Design considerations for an ultra-low-head Kaplan turbine system

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Abstract. Interests in utilizing small hydraulic systems are increasing recently, and new technology was developed worldwide. In comparison to large hydraulic systems which are normally equipped with a high dam and sizeable turbine system, small hydraulic has significant environmental and economic advantages. Most of the research focuses on small hydraulic systems that operating between a 2m-30m head condition or zero head hydraulic kinetic conversion system. However, the ultra-low head condition (head condition between 1m-3m) is another attractive, renewable and effective resource for developing a hydraulic system. Traditional hydraulic turbines like the Francis, Crossflow, and Kaplan turbines were all been modified and designed for small hydraulic systems. However, designing for ultra-low head (ULH) condition required some non-traditional design considerations which had not been yet proposed. This paper proposes several overall geometry design considerations for developing a ULH Kaplan turbine.

1. Introduction
Traditionally, a water turbine is classified into two main categories: impulse turbines and reaction turbines, and each turbine type has its specific operation range and application. However, the sustainability of a large hydraulic system was questioned because of the environmental impact of dams and reservoirs on the local environment [1] [2]. Additionally, high initial investment and cost of manpower are also major barriers for developing countries or remote areas to construct a large hydraulic system. So, interests in utilizing a small hydropower system are increasing substantially, and different models and technology were developed worldwide. Compared to a large hydraulic system, a small hydropower system is both economically and environmentally friendly to the developer. Most of the current published research focuses on small hydropower systems that operated in low head condition (head between 3m-30m) or zero head hydraulic kinetic conversion technology [3]. Not much attention is paid to an ultra-low head (ULH) condition (head between 1m-3m) which will become a new attractive and efficient way for developing hydraulic systems with advanced design considerations and methods [3]. Compared to other small hydraulic systems, ULH could simplify civil works and reduce project work cost and time. It also could be installed near human population with less impact on the local environment [3]. Specifically, the low environmental impact of ULH hydropower is reflected in two main points: 1) the wide blade passage and low rotating speed (below 100rpm) can significantly reduce
collision damage for fish; and 2) because no dam or a very low dam is involved, barriers for fish migration and navigation are avoided and water flow downstream are ensured [3].

Currently, Francis, Kaplan, and the cross-flow turbine are three major turbine types for ULH application. Each type has its own strengths and drawbacks. The Francis turbine normally has fix blade geometries and high efficiency and stable performance on design conditions but tends to be less efficient when on off-design conditions [3]. For ULH sites, flow conditions tend to change from season to season, which means the Francis turbine only suitable if the site has a stable flow condition. Besides, with fixed blade geometries and small size, Francis turbine for ULH conditional is expensive and is difficult to machine [3]. The Kaplan turbine which is normally equipped with adjustable stators and blades (commonly refer as double regulated Kaplan turbine), is a great candidate for the ULH application with stable and efficient performance over wide flow conditions. However, the regulation mechanism is complicated and expensive that makes the Kaplan turbine units expensive and more suitable for larger flow rate conditions for the ULH situation. The Cross-flow turbine is another suitable candidate for the ULH application, even though the overall efficiency is less than the Kaplan or Francis turbines, it tends to have a flat performance curve under various loading conditions.

Based on increasing needs for the ULH turbine and lacking design considerations or methods, this paper focuses on proposing several design considerations for developing a ULH damless Kaplan turbine system with some CFD validation results. The topic covered in this paper could provide some guidelines for future ULH turbine system design.

2. Design Methodology and CFD modelling Considerations

All Geometries considerations and results mentioned in this paper were based on the design methodology and CFD modelling methods published by the author in reference [4] and [5]. Table 1 and 2 show some basic information about the design specifications and CFD methods.

| Table 1. Design Specification of the models. |
|---------------------------------------------|
| Design Specifications                       |
| Design Head [m] | Tip Diameter [m] | Design mass flow rate [kg/s] |
| 2.5 | 3.5 | 2000-5000 |

| Table 2. CFD modelling methods. |
|----------------------------------|
| CFD Methods                      |
| CFD Turbulence Model | Mesh Type | Mesh Size | Boundary Condition          | Convergence Rate |
| Steady State k-ω SST          | Tetrahedrons Mesh +10 layer near the wall | ~14 million | Pressure inlet + Pressure Outlet | $10^{-4}$ |

3. General Considerations of the ULH Kaplan Turbine

As mentioned in the introduction, the ULH hydropower system seeks to minimize civil costs, as a result, it normally has no dam and almost always a “run-of-the-river” installations (water storage capabilities are small or nonexistent). Therefore, in order to provide an ultra-low head condition, the turbine structure is considered equal to the dam and providing enough head for the system; and the positioning of the turbine structure is also related to turbine head conditions.

Traditionally, for low head applications, the turbine trends to have a relatively smaller overall size and operated at a high rotational speed (Typical bulb Turbine). However, for the ULH application, since the turbine is directly installed into the water, a smaller size and high rotational speed are significantly
dangerous to the local environment and disobey the main design principle of the ULH system. As a result, larger turbine area and low rotational speed are the most important design considerations for any ULH system design. Figure 1 shows an example of an ULH Kaplan system with turbine structure and turbine components. For a given head condition, there are four geometry parameters and one operation condition that need to be considered: 1) Turbine structure length, 2) Turbine positioning angle ($\theta$), 3) Turbine overall diameter, 4) Turbine hub diameter, 5) Rotational Speed.

![Figure 1](image)

**Figure 1.** An example model of a ULH Kaplan system from two different points of view.

3.1. **Turbine Structure and Positioning Considerations**

The turbine structure minimum length and turbine positioning angle ($\theta$) are related to the head condition as the following:

$$\text{Turbine Structure minimum length} \approx \frac{H}{\cos\theta}$$

(1)

For the Kaplan turbine there are three positioning options: 1) Vertically [$\theta = 90^\circ$], (2) Horizontally [$\theta = 0^\circ$], 3) Inclined [$15^\circ < \theta < 45^\circ$]. Figure 2 [6] shows the example of each positioning option. The vertically positioning option is usually for a traditional Kaplan turbine with inlet water channel and other water regulation system which is not suitable for ULH system. Horizontal and inclined are two suitable options for the ULH system; however, the inlet flow angle is equal to the turbine inclined angle. This angle is critical for designing the stator geometry. Figure 3 shows two stator geometries designed for the same ULH condition, the left one is the stator when it was horizontally positioned, the right one is the stator when it was positioned with $45^\circ$ angle. It was clear that when it was in the horizontal position, the stator had a larger deflect angle that can cause more flow separation around the stator that could lead to unevenly fluid distribution to the blade area and would affect the blade performance. As a result, for ULH Kaplan system, a general-inclined angle needed to be considered, Figure 4 shows the overall performance for three design conditions. Under each design conditions, three inclined angles were chosen ($\theta = 40^\circ, 45^\circ, 50^\circ$). The results show that when the mass flow rate is relatively low, the efficiency difference among three general inclined angles is small; however, when the mass flow rate increases, the difference becomes noticeable, and $40^\circ$ models have the best performance.

![Figure 2](image)

**Figure 2.** Kaplan turbine positioning option; From left to right: Vertical, Horizontal, Inclined.

![Figure 3](image)

**Figure 3.** Two Stator geometries.
3.2. Turbine Overall Diameter Considerations
The turbine’s overall diameter is another key geometry factor that needs to be carefully considered. De Siervo [7] proposed an empirical relation between the Kaplan turbine’s outer diameter and peripheral velocity coefficient $k_u$ based on 130 different Kaplan turbines back 1977:

$$D_{tip} = \frac{84.5k_u H_n}{n}; k_u = 0.79 + 1.61 \times 10^{-3} n_s; n_s = n P_t^{0.5} H_n^{-1.25}$$

Where $n_s$ is specific speed, $H_n$ is the design head, $n$ is rotational speed in RPM, $P_t$ is the Power output in Kw

Based on the equation, for example, a ULH system with a 2.5m head condition, the specific speed $n_s$ is between 83 to 128 when the flow rate is between $2m^3$ to $5m^3$. When operating under a low rotational speed condition, which is required for ULH system, the tip diameter (Turbine overall diameter) is between $3m$ to $3.5m$. This empirical relation is very useful for initially determining the overall diameter. Moreover, in order to minimize any head loss between the turbine structure and turbine, and also to avoid blockage of the water, the turbine’s overall diameter should be close to the turbine structure’s minimum length. Generally, those two considerations could be used as a gridline for initially determining the turbine’s overall diameter.

3.3. Turbine Hub Diameter Considerations
The turbine hub diameter is another key geometry parameter that needs more considerations. It could affect two major factors: 1) Hub volume and 2) Blade geometry. The hub consists of two major parts of the Kaplan turbine system: A generator and an adjustable blade control mechanism. A small hub diameter means a large turbine area with a small hub volume, which could be a benefit for the environment but may cause difficulties for fitting all the turbine parts. On the other hand, a large hub diameter may have the opposite problem. Additionally, the hub diameter has a major influence on the turbine blade geometry. Figure 5 shows three examples of blades with small, medium, and large hub diameters. It is clear that a smaller hub diameter means more twisted blade geometry because of the low

![Figure 4. The relations between Performance and inclined angle under three hub diameters.](image-url)
peripheral velocity at a low radius span. This would cause a higher machining cost and higher separation possibilities. So it is important to define a minimum hub diameter for preventing a large blade twist angle. If we assume a free vortex condition, the minimum hub diameter is defined with the blade angle ($\beta_2$) equal to zero; this diameter could be calculated as

$$D_{hub min} = \frac{2 \sqrt{\eta g H n}}{n}; \eta \text{ is turbine efficiency}$$  \hspace{1cm} (3)

This equation would only provide a limited guideline for choosing the hub diameter, and the hub diameter is also related to the design mass flow rate. Figure 6 shows the relation between turbine efficiency and the mass flow rate for three hub diameters. Results show that smaller hub diameter has a flatter performance curve over the mass flow rate ranges than the larger hub diameter. The larger hub diameter shows a significantly better performance over the small mass flow rate conditions. So generally, determining turbine hub diameter is a balancing process between turbine areas, hub volume, and flow rate. Different river conditions and design conditions could result in different geometry configurations, this section only provides some basic considerations for hub diameter.

![Figure 5. Three blade geometry with different hub diameter; From left to right: small, medium and Large.](image)

![Figure 6. The relation between performance and three hub diameter configurations over a certain mass flow rate range.](image)

3.4. **Turbine Operation Condition Considerations**

Being environmental friendly is an essential feature for any ULH turbine design, in order to reduce collision possibilities, the rotational speed of a ULH Kaplan turbine is normally lower than a traditional design. The US Department of Energy published a summary of environmentally friendly turbine design concepts in 1999 [8], which show some guidelines for choosing an appropriate rotational speed. This paper states that the peripheral runner speed should be less than 40ft/s (12.2 m/s) and preferably 20ft/s (6 m/s). So for a 3.5m diameter turbine, the rotational speed should be between 32rpm to 64rpm. Additionally, having various rotational speeds is a good regulation method for an off-design condition, so it is always better not to choose an extremely high rotational speed for a design point, and leave some margins for future off-design considerations.
4. Turbine Blade Geometry Considerations

4.1. Velocity Assumptions Considerations

All turbine designs start with velocity calculation at each span location, so it is critical to define the velocity component. Figure 7 shows a typical velocity triangle for a Kaplan turbine. Correctly determining the velocity component at each radial span location is the most important part of the turbine blade design. Conventionally, there are three widely used velocity assumptions:

1) Free vortex \[ rC_u = \text{constant} \]
2) Force vortex \[ C_u = Kr \]
3) Constant vortex \[ C_u = \text{constant} \]

Figure 8 shows an example of three blade profiles constructed by three different velocity assumptions. Even though the blade geometries are completely different, the performances are very similar with only \( \pm 0.5\% \) difference. However, Abdul Muis and Priyono Sutikno [9] state that even though the performances were similar, three different velocity assumptions have distinct pressure distribution patterns, which need to be carefully examined when conducting cavitation and stress analysis.

There is another velocity assumption that has a potential for turbine designed proposed by Albuquerque [10]. This method includes an energy balance across the turbine blade and introduces a pressure loss component into the radial equilibrium equation. So, any loss mechanism could cause the total pressure loss across the blade row, this could be written as:

\[
P_{t1} - P_{t2} = \rho Y_{LS}
\]

where \( Y_{LS} \) is the mechanical loss per unit mass
subscript 1 is blade inlet, subscript 2 is blade outlet

And a typical radial equilibrium equation for the blade is

\[
\frac{1}{\rho} \frac{dP_c}{dr} = \frac{c_u}{r} \frac{d}{dr} (rc_u) + c_x \frac{d c_x}{dr}
\]

(5)

By introducing the total pressure loss equation into the radial equilibrium equation, the new equilibrium equation could be written as:

\[
c_u^2 \frac{d(r c_u^2)}{dr} + r c_x^2 \frac{dc_x^2}{dr} + r \frac{d Y_{LS}}{dr} = 0
\]

(6)

If one neglects the radial loss variation term \( \frac{d Y_{LS}}{dr} \), the above equation becomes a free vortex condition. By integrating equation (6), the distribution of normal velocity \( (c_x) \) could be related to the velocity torque \( (r c_u^2) \) as

\[
c_x^2(r) - c_{x,hub}^2 = 2[Y_{LS}(r_h) - Y_{LS}(r)] + \int_{r_h}^{r} \frac{1}{r^2} \frac{d(r c_u^2)}{dr} \, dr = I_s(r)
\]

(7)
And by using the continuity equation [equation (8)] the velocity component could be solved iteratively if the loss is known.

\[ \int_{r_1}^{r_2} \sqrt{\frac{c_{hub}^2}{r^2}} + 1_s(r) \, rdr = \frac{Q}{2\pi} \]  

(8)

This method is not stable enough for every ULH condition and will end up unconverged for some conditions because of lacking loss models. However, if there are enough loss models developed for the ULH condition, this velocity method could be perfect for velocity calculation and overall performance prediction.

4.2. Blade Camber-Line Construction Considerations

After determining velocity components, the blade camber line could be determined. Since the camber line is the key to any blade geometry design, the method needs to have specific consideration. Traditionally, the Bezier curve is commonly used for camber line design. Figure 10 shows the Bezier curve for constructing the runner blade camber line. The 3-Point Bezier curve is commonly used with point \( P_0, P_2, P_4 \) shows in Figure 10.

The position of those points is dependent on the velocity angle and the blade stagger angle. For a standard unit-length blade camber line, \( P_0 = (0,0) \), \( P_4 = (1,0) \) then, in order to match the velocity direction, two lines are drawn that are both tangential to the curve at inlet and outlet. The intersection point of two lines is the \( P_2 \),

\[ P_2 = (P_{2x}, P_{2y}) = \left( \frac{1}{\cot \phi_{1c} + \cot \phi_{2c}}, \frac{\cot \phi_{1c}}{\cot \phi_{2c}} \right) \]

(9)

where, \( \phi_{1c} \) and \( \phi_{2c} \) are auxiliary angles, for runner blade design:

\[ \phi_{1c} = \gamma_r + \beta_2 \]

\[ \phi_{2c} = 180^\circ - \gamma_r - \beta_3 \]

(10)

In order to construct a continuous curve:

\[ 90^\circ - \beta_3 < \gamma_r < 90^\circ - \beta_2; \gamma_r = (90^\circ - \beta_3) + C_{RSA}(\beta_3 - \beta_2) \]

\( C_{RSA} \) is Runner Blade Stagger Angle setting constant, and \( 0 < C_{RSA} < 1 \) (11)

This Stagger angle setting constant is critical for turbine blade design and has a major influence on turbine overall performance. Figure 9 shows examples of runners and stators with different Stagger angle setting constant. And Figure 11 shows how both runner stagger angle and stator stagger angle influence the overall performance for one selected design condition. It is clear that stagger angle setting constant has a major impact on turbine design and performance. There is no optimum value for stagger angle setting constant since it depend on flow rate and other geometry, but a similar trend as in Figure 11 has been observed. Therefore, when designing a blade for Kaplan turbine system, stagger angle need special considerations, and above is some guideline for it.

Even though the 3-point Bezier curve is enough for blade design, it has one major disadvantage: for a given velocity condition (which means \( \beta_2 \) and \( \beta_3 \) are given), blade profile is only the function of stagger angle. This means, for a fixed stagger angle, the blade profile could not be changed, and cause less fixable control for further optimizations. So instead of second 3-point Bezier curve, 5-point Bezier curve is used by adding two more points for constructing blade camber line. One point \( P_1 \) is on the straight line \( P_0P_2 \):

\[ P_1 = (c_1P_{2x}, c_1P_{2y}) \]

(12)

Another point \( P_3 \) is on the straight line\( P_2P_4 \):

\[ P_3 = (1 - (1 - P_{2x})c_2, c_2P_{2y}) \]

(13)

\( C_1 \) and \( C_2 \) are two coefficients to control the points’ location. By adding those two points, not only the velocity conditions could be met, but also increase the flexibility for further optimization.
4.3. Runner Blade Number And Solidity Considerations

For the ULH Kaplan turbine, there is no clear relationship between runner blade number and general performance. However, since the number of runner blade is important for both performance and economic purpose, it should be traded carefully. Additionally, the blade chord length is also related to blade number as:

\[
\text{Blade Chord length} = \frac{\sigma \pi D_{\text{tip}}}{N}; \text{where } \sigma \text{ is blade solidity, } N \text{ is blade number}
\]  \hspace{1cm} (14)

So it is important to consider the blade solidity and the blade number together for any blade design. Zweifel proposed the famous Zweifel number for finding the optimum solidity for a given velocity condition [11]. And, Meinhard T. Schobeiri [12] states that when the solidity equal to one, the profile loss of the blade is at its lowest. For ULH design, solidity and blade number are important, since less and shorter blade could reduce bearing lost, overall cost and complexity of the whole system.

Figure 12 shows how the runner blade number affect turbine performance for three design mass flow rates. Six different blade numbers were chosen (4,5,6,8,10,12), and it was clear that the 8-blade model has the highest efficiency which is reasonable since less blade would not have enough fluid guide and would cause flow separation; too many blades would have larger skin friction. Therefore, there should be a balance between flow separation and skin friction, and an 8-blade configuration looks like a great choice for blade number especially for high mass flow rate condition. For small and medium mass flow rate condition, a 4-blade model has around 2% less efficiency than an 8-blade model which is acceptable in order to reduce the turbine overall costs.

Figure 13 shows how the runner blade solidity affect turbine performances for a low flow rate condition under two different blade number configurations (5, 8). Six different blade solidities were chosen (0.7, 0.8, 0.9, 1, 1.1, 1.2), and it was clear that the optimum blade solidity occurs between 0.9 and 1 which is close to Meinhard’s claims. Shorten or prolonging the runner blade could affect overall performance dramatically.

Figure 9. Runner Blade and Stator Blade with different stagger angle setting constant. From left to right: \(C_{SA} = 0.3; 0.5; 0.7\) for both runner and stator blade.
4.4. Stator Blade Number And Solidity Consideration

The stator blade number and solidity are equally important for turbine system design. Since the ULH turbine system often installs into the water system directly, the stator also acts as trash rack for preventing random floating object damaging the blade. Less stator may lead to less fluid guidance for turbine runner; too much stator may lead to a less open area for the fish environment. The stator number depends on site condition hugely. Figure 14 shows how the stator number affects overall performance for three design mass flow rate conditions. Six different stator numbers were chosen (20, 30, 40, 50, 60), it shows that depend on the design mass flow rate, the efficiency peak occurs at around 30-40 stator number, and the efficiency difference is about 2%. This would provide some guideline for choosing a right stator number.

Figure 15 shows how stator solidity affects turbine performance for three selected flow conditions, a dozen different blade solidity were chosen (between 0.7 and 3). Unlike runner blade, the optimum solidity is not near one. When solidity larger than one, the efficiency trends to maintain a high value over a large range of solidity, then drop quickly because of increase of the thickness. Figure 16 and 17 show how solidity affect both runner and stator geometry. Since the runner blade has a thicker leading edge, large solidity means the adjacent blade’s trailing edge is very close to the next blade leading edge. This makes large runner blade solidity has a large fluid blockage, thus cause an efficiency drop. However, for stator blades, the thickness is relatively small, thus the solidity has less effect on overall performance. One thing worth notice, larger stator blade solidity means longer blade which leads to increase stator thickness, this will reduce flow passage inside the stator which would cause lower design mass flow rate and may cause harm to fish immigration.
Figure 12. The relation between Runner Blade Number and performance under three mass flow rate conditions.

Figure 13. The relation between Runner Blade Solidity and performance under two blade number configurations.

Figure 14. The relation between Stator Blade Number and performance under three mass flow rate conditions.

Figure 15. The relation between Stator Blade Solidity and performance for one selected condition.
5. Conclusions
Designing an ULH hydraulic system is a balancing process, all geometries, design specifications and operation conditions are related and have a major influence on turbine overall performance. The untraditional nature of the ULH system design means various geometry should be considered with care, and different approaches need to be developed. This paper provides some important design considerations and approaches of some essential geometry parameters include three parts: general geometry; general positioning; general blade geometry. The ULH hydraulic system is a promising new technology for utilizing the low head hydraulic resource and this paper proposes serval considerations that could be benefited for future researches.

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