sides on which connection between the domains surface exists. This is done by selecting individual domain name and the location of both domains. For the first interface side the volute domain is selected and “volute interface‟ surface is selected as location whereas the “impeller interface 1” surface on the impeller domain is selected for the second interface side.

The General Grid Interface (GGI) connection option is chosen as interface model to be used on the two separate individual rotating and stationary domains. Frozen rotor is used for the frame change/mixing model. The same technique is used to create the interface between draft tube and impeller domain.

Solver Control and Output

Setting solver control in Ansys CFX involves defining advection scheme, turbulence numeric and convergence criteria. The default advection scheme, high resolution, is chosen because it is recommended by ANSYS. For simulations involving liquid pumps and turbines.

First order upwind is used for turbulence numeric option since this setting gives the most robust performance of the CFX solver. The residual is the most important measure of convergence, as it relates directly to whether or not the equations have been solved. A residual level of $10^{-4}$is considered as a relatively convergence criteria by ANSYS.

Solution Initialization

Initialization in Ansys CFX is done by providing initial guess values to solve the governing equation so that the flow field variables can be solved by iteration toward the solution. By clicking the “Global Initialization” t=0 icon on the toolbar, the initial values are specified. The default automatic initialization for the velocity and static pressure is used to provide a start point to the solution.

3.6.5 Processing in Ansys CFX Solver

After defining the required boundary condition and solution initialization on the PAT fluid domain, the preprocess stage in Ansys CFX is completed. In this case, setup cell displays a “right mark‟ to show setup is finished. Then the setup files consisting of the necessary information to solve the PAT model is transferred to solution cell of the project scheme. When solution starts, plots of residual and other solution monitors are displayed on separate tab. One of the most important features of ANSYS CFX is, it uses a coupled solver in which all the hydrodynamic equations are solved as a single system. The coupled solver is faster thus less iteration is required to obtain a converged flow solution.

3.6.7 Post Processing in Ansys CFX Post

The CFD-Post for ANSYS CFX, is a flexible, state-of-the-art post-processor for CFD products. It is designed to allow easy visualization and quantitative analysis of the results of CFD simulations once the solution converged. It supports a variety of graphical objects and locator objects that are used to create post-processing plots and quantitative calculation.

The quantitative results for PAT in Ansys CFX post can be obtained using the built-in function calculator. By setting the locations, function and variable to be calculated, the numerical value of the variable can be obtained.

The Ansys CFX outputs used to evaluate the performance of PAT are:-

- Total efficiency
- Shaft power/ output power

4. Result and Discussion

4.1 Site selected

For this study Kality, Chefy, Bole Arabsa 2b and Bole Bulbula sites were selected and analyzed due to the high value of the specific speed of the pumps. Note that pumps of specific speed $n_{qp} < 15$ are not used as turbines. Efficiency of such impellers are low (long and narrow passages incurring friction losses) and the performance of such pumps as turbine cannot be predicted accurately. Also, during site visits it was observed that the topographic features on upstream side of the WWTP does not allow power generation options. When we see the location of these treatment plants, they are found near sewerage network and the elevation difference between the main line and the treatment plant is very small. Hence, this study mainly focuses only on downstream side of the treatment plants for power generation option.

4.2 Selection of turbines

Among the various available turbines, a comparison was done by evaluating their production capacity, efficiency, cost and flexibility of the technology. To help with the comparison, turbines with production capacity in the range between 5-6 kW were selected and analyzed.

Table 13: Turbine cost for between 5-6 KW powers in USD

turbine type	**head (m)**	**flow (l/s)**	**Efficiency %**	**output power (kw)**	**price (USD)**
Vertical Tubular	6	151	60	5	9834
Propeller	1-5.5	140	65-75	5.84	3700-4500
Cross flow	5	180	60	5	5000
Gravitational Water Vortex	1.2	700l/s	80	5	66905
PAT	45-63	50-60	80	5	1500

From selected and analyzed turbine types, Pump as turbine is the least cost turbine type.

The cost prediction formulas of Ogayar and Vidal (2009) was used to determine the cost per kW for various common turbines.

Table 11: Cost prediction results

Turbine	Initial Capital Cost of electro-mechanical equipment (Euro/W)
Pelton	5.94
francis	7.17
Kaplan	5.08

4.3 Simulation for Design Condition

4.3.1 Pump Mode Performance

In order to begin the investigation of the PAT performance, it is mandatory to check the pump mode performance of the model under consideration whether it coincides with the pump curve data. For this reason, the simulation of the model operating in pump mode is performed at the design rotation speed of 2900 RPM for all sites except kality site which was 760RPM in ANSYS CFX.

Then, comparison where made between the pump curve data and numerical result from the CFD.

Q-H pump characteristic

In a first phase, several simulations were made in the pump mode and the results were compared with the data declared by manufacturer. This first validation phase was useful to validate the simulation model. For this different head and flow data was simulated in ansys cfd and the pump curve is drawn. Here below are the results in table form.

Table 12: Pump curve and simulated pump curve efficiency comparison

H(m)	Q (m^3/hr)	Ansys CFD η (%)	Manufacturer pump curve η (%)
37	28.5	60.84	60
36.4	40	69.9	70
35.6	50.3	75.49	75
32.5	73	79.3	80
31	82.3	80.8	80.8
29.3	90.3	79.7	80

Bole Bulbula

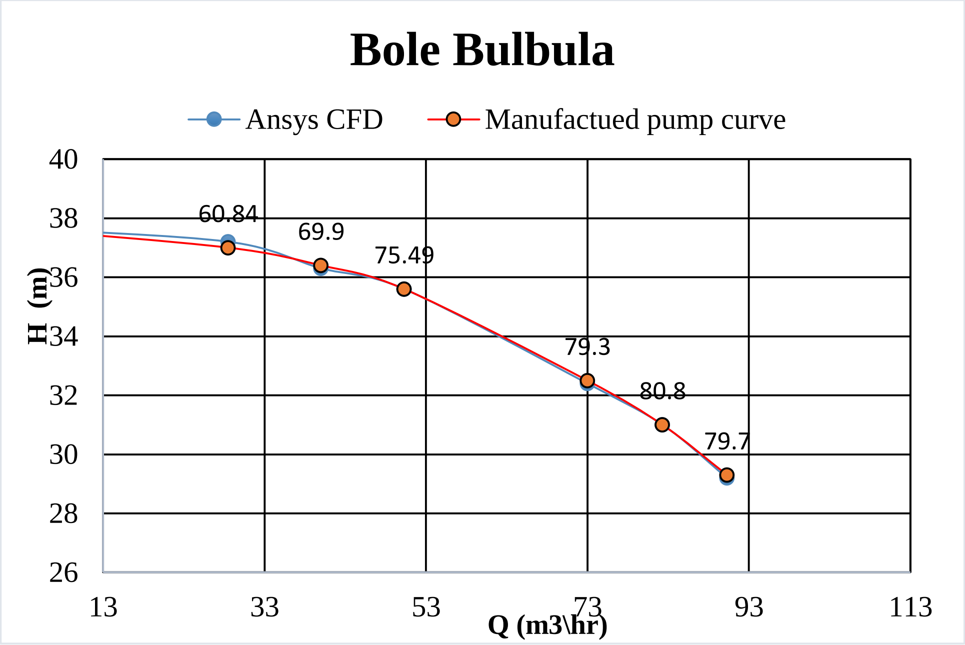
H(m)	Q (m^3/hr)	Ansys CFD η (%)	Manufacturer pump curve η (%)
26.8	104.5	79.72	80.9
30	77	77.14	78
30.2	66	76.47	75
30.5	55	70.24	70
30.8	40	59.66	60

Bole Arabsa 2B

H(m)	Q (m^3/hr)	Ansys CFD η (%)	Manufacturer pump curve η (%)
24	**72**	**49.45**	**50**
23.4	**98**	**60.3**	**60**
21.3	**131**	**70.67**	**70**
17	**192**	**77.1**	**76.4**

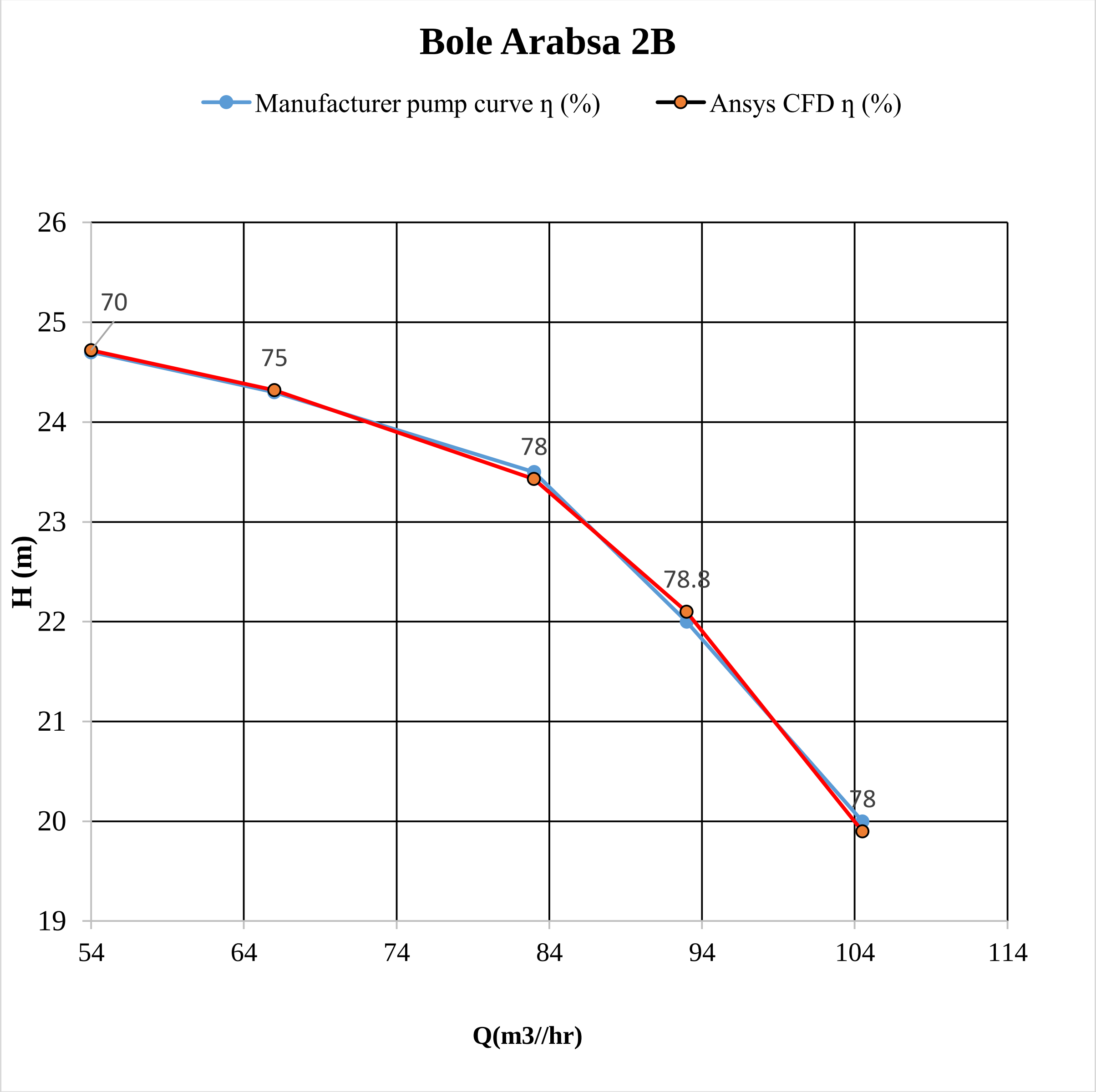
Chefy

In the tables above, results in terms of total head versus volumetric flow-rate are shown. The manufactured pump curve and the simulated pump curve are in very good agreement and the error, in all examined conditions, it is less than 2%. The model accuracy is verified.



(a)

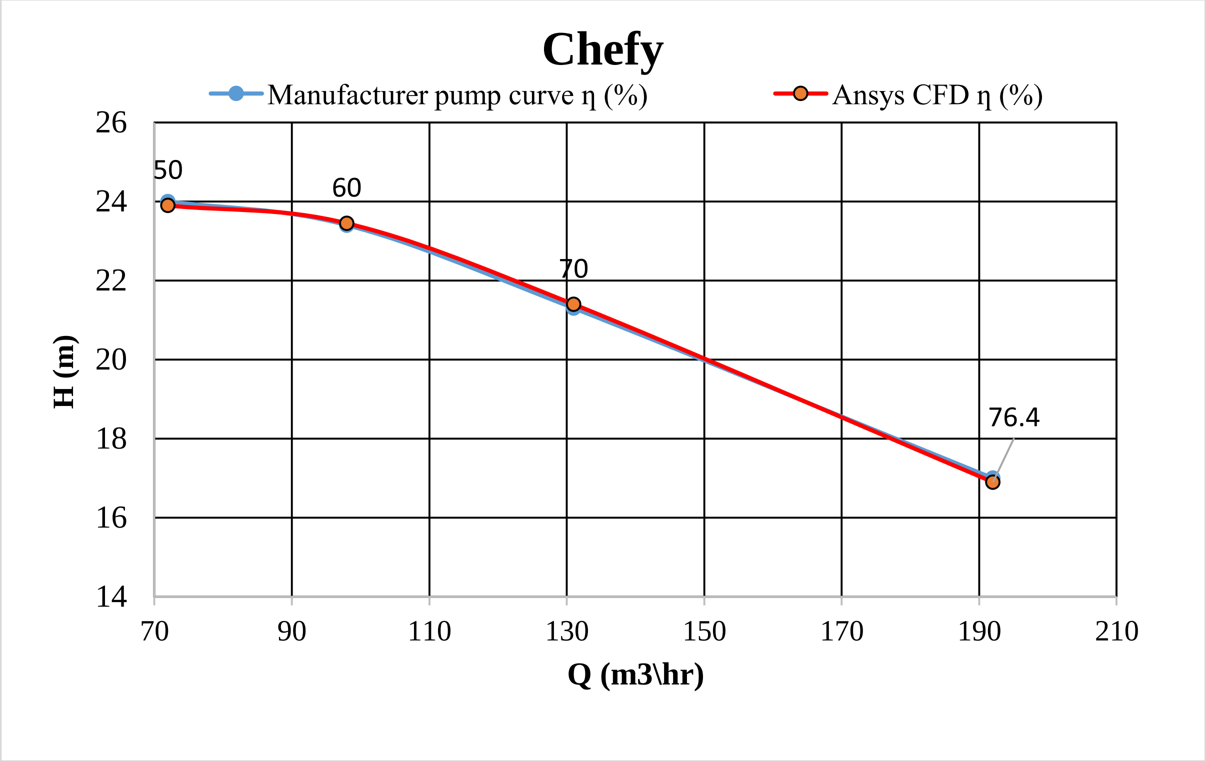
For the direct mode, CFD models have been validated using the data supplied by the pump manufacturers. In Figure 24(a), the head vs. flow rate plots (as the blue curves) are shown. Across the range of flow rates (13–93) m^3/h, the head varies from 29 m to 37 m. In the plots in Figure 24(a) , the model results are shown in red. The comparison in Figure 24(a) demonstrates the accuracy of the adopted methodology; in fact, the percentage error is always less than 2% while for many points the error is near zero.



(b)

For the direct mode, CFD models have been validated using the data supplied by the pump manufacturers. In Figure 24(b), the head vs. flow rate plots (as the blue curves) are shown. Across the range of flow rates

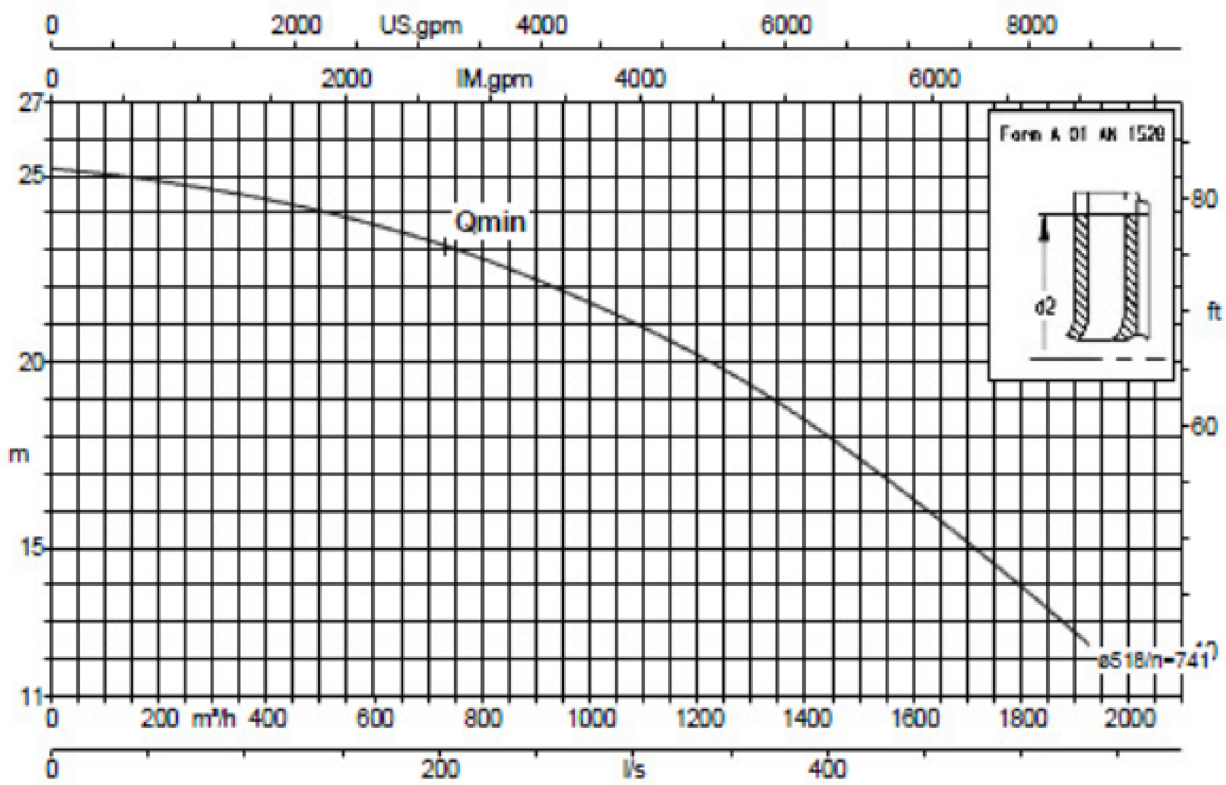
(54–104) m^3/h, the head varies from 20 m to 5 m. In the plots in Figure 24(b) , the model results are shown in red. The comparison in Figure 24(b) demonstrates the accuracy of the adopted methodology; in fact, the percentage error is always less than 2% while for many points the error is near zero



(c)

For the direct mode, CFD models have been validated using the data supplied by the pump manufacturers. In Figure 24©, the head vs. flow rate plots (as the blue curves) are shown. Across the range of flow rates (70–192) m^3/h, the head varies from 17 m to 4 m. In the plots in Figure 24(c) , the model results are shown in red. The comparison in Figure 24 (c) demonstrates the accuracy of the adopted methodology; in fact, the percentage error is always less than 2% while for many points the error is near zero.

For Kality site two axial flow pump were used. Each with 1560m^3/day capacity. The specific speed for the pump was 741rpm. The head for this pump was 17.4m.



(d)

Figure 24: Model validation pump mode

Using the validated model, the reverse mode was analyzed. The boundary conditions were changed in order to running the pump as a turbine.

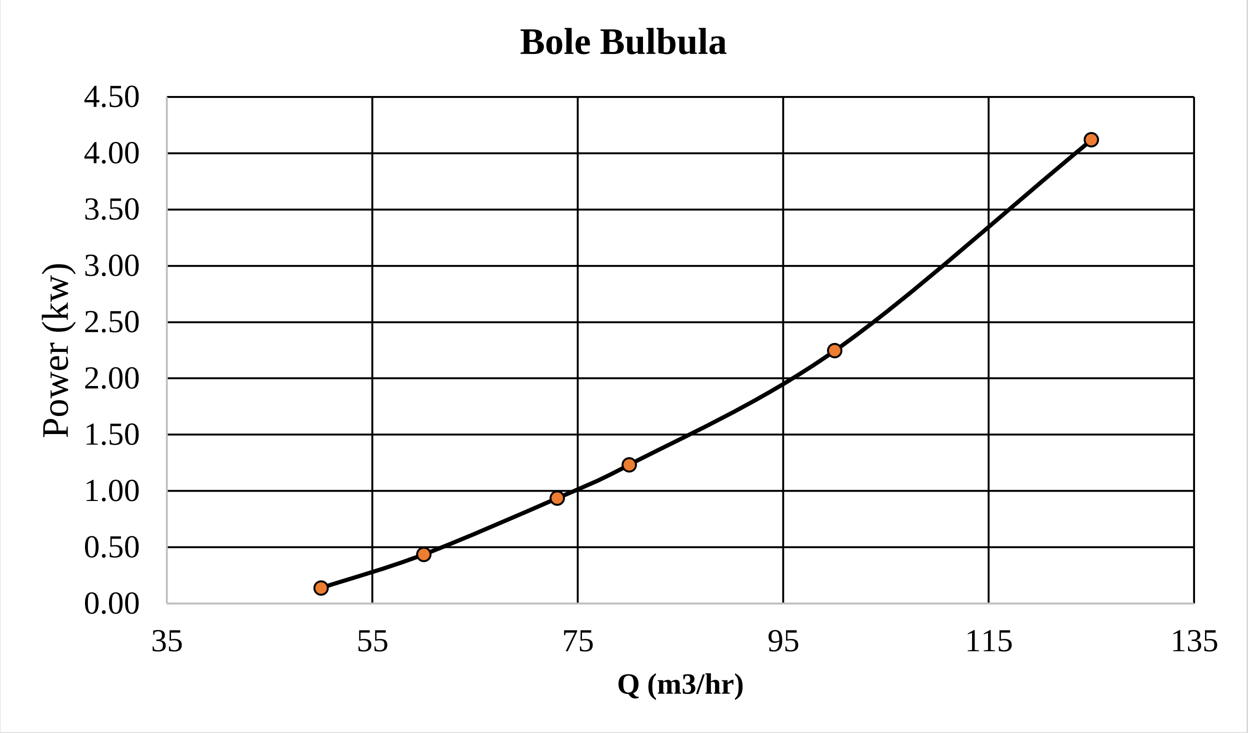
4.3.2 Predicted Performance of the PAT

After simulating the behavior of the pump operating in turbine mode at different flow rate values, the results obtained are graphically and numerically displayed. Based on the discussion made on the selection of PAT for selected Pico hydro site, the PAT is simulated in accordance to the selected synchronous speed of a generator. Thus simulations of PAT were carried out at fixed rotational speed of 760 and 1500 RPM (157.08 rad/s) and with same procedure.

The numerical results obtained from Ansys CFX are used to calculate the performance parameters of the PAT. These parameters are useful to identify the behavior of the PAT at different flow rate conditions.

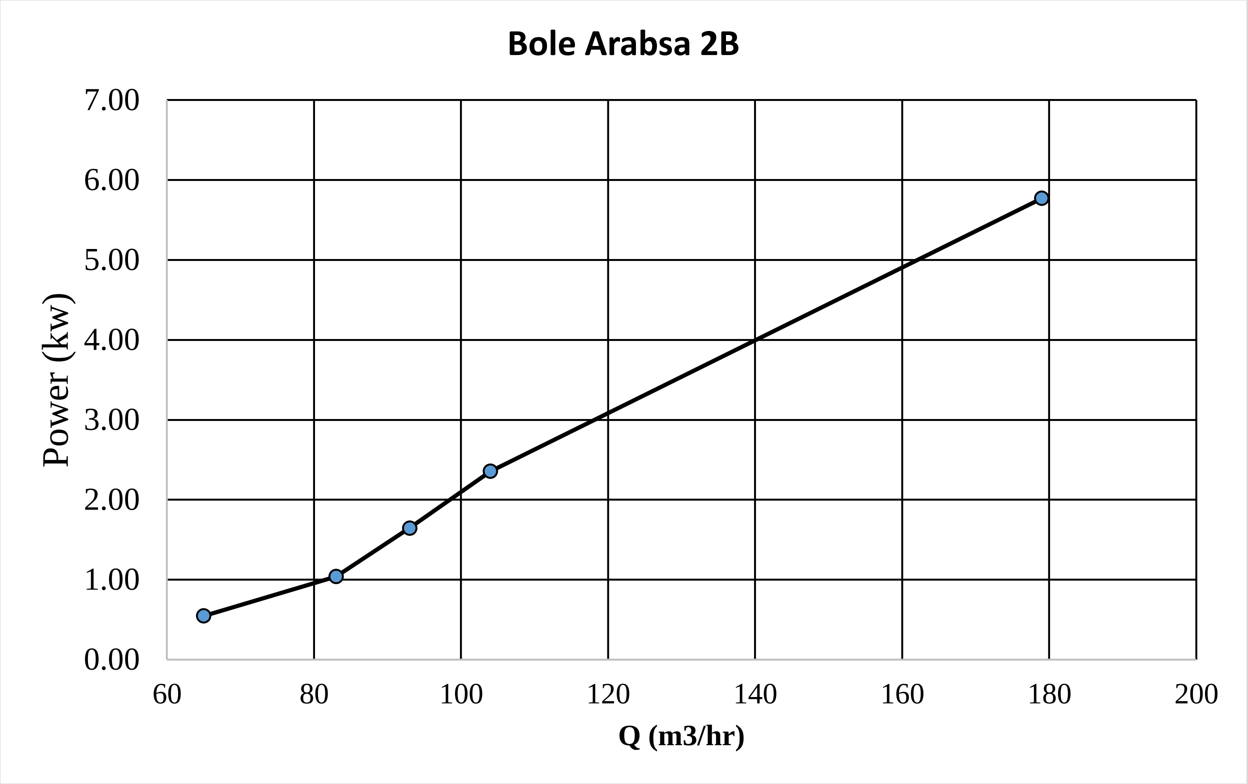
These parameters include the head, power, and the non dimensionless coefficients like head, flow, and power coefficient.

Figure 25: Pat power Vs flow rate



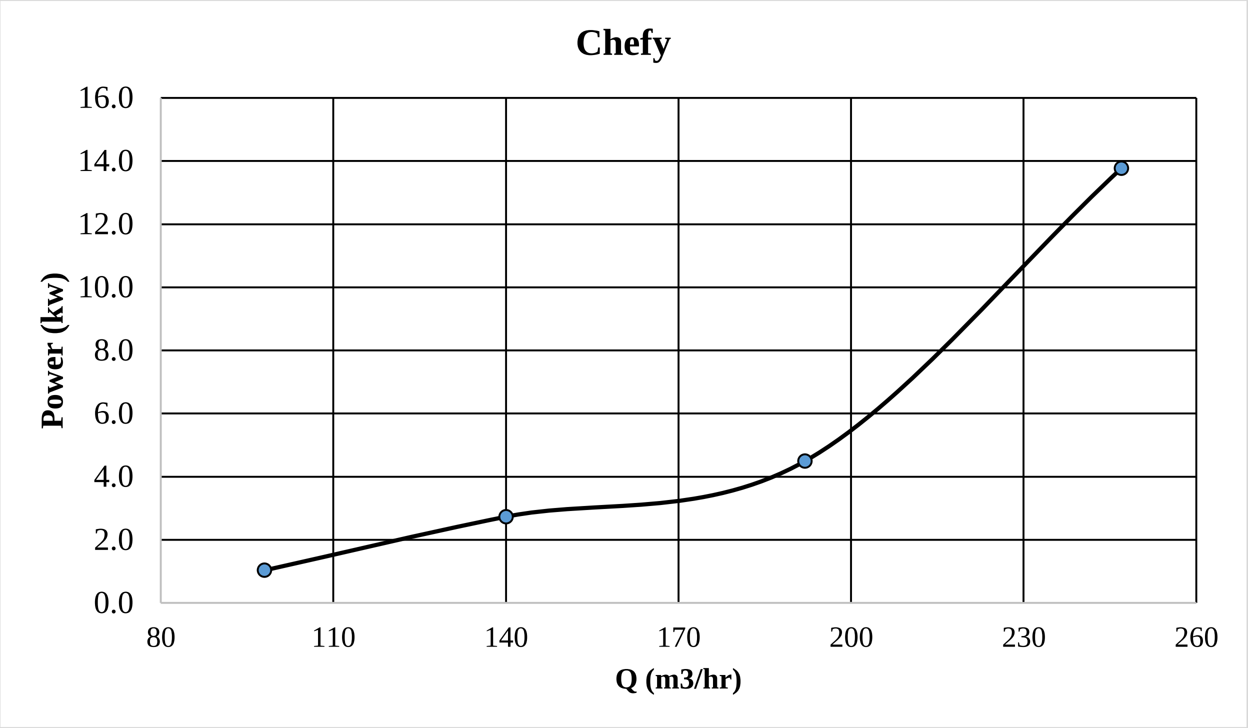
(a)

For the revers mode, CFD models for bole bulbula has been validated using the chapallaz design results. The result from chapallaz method is within the CFD result range. In Figure 25(a), the power vs. flow rate plots are shown. Across the range of flow rates (50–125) m^3/h, the potential power output is upto 4.1 Kw.



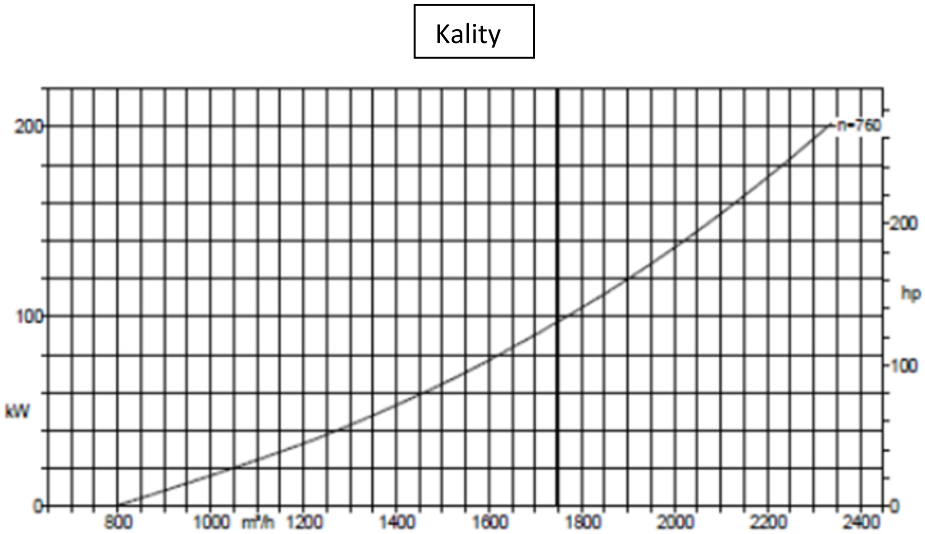
(b)

For the revers mode, the CFD models for bole arabsa 2b has been validated using the chapallaz design results. The result from chapallaz method is within the CFD result range. In Figure 25(b), the power vs. flow rate plots are shown. Across the range of flow rates (65–180) m^3/h, the potential power output is upto 5.77 Kw.



(c)

For the revers mode, the CFD models for chefy site has been validated using the chapallaz design results. The result from chapallaz method is within the CFD result range. In Figure 25(c), the power vs. flow rate plots are shown. Across the range of flow rates (100–250) m^3/h, the potential power output is up to 13.8kw for the one unit. Totally we have two units so our total potential power production will be 27.6kw.



For Kality site because the pump manufacturer has provided the turbine performance curve. The power production potential for this site was compared with chapallaz design results and the result is found to be in good agreement or in the range of the chapallaz method. According to chapallaz result for two turbine unites the total power generated was (122.68-174.2) kw. And according to the CFD simulation the power production capacity was 147kw.

4.4 Selected sites Installed capacity and potential power production

Assuming the power plan will work throughout the year. The following calculations were done,

Table 13:- Annual power generated in KWh & % of coverage of the power demand of wwtp

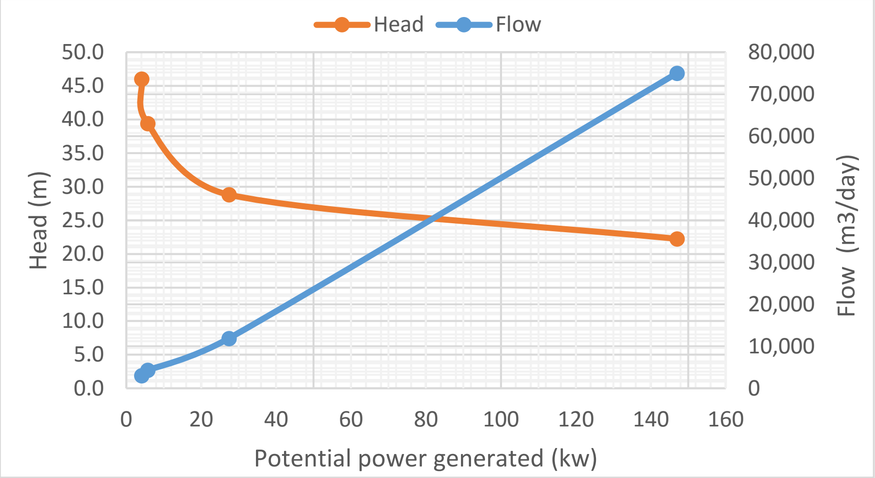
Site Name	Installed capacity (kw)	Annual Power demand (kwh)	Potential power generation from wwtp (kw)	Annual power generation (kwh)	Percentage of coverage by the generated power
Bole Bulbula	240	2,102,400	4.1	35,916	1.71
Bole Arabsa 2B	309	2,706,840	5.7	49,932	1.84
Chefy	213	1,865,880	27.4	120,012	6.43
Kality	250	2,190,000	147	1,287,720	58.80

In the above table, its clearly Shown that the power generated by the energy recovering turbines (Pat)s especially for kality site can generate more than half of the power demand of the treatment plant.

4.5 Relationships between head, flow and power generated

The graph below shows the relationship between head and flow with power. The graph was drawn by keeping the tariff constant. The two lines intersect at 25m head, at 40,000 m^3/day and 82 kW. This point and the region above the point is the optimal location to make the site economically feasible.

Figure 26:- Relationship between head, flow and power generated



4.6 Comparison of power generation from wwtp with similar equal capacity renewable energy sources

In order to generate a continuous electric power using (Irrigation canals, wind turbines and solar panels), all of the listed renewable energy sources have their own limitations. For instance hydropower generation in irrigation canals is greatly affected by the stream size and the crop water requirement, as the flow varies depending on the irrigation schedule and seasonal demand, the power output from these infrastructures will fluctuate greatly. Usually, summer months tend to present less water flow which results in less energy produced

When we see other renewable sources like solar and wind energy sources they are too have limitation on power generation. For instance for power generation from winds, these type of source of energy will fluctuate greatly due to the unreliability of winds and because of these the turbines usually function at about 30% capacity. And these intern will result in less energy production. Also when we look into the performance of solar energy source, it depends greatly on the radiation energy which is only found during day time and also it will be affected by the weather condition of the day. These factors limit the amount of power productions.

5. Conclusion and Recommendation

5.1 Conclusion

This paper presented a method to estimate the energy potential of pumps used as turbines (PATs) in order to exploit the available hydraulic energy from WWTP. The methodology to analyze a Pump as Turbine consists of building up a tridimensional CFD model of a pump and to obtain the reverse mode characteristics, running the model as a turbine.

The simulation model was validated both in direct and reverse mode. In direct mode, the model was validated using the pump manufacturer data, available in the pump data-sheet, while in reverse mode the data is validated by comparing the simulation result with Chapallaz design results. According to the results obtained following the design procedures of chapallaz, the pat power output capacity was found to be in the good range with the simulated results.

The potential power generation capacity of the selected four WWTP were in the range of (4.1-147) kw. Also the potential generated power can cover (1.7-58.8) % of the power demand of these WWTP. By doing these it will be possible to cover more than half of the power demand of Kality wastewater treatment site. Also we can save up to 842,168.9 Birr and more per year from power production from these facilities.

Also by keeping the tariff constant a relationship was developed between flow rate, head and power. For those sites which has a flow rate of 0.463 m^3/s and a head of 25m and above are economically feasible site for hydropower generation.

When we compare the power generated from wastewater treatment plants with other renewable energy sources such as (hydropower in irrigation canals, solar energy and wind energy), the power generated from wastewater treatment plants have a great advantage over the other type of energy sources, as wastewater treatment plants will run continuously throughout the year due to availability of water in these infrastructures. Which means wastewater treatment plants can generate power throughout the year and this make them a preferable choice among other renewable energy sources.

5.2 Recommendation

- The wastewater treatment plants are not working with their full capacity, more detailed study should be done in the coming years.
- The current electricity tariff price should be increased more than the current rate.
- Though the implementation of a hydroelectric turbine system may not be feasible for every wastewater treatment facility, there are other options for implementing renewable energy. Like biogas systems that could make the investment more worthwhile.
- Enacting Policies that favorers renewable energy sources.
- Experimental investigations are still indispensable when an exact knowledge of pump as a turbine characteristics is required.

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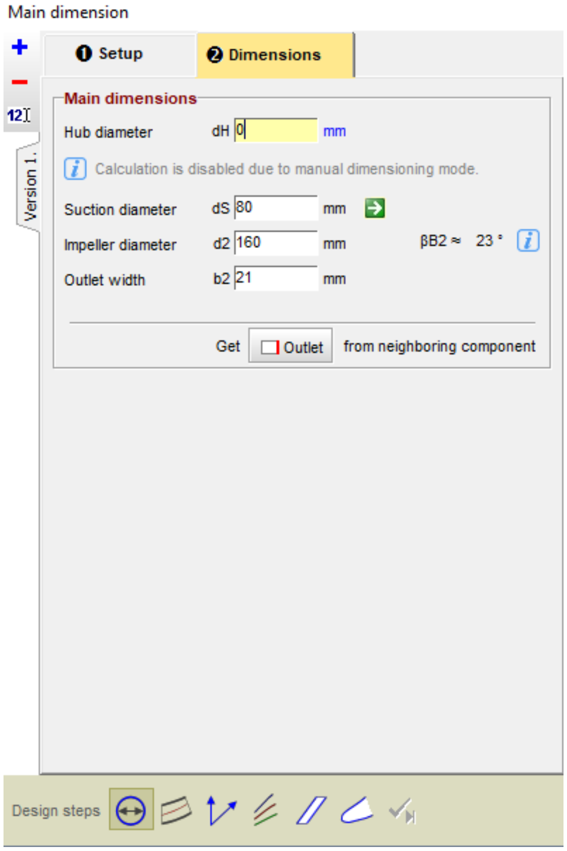
Appendix-1: Geometric modeling of PAT components

1.1 Impeller and Volute modeling using CF turbo

When the 3D impeller geometry is modeled, the main dimensions menu item is used to define main dimensions of the impeller. By clicking “main dimension” icon on toolbar, the main dimensions of the impeller like hub diameter, suction diameter, and impeller diameter are manually inserted in the box using the manual dimensioning mode. All these values are user defined input values.



Main dimensions

Hub diameter dH 0 mm

Calculation is disabled due to manual dimensioning mode.

Suction diameter dS 80 mm

Impeller diameter d2 160 mm βB2 ≈ 23 °

Outlet width b2 21 mm

Get Outlet from neighboring component

Figure 27: Main dimension GUI in CF turbo

The meridional contour is the second important step to define the geometry of impeller. In this way the blade position between the hub and shroud can easily be created with a reasonable accuracy. In Meridional contour menu, Graphical elements can be manipulated by the computer mouse per drag and drop .The final geometry of meridional of the meridional view looks like in Figure A-2. The trailing edge is fixed on meridional outlet.

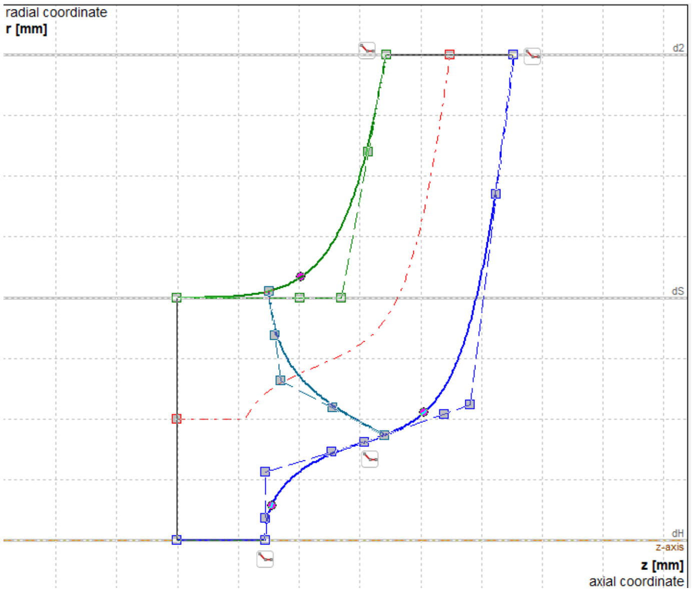
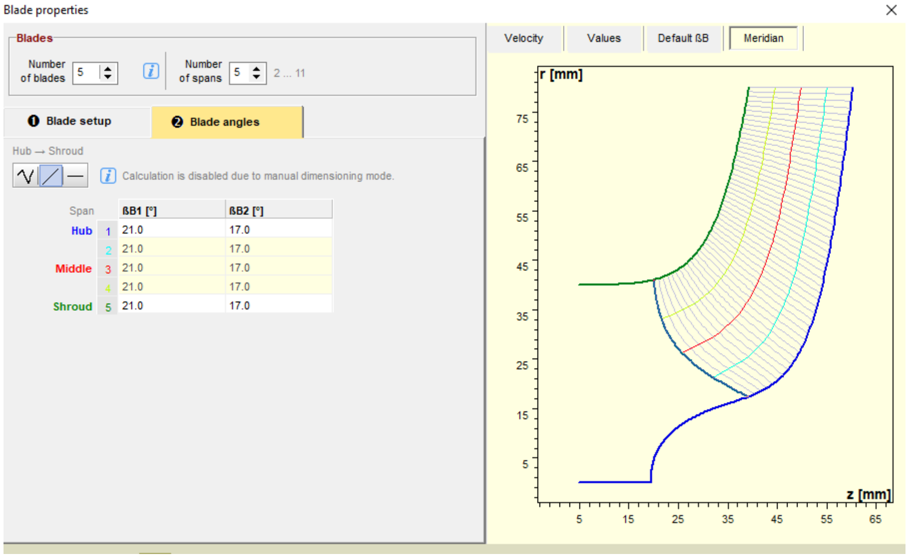


Figure 28: Meridional view of the selected pump impeller

The next step is to define the blade thickness, blade angle and blade number using the “blade property” icon on the toolbar. Definition of blade properties is made in two steps; first the blade angle is defined in the “blade angle” tab while the blade number and blade thickness at the leading and trailing edge of the corresponding hub and shroud is defined in “blade setup” tab. Figure A-3 shows the dimensions set for the blade angles and blade thickness at the leading edge and trailing edge on the blade angel and blade setup tab respectively.


Blade properties

Blades

Number of blades: 5

Number of spans: 5 (2 ... 11)

❶ Blade setup ❷ Blade angles

Hub → Shroud

Calculation is disabled due to manual dimensioning mode.

Span		ßB1 [°]	ßB2 [°]
Hub	1	21.0	17.0
	2	21.0	17.0
Middle	3	21.0	17.0
	4	21.0	17.0
Shroud	5	21.0	17.0

Figure 29: Blade properties GUI in CFturbo

Then we proceed in defining the blade edge of the impeller at the leading and trailing edge. This is done by using the "blade edge" icon on the toolbar. The blade edge at the leading edge is created using elliptical element by specifying the ellipse axis ratio. Thus the leading edge is made into curved edge while trailing edge is provided with a trimmed or sharpened tip. The edge geometry at the leading edge is as described in Figure A-4.

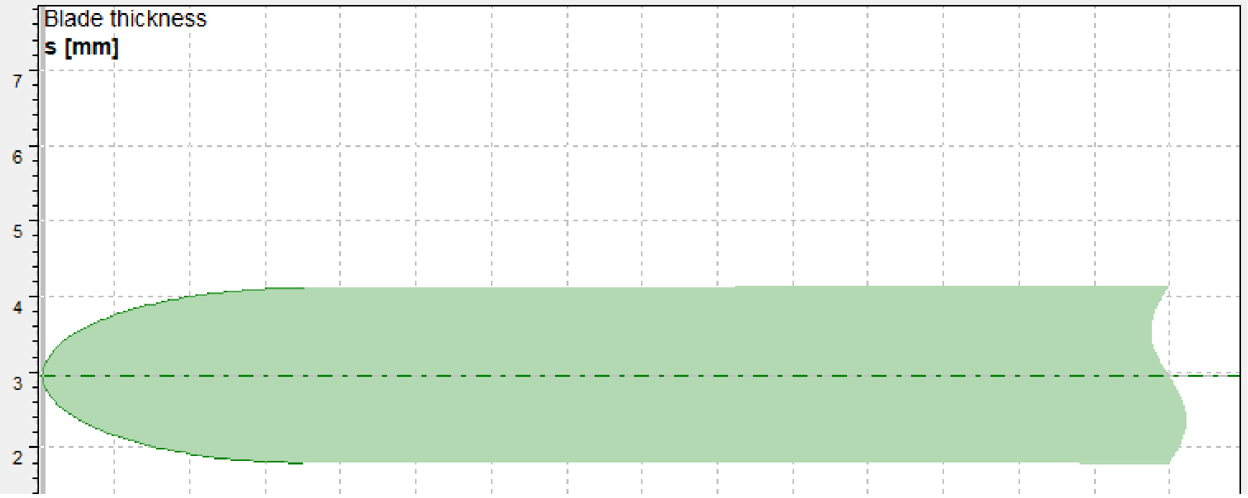


Figure 30: Leading edge profile of impeller blade

Finally after all the necessary dimensions are fed, the complete geometry of the impeller is created. The final step is to add CFD setup extension to the Impeller. The extension defines the impeller-volute interface .The extension is added in meridional direction at the outlet. Figure A-5 represents the final modeled geometry and CFD fluid volume extracted from the impeller model.

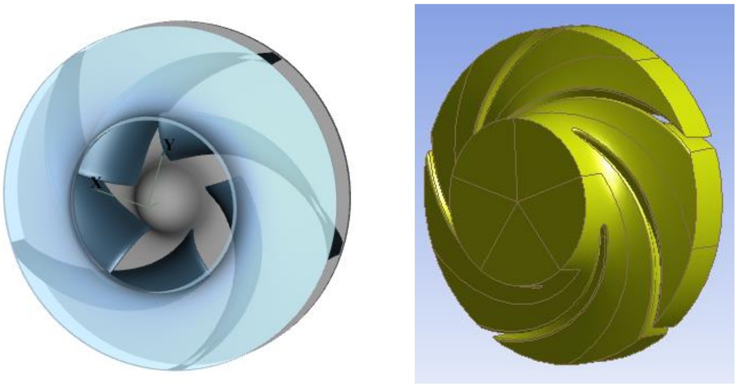
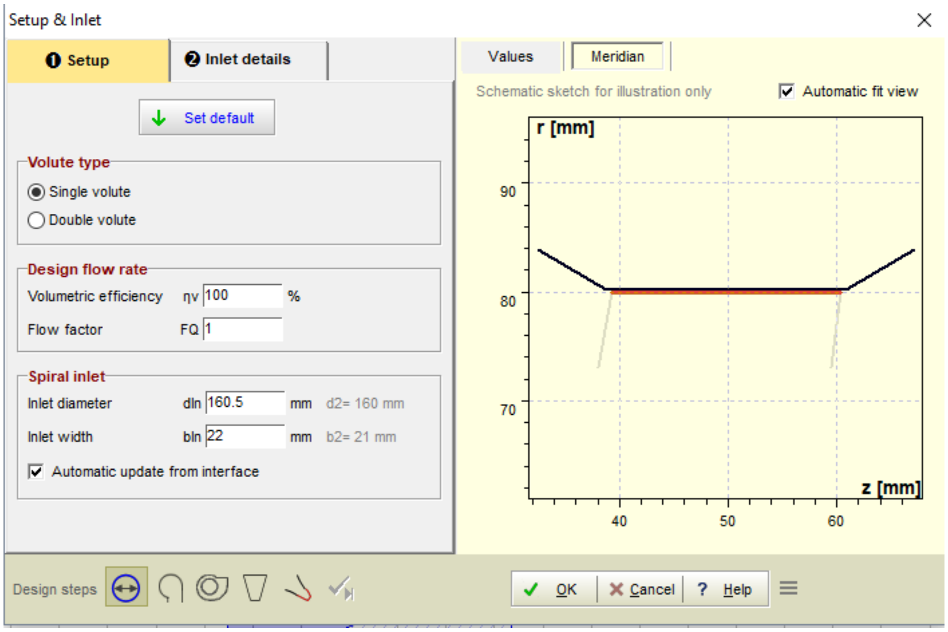


Figure 31: 3D model of a centrifugal pump impeller and CFD fluid volume

The second component to be modeled is the volute casing. When the geometry of the volute is created, the cross section, spiral profile, cut water, and the diffuser should be provided to get the required geometry. By clicking "Project" tab on the main window menu and using "add component" button, the volute component is added

The first step to model the volute is to define the inlet side using the inlet definition icon. It consists of two steps: defining the details inlet interface is first specified on the "Inlet" tab page; this is done by manually putting the radial and z-axis coordinates of hub and shroud tips of the impeller. Defining the volute diameter and width of the spiral design is the second step is to be specified on the "Volute" tab; this is used to define the spiral inlet diameter and width. The GUI for the Inlet definition tab is show in Figure A-6.



Volute type
● Single volute
○ Double volute

Design flow rate
Volumetric efficiency ηv 100 %
Flow factor FQ 1

Spiral inlet
Inlet diameter dIn 160.5 mm d2= 160 mm
Inlet width bIn 22 mm b2= 21 mm
☑ Automatic update from interface

Figure 32: Inlet and volute definition GUI of CF Turbo

Definition of the cross-section of the volute is the succeeding step in defining the volute. The shape of the cross-section can be selected using the "cross-section" icon on volute toolbar menu. The cross-section of the volute is made horseshoe shaped. Figure A-7 shows the cross-section of the volute casing. The shape of the horseshoe cross-section of the volute is created by selecting the "**Round**" from the cross-section type option at the right side.

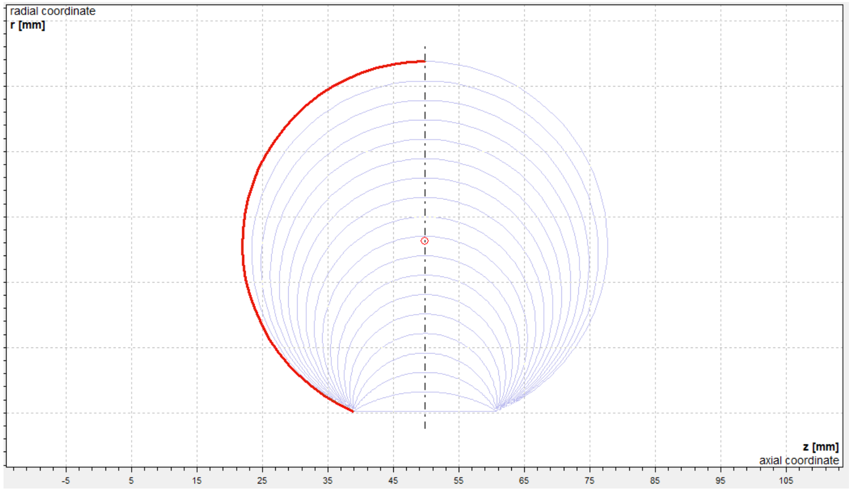
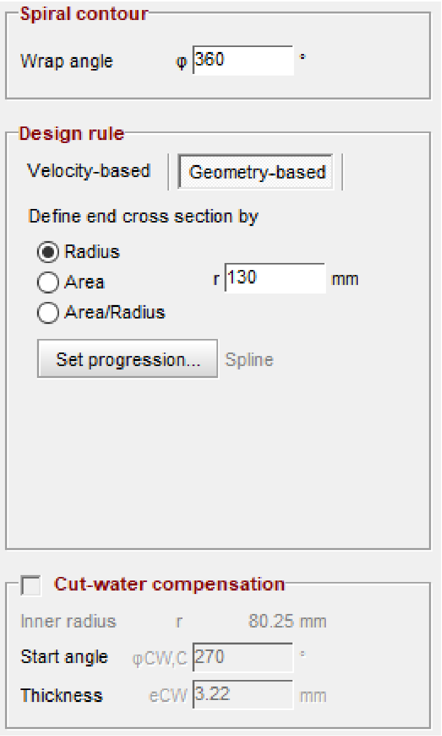


Figure 33: Cross-section of the volute

The third step is to define the spiral profile of the volute. The “spiral area” icon is used to development spiral profile. In spiral development first the wrap angle is set in to 360°.The end cross- section of the spiral is defined by specifying the radius. Figure A-8 shows user defined values set within the dialogue box for the wrap angle and radius based end cross-section definition. Figure A-9 shows final spiral profile of the volute casing.



Spiral contour

Wrap angle φ 360 °

Design rule

Velocity-based | Geometry-based

Define end cross section by

● Radius

○ Area r 130 mm

○ Area/Radius

☐ Cut-water compensation

Inner radius r 80.25 mm

Start angle φCW,C 270 °

Thickness eCW 3.22 mm

Figure 34: Dialogue box to define wrap angle and spiral design

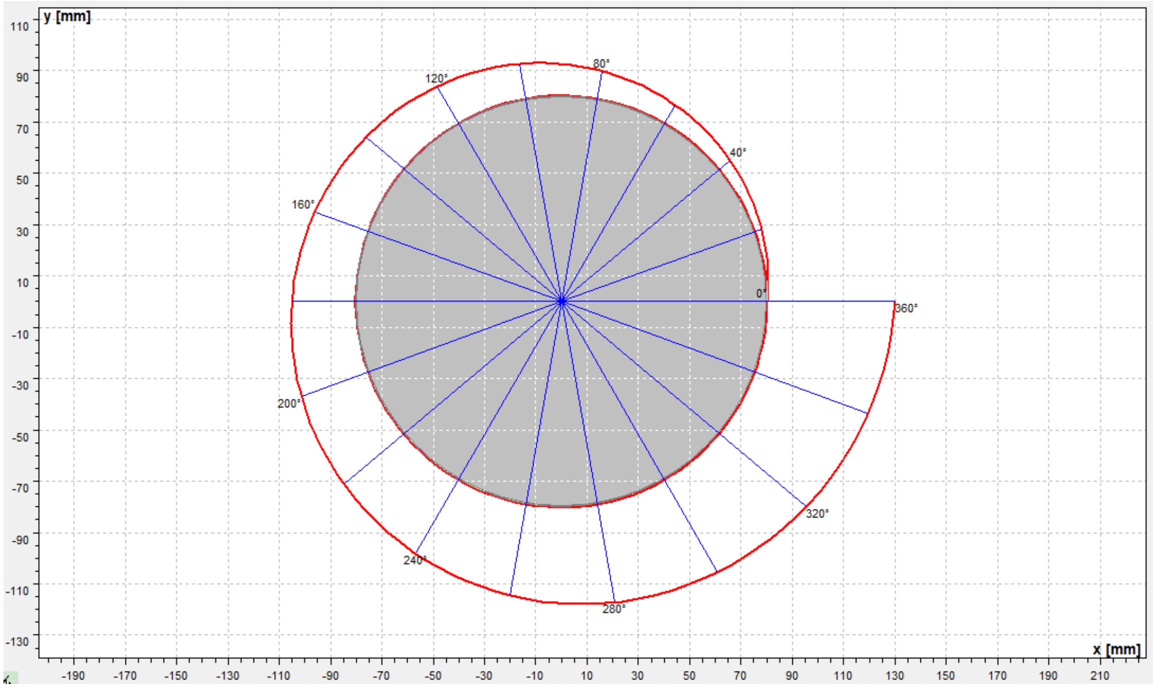
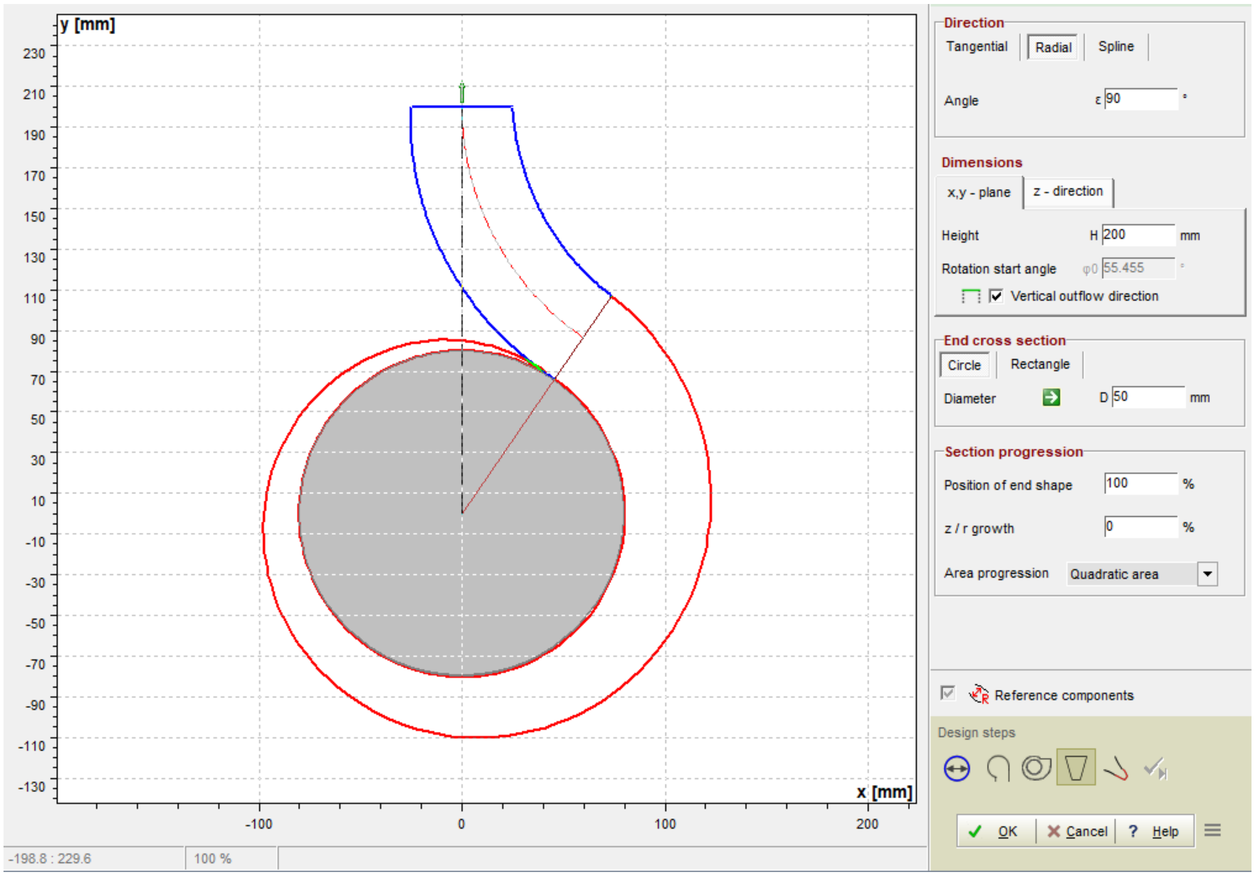


Figure 35: Spiral profile of the volute

The fourth step is to model the diffuser of the pump using “diffusor icon” on the volute toolbar. The geometry of the outlet diffuser is made to extend radial direction. In the case of a radial diffuser, the angle between the outlet branch and the line connecting impeller-center and outlet branch center should be specified. The angle is made 90^{o} whereas the diameter of the diffuser is set to 160 mm from the center line of the pump and by making the diffuser outlet a circular cross section, diameter is set to 50 mm. The final step is to define the cut water by setting the fillet radius and the angular position. Figure A-10 shows the GUI used to insert the necessary dimension of the diffuser. The complete process of modeling of the volute ends when the fluid volume is extracted from the volute casing. Figure A-11 shows the fluid volume defined with the volute casing.



Angle ε 90 °

Height H 200 mm

Rotation start angle φ0 55.455 °

☑ Vertical outflow direction

Diameter D 50 mm

Position of end shape 100 %

z / r growth 0 %

Area progression Quadratic area

☑ Reference components

-198.8 : 229.6

100 %

Figure 36: GUI for defining pump diffuser

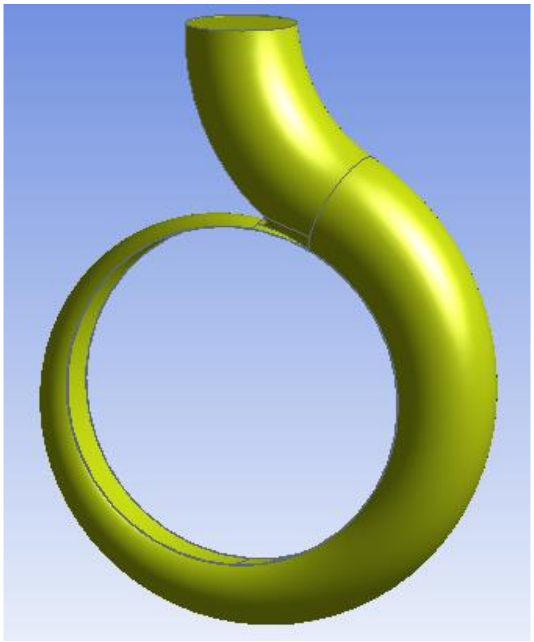


Figure 37: Fluid volume of volute casing

Appendix-2: Mat lab Codes for selecting a pump to be used as turbine

2.1 Pump Selection Procedure proposed by Chapallaz for bulubla

```
>> g=9.81;                                  %Gravitational acceleration m/s^2
rho=1000;                                   %Density of water in kg/m^3
Qnt=0.035;                                  %Nominal turbine flow rate in m^3/s
Hnt=46;                                     %Available site head in m
Nt=1450;                                    %selected nominal PAT speed in RPM
nst=(Nt*(Qnt^0.5))/(Hnt^0.75);              %specific speed of PAT
nsp=nst/0.89;                               %Pump mode specific speed
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Efficiency determination
%knowing the pump mode flow rate and the pump mode specific speed, the
%efficiency can be determined from Figure
Qnp1=Qnt/1.3;                               %Pump mode flow rate
Etap=0.70;                                  %Maximum pump efficiency from Figure D3
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Converting Turbine design condition to Pump design using specific speed and
%efficiency of pump
%The conversion factor for head and flow are obtained using Figure D2 and D3
%respectively
Ch=1.37;                                    %Conversion factor for head using Figure D2
Cq=1.56;                                    %Conversion factor for flow rate using D3
Hnp=Hnt/Ch;                                 %Pump mode nominal head
Qnp=Qnt/Cq;                                 %Pump mode nominal flow rate
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%
%Converting pump design condition at turbine nominal speed to pump nominal
%speed
Np=2900;                                              %Selected pump rated speed in RPM
Hp=Hnp*((Np/Nt)^2);                                 %Nominal head at pump rated speed 2900 RPM
Qp=Qnp*(Np/Nt);                                     %Nominal flow at pump rated speed 2900 RPM
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%
fprintf('rated Flow rate of pump in (m^3/s) is %0.2f \n\n ',Qp)
fprintf('rated Head of the pump in (m) is %0.2f \n\n ',Hp)
fprintf(' rated  pump specific Speed in (m,m^3/s)is %0.2f \n\n ',nsp)
fprintf(' Pump mode nominal flow rate in (m^3/s)is %0.2f \n\n ',Qnp)
fprintf(' Pump mode nominal head in (m)is %0.2f \n\n ',Hnp)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
Rated Flow rate of pump in (m^3/s) is 0.04
 Rated Head of the pump in (m) is 134.31
 Rated pump specific Speed in (m,m^3/s)is 17.26

Pump mode nominal flow rate in (m^3/s)is 0.0224
Pump mode nominal head in (m)is 33.58
```

converting the best efficiency point of the selected pump into turbine mode

```
Np=2900;                                          %selected nominal Pump speed in RPM
Hnp=33.58;                                        %Pump mode nominal head in (m)
Qnp=0.0244;                                       %Pump mode nominal discharge in (m^3/s)
nsp=(Np*(Qnp^0.5))/(Hnp^0.75);                    %specific speed of Pump
%The conversion factor for head and flow
Ch=1.39;                                          %Conversion factor for head
Cq=1.27;                                          %Conversion factor for flow rate
% the following scattering factors are proposed
%on conversion factor for head: = + or - 10%
%on conversion factor for flow: = + or - 7.5%
Chmax=1.1*Ch;                                     %maximum Conversion factor for head
Chmin=0.9*Ch;                                     %minimum Conversion factor for head
Cqmin=0.925*Cq;                                   %minimum Conversion factor for flow
Cqmax=1.075*Cq;                                   %maximum Conversion factor for flow
% calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors
Hnt1=Chmax*Hnp;              % maximum nominal turbine head at pump speed Np
Hnt2=Chmin*Hnp;              % minimum nominal turbine head at pump speed Np
Qnt1=Cqmax*Qnp;            % maximum nominal turbine flow at pump speed Np
Qnt2=Cqmin*Qnp;             % minimum nominal turbine flow at pump speed Np
Nt=1450;                                 %selected nominal turbine speed in RPM
%converting turbine bep into nominal turbine speed nt using the affinity law
Hnt3=Hnt1*(Nt/Np)^2;         % maximum nominal turbine head at turbine speed Np
Hnt4=Hnt2*(Nt/Np)^2;         % minimum nominal turbine head at turbine speed Np
Qnt3=Qnt1*(Nt/Np)^2;         % maximum nominal turbine head at turbine speed Np
 Qnt4=Qnt2*(Nt/Np)^2;    % minimum nominal turbine head at turbine speed Np
  g=9.81;                        %Gravitational acceleration m/s^2
rho=1000;                        %Density of water in kg/m^3
P1=rho*g*Hnt3*Qnt3*(0.813-0.03);               % maximum power
```

```
P2=rho*g*Hnt4*Qnt4*(0.813-0.03);                    % minimum power

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%

fprintf('rated Maximum power in (w) is %0.2f \n\n ',P1)

fprintf('rated Minimum power in (w) is %0.2f \n\n ',P2)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%
```

```
rated Maximum power in (w) is 821.11

rated Minimum power in (w) is 578.07
```

2.2 Pump Selection Procedure proposed by Chapallaz for Bole arabsa 2B

```
g=9.81;                                      %Gravitational acceleration m/s^2

rho=1000;                                    %Density of water in kg/m^3

Qnt=0.05;                                    %Nominal turbine flow rate in m^3/s

Hnt=40.33;                                   %Available site head in m

Nt=1450;                                      %selected nominal PAT speed in RPM

nst=(Nt*(Qnt^0.5))/(Hnt^0.75);               %specific speed of PAT

nsp=nst/0.89;                                 %Pump mode specific speed

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%

%Efficiency determination

%knowing the pump mode flow rate and the pump mode specific speed, the

%efficiency can be determined from Figure D1

Qnp1=Qnt/1.3;                               %Pump mode flow rate

Etap=0.77;                                  %Maximum pump efficiency from Figure D3

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%

%converting Turbine design condition to Pump design using specific speed and

%efficiency of pump

%the conversion factor for head and flow are obtained using Figure D2 and D3

%respectively

Ch=1.49;                                     %Conversion factor for head using Figure D2

Cq=1.36;                                     %Conversion factor for flow rate using D3

Hnp=Hnt/Ch;                                 %Pump mode nominal head

Qnp=Qnt/Cq;                                 %Pump mode nominal flow rate

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%
```

```
%converting pump design condition at turbine nominal speed to pump nominal
%speed
Np=2900;                                  %Selected pump rated speed in RPM
Hp=Hnp*((Np/Nt)^2);                       %Nominal head at pump rated speed 2900 RPM
Qp=Qnp*(Np/Nt);                           %Nominal flow at pump rated speed 2900 RPM
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
fprintf('rated Flow rate of pump in (m^3/s) is %0.2f \n\n ',Qp)
fprintf('rated Head of the pump in (m) is %0.2f \n\n ',Hp)
fprintf(' rated  pump specific Speed in (m,m^3/s)is %0.2f \n\n ',nsp)
fprintf(' Pump mode nominal flow rate in (m^3/s)is %0.2f \n\n ',Qnp)
fprintf(' Pump mode nominal head in (m)is %0.2f \n\n ',Hnp)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
Rated Flow rate of pump in (m^3/s) is 0.07
 Rated Head of the pump in (m) is 108.27
  Rated pump specific Speed in (m,m^3/s) is 22.76


Pump mode nominal flow rate in (m^3/s) is 0.0368 (132.48m3/hr)
  Pump mode nominal head in (m) is 27.07
```

```
%converting the best efficiency point of the selected pump into turbine mode
Np=2900;                                        %selected nominal Pump speed in RPM
Hnp=27.07;                                      %Pump mode nominal head in (m)
Qnp=0.0368;                                     %Pump mode nominal discharge in (m^3/s)
nsp=(Np*(Qnp^0.5))/(Hnp^0.75);                  %specific speed of Pump
%The conversion factor for head and flow
Ch=1.36;                                        %Conversion factor for head
Cq=1.29;                                        %Conversion factor for flow rate
% the following scattering factors are proposed
%on conversion factor for head: = + or - 10%
%on conversion factor for flow: = + or - 7.5%
Chmax=1.1*Ch;                                   %maximum Conversion factor for head
Chmin=0.9*Ch;                                   %minimum Conversion factor for head
Cqmin=0.925*Cq;                                 %minimum Conversion factor for flow
Cqmax=1.075*Cq;                                 %maximum Conversion factor for flow
% calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors
Hnt1=Chmax*Hnp;              % maximum nominal turbine head at pump speed Np
Hnt2=Chmin*Hnp;            % minimum nominal turbine head at pump speed Np
Qnt1=Cqmax*Qnp;                                % maximum nominal turbine flow at pump speed Np
Qnt2=Cqmin*Qnp;              % minimum nominal turbine flow at pump speed Np
Nt=1500;                     %selected nominal turbine speed in RPM
%converting turbine bep into nominal turbine speed nt using the affinity law
Hnt3=Hnt1*(Nt/Np)^2;         % maximum nominal turbine head at turbine speed Np
Hnt4=Hnt2*(Nt/Np)^2;                         % minimum nominal turbine head at turbine speed Np
Qnt3=Qnt1*(Nt/Np)^2;         % maximum nominal turbine head at turbine speed Np
 Qnt4=Qnt2*(Nt/Np)^2;        % minimum nominal turbine head at turbine speed Np
  g=9.81;                    %Gravitational acceleration m/s^2
rho=1000;                    %Density of water in kg/m^3
P1=rho*g*Hnt3*Qnt3*(0.834-0.03);             % maximum power
```

```
P2=rho*g*Hnt4*Qnt4*(0.834-0.03);                    % minimum power

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%

fprintf('rated Maximum power in (w) is %0.2f \n\n ',P1)

fprintf('rated Minimum power in (w) is %0.2f \n\n ',P2)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%
```

```
rated Maximum power in (w) is 1166.71

rated Minimum power in (w) is 821.39
```

2.3 Pump Selection Procedure proposed by Chapallaz for chefy

```
>> g=9.81;                                              %Gravitational acceleration m/s^2

rho=1000;                                               %Density of water in kg/m^3

Qnt=0.0685;                                              %Nominal turbine flow rate in m^3/s

Hnt=28.78;                                              %Available site head in m

Nt=960;                                                 %selected nominal PAT speed in RPM

nst=(Nt*(Qnt^0.5))/(Hnt^0.75);                         %specific speed of PAT

nsp=nst/0.89;                                           %Pump mode specific speed

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%

%Efficiency determination

%knowing the pump mode flow rate and the pump mode specific speed, the

%efficiency can be determined from Figure D1

Qnp1=Qnt/1.3;                                        %Pump mode flow rate

>>Etap=0.80;                                            %Maximum pump efficiency from Figure D3

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%

%Converting Turbine design condition to Pump design using specific speed and

%efficiency of pump

%The conversion factor for head and flow are obtained using Figure D2 and D3

%respectively

Ch=1.37;                                             %Conversion factor for head using Figure D2

Cq=1.52;                                                %Conversion factor for flow rate using D3

Hnp=Hnt/Ch;                                            %Pump mode nominal head

Qnp=Qnt/Cq;                                           %Pump mode nominal flow rate

%Converting pump design condition at turbine nominal speed to pump nominal

%speed

Np=1450;                                              %Selected pump rated speed in RPM

Hp=Hnp*((Np/Nt)^2);                                  %Nominal head at pump rated speed 2900 RPM

Qp=Qnp*(Np/Nt);                                      %Nominal flow at pump rated speed 2900 RPM
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%

fprintf('rated Flow rate of pump in (m^3/s) is %0.2f \n\n ',Qp)

fprintf('rated Head of the pump in (m) is %0.2f \n\n ',Hp)

fprintf(' rated  pump specific Speed in (m,m^3/s)is %0.2f \n\n ',nsp)

fprintf(' Pump mode nominal flow rate in (m^3/s)is %0.2f \n\n ',Qnp)

fprintf(' Pump mode nominal head in (m)is %0.2f \n\n ',Hnp)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%
```

```
rated Flow rate of pump in (m^3/s) is 0.07

rated Head of the pump in (m) is 47.93

rated  pump specific Speed in (m,m^3/s)is 22.72


 Pump mode nominal flow rate in (m^3/s)is 0.045

 Pump mode nominal head in (m)is 21.01
```

```
%converting the best efficiency point of the selected pump into turbine mode
Np=1450; %selected nominal Pump speed in RPM
Hnp=21.01; %Pump mode nominal head in (m)
Qnp=0.045; %Pump mode nominal discharge in (m^3/s)
%specific speed of Pump
nsp=(Np*(Qnp^0.5))/(Hnp^0.75);
%The conversion factor for head and flow
>> %The conversion factor for head and flow
Ch=1.27;    %Conversion factor for head
Cq=1.38; %Conversion factor for flow rate
% the following scattering factors are proposed
%on conversion factor for head: = + or - 10%
%on conversion factor for flow: = + or - 7.5%
Chmax=1.1*Ch; %maximum Conversion factor for head
Chmin=0.9*Ch; %minimum Conversion factor for head
Cqmin=0.925*Cq; %minimum Conversion factor for flow
Cqmax=1.075*Cq; %maximum Conversion factor for flow
% calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors
Hnt1=Chmax*Hnp;  % maximum nominal turbine head at pump speed Np
Hnt2=Chmin*Hnp;  % minimum nominal turbine head at pump speed Np
Qnt1=Cqmax*Qnp;  % maximum nominal turbine flow at pump speed Np
Qnt2=Cqmin*Qnp;  % minimum nominal turbine flow at pump speed Np
Nt=960; %selected nominal turbine speed in RPM
%converting turbine bep into nominal turbine speed nt using the affinty law
Hnt3=Hnt1*(Nt/Np)^2; % maximum nominal turbine head at turbine speed Np
Hnt4=Hnt2*(Nt/Np)^2; % minimum nominal turbine head at turbine speed Np
Qnt3=Qnt1*(Nt/Np)^2; % maximum nominal turbine head at turbine speed Np
 Qnt4=Qnt2*(Nt/Np)^2; % minimum nominal turbine head at turbine speed Np
  g=9.81;  %Gravitational acceleration m/s^2
```

```
rho=1000; %Density of water in kg/m^3
P1=rho*g*Hnt3*Qnt3*(0.78-0.03); % maximum power
P2=rho*g*Hnt4*Qnt4*(0.78-0.03); % minimum power
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%
fprintf('rated Maximum power in (w) is %0.2f \n\n ',P1)
fprintf('rated Minimum power in (w) is %0.2f \n\n ',P2)
```

```
rated Maximum power in (w) is 2769.91
rated Minimum power in (w) is 1950.06
```

Two lines = (3.9-5.5)kw

3.3 Pump Selection Procedure proposed by Chapallaz for Kality

```
g=9.81;                                          %Gravitational acceleration m/s^2
rho=1000;                                           %Density of water in kg/m^3
Qnt= 0.434;                                         %Nominal turbine flow rate in m^3/s
Hnt=22.23;                                         %Available site head in m
Nt=760;                                           %selected nominal PAT speed in RPM
nst=(Nt*(Qnt^0.5))/(Hnt^0.75);                      %specific speed of PAT
nsp=nst/0.89;                                       %Pump mode specific speed
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%
%Efficiency determination
%knowing the pump mode flow rate and the pump mode specific speed, the
%efficiency can be determined from Figure D1
Qnp1=Qnt/1.3;                                    %Pump mode flow rate
Etap=0.86;                                      %Maximum pump efficiency from Figure D3
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%
%Converting Turbine design condition to Pump design using specific speed and
%efficiency of pump
%The conversion factor for head and flow are obtained using Figure D2 and D3
%respectively
Ch=1.28;                                       %Conversion factor for head using Figure D2
Cq=1.42;                                          %Conversion factor for flow rate using D3
Hnp=Hnt/Ch;                                      %Pump mode nominal head
Qnp=Qnt/Cq;                                      %Pump mode nominal flow rate
%Converting pump design condition at turbine nominal speed to pump nominal
%speed
Np=741;                                         %Selected pump rated speed in RPM
Hp=Hnp*((Np/Nt)^2);                              %Nominal head at pump rated speed 2900 RPM
```

```
Qp=Qnp*(Np/Nt);                              %Nominal flow at pump rated speed 2900 RPM

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%

fprintf('rated Flow rate of pump in (m^3/s) is %0.2f \n\n ',Qp)

fprintf('rated Head of the pump in (m) is %0.2f \n\n ',Hp)

fprintf(' rated  pump specific Speed in (m,m^3/s)is %0.2f \n\n ',nsp)

fprintf(' Pump mode nominal flow rate in (m^3/s)is %0.2f \n\n ',Qnp)

fprintf(' Pump mode nominal head in (m)is %0.2f \n\n ',Hnp)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%
```

```
rated Flow rate of pump in (m^3/s) is 0.30

rated Head of the pump in (m) is 16.51

rated  pump specific Speed in (m,m^3/s)is 54.95


Pump mode nominal flow rate in (m^3/s)is 0.31 (1116m3/hr)

Pump mode nominal head in (m)is 17.37
```

```
>>
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%
%converting the best efficiency point of the selected pump into turbine mode
Np=741; %selected nominal Pump speed in RPM
Hnp=17.37; %Pump mode nominal head in (m)
Qnp=0.31; %Pump mode nominal discharge in (m^3/s)
%specific speed of Pump
nsp=(Np*(Qnp^0.5))/(Hnp^0.75);
>> %The conversion factor for head and flow
Ch=1.22;    %Conversion factor for head
Cq=1.26; %Conversion factor for flow rate
% the following scattering factors are proposed
%on conversion factor for head: = + or - 10%
%on conversion factor for flow: = + or - 7.5%
Chmax=1.1*Ch; %maximum Conversion factor for head
Chmin=0.9*Ch; %minimum Conversion factor for head
Cqmin=0.925*Cq; %minimum Conversion factor for flow
Cqmax=1.075*Cq; %maximum Conversion factor for flow
% calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors
Hnt1=Chmax*Hnp;  % maximum nominal turbine head at pump speed Np
Hnt2=Chmin*Hnp;  % minimum nominal turbine head at pump speed Np
Qnt1=Cqmax*Qnp;  % maximum nominal turbine flow at pump speed Np
Qnt2=Cqmin*Qnp;  % minimum nominal turbine flow at pump speed Np
Nt=760; %selected nominal turbine speed in RPM
%converting turbine bep into nominal turbine speed nt using the affinty law
Hnt3=Hnt1*(Nt/Np)^2; % maximum nominal turbine head at turbine speed Np
Hnt4=Hnt2*(Nt/Np)^2; % minimum nominal turbine head at turbine speed Np
Qnt3=Qnt1*(Nt/Np)^2; % maximum nominal turbine head at turbine speed Np
```

```
 Qnt4=Qnt2*(Nt/Np)^2; % minimum nominal turbine head at turbine speed Np

  g=9.81;  %Gravitational acceleration m/s^2

rho=1000; %Density of water in kg/m^3

P1=rho*g*Hnt3*Qnt3*(0.85-0.03); % maximum power

P2=rho*g*Hnt4*Qnt4*(0.85-0.03); % minimum power

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%

fprintf('rated Maximum power in (w) is %0.2f \n\n ',P1)

fprintf('rated Minimum power in (w) is %0.2f \n\n ',P2)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
rated Maximum power in (w) is 87127.94

rated Minimum power in (w) is 61339.5
```

TWO turbines in parallel =(122.68-174.2)kw

Appendix-3: Selected sites pictures

Figure 38 C-1: Bole Bulubla WWTP site layout

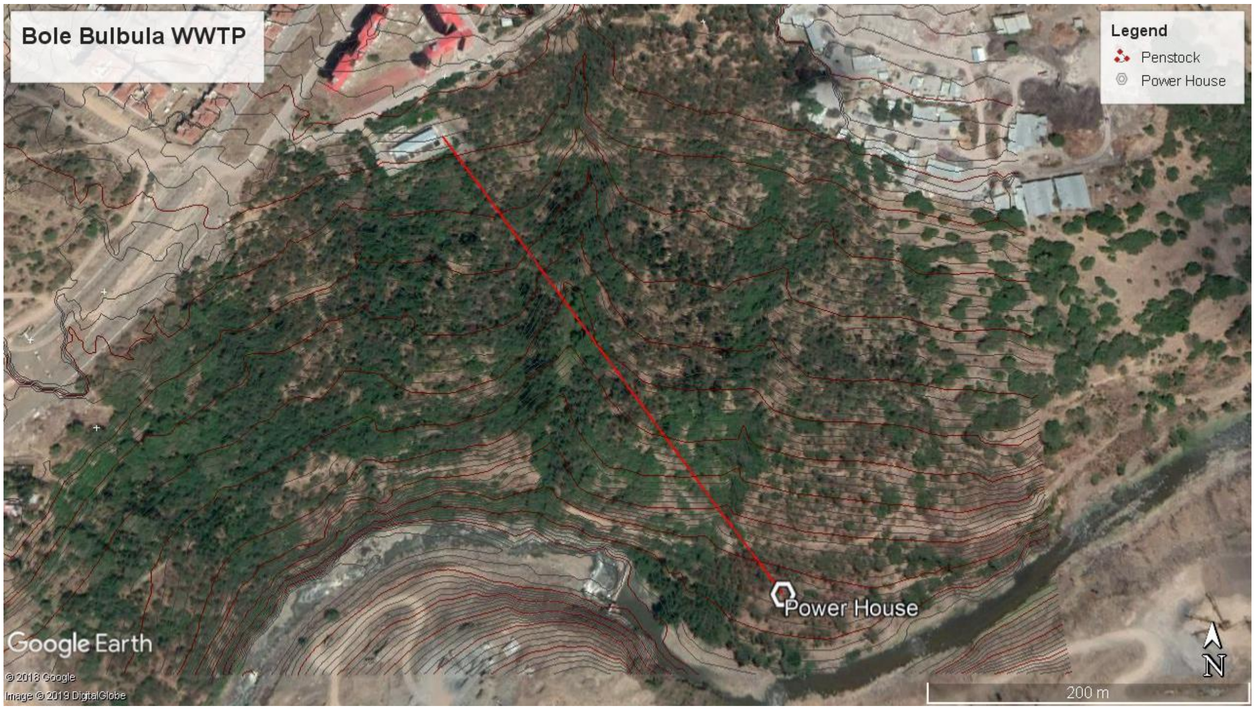


Figure 39 C-2: Bole Bulubla WWTP selected pump curve

Etanorm 065-050-160, n = 2900 rpm

Etanorm SYT, Etanorm V, Etabloc, Etabloc SYT

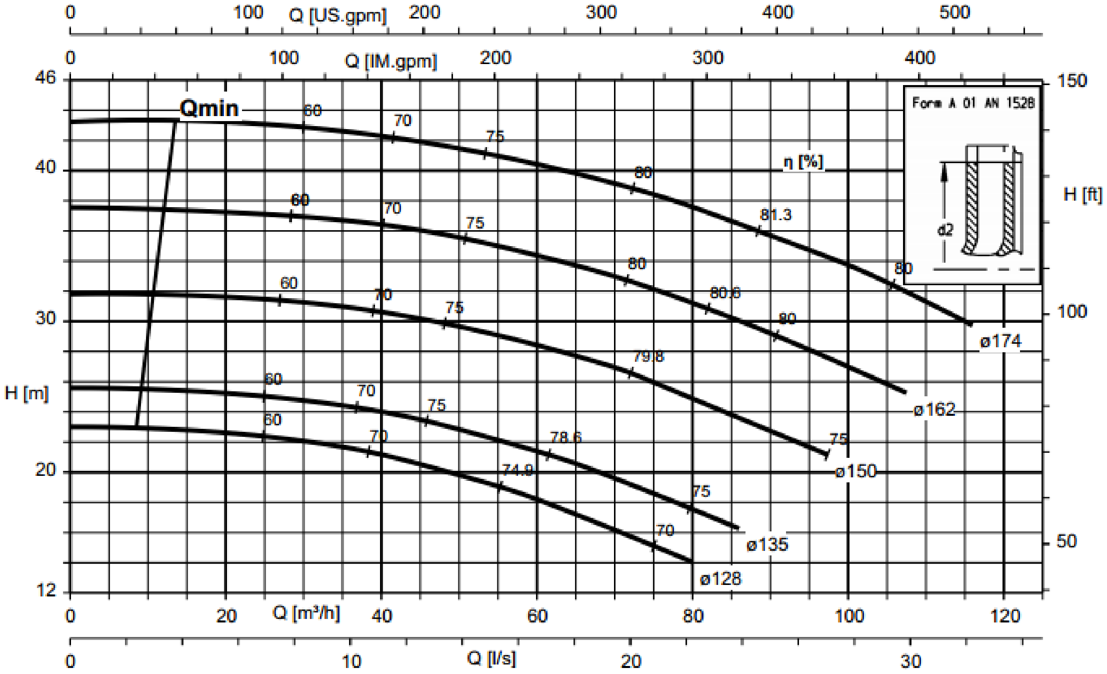


Figure 40 C-3: Bole Bulubla WWTP site profile

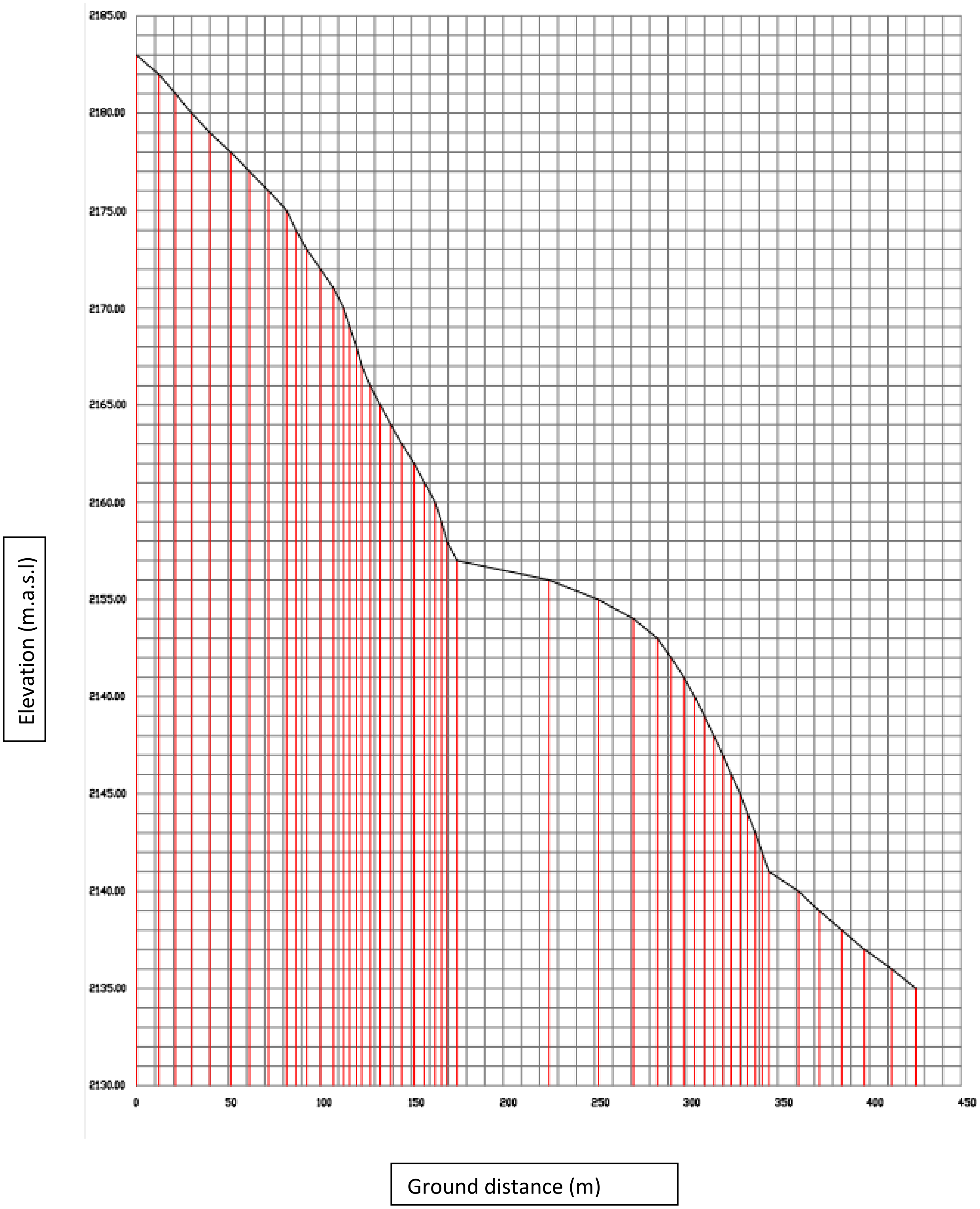


Figure 41 C-4: Bole Arabsa 2B WWTP site layout

Figure 42 C-5: Bole Arabsa 2B WWTP selected pump curve

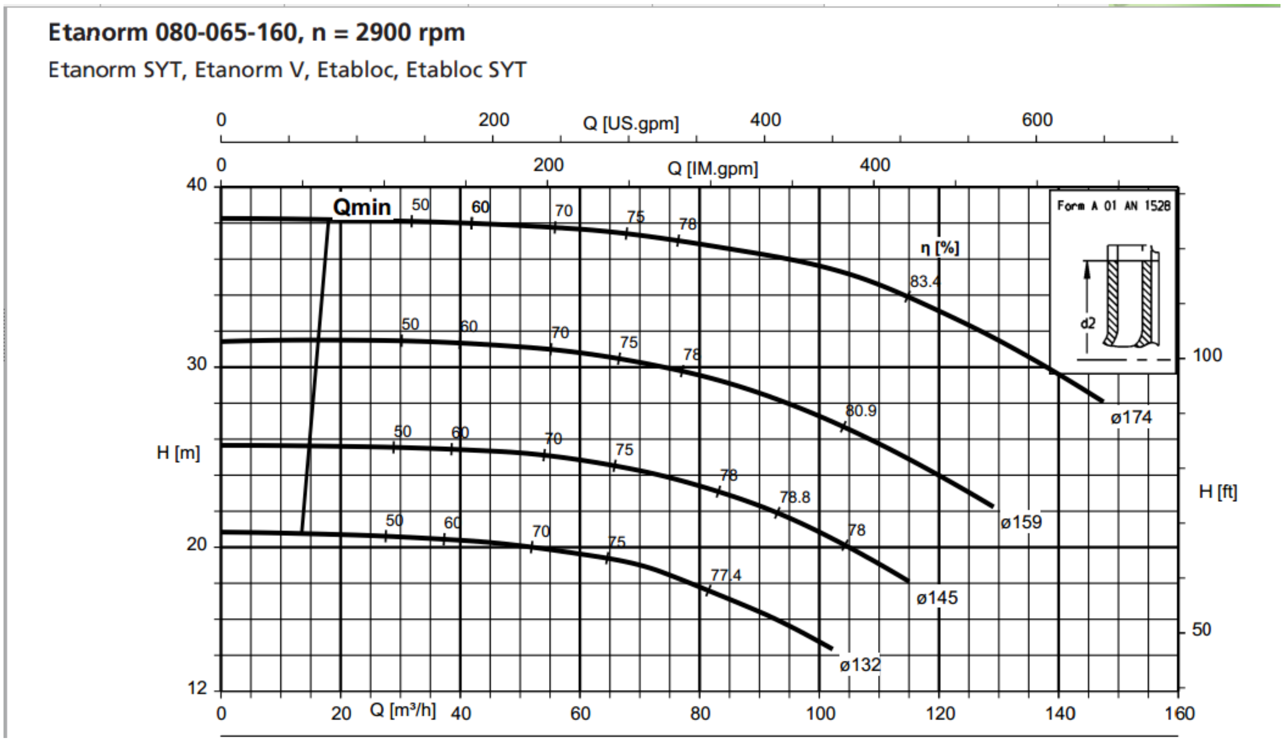


Figure 43 C-3: Bole Arabsa 2B WWTP site profile

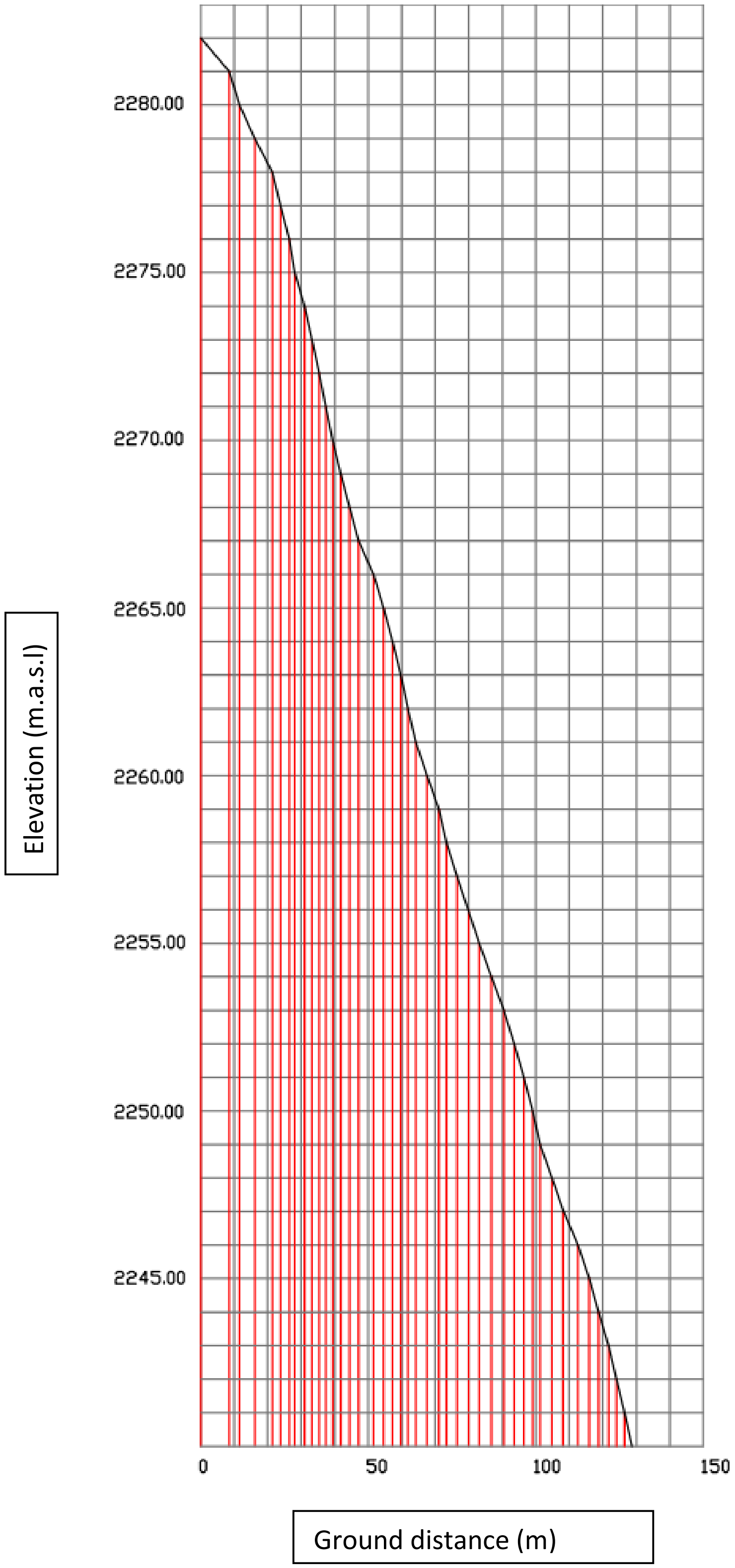


Figure 44 C-7: Chefy WWTP site layout

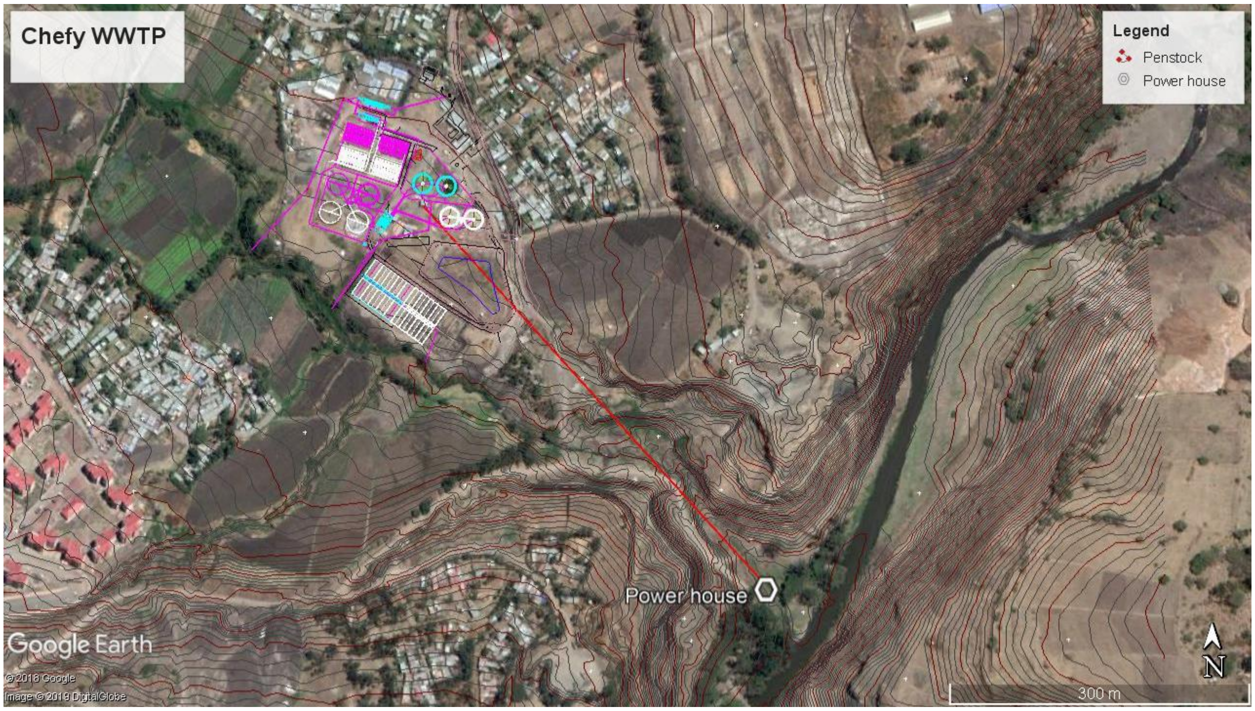


Figure 45 C-8: Chefy WWTP selected pump curve

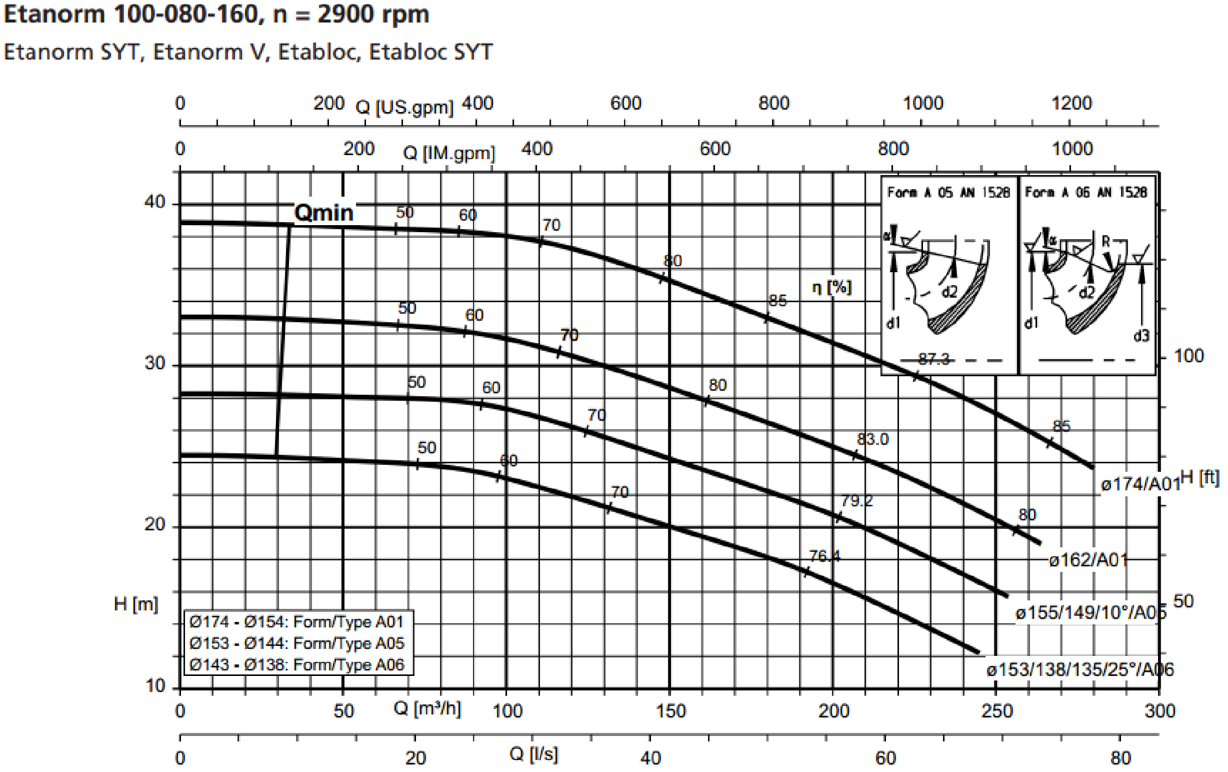


Figure 46 C-9: Chefy WWTP site profile

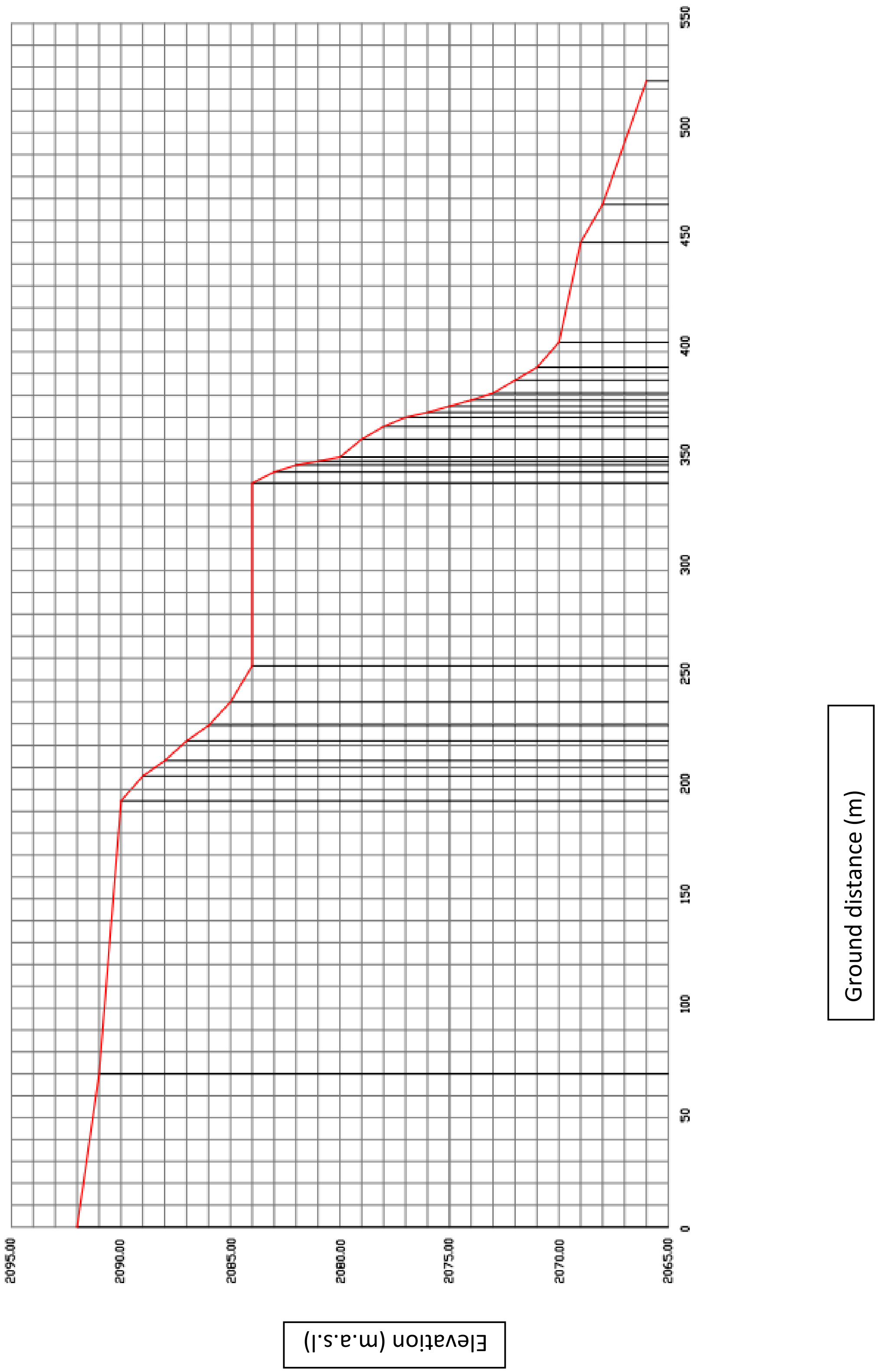


Figure 47 C-10: Kality WWTP site layout

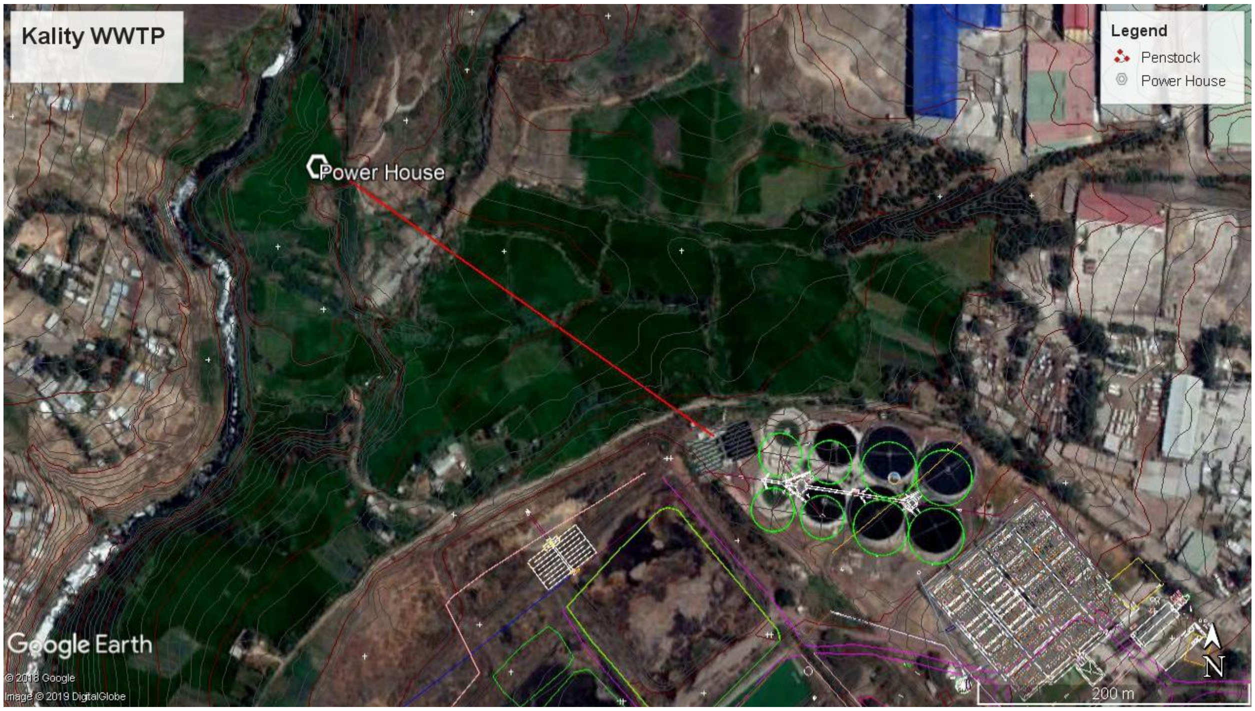


Figure 48 C-11: Kality WWTP site profile

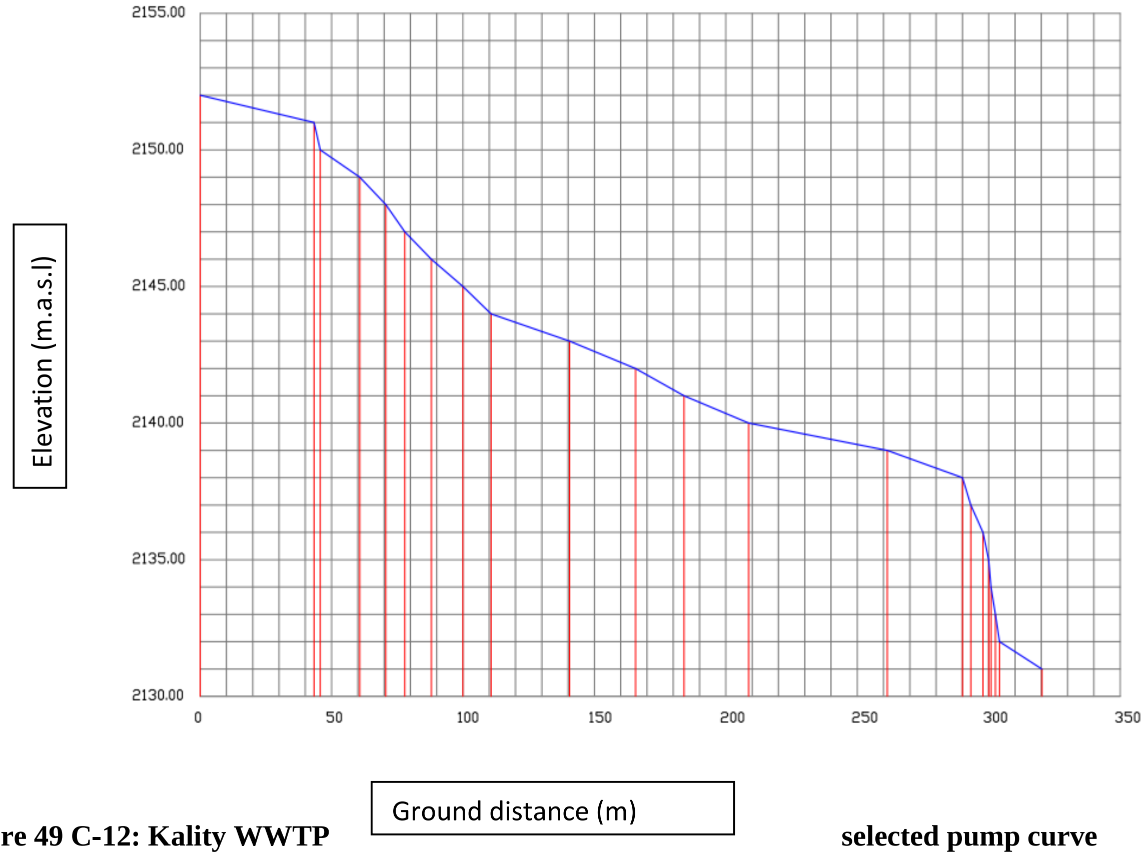
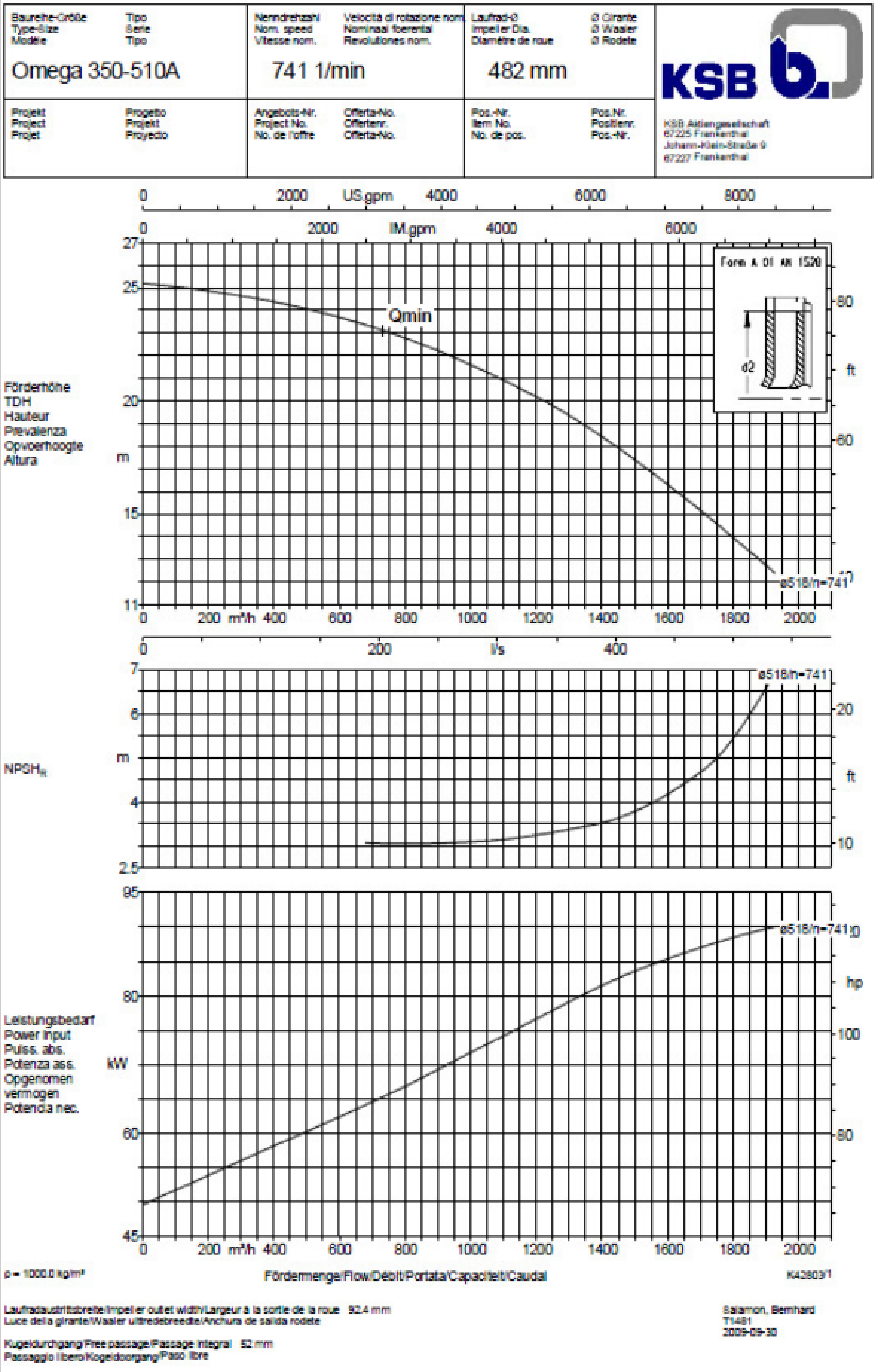


Figure 49 C-12: Kality WWTP selected pump curve



Baureihe-Größe Type-Size Modèle	Tipo Serie Tipo	Nenndrehzahl Nom. speed Vitesse nom.	Velocità di rotazione nom. Nominaal toerental Revoluciones nom.	Laufrad-Ø Impeller Dia. Diamètre de roue	Ø Girante Ø Waaier Ø Rodete	KSB Aktiengesellschaft 67225 Frankenthal Johann-Klein-Straße 9 67227 Frankenthal
Omega 350-510A		741 1/min		482 mm		
Projekt Project Projet	Progetto Projekt Proyecto	Angebots-Nr. Project No. No. de l'offre	Offerta-No. Offertenr. Oferta-No.	Pos.-Nr. Item No. No. de pos.	Pos.Nr. Positienr. Pos.-Nr.	

ρ = 1000.0 kg/m³

Fördermenge/Flow/Débit/Portata/Capaciteit/Caudal

K42803/1

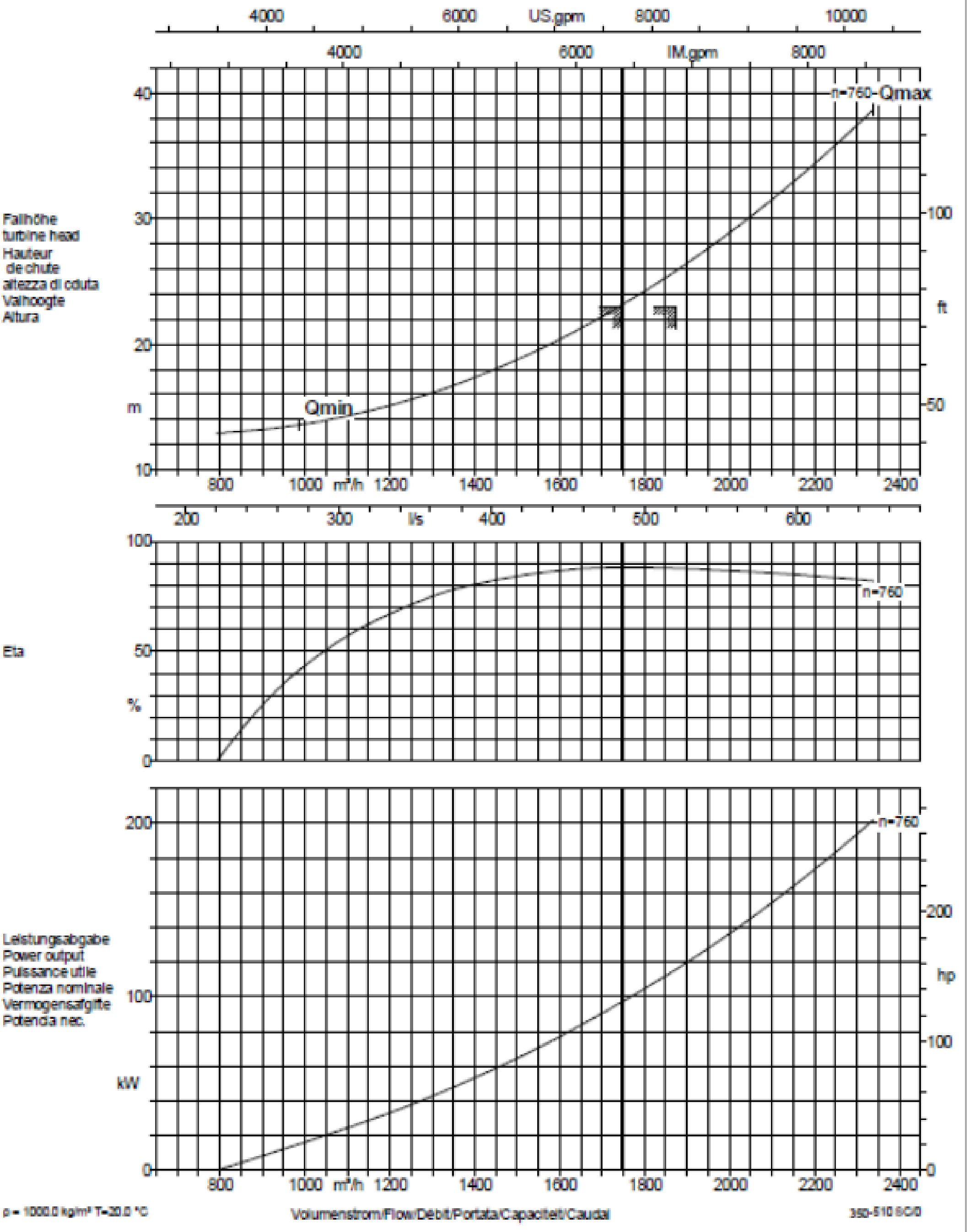
Laufradaustrittsbreite/Impeller outlet width/Largeur à la sortie de la roue 92.4 mm
Luce della girante/Waaier uittredebreedte/Anchura de salida rodete

Kugeldurchgang/Free passage/Passage integral 52 mm
Passaggio libero/Kogeldoorgang/Paso libre

Salamon, Bernhard
T1481
2009-09-30

Figure 50 C-13: Kality WWTP selected PAT curve

Baureihe-Größe Tipo Type-Size Serie Modèle Tipo Omega 350-510A Turbine	Nenndrehzahl Velocità di rotazione nom. Nom. speed Nominaal toerental Vitesse nom. Revoluciones nom. 760 1/min	Laufrad-Ø Ø Girante Impeller diameter Ø Waaier Diamètre de roue Ø Rodete 518 mm	KSB
Projekt Progetto Project Projekt Projet Proyecto Roman Bay Sea Farm	Angebots-Nr. Offerta-No. Project No. Offertenr. No. de l'offre Oferta-No. 2009-09-125	Pos.-Nr. Pos.Nr. Item No. Positienr. No. de pos. Pos.-Nr.	KSB Aktiengesellschaft 67225 Frankenthal Johann-Klein-Straße 9 67227 Frankenthal



ρ = 1000.0 kg/m³ T=20.0 °C

Volumenstrom/Flow/Débit/Portata/Capaciteit/Caudal

350-510 8C/0

tolerance:
tQ = +/- 9%
tH = +/- 7%
tETA = -7%

Salamon, Bernhard
T1481
2009-09-22